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Design and Manufacturing of Gear Driven Screwdriver

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Abstract - *This study investigated the certain important aspects of screwdriver use in occupational work situations, with an emphasis on comfort/discomfort in using screwdriver according to users. Descriptors of comfort/discomfort in mistreatment hand tools were collected from literature and interviews. Six comfort factors may well be distinguished (functionality, posture and muscles, irritation and pain of hand and fingers, irritation of hand surface, handle characteristics, aesthetics). Thus the designers can use to address the appropriate comfort descriptors in the hand tool design process; an attempt is made to illustrate the relevance of anthrometric data in design of handle of hand tools using ergonomics principle. The work is about making the use of gear mechanism in the screwdriver such that the disadvantages of conventional and electrical screwdriver could overcome. By using this screwdriver the dependency on electric and fluid power to drive a screwdriver will be eliminated.*

***Key Words*: Hand Tool, Ergonomics, Gear Driven, Manual Screwdriver, Gear Mechanism …**

1. INTRODUCTION

Hand tools have been in use for a very long time and have developed in an almost evolutionary manner. The first recorded use of the screw principle was an invention by Archimedes about 300 BC. By the first century BC, very large wood screws were in use, while metal screws began to appear during the fifteenth century. With the increasing use of wood, screws came the need for a screwdriver, which at first was a slot-bladed bit used with a drill. A screwdriver may well be a tool, manual or battery-powered, for tightening and unscrewing (inserting and removing) screws. A typical straightforward screwdriver includes a handle and a shaft, ending in a tip the user puts into the screw head before turning the handle. These typically have a hollow handle that contains varied sorts and sizes of tips, and a reversible ratchet action that enables multiple full turns while not location the tip or the user's hand. Hand torque exertions are required for many activities of work, daily living and recreation.[1]

And the power screwdrivers are driven by electrically, pneumatically or hydraulically. The power screwdriver requires electrical supply or pneumatic or hydraulic energy to drive the tool of the screwdriver. A recent innovation in screwdriver bit design is the ECXTM bit developed by Milwaukee Electric Tool, which features elements of both the straight blade and Philips head. This combination is intended to allow the bit to have increased retention in the fastener, which may have the added benefit of decreasing the push force required for proper bit retention. This can be significant, as when a driver bit does not stay in the fastener, the user must apply a forward “push” force in an attempt to increase bit retention. Reducing push force will minimize user fatigue and may increase productivity.[2]

To reduce the human efforts and to eliminate the power usage, the gear driven screwdriver will be beneficial. This project is about making the use of gear mechanism in a screwdriver such that the disadvantages of conventional and electrical screwdriver could overcome. This screwdriver contains gear mechanism using rack and pinion and bevel gears. By using this gear mechanism linear motion is converted into rotary motion so that the twisting moment will be eliminated which is the main disadvantage of the conventional screwdriver.

2. METHODOLOGY

- Aim of the study is to reduce efforts in operation of conventional screwdriver.
- For this purpose gear mechanism is to be used.
- To start with gear design we need to know the input power, input rpm and output rpm.
- We found that, for a screwdriver average torque is 3-6 Nm and speed is 500-1000 rpm. So to find out input power and rpm we will assume efficiency of gear train as 92%.
- After finalizing input parameters we can start with the calculations assuming suitable materials (less weight high strength materials such as EN24, Aluminium, etc)
- Based on the calculations standard dimensions for gears, bearings, shafts, etc can be selected.

- For flexibility, the output shaft is to be provided with internal splines to accommodate various tools with external splines.

