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FABRICATION of CHAIN REPLACING by SHAFT DRIVE

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Abstract:A chainless bike is a bike that uses a determined shaft rather than a fasten to transmit control from the pedals to the wheel. As of late, because of headways in inner apparatus innovation, few present day shaft-driven bikes have been presented. The pole drive just needs intermittent oil utilizing an oil weapon to keep the riggings running tranquil and smooth. This "chainless" drive framework gives smooth, very and proficient exchange of vitality from the pedals to the back wheel. It is alluring in look contrast and chain driven bike. It replaces the conventional strategy. This task is produced for the clients to pivot the back wheel of a bike utilizing propeller shaft. As a rule in bikes, chain and sprocket technique is utilized to drive the back wheel. Be that as it may, in this undertaking, the Engine is associated at the front piece of the vehicle.

Keywords--- *Bevel gears, Fabrication, Propeller shaft,Shaft Driven Bicycle*

1. INTRODUCTION

A pole driven bike is a bike that uses a drive shaft rather than a tie to transmit control from the pedals to the wheel course of action showed in the accompanying fig Shaft drives were presented over a century back yet were generally supplanted by affix driven bikes because of the rigging ranges conceivable with sprockets and derailleur. As of late, because of progressions in inward rigging innovation, few present-day shaft-driven bikes have been presented. Shaft driven cycle have a substantial slope outfit where a traditional cycle would have its chain ring. This cross section with another slant outfit mounted on the drive shaft. The goal of this work is to decrease the human exertion and satisfying the excitement of riding bike by supplanting the current chain drive framework with incline gears. The slope gears are put at the backwheels.

The back wheels will turn with the assistance of torque transmitted from the pedals to the drive shaft with 90° pivot of slant gears. The speed proportion is three that implies for each one turn of the apparatus the pinion finishes three insurgencies. Here the power transmission is opposite way as pinion and rigging pivot is opposite to each other.

2. LITERATURE REVIEW

The principal shaft drives for cycles seem to have been developed autonomously in 1890 in the United States and England. The Drive shafts are transporters of torque; they are liable to torsion and shear pressure, which speaks to the distinction between the info compel and the heap. They along these lines should be sufficiently solid to endure the worry, without forcing excessively extraordinary an extra inactivity by goodness of the heaviness of the pole. Most vehicles today utilize unbending driveshaft to convey control from a transmission to the wheels. A couple of short driveshaft is ordinarily used to send control from a focal differential, transmission, or transaxie to the wheels.

3. CONSTRUCTION AND WORKING

The term Drive shaft is utilized to allude to a pole, which is utilized for the exchange of movement starting with one point then onto the next. While the poles, which drive is alluded to as the propeller shafts. However the drive shaft of the car is additionally alluded to as the propeller shaft on the grounds that separated from transmitting the rotational movement from the front end to the backside of the vehicle, these poles likewise move the vehicle forward. The pole is the essential association between the front and the backside, which performs both the employments of transmitting the movement and driving the front end.

Accordingly the terms Drive Shaft and Propeller Shafts are utilized reciprocally. At the end of the day, a drive shaft is a longitudinal power transmitting, utilized as a part of vehicle where the pedal is arranged at the human feet. A drive shaft is a get together of at least one tubular shafts associated by general, steady speed or adaptable joints. The quantity of tubular pieces and joints relies upon the separation between the two wheels.

Fig 1 : Construction & Working

4. SELECTION OF METHODOLOGY

4.1 Selection of bevel gear

Fig 2 : Position Of Bevel Gear

4.2 Selection of Drive Shaft

Fig 3 : Drive Shaft

4.3 Placing of bevel gear

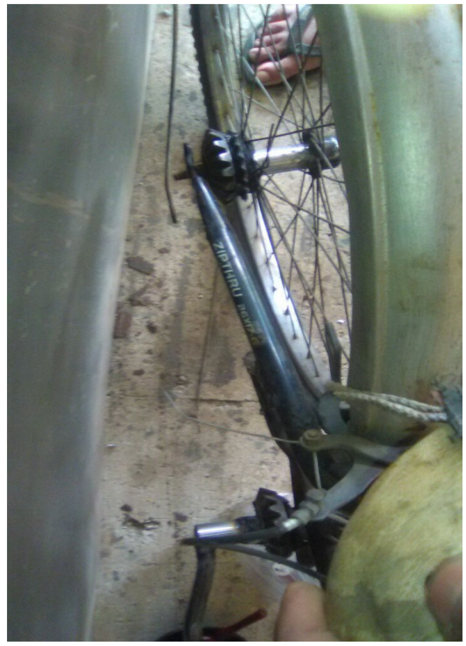

Fig 4 : Placing of Bevel Gear & Drive Shaft

5. Mechanical properties of Cast iron

Sr. No.	Mech. Properties	Symbols	Units	Cast Iron
1.	Youngs Modulus	E	GPa	101.0
2.	Shear Modulus	G	GPa	35.75
3.	Poisson Ratio	V	----	0.20
4.	Density	ρ	Kg/m^3	7202
5.	Yield Strength	S_y	MPa	131
6.	Shear Strength	S_s	MPa	168

