

# Design Consideration in an Automatic Can/Plastic Bottle Crusher Machine

Vishal N. Kshirsagar<sup>1</sup>, Dr. S..K Choudhary<sup>2</sup>, Prof. A.P Ninawe<sup>2</sup>

<sup>1</sup>Research Scholar (M.Tech Pursuring, M.E.D), KDK College of Engineering, Nagpur, Maharashtra, India

<sup>2</sup> KDK College of Engineering, Nagpur, Maharashtra, India

E-mail- [vishalk031@gmail.com](mailto:vishalk031@gmail.com)

**Abstract:** This paper describes about the design of various components of can or plastic bottle crusher machine. This machine is widely used in beverage industries or in scrap dealers shop to reduce the volume of the cans/bottles. Hence in this design of various parts are necessary, and design of various parts due to which the design quality of those parts will be improved. There are so many researchers who have done work on design and analysis also, but still there are so many areas of scope regarding this design. Overall, this project involves processes like design, fabrication and assembling of different components etc. After all process has been done, this crusher may help us to understand the fabrication and designing that involved in this project.

**Keywords:** Design consideration, calculations, design procedure, Crushing force, force analysis, load diagram.

## I. INTRODUCTION

The sole purpose of this paper is to understand the fundamental knowledge of design and mechanism. The design is an environment friendly and uses simple mechanism properties such as fulcrum system, single slider crank mechanism and automation properties etc. In this, some crushing force is needed to crush the cans/bottles to reduce its volume by large extent. The design is so done that the knowledge of designing, mechanism and forces are increased. This project consists of designing and fabrication of an automatic can crusher machine considering various important parameters. In this project, development of a recycle bin can/bottle crusher so the Can might crush as flat and look as symmetrically as possible and inserted the bin. As well as the study of manufacturing was very important in order to carry out this project to ensure that what are needs to do. This project involves the process of designing the different parts of this crusher machine considering forces and ergonomic factor for people to use. This project is mainly about generating a new concept of can/bottle crusher that would make easier to bring anywhere and easier to crush the can or bottle. After the design has completed, it was transformed to its real product where the design is used for guideline.

## II. DESIGN PROCEDURE

The aim of this is to give the complete design information about the Can Crusher machine. In this, the explanations and some other parameters related to the project are included. With references from various sources as journal, thesis, design data book, literature review has been carried out to collect information related to this project.

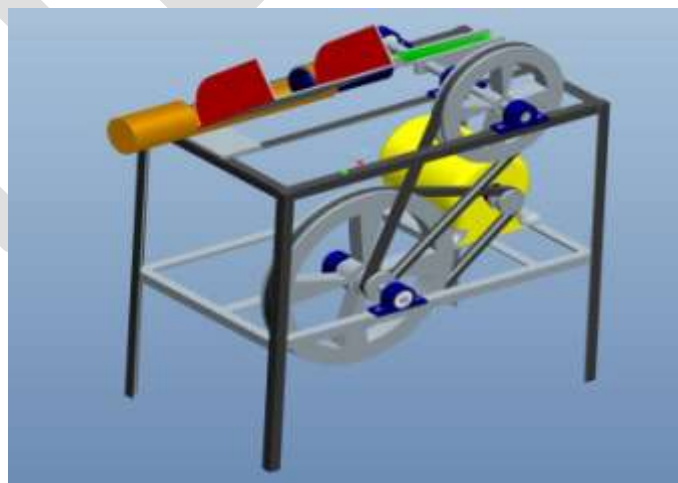


Fig.-1: Modeling of machine

## A. Design consideration

- Maximum Force required to crush the Can/Plastic bottles
- Considered elements
- Standard size of cans/plastic bottles
- Material of Cans Aluminum

## B. Design calculations

Determination of crushing force experimentally

- Force required to crush the plastic bottle
- Force required to crush the Soda/Pepsi Can

So, we considering maximum of it.

$$\text{Torque, } T = F \times r$$

Where,  $r$  is radius or length of the crank.

$F$  is required crushing force.

Power is given by,

$$P = \frac{T\omega}{60}$$

$T$  is torque required

$\omega$  is angular velocity =  $2\pi N/60$

where,  $N$  is speed of the crank.