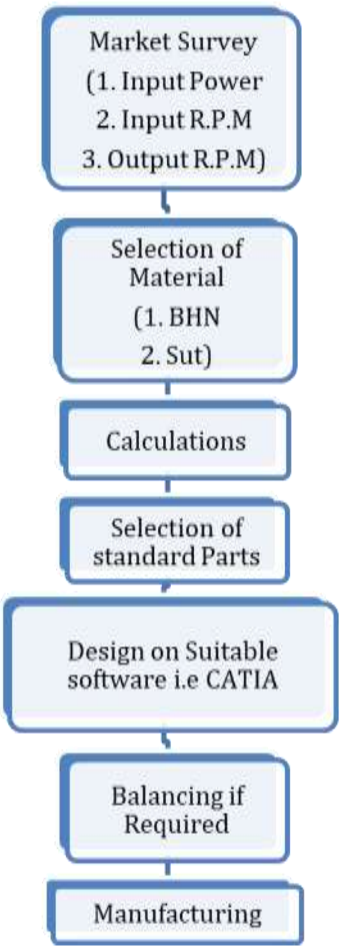

Fig -1: Process Flow Chart

3. CALCULATIONS

Abbreviations

Symbol	Description
α	Pressure Angle
Γ_p	Pitch Cone Angle
G	Gear Ratio
Z_p	Number of teeth on Pinion
Z_g	Number of teeth on Gear
M_t	Torque
N_p	Speed of Pinion
C_s	Factor of safety
S_{ut}	Ultimate yield strength
σ_b	Beam Strength
Y_p	Lewis Form Factor for Pinion
A_o	Cross Section Area
m	Module
D_p	Diameter of Pinion
D_g	Diameter of Gear
b	Face width
F_b	Bending Strength
Q	Ratio Factor for external Gear Pair
K	Load Stress Factor
F_w	Wear Strength
P_{eff}	Effective Load
V	Velocity
P	Power
P_t	Tangential Force
C_v	Velocity factor
H_a	Addendum
H_f	Dedendum
P_r	Radial Force
M	Moment
R	Reaction
D	Diameter
w	Width
H	Height

1. Design of Bevel Gear
1] Beam Strength,
As the gear and pinion are made up of same material then the pinion is weaker than gear in bending. Hence it is necessary to design pinion for bending.
$\sigma_b = S_{ut}/3$
$= 850/3$
$\boldsymbol{\sigma_b = 283.33\ N/mm^2}$**,**
$\tan\Gamma_p = Z_p/Z_g$
$= 18/36$
$\tan\Gamma_p = 0.5$,
$\Gamma_p = \tan^{-1}(0.5)$

***Γ_p* = 26.56°,**
$D_p = m*Z_p = 18m$,
$D_g = m*Z_g = 36m$,
$A_0 = [(D_p/2)^2+(D_g/2)^2]^{1/2}$,
$= [(18m/2)^2+(36m/2)^2]^{1/2}$,
A_0 = 20.1246*m,
b = A_0/3 or 10m … (Whichever is smaller),
= 20.1246m/3 or 10m,
= 6.7083m or 10m,
b = 6.7083m
$F_b = \sigma_b *b*m* \Gamma_p'*[1-(b/A_0)]$,
= 283.333*6.7082m*m*0.3079*[1-(6.7083m/20.1246m)],
F_b= 390.140999m^2 … (N)
2] Wear Strength,
$Z_p' = Z_p/ \cos \Gamma_p$
Z_p' = 18/ Cos(26.56)
Z_p' = 20.1287,
$Z_g' = Z_g / \cos \Gamma_p$
Z_g' = 36/ Cos(26.56)
Z_g' = 40.2424,
$Q' = 2*Z_g/[Z_g+Z_p]$
= 2*36/[18+36]
Q' = 1.333

$K = 0.16[BHN/100]^2$
$= 0.16[300/100]^2$
K = 1.44,

$F_w = 0.75*D_p*b*Q*K/ \cos \Gamma_p$,
= 0.75*18m*6.7082m*1.333*1.44/ Cos(26.56),
F_w = 194.3427m^2 … (N),
As, $F_w < F_b$ the gear pair is weaker in pitting hence it should be designed for safety against pitting failure.
3] Effective Load,
$V= 3.142*D_p*N_p/60000$
= 3.142*18m*750/60000 = 0.7068m,
$P_t = P/V$,
But, $P = 2*3.142*N*M_t/60000$,
= 2*3.142*750*6/60000,
= 0.471238 KW,
P_t = 0.471238/0.7068m,
P_t = 0.667/m,
$C_v = 5.6/ [5.6+(V)^(1/2)]$,
$P_{eff} = [C_s/C_v]*P_t$
$P_{eff} = 2/\{5.6/[5.6+(0.7068m)^{0.5}]\}*[0.667/m]$,

