

# **DESIGN AND ANALYSIS OF A HORIZONTAL SHAFT IMPACT CRUSHER**

Thesis submitted in partial fulfillment of the requirements for the degree of

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In

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By

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Certificate of Approval

This is to certify that the thesis entitled DESIGN AND ANALYSIS OF HORIZONTAL SHAFT IMPACT CRUSHER submitted by Sri Deepak Gupta (Roll No. 107ME037) has been carried out under my supervision in partial fulfillment of the requirements for the Degree of Bachelor of Technology (B. Tech.) in Mechanical Engineering at National Institute of Technology, NIT Rourkela, and this work has not been submitted elsewhere before for any other academic degree/diploma.

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## **ABSTRACT**

Crushers are one of the major size reduction equipment that is used in metallurgical, mechanical, and other similar industries. They exist in various sizes and capacities which range from 0.1 ton/hr. to 50 ton/hr. They can be classified based on the degree to which they can fragment the starting material and the way they apply forces. Based on the mechanism used crushers are basically of three types namely Cone crusher, Jaw crusher and Impact crusher. Our objective is to design various components of an Impact crusher like drive mechanism, shaft, rotor, hammers, casing, and discharge mechanism which will be useful in minimizing weight, cost and maximizing the capacity and also do their analysis. Impact crushers involve the use of impact rather than pressure to crush materials. Here the material is held within a cage, with openings of the desired size at the bottom, end or at sides to allow crushed material to escape through them. This type of crusher is generally used with soft materials like coal, seeds or soft metallic ores. The mechanism applied here is of Impact loading where the time of application of force is less than the natural frequency of vibration of the body. Since the hammers/blow bars are rotating at a very high speed, the time for which the particles come in contact with the hammers is very small, hence here impact loading is applied. The shaft is considered to be subjected to torsion and bending. The grinding screen is also designed for optimal output from the crusher. A performance model is also considered for the horizontal shaft impact crusher so as to find out the relation between the feed, the crusher parameters and the output parameters.

**INTRODUCTION**

A crusher is a device that is designed to reduce large solid chunks of raw material into smaller chunks.

Crushers are commonly classified by the degree to which they fragment the starting material with primary crushers that do not have much fineness, intermediate crushers having more significant fineness and grinders reducing it to a fine power.

A crusher can be considered as primary, secondary or fine crusher depending on the size reduction factor.

a) Primary crusher – The raw material from mines is processed first in primary crushers. The input of these crushers is relatively wider and the output products are coarser in size. Example - Jaw crusher, Gyratory crusher, Impact Crushers, etc.

b) Secondary crusher- The crushed rocks from primary crusher are sent to these secondary crushers for further size reduction. Example:-reduction gyratory crusher, Cone crusher, disk crushers etc.

c) Fine crushers- Fine crushers have relatively small openings and are used to crush the feed material into more uniform and finer product. Example - Gravity stamp.

Table 1. Comparison of different types of crushers [6]

<b>Type</b>	<b>Hardness (input material)</b>	<b>Abrasion limit</b>	<b>Reduction ratio</b>	<b>Use</b>
Jaw crusher	Soft - very hard	No limit	3:1 to 6:1	Extracted materials, sand and gravels
Conical crusher	Medium hard - very hard	Abrasive	3:1 to 6:1	Sand and gravels
Horizontal shaft impact crusher	Soft - medium hard	Slightly abrasive	10:1 to 25:1	Extracted material, sand and gravels
Vertical shaft impact crusher	Medium hard - very hard	Slightly abrasive	6:1 to 8:1	Sand and gravels

## IMPACT CRUSHERS

These crushers involve the use of impact rather than pressure to crush materials. Here the material is held within a cage, with openings of the desired size at the bottom, end or at sides to allow crushed material to escape through them. Here the breakage can take place in a much shorter scale compared to fragmentation process used in cone or jaw crushers [7].

An impact crusher can be further classified as Horizontal impact crusher (HSI) and vertical shaft impact crusher (VSI) based on the type of arrangement of the impact rotor and shaft.

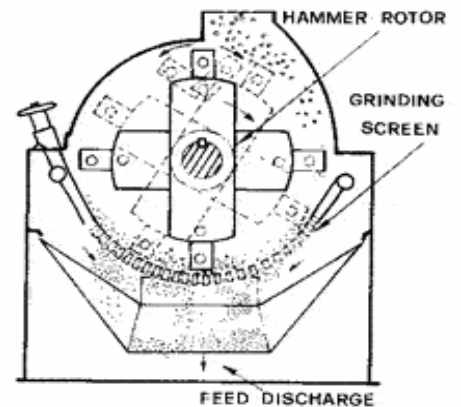


Fig. 1 A typical Impact crusher [6]

### Horizontal shaft impact crusher

These break rock by impacting the rock with hammers/blow bars that are fixed upon the outer edge of a spinning rotor. Here the rotor shaft is aligned along the horizontal axis. The input feeded material hits the rotating hammers of the rotor and due to this sudden impact it breaks the material and further breaks the material by throwing it on to the breaking bar/anvils. These have a reduction ratio of around 10:1 to 25:1 and are hence used for the extracted materials, sand, gravels etc. [6].

### Vertical shaft impact crusher

These crushers use a high speed rotor that has its axis along the vertical axis. The vertical-shaft impact crusher can be considered a stone pump that can operate like a centrifugal pump. The material is fed through the center of the rotor, where it is augmented to high speeds before being cleared through openings in the rotor sideline. The material is crushed as it hits the outer body/anvils at high speed and also due to the head on head collision action of rocks. It uses the velocity rather than the surface force as the active force to break the material fed. These have a comparative lower reduction ratio of 6:1 to 8:1 and hence are used generally for sand and gravels.



## HORIZONTAL SHAFT IMPACT CRUSHER

Here the feed material is crushed by highly rigorous impacts originating in the quick rotational movement of hammers/bars fixed to the rotor. The particles are then crushed inside the crusher as they collide against crusher parts and against each other, producing finer, better-shaped product. Adjusting the distance between impact frame and rotor frame can change the shape and size of the output.

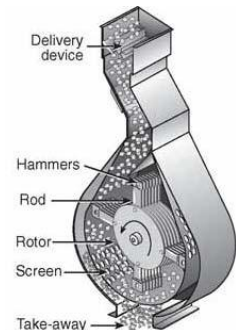


Fig. 2 Horizontal shaft impact crusher [8]

In an impact crusher the breakage takes place in a lesser time span as compared to the conical or jaw crushers. So here the nature and magnitude of forces as well as the energy dissipated due to impact breakage is different from that of the relative slow breaking that occurs due to compression or shear in other type of crushers.

### OPERATING PRINCIPLE OF THE HORIZONTAL SHAFT IMPACT CRUSHER

The Impact Crusher Machine rotor revolves in fixed direction by means of driving action of triangle belt that connects with motor. Above rotor, there are sets of suspended impact plates. Material enters into the crushing chamber through the charging hole and feeding guide plate. The blow bars fixed on rotor strikes the feed material onto impact plate and then fall from it to mutually shock material blocks. Therefore, material will be moved recurrently and repeatedly in the crushing chamber that is composed of rotor, impact plate/ anvils, hammers/ blow bars , by means of which intense shock phenomenon will act predominantly, and the material will be crushed along its natural crack and hence bulge. The gap between impact plate and hammer/blow bar can be adjusted according to practical requirement by adjusting the angle and distance of the impact anvils. Product output is easily controlled by varying the rotor speed, input feed rate and the grinding screen configuration. [8]

## **DESIGN PARAMETERS:**

For good performance, all the factors below should be taken into account:

- Assortment of a proper crushing chamber for the material.
- Feed rate control.
- Apt dimensioning of the discharge conveyor with regards to crusher's capacity.
- Selection of proper material and size for the impacting members.
- Setting of the optimum number of hammers, rotor speed, etc.
- The input material properties like density, strength, etc.

The factors below, when not taken care of may affect the performance of a crusher.[4][9]

- Occurrence of humid material in the crushers' feed.
- Extreme humidity.
- Isolation of feed in the crushing chamber.
- Irregular dispersal of feed over the crushing chamber
- Deficiency of feed control.
- Incorrect motor size.
- Deficient capacity of the crushers' discharge conveyor.
- Extremely hard material for crushing.
- Crusher functioning at a rotation speed below required conditions.

### **PERFORMANCE MODEL FOR IMPACT CRUSHER**

#### **General scheme of breakage process**

Impact breakage takes place in a very small time scale and results into a dynamic crack propagation that leads to a much faster failure of particles. The impact generates compressive and tensile shock waves that travel throughout the particle. The existence of a noteworthy, quickly growing tensile stress may help the particles to break from within. [1]

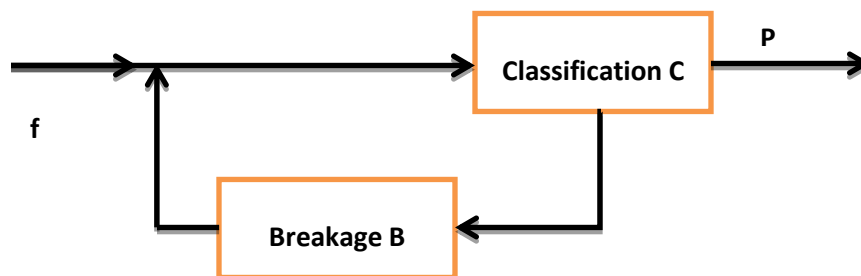


Fig.3 Scheme of breaking process in crusher [1]

#### **Mass Balance (Size distribution)**

According To size distribution model given by *whiten (1972)* the particles are represented in a discrete form of vectors [1] ( $f$ ) and ( $p$ ) where

$f$  = feed vector

$P$  = product vector

$C$  = Classification operator, computes the probability of breakage of each particle size.

