International Journal of Scientific Research in Engineering and Management (IJSREM)
Volume: 04 Issue: 07 | July -2020 ISSN: 2582-3930

Design and Modelling of Cutter Drive for Modified Leather RollGrooving Machine

Dhiraj Nagdeve[1], Nikhil Shrikhande[2], Dr. A.V. Vanalkar[3]

[1]*PG-Student, Department of Mechanical Engineering, KDKCE, Nagpur, Maharashtra, India*
[2]*Assistant Professor,Department of Mechanical Engineering, KDKCE, Nagpur, Maharashtra, India*
[3]*Professor,Department of Mechanical Engineering, KDKCE, Nagpur, Maharashtra, India*

---***---

Abstract -This paper presents the design procedure of cutter drive required for leather roll grooving machine. The cutter drive is needed to cut the grooves on the cylindrical surface of leather roll which is used in roller ginning machine i.e. Single Roller Ginning Machine and Double Roller Ginning Machine. The cutter drive contains an electric motor having a circular saw blade and motor mounting stand. The whole cutter drive assembly moves throughout the length of roll while cutting grooves. The 3D modelling is created using the Autodesk Inventor software.

***Key Words*:**Roll grooving machine, Roller Ginning, Leather Roll, Cutter drive.

1. INTRODUCTION

The leather roll grooving machine is used for cutting the grooves on the cylindrical surface of the leather roll which is used in single or double roller ginning machine for separating the cotton fiber from the seed by the mechanical movement of roller and moving knife. The separated fiber then pulled out by the rooves on the leather roll. Hence the groove parameters affect the performance of leather roll and hence of roller ginning machine. In old days, the groove cutting was performed manually by the hand grinder. The old procedure was too much time consuming, and improper grooves were cut due to due to which staple length of cotton fiber were affected. It was also harmful for the human health who were cutting the grooves. To overcome these problems, the leather roll grooving machine is redesigned and modified by considering all the necessities and easiness to use.

The newly designed grooving machine contains the mechanical components like V-belts drives for power and motion transmission, roll holder shaft, power screw for motion conversion and cutter drive to cut the grooves on the cylindrical surface of leather roll. The cutter drive plays an important roll in groove cutting machine. The cutter drive contains the following components

i) An electric motor
ii) Circular saw blade of 100mm to 120 mm diameter 1.2mm thick.
iii) Adjustable Mounting stand for height adjustment.
iv) Guide pillars for support.

2. CONSTRUCTION & WORKING

An electric motor is mounted on the top plate of the adjustable stand assembly as shown in fig. The top and bottom plates are attached by the eight moving strips which are connected to each other and to top and bottom by means of screw having clearance to move freely. Both side of strips are hold by two horizontal plates by screw and these to plates are connected by means of a square threaded lead screw to which a handle is attached for manual height adjustment. Two guide pillars are used for supporting purpose as shown. The bottom plate has two angular slots for angular adjustment of the drive for setting required angle helical grooves on roll.

Working:

For cutting the grooves first the leather roll is coupled with the roll holder shaft and indexing is done by inserting a pin in the hole for the number of grooves to be cut on the leather roll. The height of cutter drive is adjusted by rotating the handle manually just before the end of leather roll and the cutter motor is started on which a circular saw blade is mounted which rotates at 1500 rpm. The cutter drive is set at a required angle (generally120 degree) which produces helical grooves on leather roll. The cutter drive moves along the leather roll from one end to another end. As it reaches the other end of the roll the limit switch which is fixed at a defined distance on the machine structure is actuated by the cutter drive as it touches the limit switch which stop the both roll and the cutter drive for some seconds. The both roll drive and cutter drive is again started automatically by the timer circuit set for and the same thing happens at the both end until the complete roll is grooved. The linear motion and speed of the cutter drive is controlled by the lead screw which is powered by an electric motor. In this way the cutter performed the grooving process. The below diagram shows the 3D model of roll grooving machine and a cutter drive.

