iJRASET
International Journal For Research in Applied Science and Engineering Technology

INTERNATIONAL JOURNAL FOR RESEARCH
IN APPLIED SCIENCE & ENGINEERING TECHNOLOGY

Volume: 3 **Issue:** X **Month of publication:** October 2015

DOI:

International Journal for Research in Applied Science & Engineering Technology (IJRASET)

Development of a Mathematical Model for Kinematic Analysis of a Crusher Mechanism Using Matlab as a Tool

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Abstract*— Objective of this thesis is to develop a mathematical model for a crusher mechanism, which can calculate Kinematic Parameters like Velocity and Acceleration of any point on Jaw Link of a crusher which in turn is required to calculate torque required to run whole the crushing mechanism to crush a specific bolder from a given size and to a given size of finish product. Torque calculation helps to calculate or select other machine parts' dimensions to assemble the whole machine. It also helps immensely to optimize various machine parts size to make the machine perform optimally at minimum weight.
***Keywords*— Four bar mechanism, Crusher, Kinematic analysis, MATLAB**

I. INTRODUCTION

Jaw crushers are designed to impart an impact on a rock particle placed between an fixed an moving plate (jaw). The faces of the plate are made of hardened steel .Both plates could be flat or the fixed plate flat and the moving plate convex. The surfaces of both plates could be plain or corrugated. The moving plate applies the force of impact on the particle held against the stationary plate. Both plates are bolted on to a heavy block. The moving plate is pivoted at the top end (Blake Crusher) or at the bottom end (Dodge type crusher) and connected to an eccentric shaft. In universal crushers the plates are pivoted in the middle so that both the top and bottom ends can move.

The Blake crushers are single or double toggle drivers. The function of the toggle is to move the pivoted jaw. The retrieving action of the jaw from its furthest end of travel is by springs of small crushers or by a pitman for large crushers. As the reciprocating action removes the moving jaw the broken rock particle slips down but are again caught by the next movement of the swinging jaw and crushed. This process are repeated until the particle size are smaller than the smallest opening between the crusher plate at the bottom of the crushers(the closed set).For a smooth reciprocating action of the moving jaws, heavy fly wheels are used in both type of crushers.

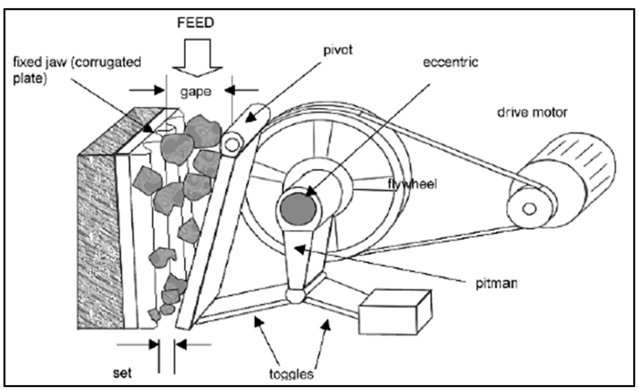

Fig1: Working of a Jaw Crusher.

Fig 1 shows a sketch of Blake crushers operated by double toggles and controlled by a pitman. These are commonly used as primary crushers in the mineral industry. The size of the feed opening is referred to as the gape. The opening at the discharged end of the jaws is referred to as the set

The factors of importance in designing the size of primary crushers, like a jaw crushers are:

Vertical height of crushers = 2 * Gape

Width of jaw > 1.3 * Gape (1)

< 3.0 * Gape (2)

Throw = 0.0502 (Gape)^0.85

where the crusher gape is in meters.

These dimensions vary as individual manufacturers have their own specification and their catalogues are a good guide to the geometry and design of individual makers.

For sizing a crushers and ancillaries for open circuit operations. Eqs (1) and (2) are helpful as an approximation .From the equation it can be seen that once the gape has an assigned value the rest of the dimension follows .To size the gape of the largest particle to be charged is considered and the following relation applied :

Largest particle size = 0.9 * Gape (3)

Jaw Crusher Operation

The ore or rock is feed to the crusher where the jaw are furthest apart i.e at the maximum opening or gape. When the jaws come together the ore is ore is crushed into smaller size and slip down the cavity. In the return stroke further eduction of size is experienced and the ore moves down further. The process is repeated till particle have size less than the bottom opening or set pass through a product.

The rule of thumb applicable for opening a jaw crusher with respect to it's design characteristics can be summerised as follows

Feed size = 0.8 - 0.9 * gape

Reduction ratio, R = 1:4 to 1:7

Throw Lt = 1 – 7 cm

Speed = 100 to 359 rpm

Freequency of stroke,v == 100 – 300 cycles per min

Length of stroke = 0.0502 * gape ^0.085

In practice the operator has to decide on the spacing of the set at the discharge end. This setting has to include the maximum and minimum positions that the bottom has to open during the oscillation of the bottom end of the jaw. The manufacturer of the jaw crusher provides all the control to adjust the settings. The actual are best measured by taking a soft metal, like lead or a ball of aluminum foil and squeezing it between the jaws at the desired width forming a template. This piece of lead metal is used to check the change of setting during operation.

During the operation of a crusher the bulk density of the material increases and the particle size decreases. With time wear on the plate surfaces develop resulting in a change in the crusher surface profile. This could alter the size and the throughput of the crusher product.

