

Design and Development of Leather Groove Cutter Machine

Mr. Kushal J. Bahadure
Student
Department of Mechanical Engineering
K.D.K.C.E., Nagpur, India

Dr. C. C. Handa
Professor & Head
Department of Mechanical Engineering
K.D.K.C.E., Nagpur, India

Abstract

This paper presents the Design and Development for leather groove cutter machine for double roller ginning machine to designing new machine for ginning industry. The main objectives of the designing to reduced grooving time and increase productivity. The paper aims to complete fabrication of leather groove cutting machine for double roller ginning machine which consist the leather roller and grinding cutter .In first stage of designing area of maximum stress was identified related to the accuracy.

Keywords: Ginning Machine, Cotton, Leather Roller

I. INTRODUCTION

Textiles constitute an important component of India's economy. Ginning is the first and most important mechanical process by which seed cotton is separated into lint (fibre) and seed and machine used for this separation is called as gin. There are mainly two types of gins (i) roller gins- most commonly used in India, Egypt, Uganda, Tanzania etc. and (ii) saw gins- extensively used in countries like USA, China, Australia, Uzbekistan etc.. Both type of gins are noted for certain advantages and disadvantages. The roller gin is used on high quality, fine fibred, extra-long staple cottons because of its tendencies to maintain fibre length and low nep levels as opposed to the adverse effects on these characteristics by the saw gins. Double roller (DR) gins are commonly used in rural India for ginning and producing about 2.8 million tonnes of lint . In India about 50000 DR gins are working in around 4000 ginning factories and there is demand of around 1500 DR gins every year towards new addition as well as replacement of old one. During 1700's some developments using roller principle followed and in 1840 Fones McCarthy invented single roller gin . The British Middleton model of DR gin used 40 inches (1016mm) roller length while the American Foss model DR gin used 60 inches (1524mm) long roller . Britishers introduced Middleton Double Roller gin manufactured by Platt Saco Lowell (UK) Ltd and Monforts M. Gladbach (Volkart) DR gin in India in the beginning of 20th Century. After India's independence Indian manufacturers have started manufacturing DR gins similar to Middleton DR gin and Monforts DR gin and major technological design modifications were not implemented for the improvement of the machine It consists of two spirally grooved leather roller pressed against a fixed knife, are made to rotate at about 90-120 rpm. Two moving blades combined with seed grids constitutes a central assembly known as beater which oscillates by means of a crank or eccentric shaft, close to the fixed knife. When the seed cotton is fed to the machine in action, fibres adhere to the rough surface of the roller are carried in between the fixed knife and roller in such a way that the fibres are partially gripped between them. The oscillating knife beats the seed and separates the fibres. This process is repeated for number of times and due to push-pull-hit action the fibres are separated from the seed, carried forward on the roller and dropped out of machine. The ginned seeds drop down through the grid which is oscillating along with beater. Drawbacks in Present Design of DR Gin Detail design study revealed that present DR gin carries several drawbacks. One of the most important drawbacks is groove filling of leather roller with cotton after every 60 hours. Uniform pressure between fixed knife and roller plays an important role in quality and output of the lint. In present design the roller is pressed against fixed knife with the help of hanging dead weights (total weight of 1158 N i.e. 324 N and 255 N/roller on gear box side and offside respectively) mounted on the weight lever of 495 mm in length. This method does not ensure uniform pressure between roller and fixed knife, occupies more space, and also makes it difficult to remove the roller for maintenance. Presently rollers are made of chrome composite leather washers and wear rate of roller is 0.02 mm/h of working (i.e. it has life of around 1200 working hours). Besides this in rainy season it has tendency to absorb water and get swelled to reduce the life of material further. Studies revealed that chromium particles generated during the process of ginning produce deleterious effect on the people working in the vicinity . Theoretically energy required to remove 1 kg lint (fibres) varies between 1075 to 2775 joules but actual energy consumed by present DR is about 118000 joules/kg lint. This is about 60 to 120 times more. This poor energy utilization efficiency is mainly due to improper design of gearbox, unscientific way of applying pressure etc. Machine noise level is reasonably high (93 dB) due to the reciprocating action of beater and gearbox. Leather roller with spherical groove firstly removed by hand grinder which required two person one for holding the grinding cutter and other for rolling the leather roller study suggest that it required near about half hour to cut groove on the leather roller and after every 60 hours ,the groove made

by the grinding cutter are not approximate and spherical and the depth and width of the groove increases which required 2 mm both.