$B$  = breakage operator (Governs the redistribution of broken particles in the preliminary defined size classes.)

The particles entering into crushers are selected for the breakage through the classification function operator  $C$ .

But according to the distribution model by *czoke and racz (1998)* it was assumed that the particles entered crushers for a single breakage process i.e. there is no feedback between the classification and the breakage function. This was not favorable as a single breaking process

would not yield the desired result. Hence then according To *Attou (1999)* it was found that breakage process can be divided into sequence of two processes. (a) Breakage due to impact with hammers of the rotor (b) fragmentation due to particle-particle collisions. [10][11]

The product size distribution P that we get from the process is then expressed as.

$$\mathbf{P} = (\mathbf{I} - \mathbf{C}) (\mathbf{I} - \mathbf{B.C})^{-1} \mathbf{f}$$

Where, I = identity matrix

### Classification function:

We have

$$C_i(d_i) = 1 - [(d_i - k_1)/(k_2 - k_1)]^m$$

Where  $C_i(d_i)$  = probability of breakage for a particle of size  $d_i$  (mm)

$K_1$  = min, size of particles that undergo breakage

$K_2$  = max. Particle size found in product

$m$  = shape parameter

But in this fn.  $K_1$  &  $K_2$  are static variables in the impact fracture of the particles; the probability of impact breakage depends mostly on the size and impact kinetic energy which is again a dynamic variable. So,

$$C_i(d_i) = 1 - \exp[-(d_i - d_{\min})/d_{\min}^k]$$

Where,

$d_{\min}$  = min. size of particles that undergo breakage for the given operating conditions.

$k$  = controls the shape of the classification function.

So we can see that as the feed rate increases the no of particle-particle collision increases which dissipated a lot of energy and this loss of energy leads to coarser product and greater value of  $d_{min}$

For an impact crusher the  $d_{min}$  decreases with increase in impact energy. Hence  $d_{min}$  is written as

$$d_{min} = \beta(Q/Q_0)^s (E_0/E)^n \text{ where}$$

$Q_0$  = reference feed rate

$E_0$  = reference impact energy per unit mass

$n$  = material parameter

$s$  = intensity of particle – particle interaction

$\beta$  = specific particle size depending on the crusher design and granulate properties.

## BREAKAGE FUNCTION

The breakage distribution to  $b_{ij}$  represents the fraction of the debris created from breakage of identical parent particles of size  $d_j$  and passing through a screen with mesh size  $d_i$ .

$$B_{ij}(d_i, d_j) = \phi(d_i/d_j)^m + (1 - \phi)(d_i/d_j)^l$$

$\phi$  = mass fraction of fine product

$m, l$  = material co-efficient

The breakage matrix B for N screens of mesh sizes  $D_i$  ( $i=1, N-1$ )

But according to *Kings* [12]

$$B_{ij} = b_{(i-1)j}(D_{i-1}, d_j) - b_{ij}(D_i, d_j)$$

$$B_{jj} = 1 - b_{jj}(D_j, d_j)$$

Also  $d_i$  is the characteristic dimension of particles, where

$$D_i > d_i > D_{i+1}$$

# IMPACT ENERGY CALCULATION PER UNIT MASS FOR A HORIZONTAL SHAFT CRUSHER

The Basic Assumptions made here:

1. Rotor mass is much greater than mass of single particles in the feed
2. Before impact, linear velocity of the crushing bar is much more important than the particle velocity. Hence KE of particles is negligible.
3. It is also assumed that most particles enter into the collision with the rotor bars in the median region of their impact areas with the hammer.

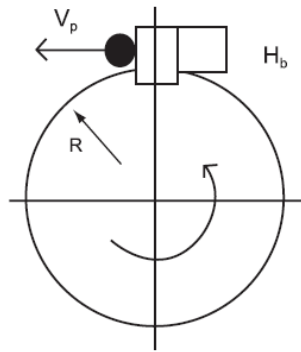


Fig.4 A Single particle just after with the rotor bar of a hammer crusher [1]

Considering the conservation of linear momentum, before and after the impact the energy/ mass is given as

$$E = 0.5 (R + 0.5H_b)^2 \cdot \omega^2$$

Where,  $R$  = Rotor radius

$H_b$  = height of impact surface of crushing bar/ hammer.

$\omega$  = rotor angular velocity

It is also found out that the kinetic energy is a dominant form of energy in an impact crusher. The amt. of specific kinetic energy (KW h/T) is found to be a function of the particle size and the rotational speeds of the rotors.

The intensity of dynamic stress induced by the rotor and by the impact into the fixed surface [2] i.e. the breaking bars/wall can be calculated as

$$S = \rho V_p V_{pp}$$

Where  $S$  = dynamic stress (Pa)

$\rho$  = density of the rock

$V_p$  = propagation velocity of the longitudinal stress wave

$V_{pp}$  = peak particle velocity = impact velocity =  $V_i$

We have  $V_i = \omega d$

It was also found out that the mean diameter of the fragment produced by the impact [2]

$$D = \left( \frac{4.472 \times Kl_c \times L}{0.133 \times \rho \times V_p \times \pi \omega d} \right)^{(2/3)}$$

Where  $\omega$  = rotational speed (rpm)

$Kl_c$  = fracture toughness of rock (Pa m<sup>0.5</sup>)

$P$  = density of rock (kg/m<sup>3</sup>)

$V_p$  = propagation velocity of longitudinal elastic wave (m/s)

$L$  = dimension of the rock sample (m)

## KINETICS OF HAMMER ROTATION

Some other performance parameters of the impact crusher are judged as:

- Fineness of the crushing
- Life of hammer

The average life of a hammer in an impact crusher depends on the kind of operation it is being used, the hardness of the material of the hammer, the usage of the crusher, depth of penetration of material into the hammer faces and the kinetics of the hammer rotation. A hammer is found to have an avg. life of around 50- 60 hours. [3]

When a lump of limestone falling through the feeding zone of the crushers reaches point  $a_1$ , it enters the impact zone. Central impact is considered to be most effective (As shown in fig. 5) but it can only come about provided the hammers in the second row travel through a distance  $S$ .

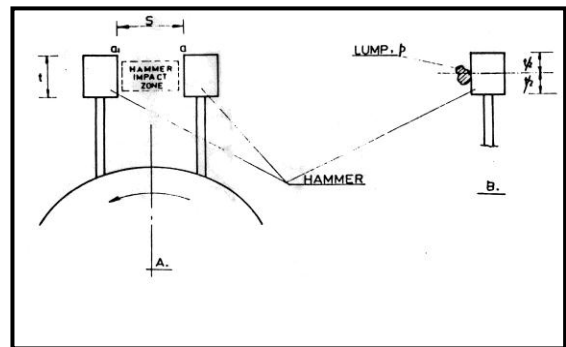


Fig. 5: Mechanism of crushing by impact in a hammer crusher [3]

In other words, the velocity  $V$  of lump  $P$  at point  $a_1$  must be equal to  $nzt/120$  where  $n$  is the rotational speed of the rotor in r.p.m. &  $z$  is the number of hammers/bars in a radial row and  $t$  is the length of the working face of a hammer/bar.

The crushing effect does not depend solely on kinetic energy of hammer ( $\frac{1}{2} MV^2$ ), where  $M$  is the mass of hammer and  $V$  is the peripheral velocity of rotor. This depends on the interchange of energy between hammer and particle or the loss of energy due to impact. Based on dynamics of non-elastic collision and the fact that “momentum of the system at the first moment of maximum deformation remains unchanged” we have

$$MV = (M + m) U \quad \dots \quad \dots \quad (1)$$

$$U = (MV / (M+m)) \quad \dots \quad \dots \quad (2)$$



Initial kinetic energy of the system before impact is,

$$T_0 = MV^2/2 \quad \dots \quad \dots \quad (3)$$

Where, M = mass of hammer;

m = mass of limestone particle;

V = velocity of hammer;

U = system velocity at the end of impact.

Final kinetic energy of the system is,

$$T = \frac{1}{2} (M+m) U^2$$

Hence we can write

$$T = \frac{M^2 v^2}{2(M+m)} \quad \dots \quad \dots \quad (4)$$

Hence, crushing effect is the amount of kinetic energy lost due to impact and is given by,[3]

$$D_m = T - T_0 = \left(\frac{MV^2}{2}\right) \left(\frac{M}{M+m}\right) \quad \dots \quad \dots \quad (5)$$

This shows that greater weight of hammers beyond an ideal wt. does not improve crushing effect.

A better depth of penetration is achieved when the collision vector passes through the particle's center and is also normal to the face of the hammer.