The operation of the jaw crusher has been best described by Whiten. According to Whiten if a certain particle size, d in the size distribution curve of a mineral is less than K1 then it would pass through uncrushed, Being smaller than the opening of the set. But all particle size greater than K2 were smaller than the largest opening between the jaw of the jaw crusher and therefore will always be crushed. The probability P of a particle being crushed or not may be written as:

$P(d) = 0 \quad \text{for} \quad d < K1$

$P(d) = 1 \text{ for } d > K2$

$P(d) = 1 - [\frac{K2-d}{K2-K1}]^2 \qquad K1 < d < K2 \qquad (4)$

The values of K1 and K2 are primarily function of the crushed set but also depend on the throughput feed size and linear length and

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are determined statistically by regression analysis of the operating data . Such relationships were initially determined by Lynch [12] and later revised by Anderson [13] as :

K1 = a0 +a1 Lmin – a2Q + a3F80 + a4 Llinear

K2 = b0 ± b1 Lmin + b2Q + b3F80 – b4 tLinear + b5Lt

Where tLinear is the age of the linear in hours and Llinear is the liner length.

Or

K1 = 0.67 Lmin ± 1.956mm

K2 = 1.131 Lmin + 58.67 q + 25.4 T(t) ± 1.8 mm

Where q = the fraction > 25.4 mm in feed and

T(t) = an interpolating spline function of tonnage.

For a single feed , k1 and K2 can be obtained by size analysis. From a alarge number of data the value of exponent of Eq (4) was found to closer to 2.3. This mathematical concept of jaw crusher operation has been developed for modeling and subsequent throughput prediction from jaw crushers.

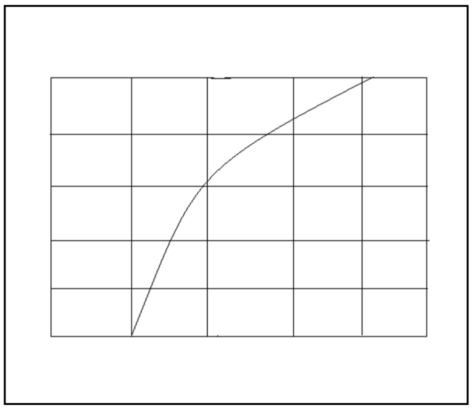

Fig 2: Particle size vs Probability of crushing particles in a Jaw Crusher (from Eq (4))

The capacity of jaw crushers is a measure of the mass and volume of crushed material produced in unit time of operation . The capacity is primarily a function of:

Crusher design characteristics like width and depth of the crushing chamber.

Open and closed side settings

Options on feeding methods eg intermitted feeding,(manual or direct by haulage trucks) and continuous by conveyor belt

Operating charecteristics like length of stroke , the number of strokes per minute, the nip angle

Mathematically the capacity can be expressed by the general formulae:

Q = f (w,L,Lmax,Lmin,Lt,n, θ,K)

Where Q = capacity

w = width

L = height (or depth of jaws).

,Lmax = set maximum (Open Set)

Lmin = set minimum (Closed Set)

Lt = length of stroke or throw

n = frequency(Revolution rpm = cycles per unit time)

θ = jaw angle

K = constant related to machine characteristics.

The mechanism of movement of rocks down the crusher chamber determine the capacity of jaw crushers. The movement can be visualized as a succession of wedges (jaw angles) that reduce the size of particles progressively by compression until the the smaller particle pass through the crushers in a continuous procession. The capacity of a jaw crusher per unit time will therefore depend on the time taken for a particle to be crushed and dropped through each successive wedge until they are discharged through the bottom. The frequency of opening and closing of the jaws therefore exerts a significant action on capacity.

Many works have already been done on different types of crusher till now. But very few works has been done on the mathematical modeling of crusher working principal. It is because the utmost dynamicity in the working of any crusher machine, Due to adverse dynamic condition it is very difficult to establish any well-defined mathematical model of crushing phenomenon to predict all the performance parameter like displacement of any point on crushing link with respect to any pre-defined origin, Velocity of any point on the crushing link, Acceleration of the same, Static and Dynamic torque playing on in various part of different link etc. In the present work an attempt has been made to predict displacement of a point on the rusher link by developing a mathematical model

In this work kinematic as well as dynamic analysis of a crusher mechanism has been performed. For the above mentioned analysis few generalized programs in Matlab have been generated. These programs are to determine-

Position of crusher link for different angle of crank

Coupler curve of any point on the crusher link.

Velocity of any point on the crusher link for an input velocity of crank.

Acceleration of any point on the crusher link.

Dynamic torque required for the mass and configuration of all the links in the crusher mechanism.

Here no specific configurations of the crusher machine have been considered as the analysis and program have been generated parametrically.

To validate the mathematical model an example of crusher mechanism has been considered from reference [7]. In this reference Dr. Modak et al have performed a dynamic analysis of the given crusher mechanism by graphical method. Graphical method is not a recursive method by virtue of which we can make the analysis independent of any particular mechanism configuration.

As per the work done by Dr. Modok et al. in reference [7], they considered a crusher mechanism with following configuration and evaluated the static and dynamic torque for that specific configuration by Graphical Method. Graphical Method is not a generic method and for the change of configuration whole the process has to be done again from the scratch. Moreover to calculate dynamic torque it is required to calculate velocity and acceleration of the link first at the given position of the crank.

In the present work few MATLAB codes have been generated to calculate velocity and acceleration of any point on the Jaw link of the crusher mechanism for any given crank angle. As in reference [7] Dr. Modok et al have not documented the values of Jaw position, its velocity and acceleration with respect to different crank angle the code generated in the present work for calculating position of the Jaw link of the crusher has been compared with the angle calculated by AutoCAD drawing for the purpose of validation.