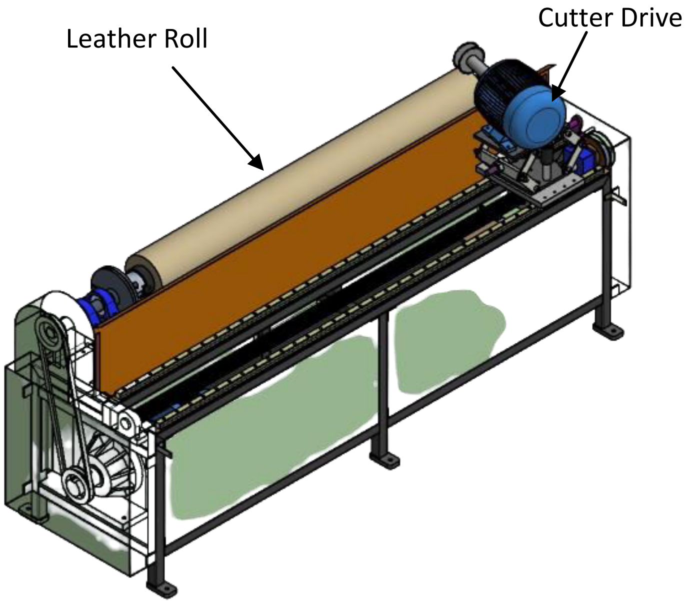

Fig. 3D Model of Roll Grooving Machine

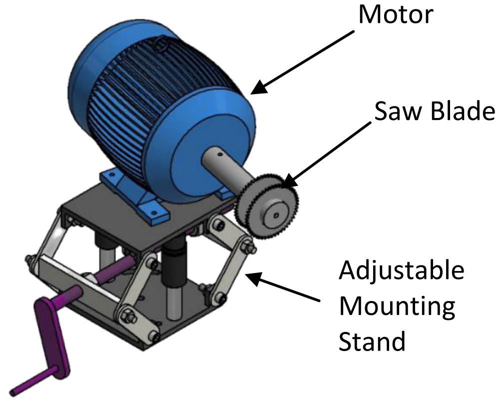

Fig. 3D Model of Cutter Drive

3. DESIGN PROCEDURE

Selection of Cutter Motor

Available Motor from the market is selected of the following specifications.
Type: 3phase Induction Motor 440VAC supply.
Rated Speed: 1500 RPM
Rated Power: 2HP
Motor Shaft Size: 24 mm Diameter.

Design considerations

Load to be lifted, W = 980N
Length of connecting link, l = 100mm
Permissible stress in tension, σ_t= 100 Mpa
Permissible shear stress, τ = 50 Mpa
Bearing pressure, Pb = 15 Mpa
Coefficient of friction between threads, $\mu = \tan\Phi = 0.2$
Assuming pitch, P = 5 mm.
The cutter drive mechanism is like toggle jack therefore we adopt the same design procedure for the cutter drive.

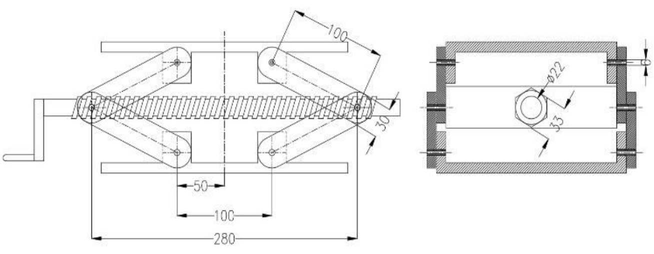

Fig. Cutter drive stand schematic drawing.

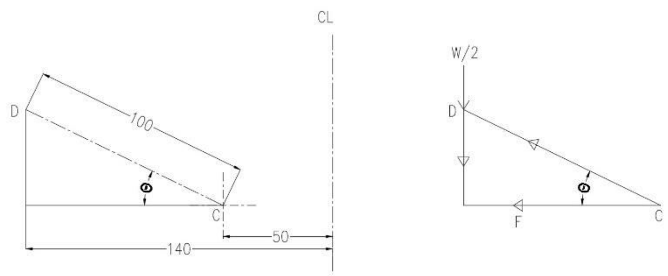

Fig. Link Geometry.

I. Design of square threaded screw:

From above geometry we get

$\cos\theta = x / l \quad x = 140-50 = 90$ & $l = 100$ (link length)

$\therefore \theta = 25.84^o$

$F = W / 2\tan\theta \quad W = 980N$

$\therefore F = 1013.6N$

Also, $W1 = 2F = 2027.2N$

Let, dc = core diameter

Load on screw, $w = (\pi/4) \times dc^2 \times \sigma_t$

$\therefore$ dc = 5.07 mm

As this value is very less and screw also subjected to torsional shear stress, therefore we take dc = 17 mm,

Also, do = dc + p

$\therefore$do = 22 mm

and. d = do – (p/2)

d = 19.5 mm i.e. 20 mm

Now, checking for principle stresses

We know, $\tan\alpha = p / \pi d = 0.079$ where, α = helix angle

and $P = W1.\tan(\alpha + \Phi)$, where, P = Effort req. to rotate screw

$= W1(\tan\alpha + \tan\Phi) / 1 - (\tan\alpha.\tan\Phi)$

$\therefore$ P = 574.66 N

Also, $T = P \times d/2$ where, T = Torque req. to rotate the screw

T = 5746.68 N-mm

$\tau = 16T / \pi (dc)^3$,where, τ = Shear stress inscrew due to torque

$\therefore \tau = 5.96$ N/mm^2

$\sigma t = W1/ (\pi/4)\, dc^2$ where, σt = Direct tensile stresses in the

$\therefore \sigma t = 8.9$ N/mm^2

$\sigma t\,(max) = (\sigma t / 2) + (1/2) \sqrt{(\sigma t^2 + 4\tau^2)} = 11.88$ N/mm^2,

where, σt (max) = max. principle stresses

$\tau(max) = (1/2) \sqrt{(\sigma t^2 + 4\tau^2)} = 7.45$ N/mm^2

Since the maximum stresses are within safe limits, thereforethe design of square threadedscrew is satisfactory.