The depth of penetration was found out as [3]

$$C = \frac{54}{nz} \times \left\{ 1 - \frac{z(B+d)}{2\pi R} \right\} \times \sqrt{2gh} \quad \dots \dots \quad (6)$$

Where d= particle dia. ; h= height of fall ; z=no of rows of hammer

n= rotor speed ; B= Length of working face along radius

The optimum value of C is found as  $C \leq t + 0.5d$

## **LEARNINGS FROM THE LITERATURE REVIEW**

It was also found out that the particle entering into the breakage process procures continuous breakage until it fails the classification function for breakage. Hence larger the parent particle the larger is the number of breakage process [1]. Due to the dynamic nature impact breaking it was found that the classification function depends on the crusher design parameters (shape parameter and impact energy) and feed rate and also on the material strength parameters. The performance model is able to predict the product size distribution with reasonable accuracy even when important variations in both the rotor velocity and feed are imposed. The specific impact energy for a Horizontal shaft crusher is very less than that for a vertical shaft crusher [2]. It was also found out that no other force acts on the particle during its free fly from the rotor hammer impact to the wall impact. It was also found out that the kinetic energy is the dominant form of energy. The depth of penetration can be increased by decreasing rotor speed or increasing the height of fall. For effective crushing the velocity of free fall of the lump should be sufficient to reach the middle of head of hammer or the impact zone. The particles with a smaller grain size have higher strength [2]. From the kinetics of the hammer/ blow bar rotation it was found out that reducing the number of blow bars on the rotor not only reduces the total weight and cost by also provides enough spaces between the two hammers so that the portion of material admitted to each row of blow bars encounters a crushing surface equal in size to a continuous bed over the entire width of the rotor and consequently a larger surface than that of the original arrangement by the magnitude of the gaps between the hammer/blow bar heads will be available. We can now easily calculate the ideal number of hammers. Also for the size of the material required we can find out the optimum speed of rotation of the rotor.

## CHAPTER 3

### DESIGN

Designing a horizontal shaft impact crusher for materials like asbestos/ aluminum ore/ clay wet/ cryolite/ lime stone / dry sand (say  $\rho = 1600 \text{ Kg/m}^3$ ) with a feed rate of about 350 mTPH and the top feed size as 1000 mm.

#### **Design of Hammer / Blow bars**

The hammers or the blow bars are subject to shear force at the point of fixation, centrifugal force due to rotation, bending force due to striking of the material.

When a sudden impact is observed by the blow bars due to input feed striking over , it experiences an impact load. The effect of impact loads differs appreciably from that of the static loads as with a suddenly applied load, both the magnitude of the stresses produced and resistance properties of materials are affected.

Hammers or blow bars can be made using different sections like, I section, T section, S section, cylindrical bars, rectangular bars etc. The shape of the hammers decides the impacting capacity as well as the strength of the crusher [9]. Hammers are mounted of the rotor plates or rotor drum using lock pin mechanism.

Let us consider a hammer or the blow bar made of Manganese steel and having a rectangular cross section.

Length of bar = 1500 mm ; Width of bar = 400 mm ; Thickness of bar = 114 mm

Material = Manganese steel ; Density  $\rho = 7.8 \text{ g/cm}^3$

Young's Modulus  $E = 165 \text{ GPa} = 165 \times 10^3 \text{ N/mm}^2$ ; Yield Stress  $\sigma_{ys} = 350 \text{ MPa} = 350 \text{ N/mm}^2$

Height of fall of material  $h = 36 \text{ inch} = 914.4 \text{ mm}$  ; Wt. of each hammer/ blow bar = 477 Kg

The hammer is considered to act like a cantilevered beam with 1/3 of its width inserted in to rotor plate slots for the fixation purpose.

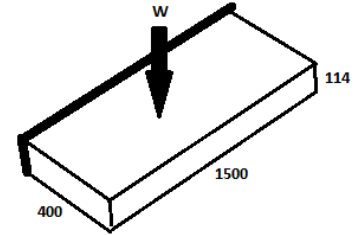
## IMPACT BENDING STRESS (STATIC)

(a) When the cantilever is subjected to a concentrated load at the mid of its span.

Total open screen area per hammer

= 67% of area of the hammer plate

$$= (67 \times 1500 \times 400) / 100 = 4.02 \times 10^5 \text{ mm}^2$$



Now from a feed rate of 350 TPH and a revolution of 480 RPM of the rotor we have 8 impacts by 4 rotors in one second. i.e. 1 rotor has 2 impacts.

$$\text{So Tonnage / impact } W = \frac{350 \times 10^3 \times 9.8}{3600 \times 8} = 119.21 \text{ N}$$

Let  $y$  be the bending

Applying impact equation [5] we get

$$W(h + y) = Py/2 \quad \text{Where P is the equivalent static force}$$

$$\therefore 119.21 \times (914.4 + y) = PY/2$$

Also for a cantilevered beam subjected to a load the deflection [5] is given as

$$y = Pl^3 \frac{1}{3EI} \quad \text{Where I is the moment of inertia}$$

$$\therefore P = \frac{3EIY}{l^3}$$

$$\therefore 119.21 \times (914.4 + y) = \frac{3EIy^2}{2l^3}$$

$$\text{here } I = \frac{bd^3}{12} = 1500 \times \frac{114^3}{12} = 1.85 \times 10^8 \text{ mm}^4$$

$$\therefore EI = 3.05 \times 10^{13} \text{ Nmm}^2$$

So we get

$$119.21 \times (914.4 + y) = \frac{3 \times 3.05 \times 10^{13} \times y^2}{2 \times 1500^3}$$

$$\rightarrow y = 2.8 \text{ mm} \quad (\text{deflection})$$

$$\therefore P = \frac{3EIY}{l^3} = \frac{3 \times 3.05 \times 10^{13} \times 2.8}{1500^3} = 75911.11 \text{ N}$$

$$\text{so Max Moment, } M_{max} = 75911.11 \times \frac{400}{2} = 1.518 \times 10^7 \text{ N mm}$$

Now we have allowable stress

$$\sigma_{ys} = 500 \text{ MPa} = 500 \text{ N/mm}^2$$

So max allowable moment

$$M_{all} = \sigma_{ys} \times Z = \sigma \times \frac{I}{\frac{d}{2}} = 500 \times 1500 \times 114 \times 114 \times \frac{2}{12} = 1.625 \times 10^9 \text{ Nmm}$$

Since  $M_{all} > M_{max}$  .... The design is safe for this condition.

**(b) When the cantilever blow bar is subjected to a concentrated load at the tip of the cantilever.**

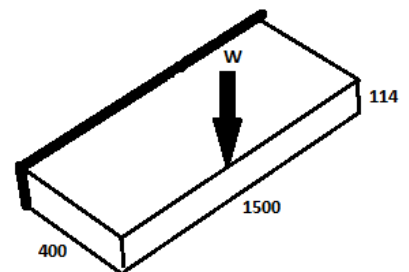
We have

$$W(h + y) = Py/2$$

$$\therefore 119.21 \times (914.4 + y) = Py/2$$

$$\text{where } P = \frac{3Ely}{l^3}$$

$$\therefore W \times (h + y) = \frac{3Ely^2}{2l^3}$$



$$\therefore 119.21 \times (914.4 + y) = \frac{3 \times 3.05 \times 10^{13} \times y^2}{2 \times 1500^3}$$

$$\rightarrow y = 2.8 \text{ mm} \quad (\text{Deflection})$$

$$\therefore P = \frac{3EIY}{l^3} = \frac{3 \times 3.05 \times 10^{13} \times 2.8}{1500^3} = 75911.11 \text{ N}$$

Max. Moment  $M_{\max} = Px l = 3.036 \times 10^7 \text{ N mm}$

Max. Allowable moment  $M_{\text{all}} = \sigma Z = 1.625 \times 10^9 \text{ N mm}$

Since  $M_{\max} < M_{\text{all}}$  hence the design is safe.

**(c) Impact bending stress due to cantilever beam subjected to uniformly distributed load.**

Total tonnage/ hammer/ impact = 119.21 N

Length of exposed blow bar  $l = 400 \times (2/3) = 267 \text{ mm}$

Height of fall  $h = 36 \text{ inch} = 914.4 \text{ mm}$

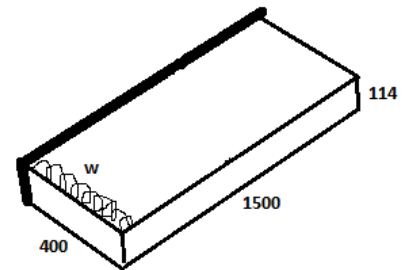
$W = 119.21 \text{ N}$

Since the weight is distributed uniformly over the length  $l = 267 \text{ mm}$

$$\text{We have } W(h + y) = Py/2$$

The Bending moment at any section X from the fixed is given as [5]

$$M = EI \times \frac{d^2y}{dx^2} = -\frac{W}{2} (l - x)^2$$



Integrating we get

$$EI \times y = -\frac{W}{24}(l-x)^4 - \frac{Wl^3x}{6} + C_1$$

At  $x=0, y=0$   $C_1 = wl^2/24$

$$\therefore y = -\frac{W}{24EI}(l-x)^4 - \frac{Wl^3x}{6EI} + \frac{Wl^4}{24EI}$$

