Methodology for Design of Tire Removing and Fitting on Rim of Wheel Machine

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Abstract— In the present context we use bevel gear with shaft arrangement for rotating the shaft on which the tire is mounted. This is done for replacement of high cost tire changer which are available in markets for large workshops. Nowadays of market situation, there is always need for better design of the equipment with the maximum reduction in cost. Added to these is the human comfort which enables a person to operate it with ease and least consumption of human effort. In accordance with this situation we design and fabricate low cost semi manual tire fitting and removing machine. Due to this the road side tire punctured remover shops or small tire retailer shops can easily remove or fit the tire on to the rim of wheel. The main objective of this research is to design and fabricate low cost manual 'tire-changer' equipment to develop self-employment opportunities and help eliminating high cost of machine so that any middle class businessmen can buy it and complete their work in less time.

Key words: Bevel Gear, Pedal, Vertical Shaft

I. Introduction

This machine is use for tire removing and fitting on Rim of wheel. The wheel rim is tightly fixed on the mounting wheel. After the wheel and tire assembly are removed from the automobile, the tire changer has all the components necessary to remove and replace the tire from the wheel. Objective is to remove or fit tire to the disk by using this machine. Tool which remove the tire from the disk required force, therefore to find out the force acting on gear and shaft should be calculated.

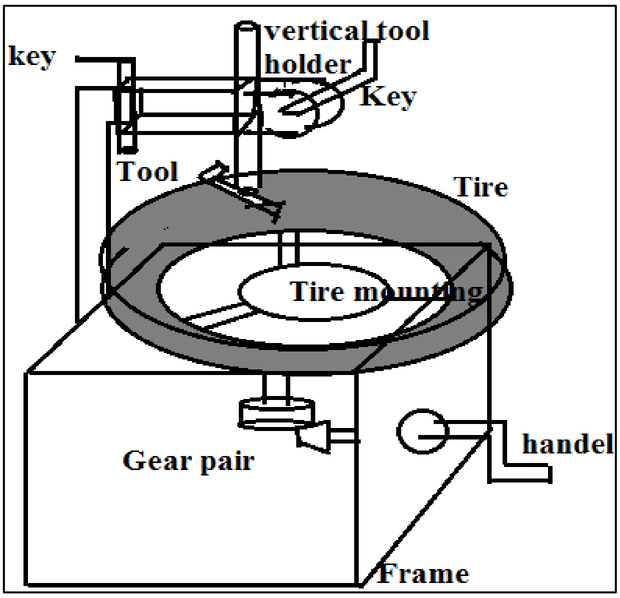

Fig. 1:

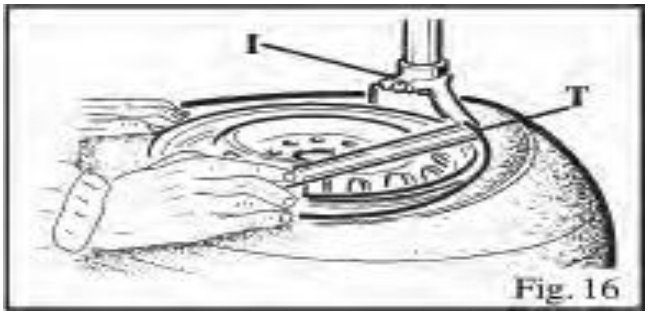

Fig. 2:

II. Design Assumption

1) The shaft rotates at constant speed about its longitudinal axis.
2) The shaft has a uniform, circular cross section.
3) The shaft is perfectly balanced.
4) All damping and nonlinear effect are excluded.

III. Calculations

A. Design Calculation for Gear

First we take gear ratio In this we used pair of bevel gear which one having t_p=7 and Second having tg=40.Therefore gear ration is 5.71, (Standard gear tooth taking by standard differential gear box)

Teeth of pinion (t_p) = 7

Teeth on gear (tg)=40

Considering

Mass of rider (m) = 60 kg

Length of the pedal lever (L) = 150 mm

Speed of gear (N_g) = 10 rpm

Step 1) Maximum torque applied on bicycle

T= (Mass of rider x g) x Length of the pedal lever

= 60 x 9.81 x 0.15

= 88.29 N-m

Fig. 3:

Step 2) Calculate Rated power

$$PR = 2\pi NT / 60$$

$= (2\pi \times 10 \times 88.29) / 60$

$= 92.457$ W

Step 3) Select suitable teeth of pinion

$T_p = 7$
$T_g = 40$
$V.R. = t_g / t_p$
$= 40/7$
$V.R. = 5.71$
Now
$V.R. = Np / Ng$
$5.71 = Np / 10$
$Np = 57.1$ rpm
Step 4) Pitch angle
For pinion, $\tan \gamma_p = t_p / t_g$
For gear, $\tan \gamma_g = t_g / t_p$
$\gamma_p = \tan{-1}\ (7/40)$
$\gamma_g = \tan{-1}(7 / 40)$
$\gamma_p = 9.926^0$
$\gamma_g = 80.07^0$
Step 5) Module
$m = D / t$
$D = m \times t$
$D_p = m \times t_p$
$D_g = m \times t_g$
$D_p = m \times 7$
$D_g = m \times 40$
Cone distance
$L = 0.5 \times \sqrt{(D_{p2}+D_g^2)}$
$L = 0.5 \times \sqrt{(49m^2+1600m^2)}$
$L = 20.3 \times$ module
Step 6) Formative number of teeth
$(t_f)_p = t_p / \cos\gamma_p$ $\quad (t_f)_g = t_g / \cos\gamma g$
$(t_f)p = 7 / \cos (9.926)$ $\quad (t_f)_g = 40 / \cos(80.07)$
$(t_f)_p = 7.5 \approx 8$ $(tf)_g = 231.9 \approx 232$
Step 7) Design power
$P_d = P_R \times K_L$
Where, $P_R = 92.457$
Load factor K_L
$K_L = 1.25$ from table xvι – 2 (midium load 3 hr per day)
Therefore
$P_d = 92.457 \times 1.25$
$P_d = 115.571$ W
Step 8) Tangential or tooth load
$F_t = P_d / V_p$
Where, $V_p = (\pi \times D_p \times N_p / 60000)$
$= (\pi \times 7m \times 57.1 / 60000)$
$= 0.0209 \times m$
Therefore
$F_t = (115.571) / (0.0209 \times m)$
$F_t = (5529) / (m)$ W ……………(1)
Step 9) Beam strength
$F_B = S_o \times C.V. \times Y \times b \times m\ (1– b/L)$
Where, S_o = beam strength
For pinion, $S_o = 56$ mPa --- (Gear material-Cast Iron, high grade)
For gear, $S_o = 56$ mPa
C.V. = Velocity factor
C.V. = 0.5 --- (Speed N<1000)
Y = Lewis Factor
For pinion,
$Y_p = 0.45- (2.87 / (t_f)_p)$
$Y_p = 0.45- (2.87 /8)$
$Y_p = 0.0912$
For gear,
$Y_g = 0.45- (2.87 / (t_f)_g)$
$Y_g = 0.45- (2.87 / 232)$
$Y_g = 0.437$
B = Face width of gear
Taking
$B = 7 \times m$
Now,
For pinion $(S_o \times Y)_p = (56 \times 00912) = 5.10$
For gear $(S_o \times Y)_g = (56 \times 0.437) = 24.5$
From above pinion section is weaker
Hence design for pinion
Therefore,
$F_B = 5.1 \times 0.5 \times 7m \times m \times (1 – 7m/20.388m)$
$F_B = 16.9xm2$ ……………………… (2)
by limiting condition from equation (1) and (2),
$FB = Ft$
$16.9 \times m2 = 5529 / (m)$
$m^3 = 327.15$
$m = 6.8$
Step 10) Value calculated
Tangential or tooth load
$F_t = (5529) / (m)$
$= (5529) / (6.8)$
$= 813$ N
Pitch velocity
$V_p = 0.0209 \times m$
$= 0.0209 \times 6.8$
$= 0.1421$ m/sec
Beam strength
$F_B = 16.14 \times m^2$
$= 16.14 \times 6.8^2$
$= 746.34$ N
Diameter of pinion
$D_p = 7 \times m$
$= 7 \times 6.8$
$= 47.6$ mm
$= 4.76$ cm
Diameter of gear
$D_g = 40 \times m$
$= 40 \times 6.8$
$= 272$ mm
$= 27.2$ cm

B. Design Calculation for Shaft

Design of gear blank
Alloy design gear blank of higher stage by considering D_p
Decide the type of construction
For solid construction $D_p \le 15m + 60$ mm
Where $15\ m + 60 = 162$
Therefore
$D_p \le 15m + 60$ mm
$47.6 \le 120$
Hence selecting solid construction