6. DESIGN ASSUMPTIONS :

- The pole pivots at a steady speed about its longitudinal hub.
- The pole has a uniform, round cross segment.
- The pole is flawlessly adjusted, i.e., at each crosssegment, the mass focus corresponds with the Geometric focus.
- All damping and nonlinear impacts are rejected.
- The pressure strain relationship for compositematerial is straight and flexible; consequently, Hooke"s law is Applicable for composite materials.

- Acoustical liquid communications are dismissed, i.e., the pole is thought to act in a vacuum.
- Since lamina is thin and no out-of-plane burdens are connected, it is considered as under the plane Stress.

7. CALCULATIONS :

A. Design calculation of Gear

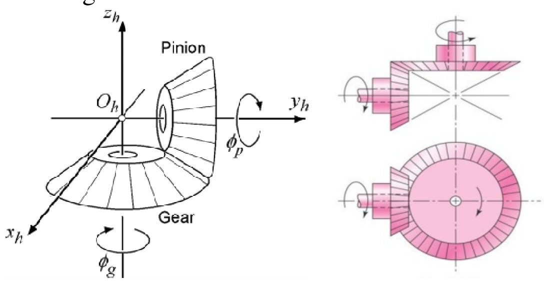

Fig 5 : Design Of Gear

Speed of apparatus (Ng) = 100rpm
Speed proportion (i) = 4
Teeth of pinion (Ze) = 8
Mass of rider (m) = 85 kg
Internal measurement of pinion = 33mm
External distance across of pinion = 41mm
Internal measurement of crown = 92mm
External distance across of crown = 132mm
Length of pedal leaver = 190mm

1. Most extreme torque connected on bike

Torque = weight of rider * length of pedal leaver

T = m*g*L
T = 85*9.81*0.190
T = 158.4315 N-m

2) Rated control (P)

P = 2*πN*T/60
P = 2*100**πN *158.4315/60
P = 1.6590 KW

3) Design control (Pd)

Pd = KL * P
Pd = 1.25*1.6590
Pd = 2.073875 KW

4) Select reasonable teeth of crown

Zp = 8 Nc = 100
i =Zc/Zpi = Np/Nc
4 =Zc/8 4 = Np/100
Zc = 32 Np = 400 rpm

5) Pitch point (γp)

For pinion
Tan (γp) = Zp/Zc
Tan(γp) = 8/32
(γp)= 14.036

Essentially for crown
Tan (γc) = 32/8
(γc) = 75.9637

6) Module (m)

Breadth = module * teeth
41 = m*8
m = 5.125mm
Ordinary module (mn)
mn = 5 mm
mn = m*cos(β)
5 = 5.125 * cos(β)
(β) = 12.68

7) Cone separate (Lc)

L = 0.5* √(Dp^2 + Dc^2)
L = .5* √(132*132 + 41*41)
L = 69.11mm

8) Pitch circle breadth (Pc)

Pc = πm
Pc = π*5.125
Pc = 16.1mm

9) Normal pitch breadth (Pn)

Pn = π*mn
Pn = π*5
Pn = 15.70mm

10) Virtual number of teeth (Ze)

For crown
(Ze)c = Zc/(cosβ)^3
(Ze)c = 33/(cos12.68)^3
(Ze)c = 35.537
(Ze)c = 36

For pinion
(Ze)p = Zp/(cos β)^3
(Ze)p = 8/(cos12.68)^3
(Ze)p = 9

11) Tangential power (Ft)

Ft = 1000*Pd *Cv/V

Speed (V)
V = π D*N/60
V = π *400*0.041/60
V = 0.857 m/s

For medium stun of administration factor
Cs = 1.50
Ft = 1000*2.0986*1.5/0.8587

Ft = 3.6658KN

12) Beam quality

Lewis condition

Ft = σd *Cv*b*Y*mn/Cw

for speed less then 5m/s

Cv = 4.5/(4.5+V)

Cv = 4.5/(4.5 +.8587)