In the present work the fact which has been intended to establish is that, for the synthesis of any mechanism it is very much required to first do the kinematic and dynamic analysis of the mechanism. Synthesis of any mechanism is not a shot job. Trial and error method type of activity is involved. To rationalize the trial and error type of activity Optimization methods are employed in mechanism design. So it is needed to vary the input parameters for optimization. Therefore graphical method is not at all employable where synthesis of a mechanism is concern. In reference [7] Dr. Modok et al have investigated a crusher mechanism with help of graphical method. In the present work a code has been generated by virtue of which one can do the kinematic and dynamic analysis of a mechanism generically. So an effort has been made to make the kinematic analysis of a mechanism applicable for any type of configuration.

In the present work first a mathematical model have been developed to find out linear as well as angular displacement of any point on the crushing link which is call Jaw as a function of angular position of crank rotation. Then this mathematical model has been converted into a MATLAB program. The output results or parameters from this program for a given set of input parameter (as per reff [7]) have been validated with the output generated by graphical method using AutoCAD.

Next the mathematical model for displacement has been converted into mathematical model for Velocity and Acceleration by differentiating once and twice respectively.

To generate a mathematical model and code for dynamic analysis and synthesis of a mechanism one can use the codes for velocity and acceleration analysis as developed in the present work.

In this chapter a detailed mathematical nitty-gritty of four bar mechanism has been presented and then that basic kinematic model of four bar mechanism has been implemented on jaw crusher mechanism to find out following output parameters out of the following mentioned input parameters.

Output parameters those will be evaluated-

Angle subtend by the Jaw of the crusher with vertical axis with rotation of crank

Variation of velocities of any point on the Jaw with rotation of crank

Variation of acceleration of a point on the Jaw of the crusher with rotation of crank.

Coupler curve of a point on the jaw

Dynamic torque calculation

Required input are-

Length of each link in the crusher mechanism

Angular interval of rotation of the crank

Mass of each link

CG and Radius of Gyration of each link

Moment of inertia of each link.

In this work few Matlab programs have been generated and those have been validated with analytical approach of kinematic analysis of a basic four bar mechanism.

Mathematical modeling for Kinematic analysis of four bar mechanism

A four link mechanism shown in the diagram in Fig 4.1 is in equilibrium. a, b, c, d represents the mechanism of the link AB,BC,CD and DA respectively θ , βand ϕ are the angles of AB,BC,DC respectively with the x axis (taken along AD) . AD is the fixed link, AB is taken as the input link whereas DC as the output link .

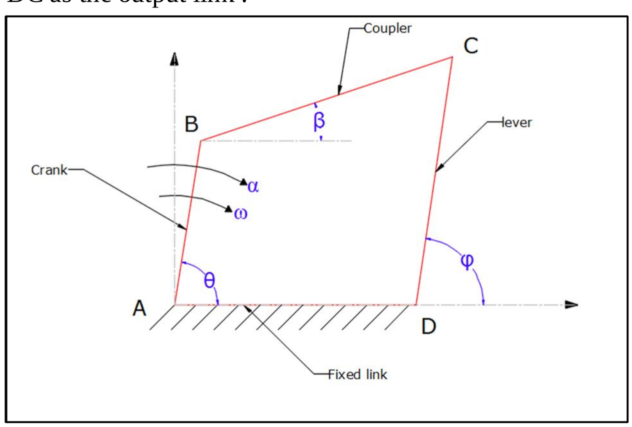

Fig 3: Shematic representation of a four bar mechanism in parametric form

$$\Phi = 2\tan^{-1}\left[\frac{-B \pm \sqrt{B^2 - 4AC}}{2A}\right] \qquad (1)$$

where

A = k – a (d-c) cosθ –cd
B = -2ac sinθ
C = k – a (d+c) cosθ +cd
$a^2 - b^2 + c^2 + d^2 = 2k$

$$\beta = \tan^{-1}\left[\frac{-E \pm \sqrt{E^2 - 4DF}}{2D}\right] \quad (2)$$

Where,
D = k' – a(d+b) cosθ +bd
E = 2ab sinθ
F= k' – a(d-b) cosθ – bd
$a^2 + b^2 - c^2 + d^2 = 2k'$

Velocity Analysis

$$\omega_c = \frac{a\,\omega a \sin(\beta - \theta)}{c sin(\beta - \varphi)} \quad (4)$$

$$\omega_b = \frac{a\,\omega a \sin(\varphi - \theta)}{b sin(\varphi - \beta)} \quad (5)$$

Acceleration analysis

$$\alpha b = \frac{a\alpha \,\mathrm{a}\sin(\varphi - \theta) - a\omega a^2 \cos(\varphi - \theta) - b\omega b^2 \cos(\varphi - \beta) + c\omega c^2}{b sin(\beta - \varphi)} \quad (6)$$

$$\alpha c = \frac{a\alpha \,\mathrm{a}\sin(\beta - \theta) - a\omega a^2 \cos(\beta - \theta) - b\omega b^2 + c\omega c^2 \cos(\beta - \varphi)}{c sin(\beta - \varphi)} \quad (7)$$

Now on the basis of the above mathematical model few MATLAB programs have been generated to calculate Position, Velocity and Acceleration of all the links of a four bar mechanism for different crank angle.

II. KINEMATIC ANALYSIS OF CRUSHER

Above mentioned analytical expressions for displacement, velocity and acceleration are of a four bar chain. Now a mathematical model has been developed in this work which can be used to create a recursive expression for the evaluation of displacement, velocity and acceleration of a point on the crusher link recursively.