Small work done due to impact distributed load  $w = W(h+y)/l dx$

So the total work done becomes  $= \int_0^l \frac{W(h+y)}{l} dx$

$$\begin{aligned} &= \frac{W}{l} \int_0^l \left( h - \frac{W(l-x)^4}{24EI} - \frac{Wl^3x}{6EI} + \frac{Wl^4}{24EI} \right) dx \\ &= \frac{W}{l} \left\{ hl + \left[ \frac{W(l-x)^5}{120EI} \right]_0^l - \left[ \frac{W(l)^3x^2}{12EI} \right]_0^l + \left[ \frac{W(l)^4x}{24EI} \right]_0^l \right\} \\ &= \frac{W}{l} \left( hl - \frac{wl^5}{20EI} \right) \\ &= \frac{119.21}{267} \left( 914.4 \times 267 - \frac{119.21 \times 267^5}{20 \times 3.05 \times 10^{13}} \right) = 109059 \text{ N mm} \end{aligned}$$

Also Static Work done  $= \int P \frac{y}{2} dx$

$$\begin{aligned} &= \int_0^l \frac{P}{2} \left[ -\frac{W(l-x)^4}{24EI} - \frac{Wl^3x}{6EI} + \frac{Wl^4}{24EI} \right] dx \\ &= \frac{P}{2} \left\{ \left[ \frac{W(l-x)^5}{120EI} \right]_0^l - \left[ \frac{W(l)^3x^2}{12EI} \right]_0^l + \left[ \frac{W(l)^4x}{24EI} \right]_0^l \right\} \\ &= \left( -\frac{P}{40} \frac{wl^5}{EI} \right) \end{aligned}$$

So we have

$$\int \frac{w}{l} (h + y) dx = \int \frac{Py}{2} dx$$
$$\therefore 109059 = \frac{-P \times 119.21 \times 267^5}{40 \times 3.05 \times 10^{13}}$$

So we get  $P = 822531.13 \text{ N}$

\* Max Moment,  $M_{\max} = Pl/2 = 109807906 \text{ N mm} = 1.098 \times 10^8 \text{ N mm}$

Max Stress Induced,  $\sigma_b = M/Z = 2M/Id = \frac{1.098 \times 10^8}{1.85 \times 10^8} \left( \frac{114}{2} \right) = 33.830 \text{ N/mm}^2$

But max allowable stress  $M_{\text{allowable}} = 500 \text{ N/mm}^2$

So the design is safe in accordance to this condition too.



## STATIC LOAD SHEARING

By using strain energy method [5] and approximating the loading to be a static one,

Shear stress produced due to force F at any distance y is

$$\tau = \frac{FAy}{Ib} = \frac{F \times b}{I \times b} \left(\frac{d}{2} - y\right) \left(\frac{d}{4} + \frac{y}{2}\right)$$

$$\tau = \frac{6F}{bd^3} \left(\frac{d^2}{4} - y^2\right)$$

Shear strain energy for the small volume

$$dv = \frac{\tau^2}{2G} (\text{volume})$$

$$= \frac{1}{2G} \left[ \frac{6F}{bd^3} \left(\frac{d^2}{4} - y^2\right) \right]^2 \times (bdy) \times (dx)$$

So the total strain energy =  $\int_0^L \int_0^{d/2} dv$

$$\begin{aligned} &= \frac{1}{2G} \int_0^L \int_0^{d/2} \left[ \frac{6F}{bd^3} \left(\frac{d^2}{4} - y^2\right) \right]^2 \times (bdy) \times (dx) \\ &= \frac{1}{2G} \times \left(\frac{6F}{bd^3}\right)^2 b \times \int_0^L \left[ \frac{d^4 y}{16} + \frac{y^5}{5} - \frac{d^2 y^3}{6} \right]^{d/2} dx = \frac{3F^2 L}{10bGd} \end{aligned}$$

Work done  $W = F \times \frac{y_s}{2}$  where  $y_s$  = displacement

So we get  $y_s = \frac{6PL}{10bGd}$

Here  $P = 119.21 \text{ N}$ ;  $G = \text{bulk modulus} = 80 \text{ GPa} = 80 \times 10^3 \text{ N/mm}^2$

$$\text{So } y_s = \frac{6 \times 119.21 \times 267}{10 \times 1500 \times 114 \times 80 \times 10^3} = 0.0000031 \text{ mm}$$

## DESIGN OF V-BELT DRIVE

A V- belt drive mechanism drives the rotor.

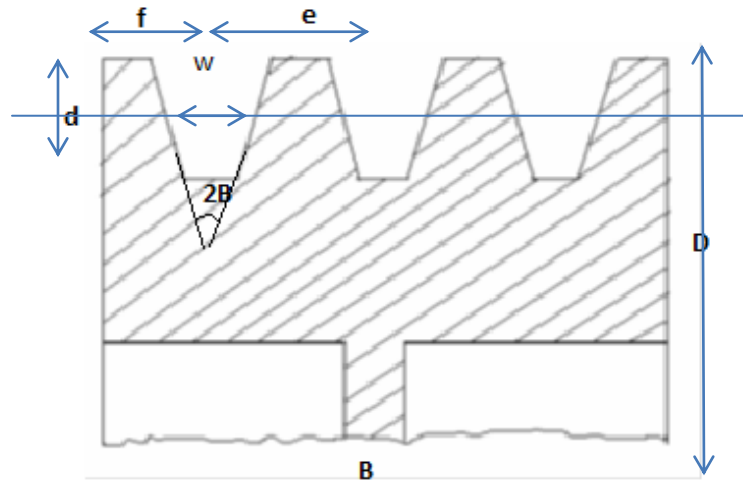


Fig. 6: Cross section of V grooved Pulley [pp 728; *A text book of machine design*, R S Khurmi & gupta, 2005]

Power to be transmitted = 450 Hp = 335 KW (calculated from the crushing requirement and its drive power required)

So according to the V belt standards [Khurmi R S, Gupta, *V-belt and rope drives, A text book of machine design*, 2005]

Minimum pitch dia. D of pulley = 500 mm

Pulley dia. at sheave  $d_2 = 300$  mm

Top width of v belt,  $b = 38$  mm

Thickness of v - belt,  $t = 23$  mm

$2\beta = 36^\circ$  (assumed)

For pulley

$$w = 32 \text{ mm} ; d = 33 \text{ mm} ; a = 9.6 \text{ mm}$$

$$c = 23.4 \text{ mm} ; f = 29 \text{ mm} ; e = 44.5 \text{ mm} ; \text{No. of sheave grooves } (n) = 20$$

$$N_2 = 480 \text{ rpm}$$

For belt:

$$\text{Coeff. of friction} = \mu = 0.25 \text{ (leather)} ; \sigma_{\text{all}} = 7 \text{ N/mm}^2 ; \rho = 1.2 \times 10^3 \text{ Kg/m}^2$$

$$N_1 = 1000 \text{ rpm}$$

$$\text{As we have} \quad N_1/N_2 = d_2/d_1$$

$$\text{So } d_1 = 144 \text{ mm}$$

Let the overhang be,  $x = 1000 \text{ mm}$

So we have

$$\sin \alpha = (r_2 - r_1)/x \quad \therefore \alpha = 22.9^\circ$$

$$\text{Angle of lap on the driving pulley } \Theta = 180^\circ - 2\alpha = 134.2^\circ = 2.34 \text{ rad}$$

$$\text{Mass of belt per length} = \text{area} \times \text{density} = 0.841 \text{ Kg/m}$$

$$\text{Velocity of belt } V = \frac{\pi d_1 N_1}{60} = 37.6 \text{ m/sec}$$

$$\text{Centrifugal tension } T_c = mv^2 = 1193.88 \text{ N}$$

$$\text{Max tension in the belt } T = \sigma \times a = 7 \times 701.5 = 4910.5 \text{ N}$$

$$\text{Tension on the tight side} = T_1 = T - T_c = 3716.6 \text{ N}$$

$$\text{Also we know that } 2.3 \log \frac{T_1}{T_2} = \mu \cdot \theta \cdot \text{Cosec} \beta$$

$$\Rightarrow T_2 = 561.6 \text{ N}$$

## DESIGN OF ANVILS

Anvils are the structures that help in crushing by further impacting with the material thrown by rotor assembly. These structures can be made up of thick plates or beams fixed at one face such that we can change the orientation as well as alignment so as to alter the distance between the rotor and the anvil. This mechanism also helps in changing the angle at which the material impacts on the anvil so as to get the required size and shape of the fragmented particles. A number of such anvils are used to get the fragmentation at different levels and angles.

Considering anvils to be rectangular beam aligned at an angle  $\Theta$  w.r.t the horizontal axis.

Force exerted by incoming particle  $F = m r \omega^2$

Where  $m$  = mass of incoming particle = 25 Kg (assumed Max)

$r$  = radius of rotor = 1633 mm = 1.63 m

$\omega$  = rotor angular velocity =  $2\pi N/60 = 16\pi$

So  $F = 102855$  N

For impact loading we multiply it with a factor of 2.5 hence force acting on the anvil during impact  $P = 2.5F = 257138$  N

Let the dimensions of the anvil be  $1500 \times 2000 \times 50$  mm<sup>3</sup>.

The anvil is made of manganese steel with  $\sigma = 500$  MPa

### (a) When the load is concentrated at the tip of the anvil

Here  $P = 257138$  N

$d = 50$  mm ;  $b = 2000$  mm ;  $l = 1500$  mm

We can see that Bending moment =  $P l \sin\Theta$

So Max Bending Moment =  $M_{\max} = P.l = 385707552$  N mm

Max Allowed bending moment =  $M_{\text{allowed}} = \sigma Z = \sigma(bd^3)/6 = 500 \times (2000 \times 50^3)/6$   
 $= 20.83 \times 10^9$  N mm

Since the allowed bending moment is higher, the design is safe for this type of impacting.