Fig. 1:



Fig. 2:

Due to all this problem we are able to design the leather roller groove cutter machine this is an proto type model.in which the leather roller is attached on the main base frame stand and grinding cutter is mounted on the rack and pinion arrangement which slides as the roller rotates with the help of motor and pulley arrangement, The leather roller rpm is maintain at 1 rpm and the grinding cutter slides along the length of the groove which is more than the length of the leather roller.as the roller rotates the the grinding cutter automatically rotates with the leather roller and the cutter rotates at 11000rpm which maintain the depth and width of the groove of roller of 2 mm.All the calculation are based on time minimum 10 to 15 sec are required to cut 1 groove on the base on roller length and no of grooves to cut. on the based of previous paper we are able to cut groove and recent industries study help us for calculating and maintaining accuracy for the leather roller.

II. LITERATURE REVIEW

Valentin Appenzeller, Kempen present paper on "method of making helically grooved roller" In a grooved drum with a working drum surface of metal consisting of a cylindrical inner drum on which at least one ribbon is helically wound, turn by turn, the radially outer boundary of which forms the working drum surface, the ribbon also forming axially extending circumferential grooves, adjacent turns of the ribbon are connected to each other in a form locking manner on their sides facing each other in the axial direction causing the adjoining turns to be immovably xed relative to each other so that in the event of a break in the ribbon the turns will not give in to the circumferential tension force generated during winding and burst open.

Ernst Grob, Zurich, Switzerland, and Benjamin Grob, Grafton present paper on "improvement for machine cutting leather" This invention relates to the art of metal rolling. A general aim of the present invention is to provide a novel method of metal rolling particularly adapted for the rapid and economical production of gears and other similar repetitively grooved essentially cylindrical shapes. A more specific object is to provide an improved rolling method for the purpose indicated in which each of the several grooves is progressively formed by the repetitive rolling engagement of a forming roller or succession of like rollers therewith. Another object is to provide a gear forming method in which the gear teeth are formed by a metal rolling action that progresses across the face of the gear or substantially lengthwise of the axis thereof. Other more specific objects and advantages will appear, expressed or implied, from the following description of a metal rolling method performed in accordance with this invention. For purposes of illustration and explanation the method of the present invention is shown applied to the rolling of spur and helical gears, although it may be utilized to advantage in the rolling of splines and other shapes of the kind hereinabove mentioned. In the accompanying drawings: Figure 1 is a sectional view of gear rolling equipment capable of use in carrying out the method of the present invention.

A. Objective

The main aim of this project is to overcome the traditional method

- To design the leather roller groove cutter machine for double roller ginning machine.
- Fabrication of leather roller using mechanical process and engg parameters.
- To sort-out the reasons of failure of leather roller groove.
- To suggest the solutions on failure.

III. RESEARCH METHODOLOGY

The research methodology will cover follow; sufficient literature is available related to ginning machine and leather roller. Initially the available literature would be reviewed for identifying the parameters responsible for the failure of the leather roller groove .

First the design and fabrication of leather roller groove cutter will be done using CATIA software and then analysis will give the exact location from which the leather roller groove fails. And then redesign and analysis of redesign leather roller groove cutter will be done using same software. After this the fabrication of the leather groove cutter machine will be done as per the design and engineering parameter and running of the machine will be done and maintain the accuracy of grinding cutter and leather roller is done. The conclusion and future scope of work will be discussed in the end.

A. Design of Leather Groove Cutting Machine

By, considering efficiency, $\eta = 85\%$

Required motor power is,

$$P = H / \eta$$

$$P = 624.88 / 0.85$$

$$P = 735.15 \text{ watts}$$

Selecting market availability of Motor:-

Power : 1H.P., 746 watts

Current 1.8 AMP

Speed 1440 RPM

Consider,

$N_1 =$ Speed of electric motor shaft = 1440 rpm

$N_1 =$ Speed of pulley-1 i.e. on motor shaft

$N_2 =$ Speed of pulley-2 i.e. on main shaft

Assuming, Velocity Ratio for shelling speed, $V_R = 3$

$$\frac{N_1}{N_2} = 3$$

So, $N_2 = 540 \text{ rpm}$

Gear reduction ratio = 1:56

Design of Shaft

A) Design Torque,

$$T_d = \frac{60 \times P \times K_L}{2\pi N}$$

Load Factor, $K_L = 1.75$ (For Line Shaft)