In the following figure a crusher mechanism has been represented as a four bar chain. In this mechanism the eccentricity of shaft has been represented by line AB. The Jaw has been represented by the line BC and the toggle link has been represented by CD. AD is fixed. In this case the origin has been placed at lower end of the fixed link. Centre of the eccentricity has been put at the coordinate as per the configuration.

For different position of the input link this mathematical model will calculate position of any point of the crusher and inclination of the crusher link with vertical axis. The program generated value has been verified with the configuration generated by AutoCAD to validate the mathematical modeling.

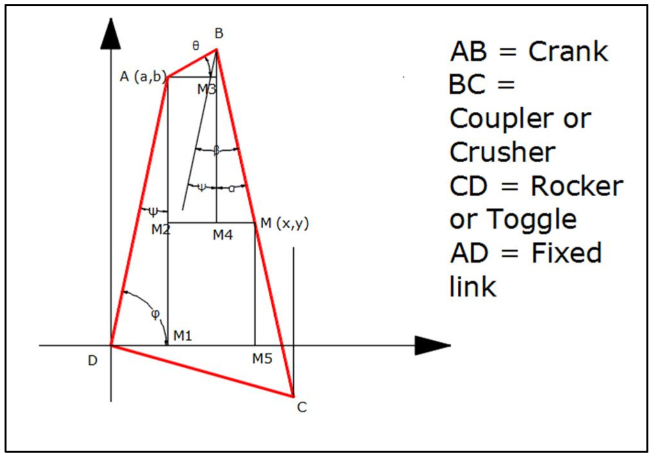

Fig 4: Shematic representation of a crusher mechanism in parametric form

Figure 4.2 above explains the working of a crusher mechanism. Now recursive expression of any point on the crusher link say point M here has been shown below.

$x_i = DM_5 = DM_1 + M_1M_4 + M_4M$

$x_i = a + AB\cos\theta + nCD\sin\alpha$ (8)

$y_i = MM_5 = AM_1 + BM_3 - BM_4$

$y_i = b + AB\sin\theta + nCD\cos\alpha$ (9)

Now from the above expressions few MATLAB program has been generated to evaluate position of any point on the crusher with respect to the origin, Angle of the crusher link with respect to the vertical axis, Graph between horizontal and vertical displacement of any point on the crusher link which is also called coupler curve.

To validate the above mathematical model an example of crusher mechanism has been considered from reference [7]. In this reference Dr. Modak et al have performed a dynamic analysis of the given crusher mechanism by graphical method. Graphical method is not a recursive method by virtue of which we can make the analysis independent of any particular mechanism configuration.

III. RESULT AND DISCUSSION

As per the work done by Dr. Modok et al. in reference [7], they considered a crusher mechanism with following configuration and evaluated the static and dynamic torque for that specific configuration by Graphical Method. Graphical Method is not a generic method and for the change of configuration whole the process has to be done again from the scratch. Moreover to calculate dynamic torque it is required to calculate velocity and acceleration of the link first at the given position of the crank.

In the present work few MATLAB codes have been generated to calculate velocity and acceleration of any point on the Jaw link of the crusher mechanism for any given crank angle. As in reference [7] Dr. Modok et al have not documented the values of Jaw position, its velocity and acceleration with respect to different crank angle the code generated in the present work for calculating position of the Jaw link of the crusher has been compared with the angle calculated by AutoCAD drawing for the purpose of validation.

In their work Dr. Modok et al [7] have considered following configuration of links of a Jaw crusher. The configurations have been presented in the table below.

Table 2: Link configurations from Ref. [7]

Sl No.	Parameters	Magnitude
1	Length of the 'Crank' in the crusher. It is actually the measure of eccentricity of the main shaft.	32 cm
2	Length of the 'Coupler' which is the length of the 'Jaw' link in the crusher.	110 cm
3	Length of 'Liver' that is length of the 'Toggle' link of the crusher	100 cm
4	Length of the fixed link	116 cm
5	Coordinate of the center of eccentricitywith respect to theother end of the fixed link.	58 cm, 100.46 cm

After using the above data in the MATLAB program 4.1 to calculate angle of Coupler link or the Jaw link of crusher with vertical we get the following result.
Following is the result generated by MATLAB Program 4.

Table 3: Result generated by MATLAB program

Crank angle w.r.t Horizontal (in Degree)	Jaw angle w.r.t direction parallel to fixed link (in Degree)	Jaw angle w.r.t direction parallel to Vertical (in Degree)
0	35.00	4.98
30	37.32	7.31
60	42.50	12.49
90	50.03	20.01
120	58.71	28.70
150	66.64	36.63
180	71.18	41.16
210	69.32	39.31
240	60.29	30.28
270	48.78	18.76
300	40.20	10.18
330	35.80	5.79
360	35.00	4.98

Now above result has been validated with the following figures generated in AutoCAD.
In the following figure
AB = Crank which is also the measure of eccentricity = 32 cm as per Reff [7]
BC= Coupler which is the Jaw link of the crusher mechanism = 110 cm as per Reff [7]
CD= Leaver link or the Toggle link of the crusher = 100 cm as per Reff [7]

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IC Value: 13.98 ISSN: 2321-9653
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DA= Fixed link of the crusher mechanism = 116 cm as per Reff [7]

Here point A is the center of eccentricity whose co-ordinate with respect to another fixed end that is end 'D' has been mentioned above and is one of the input parameter. This value as per the work of Reff [7] is (58 cm, 100.46 cm).