Now,
$F_w= C_s*P_{eff}$,
$194.3627m^2= 2*2/\{5.6/[5.6+(0.7068m)^{0.5}]\}*[0.667/m]$,
On Calculating,
m = 1.678 mm,
m = 2 mm.
Now,
Dimensions of Bevel Gear Pair,
m = 2 mm,
Z_p = 18,
Z_g = 36,
b = 6.708*m = 13.14 mm
$D_p = m*Z_p$ =2 *18 = 36mm,
$D_g = m*Z_g$ = 2*18 = 36 mm,
Addendum (h_a) = 1*m = 1 mm,
Dedendum (h_f) = 1.2*m = 1.2 mm,
***Γ_p* = 26.56 degree,**
A_0 = 20.1246*m = 40.24 mm

2. Design of Spur gear
Z_p=36
S_{ut}=800 N/mm^2
Efficiency = P_o/p_i
0.93 = 0.4712/Pi
P_i = 0.5067 KW
$P= 2*3.142*N*M_t/60000$
M_t= 12.90 N-mm
1) Beam Strength
$\sigma_b = S_{ut}/3$
= 800/3
σ_b = <u>266.667 N</u>
Y_p = 0.484-(2.87/Z_p)
Y_p **<u>= 0.4042 mm</u>**
b = 10m
$F_b = 6b*b*m*Y_p$
F_b = 266.667*10m*m*0.424
F_b = 1077.8680m^2 … (N)
$V = 3.142*d_p*N_p/60000$
V = 3.142*36m*375/60000
V = 0.768m … (m/sec)
$C_v = 3/(3+V)$
C_v = 3/(3+0.768m)
1) Effective Load
$P_{eff} = C_s/C_v*P_t$
$P_t = P/V$
P_t = 0.50667/0.7068m
P_t = 0.7168/m
P_{eff} = 2/(3/[3+0.7068m])*(0.7168/m)
Now,
$F_b = N_f*P_{eff}$
$1077.8680m^2$= 2*{1.4336/(3m/3+0.7068m)}
On Solving
m = 0.154 mm
m~ 1 mm
Now,
$D_p = m*Z_p$
D_p = 1*36
D_p = 36 mm
b = 10*m = 10*1
b = 10 mm

I. Design of Shaft
Tangential force on spur gear = 0.7168/m = 0.4778 KN
$P_r = P_t*\tan(\alpha)$
P_r = 0.4778*Tan(20)
P_r = 0.7139 KN
D_s = 36 mm
D_b = 36 mm
T_s = 0.75*0.18*Sut
T_s = 108 N/mm^2

T = 12.90 N-mm

Radial force on Bevel = F_t*Tan(⌀)*Cos(Γ)
= 0.2170 KN

Tangential force on Bevel = 0.66672 KN

Bending Moment on Shaft

I. Moment About point A

-R_{dv}*60+0.66672*40-0.1739*20=0

R_{dv} = 0.3868KN = 386.8 N

Ii) Moment at point D

-R_{av}*60-0.1739*40+0.66672*20=0

R_{av} = 338.1 N

Vertical Bending moment at point B and C

M_{bv} = -338.1*20 = -6762 N–mm

M_{cv} = -386.8*20 = -7736 N-mm

Taking moment about point A

R_{dh}*60-0.2170*40+0.4778*20=0

R_{dh} = 0.00146 KN = 14.6 N

Moment at D

R_{ah}*60-0.4778*40+0.2170*20=0

R_{ah} = 0.2462 KN = 246.2 N

M_{bh} = 0.4778*20 = 955.6 N-mm

M_{ch} = 0.2170*20 = 434 N-mm

$M_b = [M_{bv}^2+M_{bh}^2]^{0.5}$

$= [-6762^2+955.6^2]^{0.5}$

M_b = 6829.1884 N-mm

$M_c = [M_{cv}^2+M_{ch}^2]^{1/}$

$= [7736_2+434^2]$^(1/2)