Again, Power can be calculated by static force analysis,

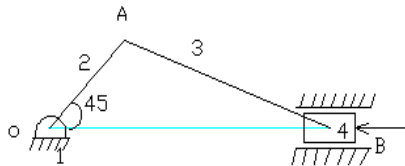
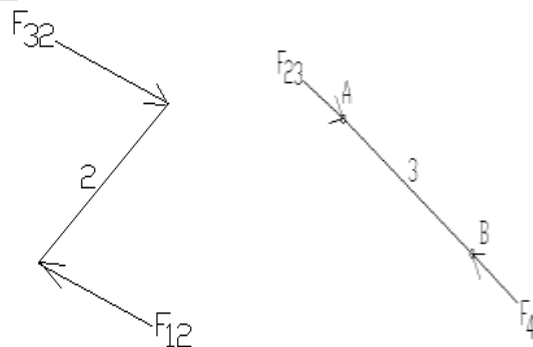


Fig.-2: Single Slider Crank Mechanism



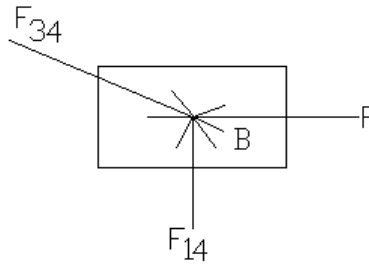


Fig.3: Force Analysis of Links 2, 3 and 4

Force  $F_{43}$  is opposite to force  $F_{34}$  and  
 Force  $F_{43}$  is also opposite to force  $F_{23}$

$$\therefore F_{43} = F_{23} = F_{34}$$

Again, Force  $F_{23}$  is opposite to Force  $F_{32}$  and Force  $F_{32}$  opposite to Force  $F_{12}$

$$\therefore F_{32} = F_{23} = F_{12}$$

Link - 2 shown in figure with Force  $F_{32}$  and  $F_{12}$  is been separated by a distance 'd' forming a clockwise couple. Therefore a resisting couple 'C' should be equal  $F_{32}d$  or  $F_{12}d$  in counter clockwise direction.

$$C = - F_{32}d \quad (\text{CCW})$$

Or

$$\text{Torque, } T = F \times r$$

Power is given by,

$$\therefore P = \frac{T\omega}{60}$$

So, from this we can decide the Crushing Power Required.

### C. Design Of V-Belt

$$\text{Design Power } (P_d) = P_R \times K_L$$

Where,

$P_R$  = rated power

Load Factor,  $K_L = 1.10$

Selection of belt on the basis of design power. Nominal width, w

Nominal thickness, t

Recommended Diameter, D

Centrifugal tension factor,  $K_C$

Bending stress factor,  $K_b$

$$\text{Peripheral Velocity, } V_p = \frac{\pi D_1 N_1}{60}$$

$D_1$  = Diameter of smaller pulley i.e. electric motor shaft pulley,

$N_1$  = Speed of electric motor shaft pulley.

If this velocity i.e.  $V_p$  is in range then, Ok.

Now, assuming Velocity Ratio, VR to calculate speed of driven pulley.

$$N_1/N_2 = VR$$

By using velocity ratio with neglecting slip,

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

$D_2$  = Diameter of larger pulley

Centre to centre distance for V-belt,

$$C = (D_1 + D_2)$$

**OR**  $C = D_2$

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,

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

Since the smaller value of ' $\theta$ ' for the pulley will governs the design.

Belt Tension Ratio,

$$\frac{F_1}{F_2} = e^{\mu\theta} \operatorname{cosec} \alpha/2$$

$\alpha$  = Groove angle =  $34^\circ$

$\mu$  = Coefficient of friction = 0.3

$F_1$  = Tension in tight side

$F_2$  = Tension in slack side

Belt Tension,  $(F_1 - F_2) = \frac{P_d}{V_p}$

$$\text{Power Rating Per Belt} = (F_w - F_c) \frac{e^{\mu\theta} / \sin \frac{\alpha}{2} - 1}{e^{\mu\theta} / \sin \frac{\alpha}{2}} \times V_p$$

Working Load,  $F_w$

$$\text{Centrifugal Tension, } F_C = K_C \times \left(\frac{V_P}{5}\right)^2$$

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

Length of the Belt,

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

$$\text{Bending Load, } F_b = \frac{K_b}{D}$$

$K_b$  = Bending stress factor,

$D$  = Diameter of pulley i.e. smaller or large.