Cv = 0.8421

Y = y*π

Where y = lewis frame fector

For ∅= 20 full profundity

y = 0.154 – 0.912/Ze

y = 0.0526

Y = y*π

Y = 0.1654

Wear and grease factor

Cw = 1.25

Ft = σd *Cv*b*Y*mn/Cw

3.6658 = 78.5*1000*0.8441*5*0.1654*b/1.25

b = 83.8mm

checkconfront width of helical apparatus

b = 12.5mn to 20mn

13) Dynamic load computation

Fd = Ft + Fi

Fd = Ft + K3*V (C*b + Ft)/K3*V □(C*b + Ft)

Where,

Fd C = dynamic load factor

C = e/K1 (1/E1 + 1/E2)

K1 = 9 For 20deg. Full profundity

K3 = 20.67

For C35 E = 490 to 580 N/mm2

For m = 5, e = 0.0555

In this way,

C = 0.0555/[9*(2/540)]

C = 1.6495

So unique load

Fd =3665.8+20.67*.8587*(1.6495*0.083+3665 .8)/[20.67*0.8587+√(()1.6495*0.083+3665. 8)]

Fd = 4496.847 N

14) Wear stack estimation

Fw = d1bQK ≥Fd

Where,

K = stack pressure factor

K = σes

2 sinσ (1/E1 + 1/E2)/1.4

σes = surface continuance point of confinement of apparatus pitch

σes = 2.75(BHN) – 70

= 2.75*300 - 70

σes = 755 N/mm2

So K = 7552 *sin20(2/540)/1.4

K = 515.76

Q = 2Z2/(Z1+ Z2)

Q = 2*33/(33+8)

Q = 1.609

In this way,

Fw = d1bQK ≥ Fd

Fw = 41*83*1.609*515.76

Fw = 2.808*106 N

Here ,

Fw ≥ Fd

B. Design Plan of shaft

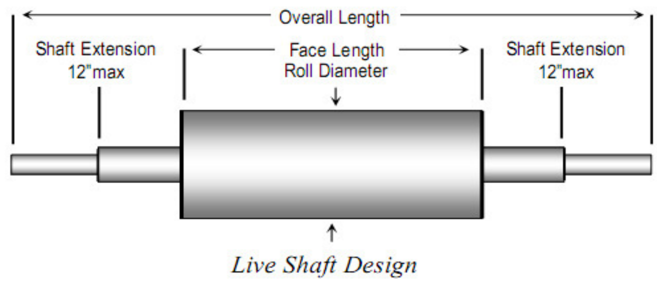

Fig 6 : Design of Shaft

Power (P) = 2.34 kW

P = 2.34 x 10^3 W

Speed (N) = 400 rpm

Mass (m) = 85 kg

Weight of first pinion (WA) = 9 N

Weight of second pinion (WB) = 5 N

RB = 24 mm

Elastic pressure (Syt) = 296 mPa

Shear pressure (**τ**) = 0.35 x Syt

τ= 103.2 mPa

Accept,

Weariness factor,

Km = 1, Kt = 0.5

Torque connected,

T = (P x 60)/(2π x N)

= (2.0738 x 103 x 60)/(2π x 400)

T = 49.50 N-m

T = 49.50 x 10^3 N-mm

Add up to vertical load acting descending on the pole at A,

Ft A = (T/RA)

= (49.50 x10^3)/(36.85)

= 1343.28N

Since the heaviness of rigging An at vertically descending along these lines the aggregate vertically acting upward of a similar shaft at A

= Ft A – WA

= 1343.28 – 9

= 1334.28 N

Expecting that the torque on the rigging B in same as that of the pole in this manner the digressive power acting vertically upward on the
same rigging B,

$Ft\ B = (T/RB)$

$= (49.50 \times 10^3)/(24)$

$Ft\ B = 2062.5$ N

Since the heaviness of rigging B at vertically descending thusly the aggregate vertically acting upward of a similar shaft at B