$$T_d = \frac{60 \times 746 \times 1.75}{2 \times \pi \times 540}$$

$$T_d = 23.944 \text{ N-m}$$

B) Forces on belt drive

$$T_d = (T_1 - T_2) D_2 / 2$$

$$23.944 = (T_1 - T_2) 2000 \times 10^{-3} / 2$$

$$(T_1 - T_2) = 232 \text{ -----1}$$

$$\frac{T_1}{T_2} = e^{\mu\theta}$$

$$\frac{T_1}{T_2} = e^{0.3 \times 2.427}$$

$$T_1 = 2.0712 T_2 \text{ -----2}$$

Using eq. 1 & eq. 2

$$T_2 = 215.528 \text{ N}$$

$$T_1 = 448.403 \text{ N}$$

C) Force calculation on main shaft 1

Weight of bigger Pulley

$$W_{PA} = 8.2 \text{ Kg}$$

$$= 8.2 \times 9.81$$

$$W_{PA} = 60.82 \text{ N}$$

Weight of main shaft with drum

$$W_{SH} = 8.4 \text{ Kg}$$

$$= 8.4 \times 9.81$$

$$W_{SH} = 82.4 \text{ N}$$

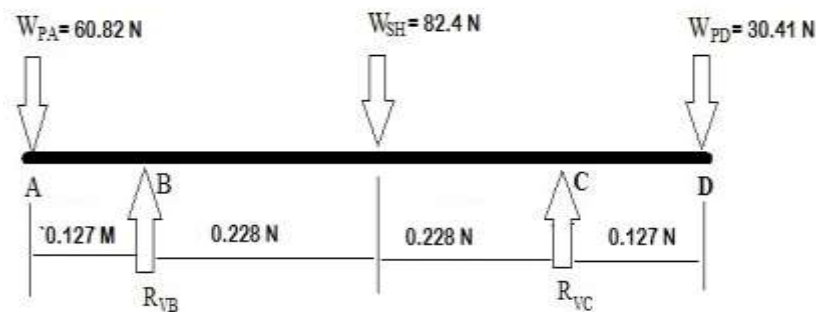
Weight of smaller Pulley

$$W_{PD} = 3.1 \text{ Kg}$$

$$= 3.1 \times 9.81$$

$$W_{PD} = 30.41 \text{ N}$$

1) Vertical Shear Force Diagram



$$\sum F = 0$$

$$W_{PA} + R_{VB} + W_{SH} + R_{VC} + W_{PD} = 0$$

$$-60.82 + R_{VB} - 82.4 + R_{VC} - 30.41 = 0$$

$$R_{VB} + R_{VC} = 173.63 \text{ N ----- (1)}$$

Movement at point B

$$\sum M_B = 0$$

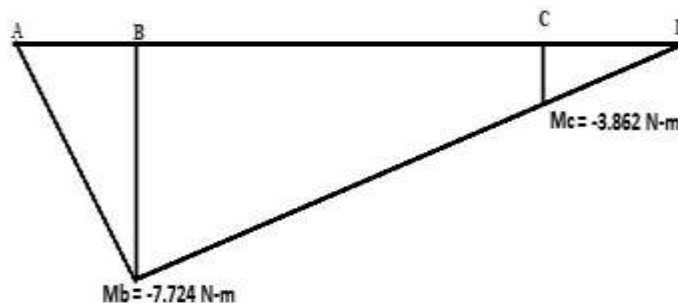
$$\sum M_B = -60.82 \times 0.127 + 82.4 \times 0.228 - R_{VC} \times 0.456 + 30.41 \times 0.583$$

$$R_{VC} = 63.141 \text{ N}$$

Putting in equation (1)

$$R_{VB} = 110.489 \text{ N}$$

2) Vertical Bending Moment



Vertical B. M. Diagram

$$M_a = 0$$

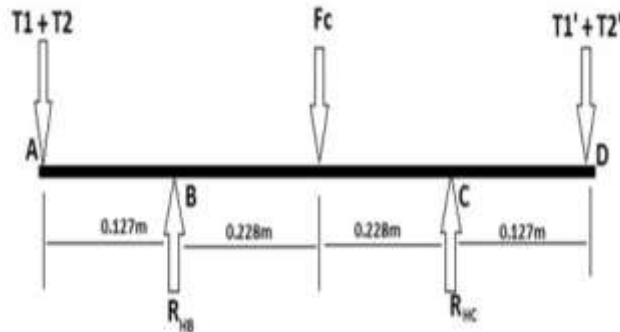
$$M_b = -60.82 \times 0.127 = -7.724 \text{ N-m}$$