M_c = 7748.1642 N-mm

M = 7748.1642 N-mm ...(Maximum of Above)

$T_e = [(K_b*M)^2+(K_t*T)^2]^{0.5}$ ... (Load gradually applied K_b = 1.5, K_t = 1)

$T_e = [(1.5*7748.1642)^2+(12.80)_2]^{1/2}$

T_e = 11622.25346 N-mm

$T_{max} = 16*T_e/3.142*d^3$

$108 = (16* 11622.25346)/3.142*d^3$

D = 8.1836 mm

D = 8 mm

IV) Dimensions of Key

Width of Key = d/4 = 8/4 = 2 mm

Height of Key = 2/3*(W) = 2/3 *(2) = 1.333 mm

II. Dimensions of Bearing

$F_a = [F_{av}^2+F_{an}^2]^{0.5}$

$F_a = [338.1^2+246.2^2]^2$

F_a = 418.24 N

$F_b = [F_{bv}^2+F_{bh}^2]^{0.5}$

$F_b = [386.8^2+14.6^2]^{0.5}$

F_b = 387.07 N

As F_b is greater than F_a

$F_r = F_a$ = 418.24 N

F_a = 0 N

$P_e = [X*V*F_r+Y*F_a]C_s$

P_e = [0.6*1.0602*418.24+0]*2

P_e = 532.10 N

$L_{10} = L_{h10}*60*N/10^6$

$L_{10} = 5000*60*375/10^6$

L_{10} = 112.5 MR

$L_{10} = (C/P_e)^3$

$112.5 = (C/532.10)^3$

C = 2568.68 N

From Dimensions and basic Capacities of ball bearing

Bearing Number = 698

Bore Diameter = 8 mm

Outside diameter = 19 mm

Width = 6 mm

Bearing Number 698 is selected.

4. EXPERIMENTAL DESIGN

The dimensions of parts by using experimental procedure are as follows:

Part Name	Specification	Dimensions
1.Bevel gear	m	2 mm
	Zp	18
	Zg	36
	B	13.14 mm
	Dp	36 mm
	Dg	36 mm
2.Spur gear	m	1 mm
	Dp	36 mm
	B	10 mm
3.Shaft	L	50 mm
	D	8 mm
4.Rack	L	120 mm
5. Key	W	2 mm
	H	1.33 mm
6.Bearing	Db	8 mm
	Do	19 mm
	W	6 mm

Table 1- Dimensions of parts

5. ANALYSIS AND INTERPRETATION

The calculations are carried out by standard procedure and standard dimensions of parts are selected. Accordingly the designs on CATIA software are made.

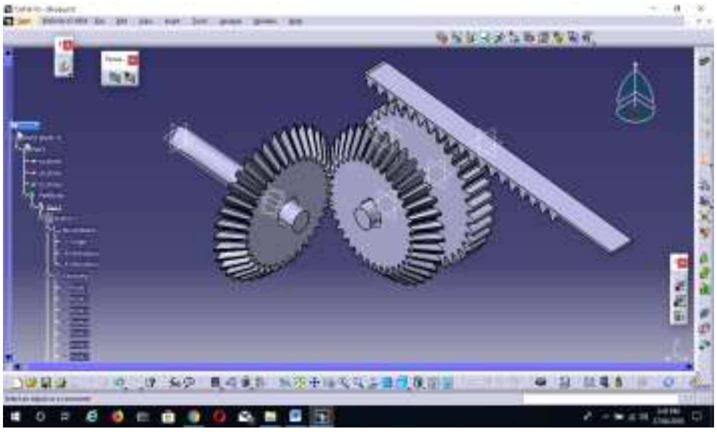

Fig -2: Assembly Design of gearbox

6. CONCLUSION

Development of new techniques in "on to go" to meet social requirements. Various types of such products are competing with one another in each specific field according to their output characteristics, weight and cost. The results will be reflecting on such design for more applicability and potentiality. However for inherent difference in construction and operating mechanism, approaches unique to this screwdriver are in process of materialization.

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