= Ft B - WB

= 2062.5 – 5

= 2057.5 N

Taking minute about D we get,

RC x 300 = 1334.28 x 360 + 2057.5 x 50 - 14x 155

RC = 1936.819 N

For the harmony of the pole,

1334.28 + RD = RC + 2062.5 + 14

RD + 1334.28 = 1936.819 + 2062.5+ 14

RD = 2679.04 N

SFD figuring,

AL = 0 N

AR = 1334.28 N

CL = 1334.28 N

CR = 1334.28 – 1936.819

= - 602.539 N

EL = - 602.539 N

ER = - 602.539 – 14

= - 616.539 N

DL = - 616.539 N

DR = - 616.539 + 2679.04

= 2062.5 N

BL = 2062.5 N

BR = 2062.5 - 2062.5

= 0 N

We realize that bowing minute at An and B = 0

BM at C

$Mc = 1936.819 \times 60$

$= 116.209 \times 10^3$ N-mm

BM at E

$ME = 1334.28 \times 205 – 1936.819 \times 145$

$= - 7.311 \times 10^3$ N-mm

BM at D

$MD = - 2062.5 \times 50$

$= - 103.125 \times 10^3$ N-mm

We see that the bowing minute is most extreme at C

Most extreme Bending Moment

$M = MC = 116.209 \times 10^3$ N-mm

We realize that the equal curving minute

Te = √[(Km *M)2 + (Kt * T)2]

Te = √[(1 *116.209 x 103)2 + (0.5 * 49.50 x 103)2]

Te = 118.815 x 103 N-mm

Additionally, we realize that the identical curving minute (Te)

Te = (π/16) x τ x d3

118.815 x 103 = (π/16) x 103.2 x d3

d = 18.03 mm

In this way,

Distance across of shaft = 18.03 mm

Presently think about mass (m) following up on shaft = mass of shaft (1.3 kg) + mass of two bearing (0.8+0.6 kg) = 2.7 kg

1) Mass snapshot of dormancy (I) I =m x R2

I = 2.7 x (0.01053)2

I = 0.00299 kg-m2

2) Polar development of dormancy (J)

J = (π/32) x d4

J = (π/32) x 0.01803

J = 1.0378 x10-8 m4

8. ADVANTAGES

- Drive framework is less inclined to end up stuck.
- The utilization of a rigging framework makes a smoother and more reliable accelerating movement.
- Lower support.
- Efficiency is more when contrasted with ordinary bike plan.
- High strength.
- Low cost of possession when made in extensive scale.

9. APPLICATIONS

1) It is utilized for hustling reason.
2) Also utilized for Off-street riding.
3) For Cycling.
4) For open and bike rental reason.

10) TROUBLESHOOTING

Problem	Caused By	What to Do
As bicycle is accelerated from stop	torque is required	Apply more torque at starting

when gears are not shifting	rusting	Clean with fluids
Vibration at speed	High speed	Maintain low speed
Noise at low speed	Universal joint	Apply grease
Gears pitch circle is not coincide	Vibrations	Adjust the position of gears
Gear backlash	Noise, Overloading, Overheating	Follow design characteristics

11. CONCLUSION

Right off the bat the venture were not able be finished with the drive shaft because of different issues around periphery of the bike ,later on this was acknowledged to run effectively with two incline gears at both end of the drive shaft. The drive shaft with the target of minimization of weight of shaft which was subjected to the requirements, for example, torque transmission, torsion clasping limit, stretch, strain, and so forth The torque transmission limit of the bike drive shaft has been ascertained by ignoring and considering the impact of outward powers and it has been watched that diffusive power will diminish the torque transmission limit of the pole.

REFERENCES

[1] Rastogi, N. (2004) Design of composite drive shafts for automotive applications. Visteon Corporation, SAE technical paper series.
[2] 73332270 Design and Analysis of a Propeller Shaft of a Toyota Qualis by “Syed Hasan”.
[3] A.M.Ummuhaani and Dr.P.Sadagopan “Design, Fabrication and Stress Analysis of a Composite Propeller Shaft, 2011-28-0013.
[4] Anup A. Bijagare, P.G. Mehar and V.N. Mujbaile “Design Optimization & Analysis of Drive Shaft”, Vol. 2 (6), 2012, 210-215.
[5] Rangaswamy, T.; Vijayrangan, S. (2005). Optimal sizing and stacking sequence of composite drive shafts. Materials science, Vol. 11 No 2., India.
[6] Rastogi, N. (2004). Design of composite drive shafts for automotive applications. Visteon Corporation, SAE technical paper series.
[7] R. P. Kumar Rompicharla1, Dr. K. Rambabu2 Sep-Oct. 2012 Design and Optimization of Drive Shaft with Composite Materials International Journal of Modern Engineering Research (IJMER) www.ijmer.com Vol.2, Issue.5, pp-3422-3428 ISSN: 2249-6645.
[8] Design Data Book ,Volume 1, Page No. 178 R.K. Jain, Khurmi
[9] Machine Design Book ,Page 353, R.S. Khurmi, J.K.Gupta

WEBSITE REFERENCES

1. Dynamic bicycles.com
2. www.engineeringtoolbox.com
3. Makeitform.com