II. Design of nut

Assuming that the load W_1 is distributed uniformly over the cross-sectional area of the nut,therefore bearing pressure between the threads (Pb),

$15 = W1 / (\pi/4)\,(do^2 - dc^2)\, n$

$\therefore$n = 1.133 i.e. 2, where n = number of threads in nut

We take ***n* = *4,*** min number of threads req to have good stability between screw and nut and to prevent locking of screw.

$t = n \times p = 20$ mm, where t = thickness of nut.

$b = 1.5\times do = 33$ mm, where b = width of nut.

To control the movement of the nuts beyond 280 mm (the maximum distance between the Centre lines of nuts), rings of 8 mm thickness are fitted on the screw with the help of setscrews.
∴ Length of screwed portion of the screw
$= 280 + t + 2 \times$ Thickness of rings
$= 316$ mm

Assumingthat a force of 100 N is applied by each person at each end of the rod,

∴lengthof the spanner required= $T / (2 \times 100) = 28.73$ mm

We shall take the length of the spanner as 200 mm in order to facilitate the operation and evena single person can operate it.

III. Design of pin in nut

Since the pins are in double shear, therefore load on the pins(F), $1013.6 = 2 \times (\pi/4) \times (d_1)2$
$\therefore d_1 = 3.5$ i.e. 4 mm
The diameter of pin $= 1.5\, d_1 = 6$ mm and thickness, d1= 3 mm

The pins in the nutsare kept in position by separate rings 3 mm thick and 1.5 mm split pins passing through the rings and pins.

IV. Design of link

Due to the load, the links may buckle in two planes at right angles to each other. For buckling inthe vertical plane,the links are considered as hinged at both ends and forbuckling in a plane perpendicular to the vertical plane, it is considered as fixed at both ends.
We knowthat load on the link $= F / 2 =$ ***506.8 N***
Assuming F.O.S = 5, the link must be designed for buckling load,
W_{cr} = load in link × 5 = ***2534 N***
$A_l = t_l \times b_l$ where, t_l=thickness of link& A_l = area of link

$= 3(t_l)^2$ $b1 = 3t_l$ width of the link

$I = (1/12) \times t_l \times (b_l)^3$ where, I = M.I. of c/s of the link

$K = \sqrt{(I/A_l)} = 0.866 t_l$ where, K =Radius of gyration

Since for buckling of the link in the vertical plane, the ends are considered as hinged, thereforeequivalent length of the link,$L = l = 100$ mm
According to Rankine's formula, buckling load

$W_{cr} = (\sigma_c \times A_l) / 1 + a(L/K)^2$

where, a= (1/7500) Rankine constant

By putting the values and solving we get the equation in following form,

$(t_1)^4 - 8.4(t_1)^2 - 14.95 = 0$

$$\therefore t1 = \frac{8.4 \pm \sqrt{(8.4)2 + 4(1 \times 14.95)}}{2}$$

$\therefore t_1 = 9.905$ i.e. 10mm (Taking +ve value)

and, $b_1 = 3 \times t_1 = 30$ mm

Now let us consider the buckling of the link in a plane perpendicular to the vertical plane.
$I = (1/12) \times b_1 \times (t_1)^3$ where, I = M.I. of c/s of the link

$\therefore I = 0.25(t_1)^4$

$A_l = t_l \times b_l$ where, t_l=thickness of link & A_l = area of link

$= 3(t_l)^2$ $b1 = 3t_l$ width of the link

$K = \sqrt{(I/A_l)} = 0.29 t_l$ where, K =Radius of gyration

$Le = l / 2 = 100 / 2 = 50$mm, Le = Equivalent length of link

Again, according to Rankine's formula, buckling load,

$W_{cr} = (\sigma_c \times A_l) / 1 + a(Le/K)2$

where, a = (1/7500) Rankine constant

∴ By putting the above values in the equation

We get, $\boldsymbol{W_{cr} = 28856.42\ N}$

Since this buckling load is more than the calculated value (i.e. **2534 N**), therefore the link is safefor buckling in a plane perpendicular to the vertical plane.
∴ We may take **$t_1 = 10$mm**; and **$b_1 = 30$ mm**

4. CONCLUSION

From the above design procedure for the cutter drive stand for roll grooving machine, we have found out that the required dimensions for the cutter drive parts is suitable up to the load of 980 N and the obtained dimensions are safe under the considered stresses. Further modification can change the dimension following the same design procedure.

5. REFERENCES

1. B.D. Shiwalkar,'' Design data for machine elements'', 2010 Denett& Company.
2. S. S Rattan, 'Theory of machine'', edition 2012, S.Chand Publication.
3. Bhandari V.B., ''Design of machine elements''.3rd edition,2010 the Tata McGraw Hill Education Private Limited
4. R.S. Khurmi& J.K. Gupta, "Machine Design".
5. Shigley, J.E., and Mischke, C.R., Mechanical Engineering Design, 5th ed., McGraw-Hill, New York, 1989.