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. 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.

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.
4. SELECTION OF METHODOLOGY

4.1 Selection of bevel gear

4.2 Selection of Drive Shaft

4.3 Placing of bevel gear

5. Mechanical properties of Cast iron

<table>
<thead>
<tr>
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</thead>
<tbody>
<tr>
<td>1.</td>
<td>Young’s Modulus</td>
<td>E</td>
<td>GPa</td>
<td>101.0</td>
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<tr>
<td>2.</td>
<td>Shear Modulus</td>
<td>G</td>
<td>GPa</td>
<td>35.75</td>
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<tr>
<td>3.</td>
<td>Poisson Ratio</td>
<td>V</td>
<td></td>
<td>0.20</td>
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<tr>
<td>4.</td>
<td>Density</td>
<td>ρ</td>
<td>Kg/m³</td>
<td>7202</td>
</tr>
<tr>
<td>5.</td>
<td>Yield Strength</td>
<td>$S_y$</td>
<td>MPa</td>
<td>131</td>
</tr>
<tr>
<td>6.</td>
<td>Shear Strength</td>
<td>$S_s$</td>
<td>MPa</td>
<td>168</td>
</tr>
</tbody>
</table>

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 composites material 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

![Design Of Gear Image]

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
\[ T = m \times g \times L \]
\[ T = 85 \times 9.81 \times \times 0.190 \]
\[ T = 158.4315 \text{ N-m} \]

2) Rated control (P)
\[ P = 2 \times \pi \times N \times T / 60 \]
\[ P = 2 \times 10 \times 100 \times \pi \times 158.4315 / 60 \]
\[ P = 1.6590 \text{ KW} \]

3) Design control (Pd)
\[ Pd = K \times L \times P \]
\[ Pd = 1.25 \times 1.6590 \]
\[ Pd = 2.073875 \text{ KW} \]

4) Select reasonable teeth of crown
\[ Zp = 8 \]
\[ Nc = 100 \]
\[ i = Zc / Zpi = Np / Nc \]
\[ 4 \times Zc / 8 = 4 \times Np / 100 \]
\[ Zc = 32 \]
\[ Np = 400 \text{ rpm} \]

5) Pitch point (\( \gamma_p \))
For pinion
\[ \tan(\gamma_p) = Zp / Zc \]
\[ \tan(\gamma_p) = 8 / 32 \]
\[ (\gamma_p) = 14.036 \]

Essentially for crown
\[ \tan(\gamma_c) = 32 / 8 \]
\[ (\gamma_c) = 75.9637 \]

6) Module (m)
2. Breadth = module * teeth
\[ 41 = m \times 8 \]
\[ m = 5.125 \text{ mm} \]

Ordinary module (mn)
\[ mn = 5 \text{ mm} \]
\[ mn = m \times \cos(\beta) \]
\[ 5 = 5.125 \times \cos(\beta) \]
\[ (\beta) = 12.68 \]

7) Cone separate (Lc)
\[ L = 0.5 \times \sqrt{(Dp^2 + Dc^2)} \]
\[ L = 0.5 \times \sqrt{(132^2 + 41^2)} \]
\[ L = 69.11 \text{ mm} \]

8) Pitch circle breadth (Pc)
\[ Pc = \pi \times m \]
\[ Pc = \pi \times 5.125 \]
\[ Pc = 16.1 \text{ mm} \]

9) Normal pitch breadth (Pn)
\[ Pn = \pi \times mn \]
\[ Pn = \pi \times 5 \]
\[ Pn = 15.70 \text{ mm} \]

10) Virtual number of teeth (Zc)
For crown
\[ (Zc) = Zc / (\cos(\beta))^3 \]
\[ (Zc) = 33 / (\cos(12.68))^3 \]
\[ (Zc) = 35.537 \]
\[ (Zc) = 36 \]

For pinion
\[ (Zc) = Zp / (\cos(\beta))^3 \]
\[ (Zc) = 8 / (\cos(12.68))^3 \]
\[ (Zc) = 9 \]

11) Tangential power (Ft)
\[ Ft = 1000 \times Pd \times Cv / V \]

Speed (V)
\[ V = \pi \times D \times N / 60 \]
\[ V = \pi \times 400 \times 0.041 / 60 \]
\[ V = 0.857 \text{ m/s} \]

For medium stun of administration factor
\[ Cs = 1.50 \]
\[ Ft = 1000 \times 2.0986 \times 1.5 / 0.8587 \]
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 + 0.8587)
Cv = 0.8421
Y = y*π
Where y = lewis frame factor
For Ø= 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 = 12.5mm to 20mm