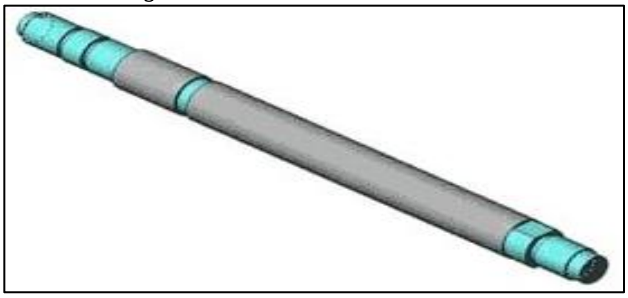

Fig. 4:

Calculations for Diameter of shaft (d_s)
$P_d = 2\Pi NT/60$
$115.57 = (2 \times \Pi \times 57.1)/ 60 \times T$
Hence $T = 19.32$ *Nm*
$T = 19.32 \times 10^3$ *Nmm*
Now
$T = \Pi/16 \times d_s^3 \times \tau_{max}$
Taking shaft material SAE 1030
From data book page no 39
$s_{yt} = 296$ *Mpa and* $s_{ut} = 257$ *Mpa*
$0.18\ S_{ut} = 0.18 \times 527 = 94.86$ *Mpa*
$0.35\ S_{yt} = 0.35 \times 296 = 103.6$ *Mpa*
Selecting lower value
τ *without keyway* $= 94.86$ *Mpa*
τ *with keyway* $= 0.75 \times \tau$
$= 0.75 \times 94.86$
$= 71.145$ *Mpa*
Therefore
$19.32 \times 10^3 = \Pi/16 \times 71.145 \times d_s^3$
$d_s^3 = 1383.58$
$d_s = 11.14$ *mm*
C design of bearing
Assuming self alignment bearing
Radial load $F_r = F_t = 813$ *N and*
Thrust load $F_a = F_B = 746.34$ *N*
Desired life = L10 life
= 170 million of revolution
Select from table T ×ııı e = 0.65
X = 1 and Y = 2.3
Life of bearing L
$L = (c/Fe)^n \times K_{ret}$
Equivalent load coming on bearing
$Fe = (xF_r + yF_a)K_sK_oK_pK_r$
Where
K_o *= constant rotational speed of races*
= 1
K_p *= Non preloaded bearing =1*
K_r *= outer race fixed inner race rotating = 1*
K_s *= moderate shock load = 2*
Now $F_e = (1 \times 813 + 2.3 \times 746.34) \times 2 \times 1 \times 1 \times 1$
$= 5059.1$ *N*
$L = (c/Fe)^n \times K_{ret}$
$170 = (C/5059.1)^3 \times 1$
$C = 28025.685$ *N (T ×ııı- 21)*
Select nearest value of C and select bore.
04xx series
Base no. 05
Bearing no. 0405
For bore no.= 05
Bore diameter = d = 25 mm
Outside diameter = D =80 mm
Width B = 21 mm

1) Advantages

1) Drive system is less likely to become jammed.
2) The use of a gear system creates a smoother and more consistent pedaling motion.
3) Lower maintenance.
4) High durability.
5) Low cost of ownership when manufactured in large scale.

C. Applications

– It is used for tire removing from rim of wheel purpose.
– Also used for tire fitting from rim of wheel.

IV. Result and Conclusion

The presented work was aimed to reduce the wastage of human power on removing the tire by traditional processes.

The presented work also deals with optimization i.e. converting linear motion into the rotary motion with aid of two bevel gears.

Instead of hydraulic means one piece drive shaft for tire mounting base have been optimally designed and manufactured for easily power transmission.

The result obtained from this work is a useful approximation to help in the earlier stage of the development, saving development time and helping in the decision making process to optimize a design.

Hence we are trying to make the transmission smooth and easy by applying the bevel gears and shaft attachment instead of traditional process or by fully automatic tire changer.

References

[1] US PATENT DOCUMENTS Kenneth S. Keyes 5,078,416 1/1992 ………..…280/260.
[2] J.C. Martin, W.W. Spirduso “Determinants of Maximal Cycling power: Crank length, Pedaling rate and Pedal speed”, Springer-Verlag, 84, 2001, 413-418.
[3] Rastogi, N. (2004) “Design of composite drive shafts for automotive applications”, Visteon Corporation, SAE technical paper series.
[4] R.S. Khurmi, J.K. Gupta “A Text Book of Machine Design”, S CHAND, 2005.
[5] B.D. Shiwalkar “Design Data for Machine Elements”,