$$\text{Initial Tension, } 2\sqrt{F_i} = \sqrt{F_1} + \sqrt{F_2}$$

$$\text{Fatigue Life of Belt, } F = F_i + F_C + F_{b\max}$$

#### D. Design of Shaft

$$\text{Design Torque, } T_d = \frac{60 \times P \times K_L}{2\pi N}$$

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

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

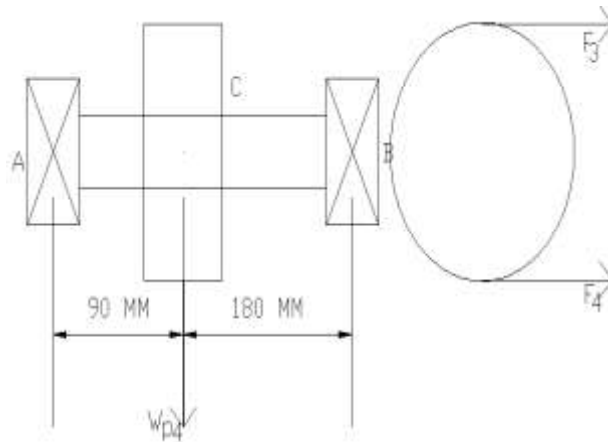
For ductile material with dynamic heavy shocks for machines like forging, shearing and punching etc.

$$\begin{aligned} \tau_{\max} &\leq 0.30 S_{yt} \\ &= 0.30 \times \frac{296}{2} \\ &= 44.4 \text{ N/mm}^2 \end{aligned}$$

$$\begin{aligned} \tau_{\max} &\leq 0.18 S_{ut} \\ &= 0.18 \times \frac{527}{2} \\ &= 47.43 \text{ N/mm}^2 \end{aligned}$$

Considering minimum of it i.e.  $\tau_{\max} = 44.4 \text{ N/mm}^2$ .

Consider Shaft-2 under loading



$W_{P4}$  = Weight of pulley.

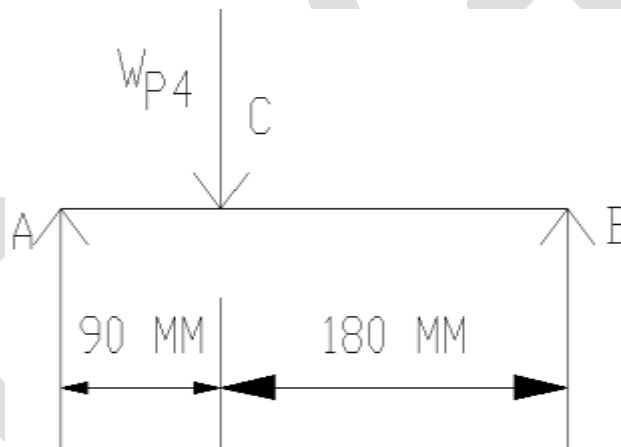


Fig.:- Vertical Load Diagram

Resolving all the force vertically,

$$R_{AV} + R_{BV} = W_{P4}$$

Taking moment about 'A'

$$W_{P4} \times 90 = R_{BV} \times 270$$

$R_{BV}$  = Vertical Reaction at B

$R_{AV}$  = Vertical Reaction at A

As we know that bending moment at A and B will be Zero.

$$\therefore M_{AV} = M_{BV} = 0$$

$M_{AV}$  and  $M_{BV}$  are the vertical bending moments at point A and B respectively.

$$\text{B. M. At C} = R_{AV} \times 90$$

Now,

Resolving all the forces horizontally,

$$R_{AH} + R_{BH} = F_3 + F_4$$

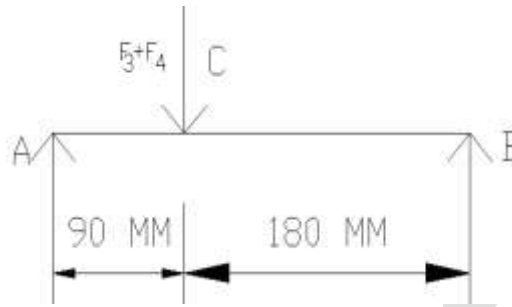


Fig.:- Horizontal load Diagram

Taking moment about 'A'

$$(F_3 + F_4) \times 90 = R_{BH} \times 270$$

We know that B.M. at A and B will be zero.