$$M_c = -60.82 \times 0.583 + 100.489 \times 0.456 - 82.4 \times 0.228$$

$$M_c = -3.862 \text{ N-m}$$

$$M_d = 0$$

3) Horizontal Shear Force Diagram



$$W_2 = (2\pi N/60)^2$$

$$= (2\pi \times 240 / 60)^2$$

$$= 631.65 \text{ r/s}$$

$$\text{Rotor radius} = 0.1778 \text{ m}$$

$$F_c = mrw^2$$

$$= 8.4 \times 0.1778 \times 631.65$$

$$= 943.38 \text{ N}$$

$$\sum F_H = 0$$

$$-661.93 + R_{HB} - 943.38 + R_{HC} - 1225.9 = 0$$

$$R_{HB} + R_{HC} = 2831.12$$

Moment at Point B

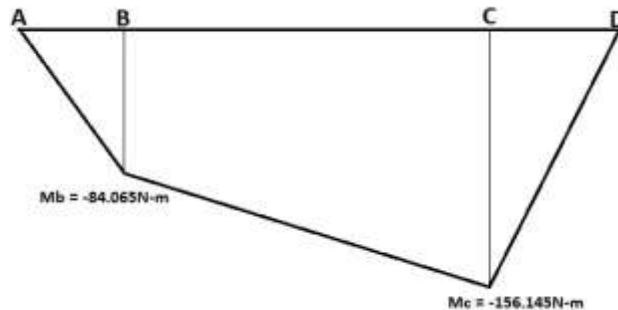
$$\sum M_A = 0$$

$$\sum M_B = -661.93 \times 0.127 + 943.38 \times 0.228 - R_{HC} \times 0.456 + 1225.9 \times 0.0583$$

$$R_{HC} = 1886.66 \text{ N}$$

Putting in equation 2

4) Horizontal Bending Movement



Horizontal B. M. Diagram

$$M_A = 0$$

$$M_B = -661.93 \times 0.127 = -86.065 \text{ N-m}$$

$$M_C = -661.93 \times 0.583 + 975.55 \times 0.456 - 943.38 \times 0.228 = -158.145 \text{ N-m}$$

$$M_D = 0$$

Resultant Bending Moment,

$$M_B = \sqrt{(M_{VB})^2 + (M_{HB})^2}$$

$$= \sqrt{(-5.724)^2 + (-86.065)^2}$$

$$= 86.42 \text{ N-m}$$

Resultant Bending Moment,

$$M_C = \sqrt{(M_{VC})^2 + (M_{HC})^2}$$

$$= \sqrt{(-3.862)^2 + (-158.145)^2}$$

$$= 158.2 \text{ N-m}$$

Hence, selecting max.movement on shaft, $M = 158.2 \text{ N-m}$

Selecting material of shaft SAE 1030,

$$S_{ut} = 527 \text{ MPa}$$

$$S_{yt} = 296 \text{ MPa}$$

$$\tau_{max} \leq 0.30 S_{yt}$$

$$\tau_{max} \leq 0.18 S_{ut}$$

Considering F.O.S. = 2 (T-I-20-A)

For ductile material with dynamic heavy shocks for machines like forging, shearing and punching etc.

$$\tau_{max} \leq 0.30 S_{yt}$$

$$= 0.30 \times \frac{296}{2}$$

$$= 46.4 \text{ N/mm}^2$$

$$\tau_{max} \leq 0.18 S_{ut}$$

$$= 0.18 \times \frac{527}{2}$$

$$= 45.43 \text{ N/mm}^2$$

Considering maximum of it i.e. $\tau_{max} = 46.4 \text{ N/mm}^2$

Now, for diameter of shaft,

$$\tau_{max} = \frac{16}{\pi d^3} \sqrt{(K_b M)^2 + (K_t T_d)^2}$$

Now, Recommended value for K_b and K_t (T-XII-3)

For rotating shaft,

Gradually applied load (Heavy shocks)

$$K_b = 1.5$$

$$K_t = 1$$

$$46.4 = \frac{16 \times 1000}{\pi d^3} \sqrt{(1.5 \times 158.2)^2 + (1 \times 51.944)^2}$$

$$= 23.2 \text{ mm}$$

Selecting std. dia. Of shaft

$$D_{sh} = 24 \text{ mm}$$

D) HUB PROPORTIONS

Hub diameter,

$$D_h = 1.5 d_s + 25 \text{ mm}$$

$$d_s = \text{Diameter of shaft} = 24 \text{ mm}$$

$$= 1.5 \times 24 + 25$$

$$D_h = 62 \text{ mm}$$

Length of Hub,

$$L_h = 1.5 d_s$$

$$= 1.5 \times 24$$

$$= 36 \text{ mm}$$

6.1.2 Design of V-Belt

A) Design Power,

$$(P_d) = PR \times K_L \text{ (T-XV-9)}$$

$$= 0.746 \times 1.10 = 0.8206 \text{ KW}$$

Load Factor,

$$K_l = 1.10 \text{ (T-XV-2)}$$

Selection of belt,

On the basis of design power i.e. 0.8206 KW from design data book (T-XV-8)

Belt designation is 'A'.