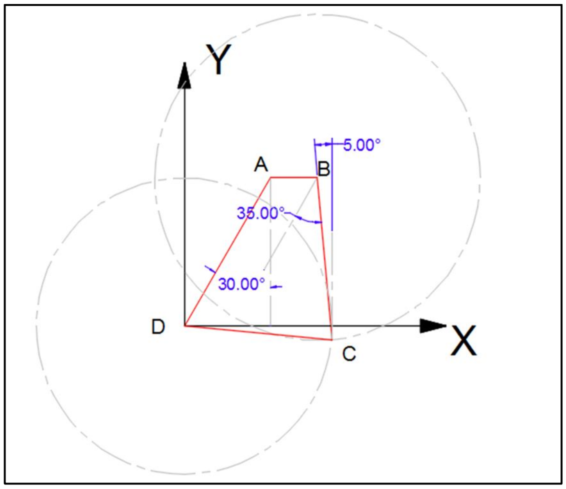

Fig 5: Angular position of Jaw link BC of the crusher with respect to the vertical direction and direction parallel to the fixed link AD when crank is at 0º with respect to the horizontal direction

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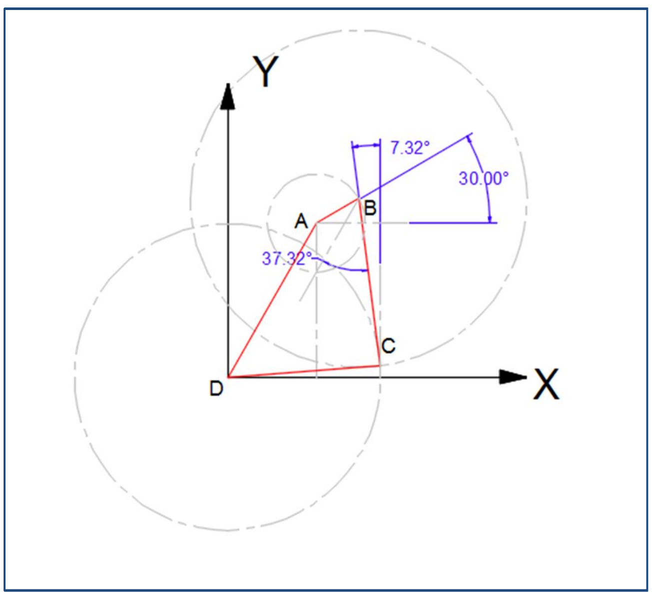

Fig 6: Angular position of Jaw link BC of the crusher with respect to the vertical direction and direction parallel to the fixed link AD when crank is at 30º with respect to the horizontal direction.

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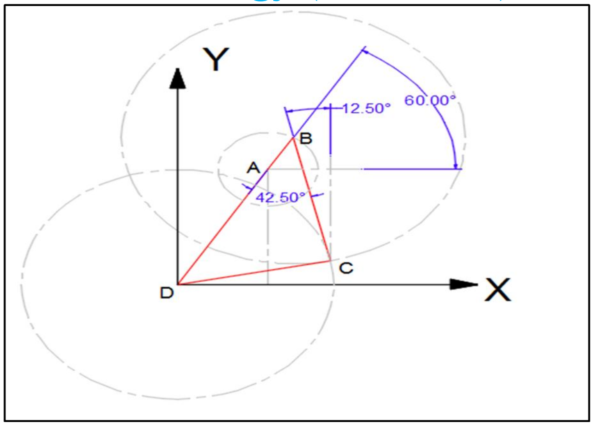

Fig 7: Angular position of Jaw link BC of the crusher with respect to the vertical direction and direction parallel to the fixed link AD when crank is at 60º with respect to the horizontal direction.

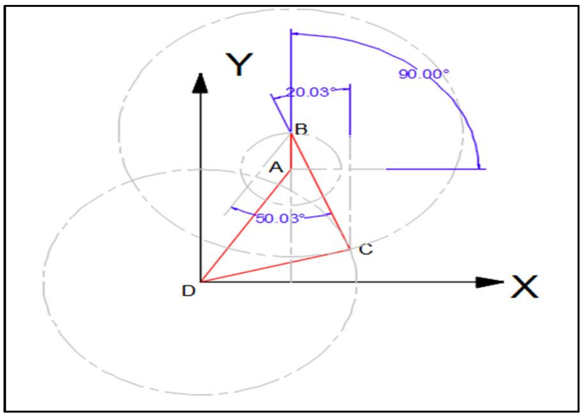

Fig 8: Angular position of Jaw link BC of the crusher with respect to the vertical direction and direction parallel to the fixed link AD when crank is at 90º with respect to the horizontal direction.

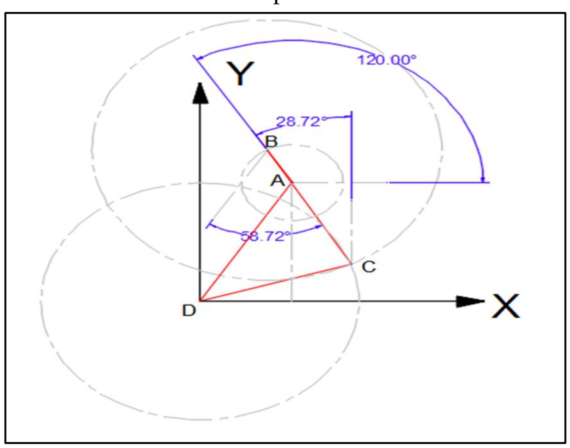

Fig 9: Angular position of Jaw link BC of the crusher with respect to the vertical direction and direction parallel to the fixed link AD when crank is at 120º with respect to the horizontal direction.