$$\therefore M_{AH} = M_{BH} = 0$$

$M_{AH}$  and  $M_{BH}$  are the horizontal bending moments at point A and B respectively.

$$\text{B.M. at C, } M_{CH} = R_{AH} \times 90$$

Resultant Bending Moment,

$$M_C = \sqrt{(M_{CV})^2 + (M_{CH})^2}$$

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  $K_t$

For rotating shaft,

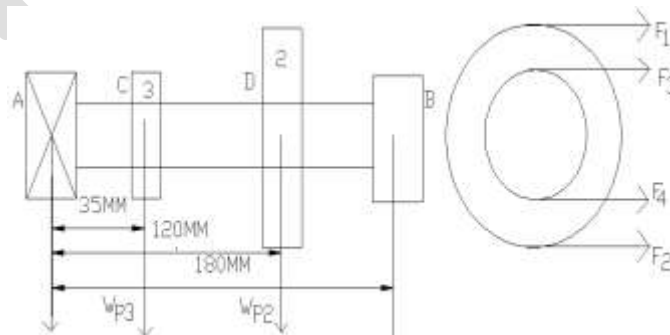
Suddenly applied load (Heavy shocks)

$$K_b = 2 \text{ to } 3 = 2.5$$

$$K_t = 1.5 \text{ to } 3 = 2.3$$

$$\tau_{\max} = 44.4 \text{ N/mm}^2$$

Consider Shaft – 1 under loading



$W_{P2}$  = Weight of pulley-2,  
 $W_{P3}$  = Weight of pulley-3.

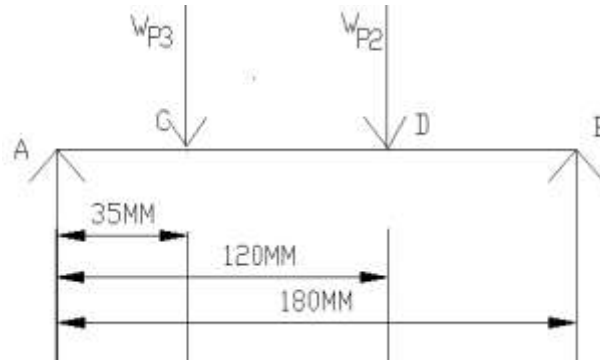


Fig.:- Vertical Load Diagram

Resolving all the forces vertically,

$$R_{AV} + R_{BV} = W_{P2} + W_{P3}$$

Taking moment about 'A'

$$W_{P3} \times 35 + W_{P2} \times 120 = R_{BV} \times 180$$

$R_{BV}$  = Vertical Reaction at B

$R_{AV}$  = Vertical Reaction at A

We know that B.M. at A and B is zero

$$\therefore M_{AV} = M_{BV} = 0$$

$M_{AV}$  and  $M_{BV}$  are the vertical bending moments at point A and B respectively.

$$\text{B.M. at C, } M_{CV} = R_{AV} \times 35$$

$$\text{B.M. at D, } M_{DV} = R_{BV} \times 60$$

Resolving all the force horizontally,

$$R_{AH} + R_{BH} = (F_1 + F_2) + (F_3 + F_4)$$

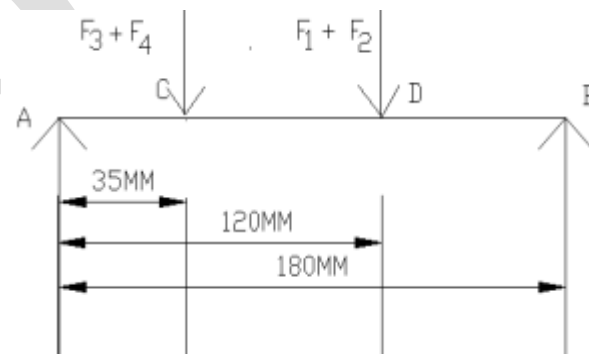


Fig.:- Horizontal Load Diagram



Taking moment about 'A'

$$R_{BH} \times 180 = (F_1 + F_2) \times 120 + (F_3 + F_4) \times 35$$

$R_{BH}$  = Horizontal Reaction at B

$R_{AH}$  = Horizontal Reaction at A.