**(b) When the load is uniformly distributed over the anvil**

Here  $P = 257138 \text{ N}$

Load distribution  $p = P/l = 171.425 \text{ N/mm}$

Now the moment in the beam with uniformly distributed load at any point is given as  $M = px^2/2$

Here bending moment will be max when  $x = l = 1500 \text{ mm}$

So maximum bending moment =  $M_{\max} = pl^2/2 = Pl/2$   
 $= 192853500 \text{ N mm}$

But max allowed BM =  $M_{\text{allowed}} = \sigma Z = \sigma(bd^3)/6 = 500 \times (2000 \times 50^3)/6$   
 $= 20.83 \times 10^9 \text{ N mm}$

Here also the allowed bending moment is higher than that of the max bending moment produced. Hence the design is considered safe in this condition too.

## DESIGN OF ROTOR SHAFT (STATIC CONDITIONS)

Material of Shaft = Cast Iron

Density of Cast Iron,  $\rho = 8000 \text{ Kg/m}^3$

Shaft dia.,  $d = 300 \text{ mm}$

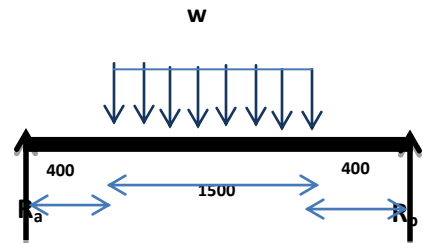


Fig . : Free Body Diag. of rotor shaft

Weights on rotor shafts

Weight of rotor plates = 12600 Kg

Weight of Rotor hammers = 4 X 477 Kg

Volume of shaft =  $\pi/4 d^2 l = \pi/4 \times 300^2 \times 2300 \text{ mm}^3 = 0.000162 \text{ m}^3$

So self-weight of rotor shaft = volume X density = 1300 Kg

Now in for the two shaft mounting points A and B... the reaction forces have the relation

$$R_A + R_B = \{1300 + 12600 + (4 \times 477)\} \times 9.8$$

As  $\Sigma M_b = 0$ .... So

$$R_A \times 2300 - \{1300 \times 2300/2 \times 9.81\} - \{9.672 \times 1500 \times 2300/2 \times 9.81\} = 0$$

$$\Rightarrow R_A = 77538.25\text{N}$$

$$\Rightarrow R_B = 77538.25\text{N}$$

We can see that, since it is a completely symmetric figure. The bending moment will be max at the center of the shaft.

Hence max. Bending moment

$$M_{max} = R_A \times \frac{2300}{2} - \frac{9.672 \times \left(\frac{2300}{2} - 400\right)^2 \times 9.81}{2} - \frac{1300 \times 9.81 \times \left(\frac{2300}{2}\right)^2}{2300 \times 2}$$

$$= 82.782 \times 10^6 \text{ N mm}$$

Now allowable bending moment  $M = \sigma Z = \sigma \pi d^3 / 32 = \frac{170 \times \pi \times 300^3}{32} = 450.4 \times 10^6 \text{ N mm}$

Hence we can see that the design is safe.

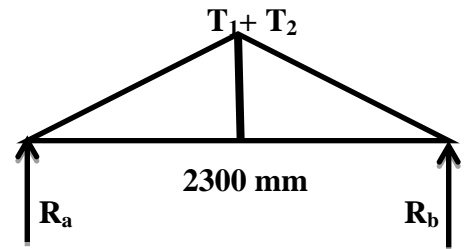
Now considering the bending moment due to tension on both sides of belt we get

$$T_1 + T_2 = 4278 = R_1 + R_2$$

$$\text{also } R_a \times 2300 = 4278 \times (2300/2)$$

$$\Rightarrow R_a = 2136 \text{ N}$$

$$\Rightarrow R_b = 2136 \text{ N}$$



$$\text{Max Moment} = R_a \times l/2 = 2456400 \text{ N mm}$$

So bending moment due to action of load on shaft as well as tension from belt

$$M = \sqrt{M_1^2 + M_2^2} = \sqrt{2456400^2 + 82782000^2} = 82.81 \times 10^6 \text{ N mm}$$

Now turning moment acting  $T = p/\omega = 53.34 \times 10^6 \text{ N mm}$

$$\text{Thus equivalent } M_e = 0.5(M + (M^2 + T^2)^{0.5}) = 90.7 \times 10^6 \text{ N mm}$$

$$\text{And equivalent } T_e = (M^2 + T^2)^{0.5} = 98.58 \times 10^6 \text{ N mm}$$

So stress induced  $= \tau_s = T_e / Z = 37.21 \text{ N/mm}^2$  Hence the design is safe when compared to the ultimate stress. With FOS =  $276 \text{ Mpa} / 37.21 \text{ Mpa} = 7.4$

## DESIGN OF GRINDING SCREEN

These screens are fitted below such that they help in segregating the output material according to their sizes and channel the outflow of the required size particles [4]. Once the particles are on the Screen there are 2 process that occurs on it

Stratification: - here the large sized particles rise to the top of the vibrating material bed due to the vibrating motion effect.

Factors that affect the stratification are material travel flow, bed thickness, screen slope, stroke characteristics like amplitude, frequency, rotation etc. and also the surface moisture.

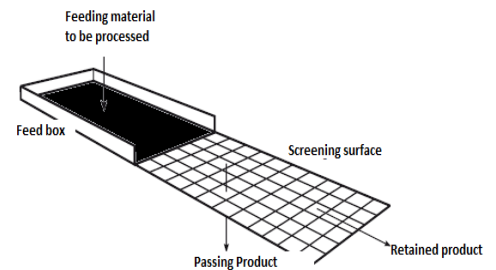


Fig. 7: Grinding screen process [4]

The vibrating motion is generally produced by the vibrating mechanism based on eccentric masses with amplitude of 1.5 to 5mm and operation in range of 800 to 900 rpm.[4]

It should have proper amplitude and frequency so that the material while travelling on the screen neither falls on the same opening nor jump over many subsequent openings.

So if we have larger openings we require higher amplitude and lower speed. But in case of smaller openings we require lower amplitude and higher speed.

The screens can be horizontal as well as inclined. In horizontal screen the motion/vibration should be capable of conveying the material without the need of gravity. So a straight line motion / vibration at an angle of  $45^\circ$  to the horizontal can produce lifting component for the stratification and conveying.



Low screening efficiency leads to over load of the closed crushing circuit as well as it may lead to products that are non-compliment with specification.

It was also studied that the efficiency of screen depends on the feed on the screen. As in the earlier lower feed the efficiency increases with increase in feed but later on the efficiency decreases with further increase in the feed rate. The mesh openings should always be slightly larger than the specified separation size.

For our assumed input feed for the screen, feed rate = 350 tph

Solid density =  $1.6 \text{ t/m}^3$

Max feed size = 100 mm

Moisture content = 3%

Particle shape = flaky; Screening process = dry

Desired products = larger than 60 mm (that are circulated back to crusher input) ; and b/w 60 mm and 40 mm.

#### **Screen selection:-**

85% of the passing material is collected in the first deck of the screen. Since the passing percentage for the deck is very high we use multislope screen. The flaky material shape leads to the choice of square opening screen [4].

We assume the use of steel screening mesh [4].

## Dimensioning

At the first deck particles with size greater than 600 mm should be retained and the rest should be passed to the conveyor.

To obtain 60 mm separation the square opening screen must be 75 mm and with an opening of 73% [Jarmo Eloranta, *Crushing and Screening Handbook*, Kirjapaino Hermes, Tampere, sept 2006, sc 4-1 4-15]

$$\text{Area} = (Q_{\text{feed}} \times P) / Q_{\text{deck}}$$

$$Q_{\text{feed}} = 350 \times 0.75 = 298 \text{ TPH}$$

$$Q_{\text{deck}} = A \times B \times C \times D \times E \times F \times G \times H \times I \times J \times K \times L$$

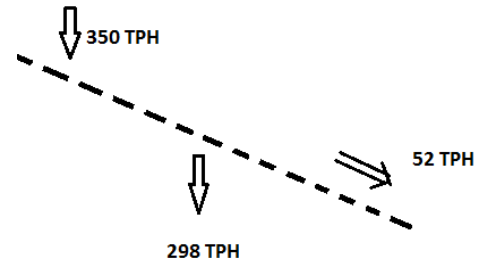


Fig. 8 : Flow chart of material on screen

Where A= capacity factor =75 [ref. 13]

B= retained material factor for 15% oversized = 1.45 [ref. 14]

C= Half size factor =1 [ref. 15]

D=1 ; E=1 (dry screening) ; F=0.6; H=1 ; J= 1; K=1.3; L=1, I=0.9 [ref. 17]

G= 1.46 [ref. 16]

So Area = 262/111.46 = 2.35 m<sup>2</sup>

$$\text{Layer thickness } D = \frac{\text{Feed}}{b_d \times w \times s \times 3600}$$

Where feed = transported capacity ; S= material travel speed

W= screen width (m) ; B<sub>d</sub> = material bulk dist.