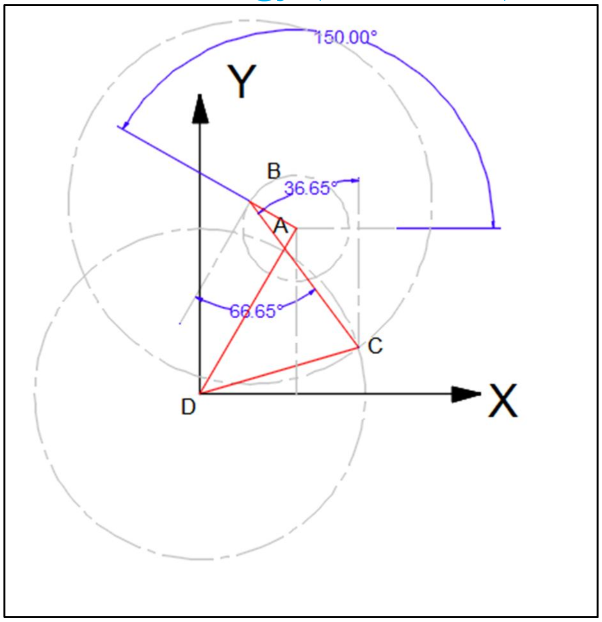

Fig 10: Angular position of Jaw link BC of the crusher with respect to the vertical direction and direction parallel to the fixed link AD when crank is at 150º with respect to the horizontal direction.

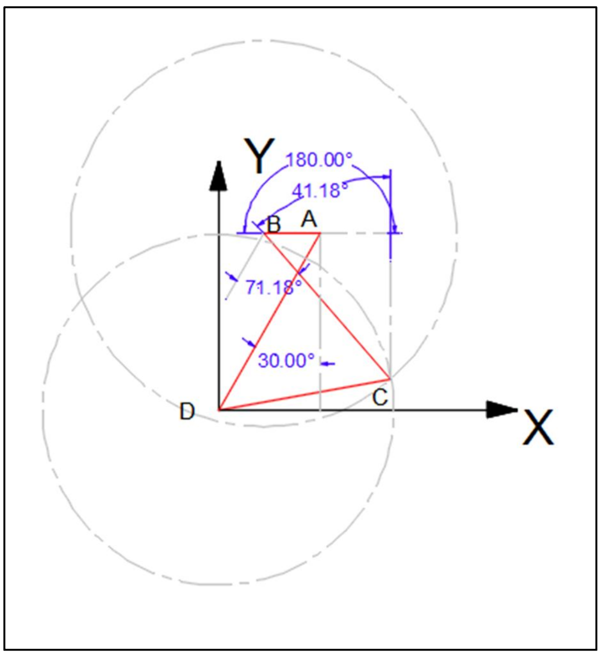

Fig 11: Angular position of Jaw link BC of the crusher with respect to the vertical direction and direction parallel to the fixed link AD when crank is at 180º with respect to the horizontal direction.

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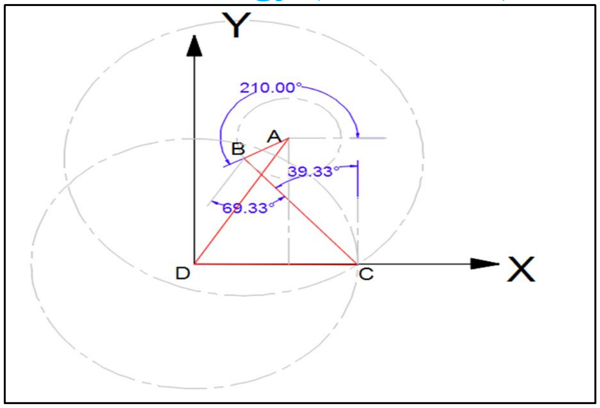

Fig 12: Angular position of Jaw link BC of the crusher with respect to the vertical direction and direction parallel to the fixed link AD when crank is at 210º with respect to the horizontal direction.

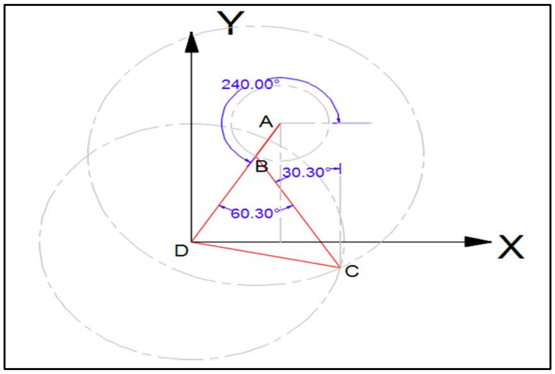

Fig 13: Angular position of Jaw link BC of the crusher with respect to the vertical direction and direction parallel to the fixed link AD when crank is at 240º with respect to the horizontal direction.