We know that B.M. at A and B is zero,

$$\therefore M_{AH} = M_{BH} = 0$$

$M_{AH}$  and  $M_{BH}$  are the horizontal bending moments at point A and B respectively.

$$\text{B.M. at C, } M_{CH} = R_{AH} \times 35$$

$$\text{B.M. at D, } M_{DH} = R_{BH} \times 60$$

Resultant Bending Moment,

$$M_C = \sqrt{(M_{CV})^2 + (M_{CH})^2}$$

$$M_D = \sqrt{(M_{DV})^2 + (M_{DH})^2}$$

Now, diameter of shaft,

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

For rotating shaft,

Suddenly applied load (Heavy shocks)

$$K_b = 2 \text{ to } 3 = 2.5$$

$$K_t = 1.5 \text{ to } 3 = 2.3$$

$$\tau_{\max} = 44.4 \text{ N/mm}^2$$

## E. Design of Pulley

$$L_p = 11 \text{ mm};$$

$$b = 3.3 \text{ mm};$$

$$h = 8.7 \text{ mm}$$

$$e = 15 \pm 0.3;$$

$$f = 9-12 = 10.5;$$

$$\alpha = 34^\circ;$$

$$\text{Min. Pitch Diameter, } D_p = 75 \text{ mm}$$

Types of construction – Web construction for pulley diameter below 150 mm

Types of construction – Arm construction for pulley diameter above 150 mm i.e. for bigger pulleys.

No. Of Arms = 4

No. Of Sets = 1

$$\text{Rim thickness, } t = 0.375 \sqrt{D} + 3 \quad (\text{Heavy Duty Pulley})$$

D = Diameter of pulley

$$\text{Width of Pulley, } W = (n - 1) e + 2f$$

Where 'n' is no. of belts = 1.

#### Hub Proportions

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

$d_s$  = Diameter of shaft = 18 mm

Length of Hub,  $L_h = 1.5 d_s$

Moment on each Arm,

$$M = \frac{(F_1 - F_2) (D - D_h)}{n}$$

n = no. of arms

$D_h$  = Hub diameter

### III. FABRICATION

#### Mechanical Components

- Shaft (2 Nos.)
- Pulley (4Nos.)
- Belt(2Nos.)
- Single Slider Mechanism.
- Separating Bin.
- Crushing Tray.
- Angles (For Frame).

### IV. CONCLUSION

The above design procedure is been adopted for the fabrication of Automatic Can/Plastic Bottle Crusher machine which will make the product durable for long time as well as make it efficient also helps to understand the concept of design. Thus, with help of this design and some other electronic components we can fabricate an automatic can/plastic bottle crusher machine to simply reduce the volume of cans/plastic bottles as well as to reduce the human fatigue.

#### REFERENCES:

- [1] Mr. Ramkrushna S. More, Sunil J. Rajpal publishes a review paper on "Study of Jaw Plates of Jaw Crusher". International Journal of Modern Engineering Research (IJMER), Vol.3, Issue.1, Jan-Feb. 2013 pp-518-522 ISSN: 2249-6645.
- [2] Mr. Shadab Husain, Mohammad Shadab Sheikh presents paper on "Can crusher machine using scotch yoke mechanism". IOSR Journal of Mechanical and Civil Engineering (IOSR-JMCE) e-ISSN: 2278-1684, p-ISSN: 2320-334X PP 60-63.
- [3] Mr. Che Mohd Akhairil Akasyah B Che Anuar Faculty of Mechanical Engineering in University Malaysia Pahang in the year Nov.2008 in his project report entitled "Development of the Can Crusher Machine".

[4]Cao Jinxi, Qin Zhiyu, Rong Xingfu, Yang Shichun presents a paper on “ Experimental Research on Crushing Force and its Distribution Feature in Jaw Crusher ” 2007 Second IEEE Conference on Industrial Electronics and Applications. 1-4244-0737-0/07/\$20.00© 2007 IEEE.

[5] Tiehong GaoJunyi Cao, Duniyu Zhu, Jinzhang Zhi presents a paper on “Study on Kinematics Analysis and Mechanism Realization of a Novel Robot Walking on Water Surface”. Proceedings of the 2007 IEEE International Conference on Integration Technology March 20 - 24, 2007, Shenzhen, China. 1-4244-1092-4/07/\$25.00 © 2007IEEE.

[6] Design data for machine elements by B.D.Shiwalkar 2013 edition and a Textbook of Machine Design by R.S. Khurmi and J.K.Gupta 14<sup>th</sup> revised edition S. Chand publication

IJERGS