Optimal speed s= 30 -35 m/min

Optimal Layer thickness, D= 83-120-163 mm

## CASING

The crusher case can be made up of welded steel construction and built in three or more sections. The lower half is made up of one piece and upper half is made up of two sections. The feed intake section is in the upper half and is bolted to the lower half resulting in a lasting dust type connection between the feed and crusher intake.

The rest of the top section is hinged for access to interior of the crusher for changing hammers, hammer pins and screens. All the mating surfaces are built-up for an accurate, dust tight fit. Single latch door is provided for easy maintenance and cleaning and a Gasket door is provided for dust tight operation. The casing of the crusher does not experiences and larger forces but still they should be able to bear abrasive forces acting on it. The impact bars are attached to the casing through a mechanism which may help in changing the angle of impact on the bar , by moving or tilting the bar.

### RESULT

#### FINAL DESIGN PARAMETERS

Density Of rock / particle  $\rho = 1600 \text{ Kg/m}^3$

Rock/feed Material = Asbestos/ aluminum ore/ clay wet/ cryolite/ limestone/ dry sand

Input feed rate = 350 TPH

Top feed size = 1000 mm

Max speed of rotor rotation  $N = 480 \text{ rpm}$

Power req. from motor = 450 HP

End size of particle = 60 mm

Dia. of rotor = 1500 mm

Width of rotor plate assembly = 1500 mm

No of rotors (plates) used = 9

Rotor material = Manganese steel

Hammer dimension = 1500 X 114 X 400 (mm)

Hammer material = Manganese steel

Density of Manganese steel used = 7.7 g/cu. cm

Weight of rotor plates (total) = 4850 Kg

Weight of hammer (each) = 477kg

Shaft Dia. for rotor = 300 mm

Young's modulus of elasticity for manganese steel  $E = 165 \times 10^3 \text{ N/mm}^2$

Yield stress  $\sigma_{ys} = 350 \text{ N/mm}^2$

Height of fall of material = 36" = 914.4 mm

Total area of hammer/ bar exposed for impact = 67 % of area of Bar surface area

Tonnage/ impact on bars = 119.21 N

Material for rotor shaft = Cast Iron

Diameter of Fly wheel / pulley at end of rotor = 1500 mm

Over hang between the driving and drive pulley = 1000mm

Number of Belts = 2

Pitch length of V- belt = 5.64 m

Dia. of Motor shaft pulley = 144mm

Hole size in square mesh of screen = 75 mm

Grinding screen area = 2.5 m<sup>2</sup>

## **DISCUSSION**

The Rotor hammers were checked for their bending and shear stress and were found within the allowable limits in the maximum load condition. The rotor plate was also checked for shear stress and was found safe. The anvils were checked for bending and shearing strengths and were found under the limits of failures. The rotor shaft was checked for torsion and bending and was found safe. The Driving mechanism of rotor was designed in such a way that the V belt was safe and was able to transmit required speed to the rotor from the motor. An appropriate casing structure is also proposed for housing the crushers' assembly.

PROPOSED DESIGNS

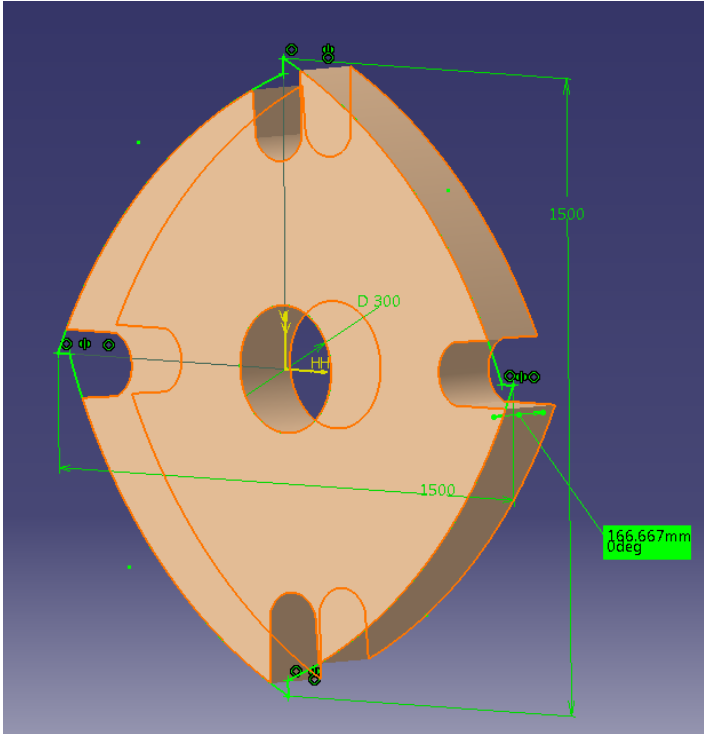


FIG. 9 : Proposed design of Rotor Plate

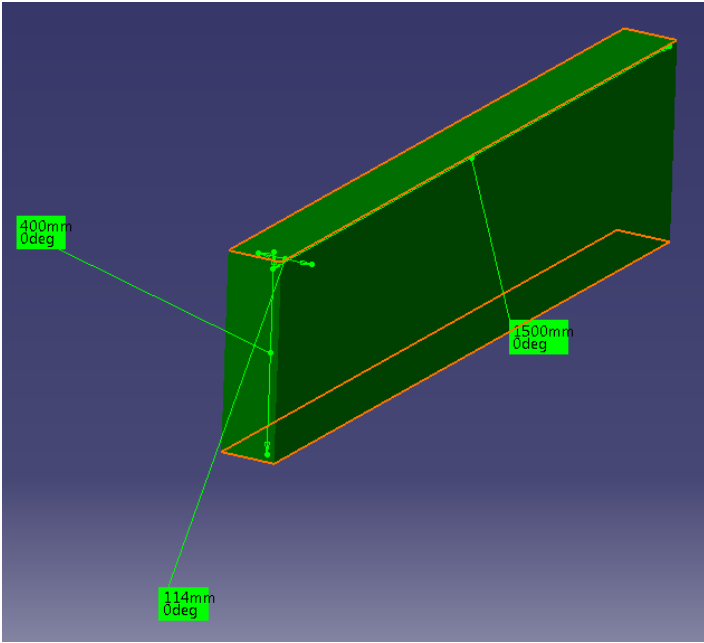
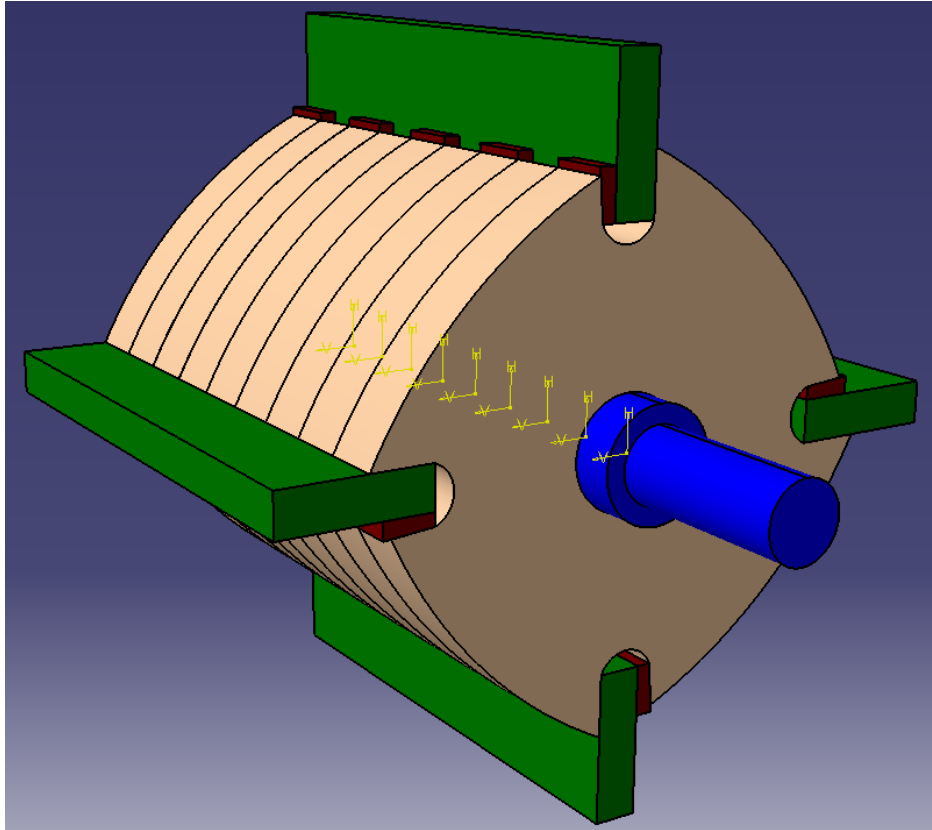
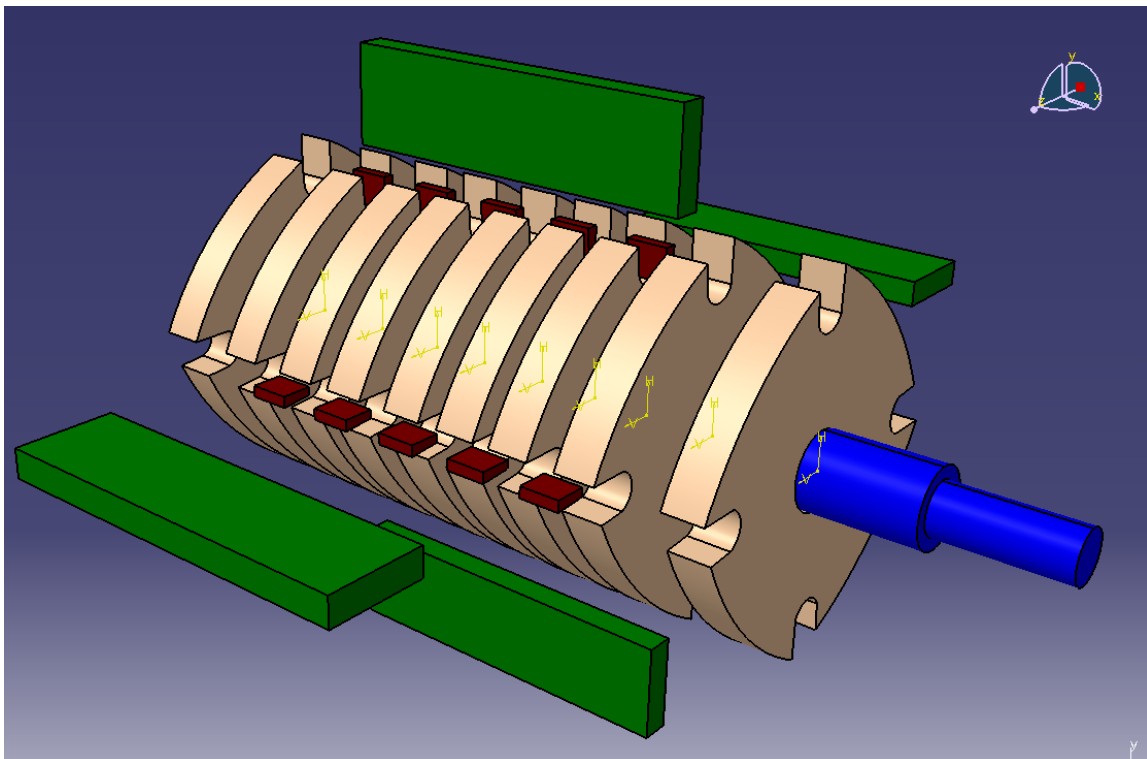