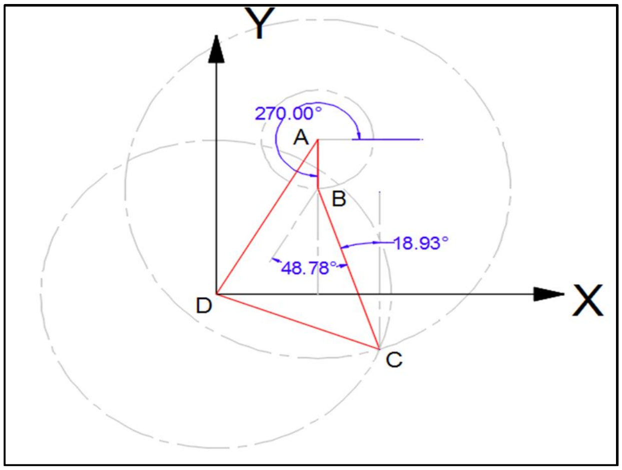

Fig 14: Angular position of Jaw link BC of the crusher with respect to the vertical direction and direction parallel to the fixed link AD when crank is at 270º with respect to the horizontal direction

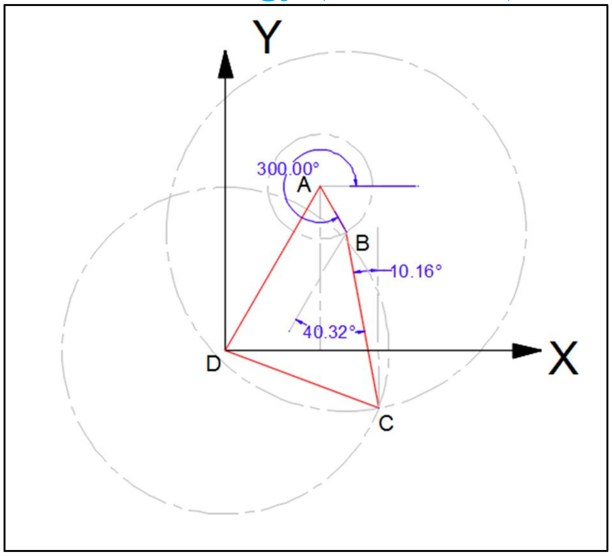

Fig 15: Angular position of Jaw link BC of the crusher with respect to the vertical direction and direction parallel to the fixed link AD when crank is at 300º with respect to the horizontal direction.

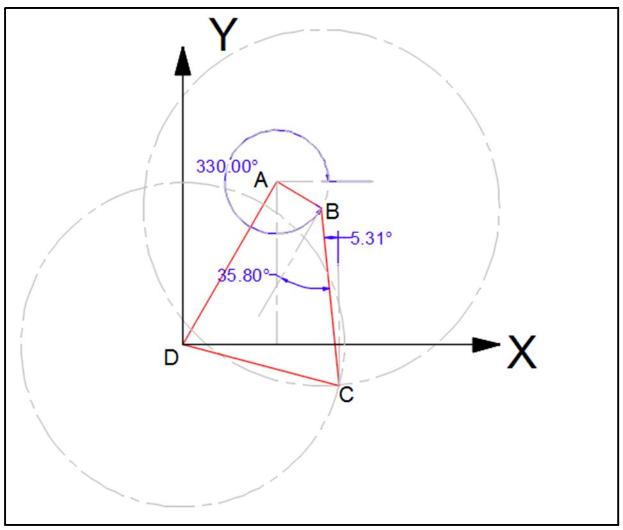

Fig 16: Angular position of Jaw link BC of the crusher with respect to the vertical direction and direction parallel to the fixed link AD when crank is at 300º with respect to the horizontal direction.

From the above figure it is evident that the code generated for the calculation of kinematic parameters are quite perfect. Now other output parameters which have been derived from the programs mentioned in the previous chapter have been presented below.
To generate the following results incremental angle of the crank has been taken as 30º. With this increment of crank's rotational angle following results have been derived from the generic equation of the kinematic analysis of a crusher mechanism.

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IC Value: 13.98 ISSN: 2321-9653
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Result 1: Coupler curve of the mid-point of the Jaw link of the crusher mechanism.

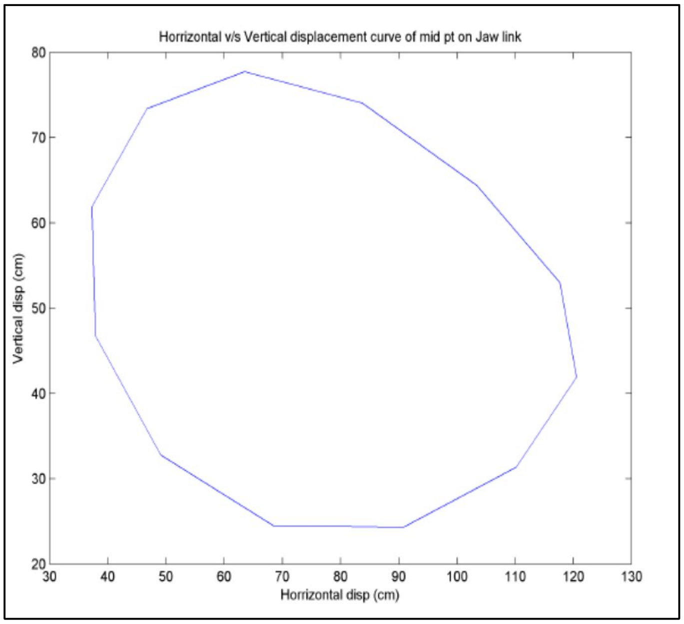

Fig 17: Coupler curve of the mid-point of the Jaw link of the crusher mechanism

Result 2: Angle of crusher's Jaw link with vertical v/s Angle of rotation of Crank.