13) Dynamic load computation
Fd = Ft + Fi
Fd = Ft + K3*V (C*b + Ft)/K3*V \(\sqrt{C*b + Ft}\)
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+\(\sqrt{(1.6495*0.083+3665.8)}\)]
Fd = 4496.847 N

14) Wear stack estimation
Fw = d1bQK \(\geq\) Fd
Where,
K = stack pressure factor
K = σes
2 sin(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 \(\geq\) Fd
Fw = 41*83*1.609*515.76
Fw = 2.808*106 N

Here ,
Fw \(\geq\) Fd

B. Design Plan of shaft

![Fig 6 : Design of Shaft](image)

Power (P) = 2.34 kW
P = 2.34 x 10³ 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)/(2\(\pi\) x N)
= (2.0738 x 103 x 60)/(2\(\pi\) x 400)
T = 49.50 N-m
= 49.50 x 10³ N-mm

Add up to vertical load acting descending on the pole at A,
Ft A = (T/RA)
= (49.50 x 10³)/(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,

\[ FT_B = \frac{T}{RB} \]

\[ FT_B = 2062.5 \text{ N} \]

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

\[ = FT_B - WB \]

\[ = 2062.5 - 5 \]

\[ = 2057.5 \text{ N} \]

Taking minute about D we get,

\[ RC \times 300 = 1334.28 \times 360 + 2057.5 \times 50 - 14 \times 155 \]

\[ RC = 1936.819 \text{ N} \]

For the harmony of the pole,

\[ 1334.28 + RD = RC + 2062.5 + 14 \]

\[ RD = 2679.04 \text{ N} \]

SFD figuring,

\[ AL = 0 \text{ N} \]

\[ AR = 1334.28 \text{ N} \]

\[ CL = 1334.28 \text{ N} \]

\[ CR = 1334.28 - 1936.819 \]

\[ = -602.539 \text{ N} \]

\[ EL = -602.539 \text{ N} \]

\[ ER = -602.539 - 14 \]

\[ = -616.539 \text{ N} \]

\[ DL = -616.539 \text{ N} \]

\[ DR = -616.539 + 2679.04 \]

\[ = 2062.5 \text{ N} \]

\[ BL = 2062.5 \text{ N} \]

\[ BR = 2062.5 - 2062.5 \]

\[ = 0 \text{ N} \]

We realize that bowing minute at An and B = 0

BM at C

\[ Mc = 1936.819 \times 60 \]

\[ = 116.209 \times 10^3 \text{ N-mm} \]

BM at E

\[ ME = 1334.28 \times 205 - 1936.819 \times 145 \]

\[ = 7.311 \times 10^3 \text{ N-mm} \]

BM at D

\[ MD = -2062.5 \times 50 \]

\[ = -103.125 \times 10^3 \text{ N-mm} \]

We see that the bowing minute is most extreme at C

Most extreme Bending Moment

\[ M = MC = 116.209 \times 10^3 \text{ N-mm} \]

We realize that the equal curving minute

\[ Te = \sqrt{[(K_m \times M) ^2 + (K_t \times T) ^2]} \]

\[ Te = \sqrt{[(1 \times 116.209 \times 103) ^2 + (0.5 \times 49.50 \times 103)]} \]

\[ Te = 118.815 \times 103 \text{ N-mm} \]

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

\[ Te = (\pi/16) \times \pi \times d^3 \]

\[ 118.815 \times 103 = (\pi/16) \times 103.2 \times d^3 \]

\[ d = 18.03 \text{ mm} \]

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

\[ I = m \times R^2 \]

\[ I = 2.7 \times (0.01053)^2 \]

\[ I = 0.00299 \text{ kg-m}^2 \]

2) Polar

development of dormancy (J)

\[ J = (\pi/32) \times d^4 \]

\[ J = (\pi/32) \times 0.01803 \]

\[ J = 1.0378 \times 10^{-8} \text{ m}^4 \]

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

<table>
<thead>
<tr>
<th>Problem</th>
<th>Caused By</th>
<th>What to Do</th>
</tr>
</thead>
<tbody>
<tr>
<td>As bicycle is accelerated from stop</td>
<td>torque is required</td>
<td>Apply more torque at starting</td>
</tr>
</tbody>
</table>
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


### WEBSITE REFERENCES

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3. Makeitform.com