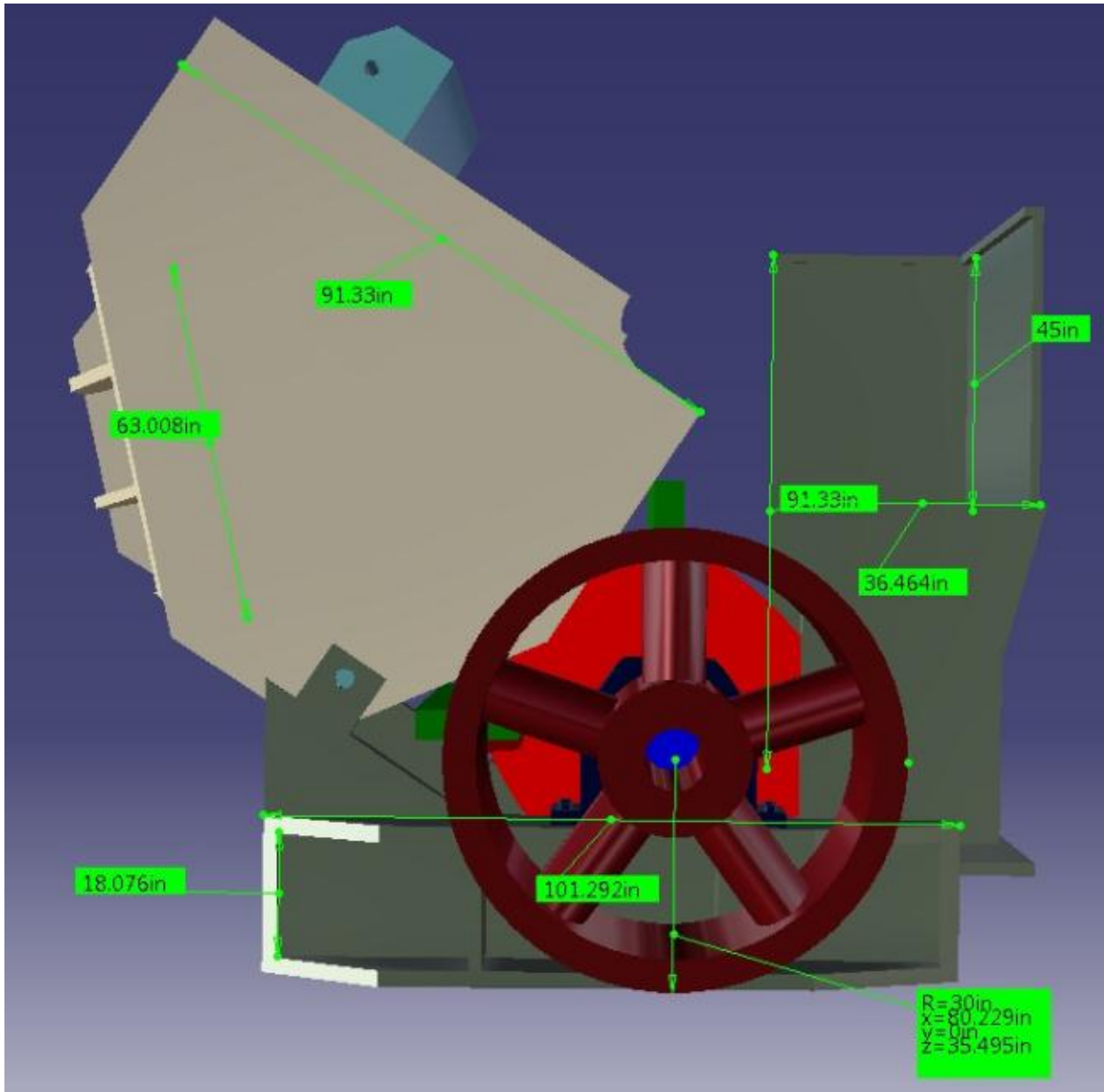
FIG. 10 : Proposed design of Hammer / Blow Bar