Nominal width, $w = 13 \text{ mm}$ or $\frac{1}{2}$ inch;

Nominal thickness, $t = 8 \text{ mm}$;

Recommended Diameter, $D = 75 \text{ mm}$ or 3 inch;

Centrifugal tension factor, $K_C = 2.52$;

Bending stress factor, $K_b = 15.6 \times 10^3$

Peripheral Velocity, $V_p = (\pi D_1 N_1) / 60$

Diameter of smaller pulley i.e. electric motor shaft pulley, $D_1 = 75 \text{ mm}$;

Speed of electric motor shaft pulley, $N_1 = 1440 \text{ RPM}$

Cross check,

$$V_p = (\pi \times 75 \times 1440) / (1000)$$

$$= 339.29 \text{ m/min. or } 5.5 \text{ m/s}$$

For V- belt drive, $V_p = 300$ to 1500 m/min

This velocity is in range .So, selected velocity ratio is correct. (T-XV-10)

By using velocity ratio with neglecting slip,

$$\frac{N_2}{N_1} = \frac{D_1}{D_2}$$

$$D_2 = \frac{1440}{540} \times 75$$

$$D_2 = 200 \text{ mm} = 8 \text{ inch}$$

D_2 = Diameter of larger pulley on shaft

$$B) \text{ Power Rating Per Belt} = (F_W - F_C) \frac{e^{\mu\theta/\sin\frac{\alpha}{2}} - 1}{e^{\mu\theta/\sin\frac{\alpha}{2}}} \times V_P$$

$$\text{Working Load, } F_W = 13^2 = 169$$

$$\begin{aligned} \text{Centrifugal Tension, } F_C &= K_C \times \left(\frac{V_P}{5}\right)^2 \\ &= 2.52 \times \left(\frac{5.5}{5}\right)^2 \end{aligned}$$

$$F_C = 3.05 \text{ N}$$

Angle of lap or contact on smaller pulley,

$$\theta_1 = \pi - \frac{D_2 - D_1}{C}$$

Angle of lap or contact on larger pulley,

$$\theta_2 = \pi + \frac{D_2 - D_1}{C}$$

Centre to center distance for V-belt, $C = D_2 = 450 \text{ mm}$

(D_2 = Diameter of larger pulley on shaft-1)

OR

Centre to center distance for V-belt,

$$C = (D_1 + D_2)$$

$$= (75 + 200)$$

$$= 275 \text{ mm}$$

$$\theta_1 = \pi - \frac{200 - 75}{525}$$

$$= 2.627 \text{ Radian}$$

Selecting, cone angle, $\alpha = 38^\circ$

$$\text{Power /Belt} = (169 - 3.05) \times \frac{e^{0.3 \times 2.4 / \sin 17} - 1}{e^{0.3 \times 2.4 / \sin 17}} \times 5.5$$

$$= 836.31 \text{ Watt}$$

$$= 0.83 \text{ KW}$$

$$\text{No. of belts} = \frac{P_d}{\text{Power/Belt}}$$

$$= \frac{0.8206}{0.83}$$

$$= 0.988$$

$$n \cong 1$$

C) Length of the Belt,

$$L = \frac{\pi}{2} \times (D_1 + D_2) + 2C + \frac{(D_1 - D_2)^2}{4C}$$

$$= \frac{\pi}{2} \times (75 + 200) + 2 \times 275 + \frac{(75 - 200)^2}{4 \times 275}$$

$$= 981.6 \text{ mm} \cong 1 \text{ m}$$

D) Bending Load,

$$F_b = \frac{K_b}{D}$$

$$\text{For smaller pulley, } = \frac{17.6 \times 10^3}{75} (D_1 = 75 \text{ mm})$$

$$= 236.7 \text{ N}$$

$$\text{For larger pulley, } = \frac{17.6 \times 10^3}{200} (D_2 = 200 \text{ mm})$$

$$= 88 \text{ N}$$

IV. CONCLUSION

The main conclusion will be to find out whether it is possible to maintain the accuracy rolling roller and grinding cutter . Also the future scope for developing design model for any profile can be identified. selection of proper design parameter has been done.

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