**FIG. 11 : Proposed design of Rotor Assembly with rotor discs, hammers, shaft and locking pin**

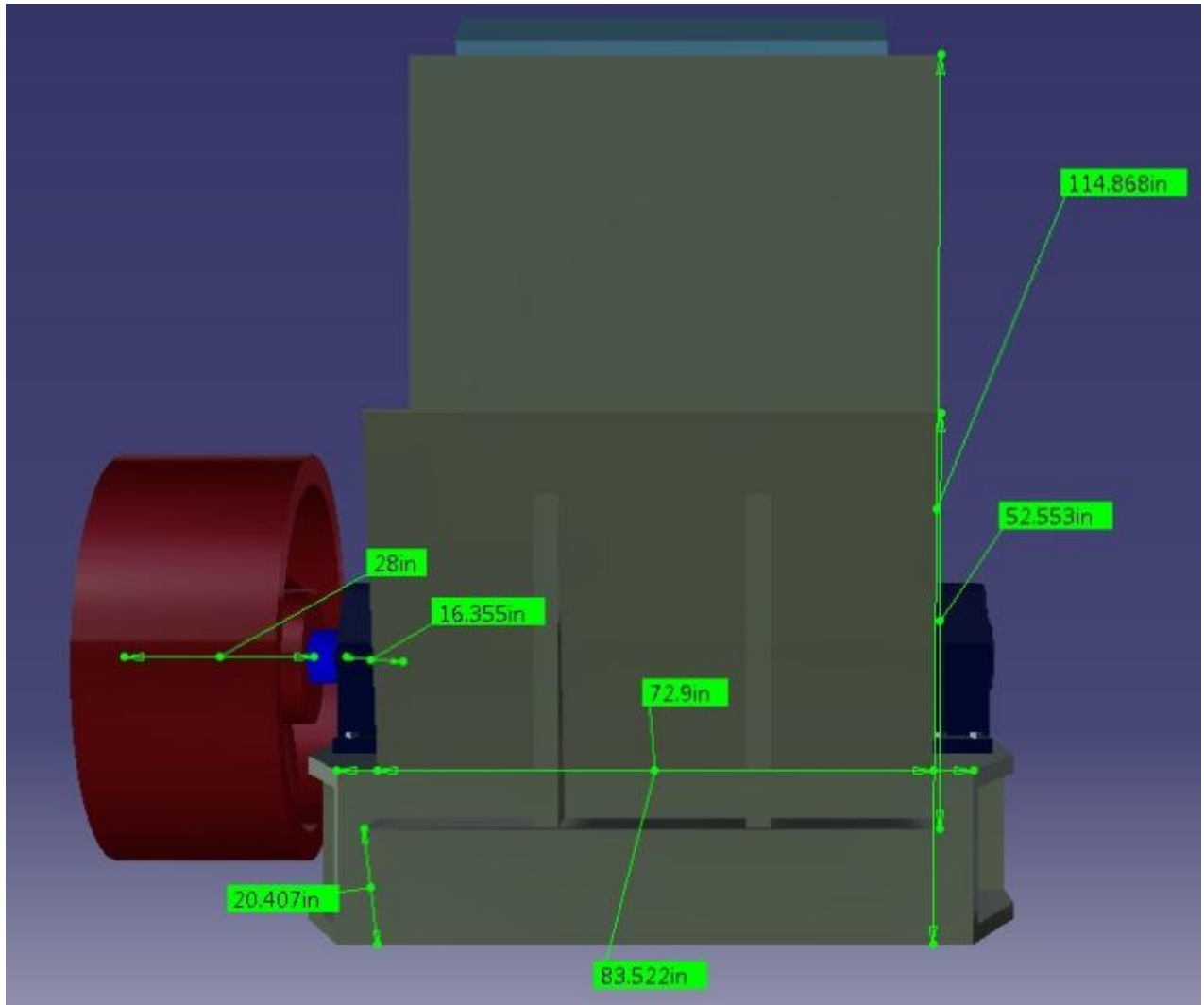


**FIG. 12 : Exploded view of Rotor Assembly with rotor discs, hammers, shaft and locking pin**

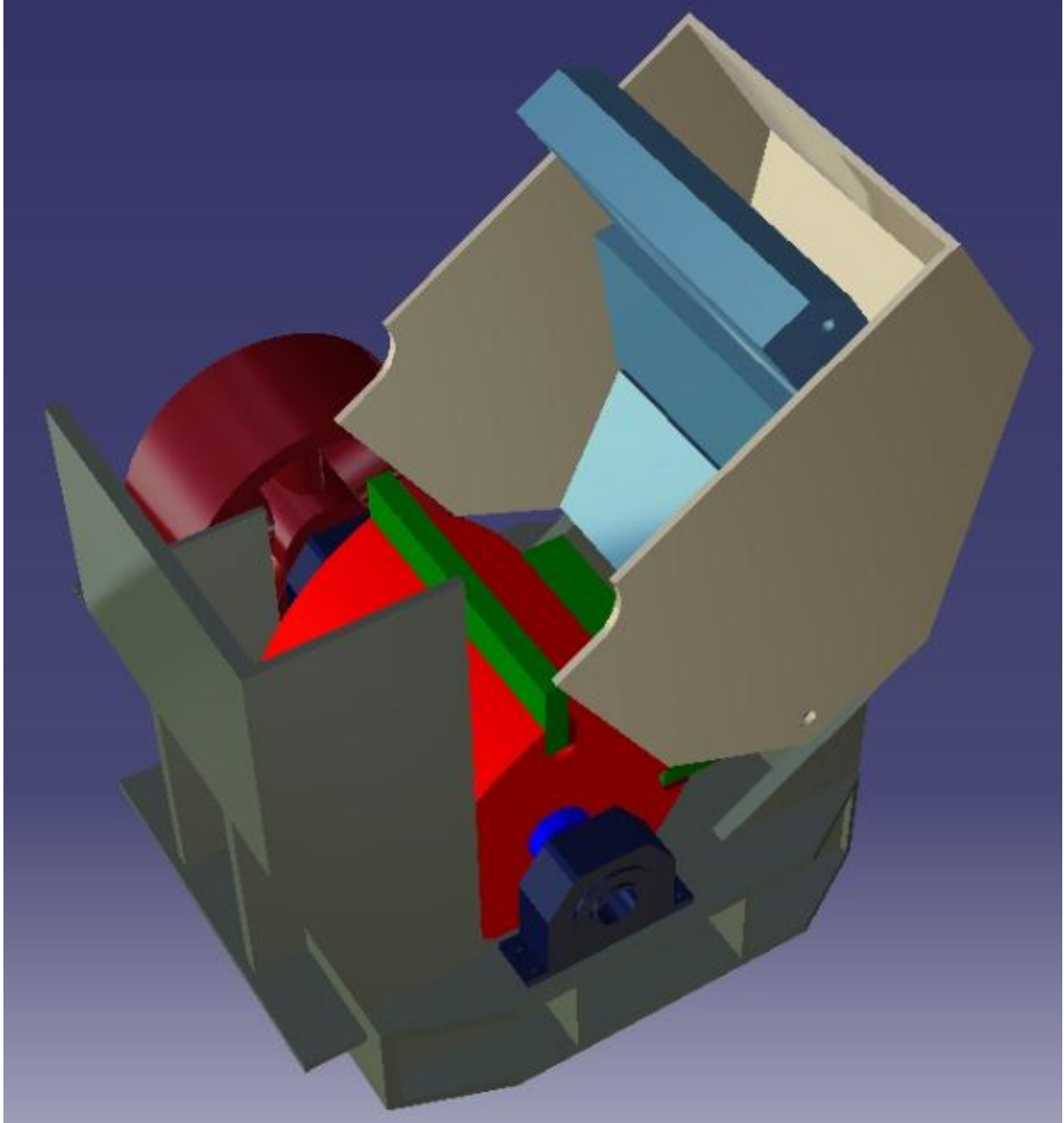


**FIG. 13 : Proposed design of Crusher Assembly with rotor, flywheel, impact bars and casing ( FRONT VIEW)**

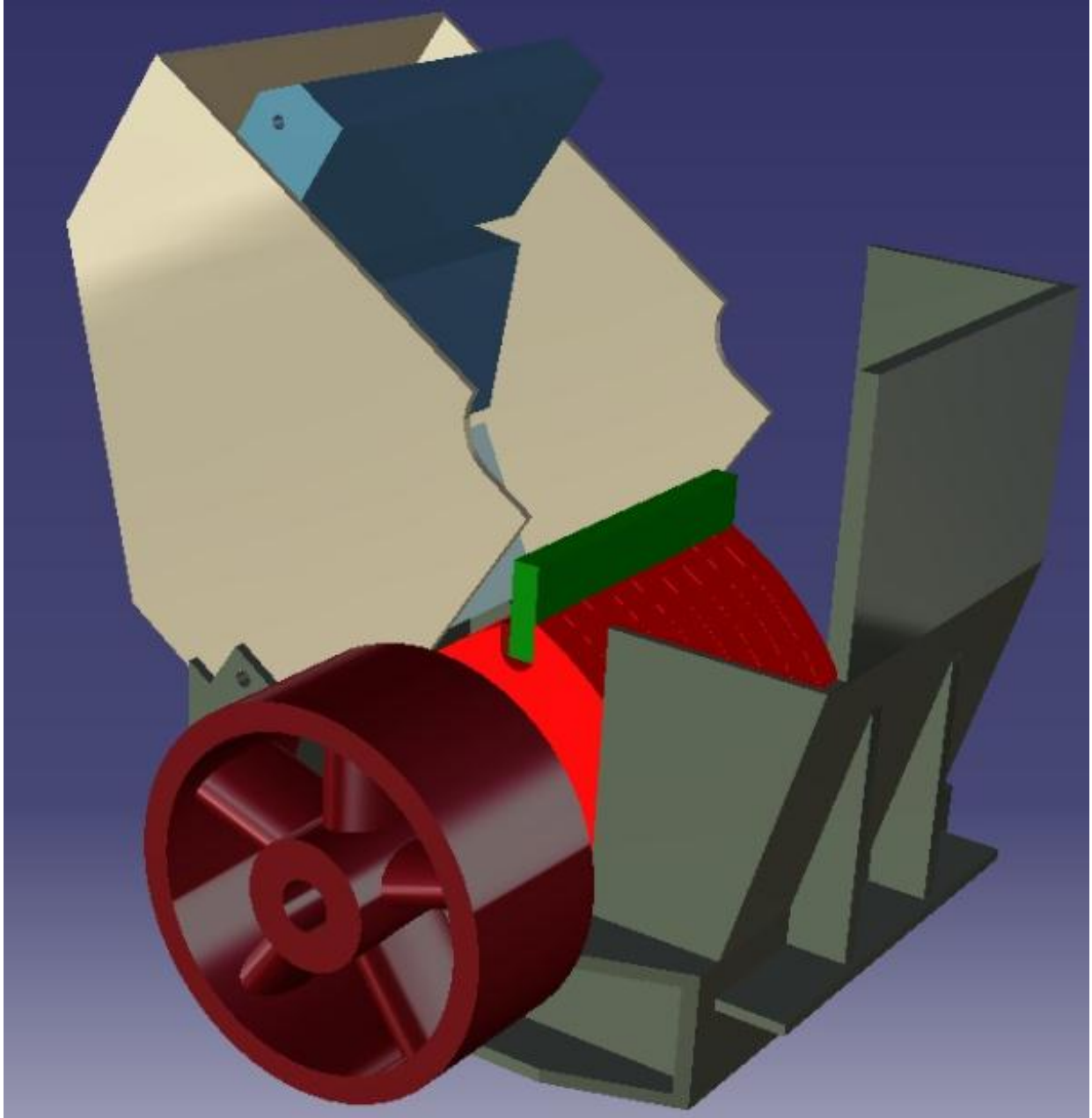




**FIG. 14 : Proposed design of Crusher Assembly with rotor, flywheel, impact bars and casing ( SIDE VIEW)**



**FIG. 15 : Crusher Assembly with rotor, flywheel, impact bars and casing  
(ISOMETRIC VIEW 2)**



**FIG. 16 : Crusher Assembly with rotor, flywheel, impact bars and casing  
(ISOMETRIC VIEW 2)**

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