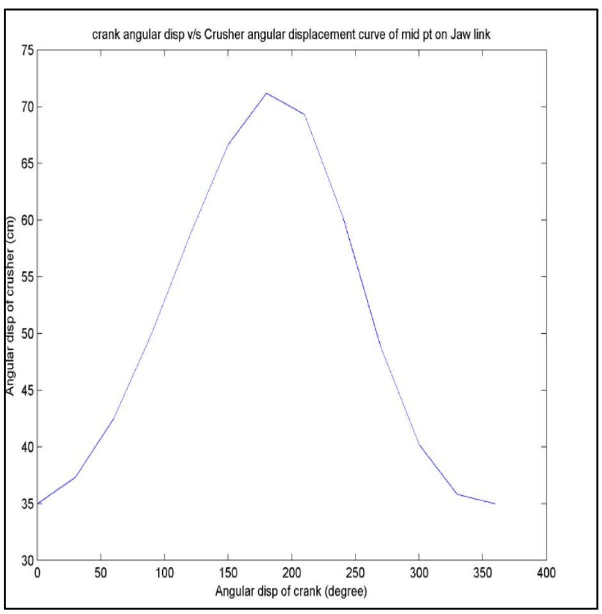

Fig 18: Angle of crusher's Jaw link with vertical v/s Angle of rotation of Crank.

Result 3: Vertical displacement of mid-point on Jaw v/s Angle of rotation of Crank.

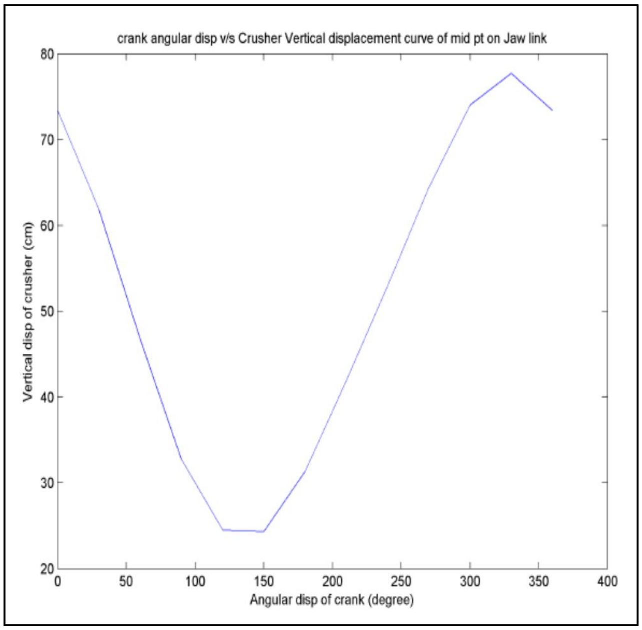

Fig 19: Vertical displacement of mid-point on Jaw v/s Angle of rotation of Crank.

Result 4: Horizontal displacement of mid-point on Jaw v/s Angle of rotation of Crank.

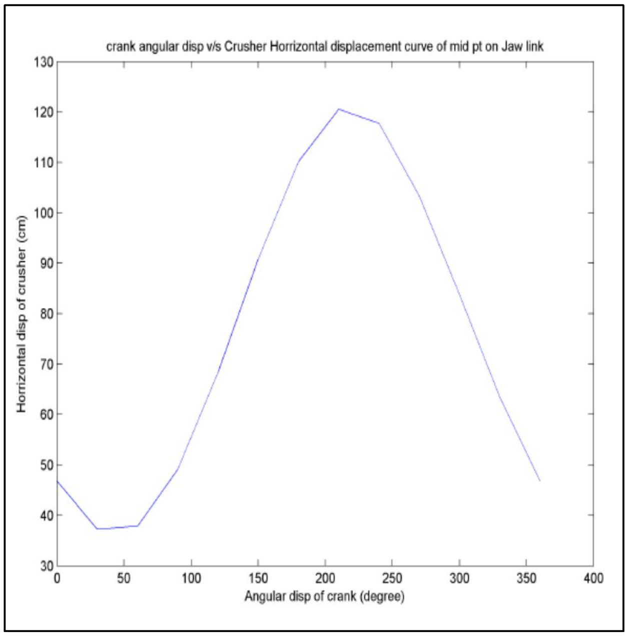

Fig 19: Horizontal displacement of mid-point on Jaw v/s Angle of rotation of Crank.

Result 5: Vertical Velocity Comp. of mid-point on Jaw v/s Angle of rotation of Crank.

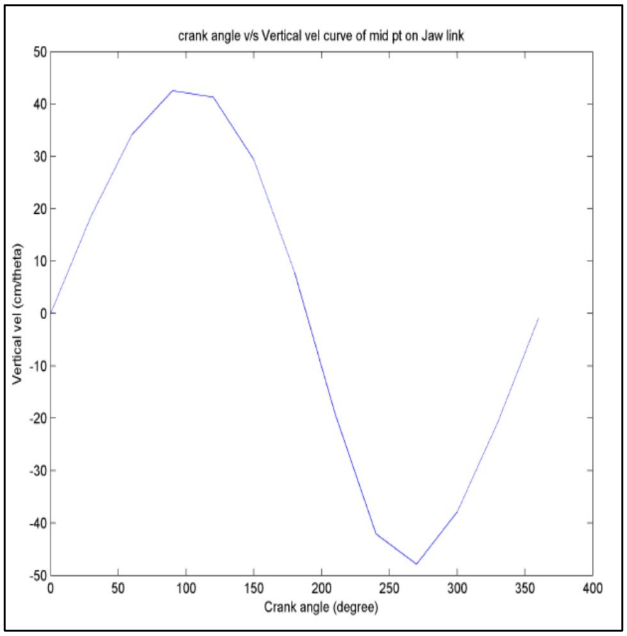

Fig 20: Vertical Velocity Comp. of mid-point on Jaw v/s Angle of rotation of Crank.

Result .6: Horizontal Velocity Comp. of mid-point on Jaw v/s Angle of rotation of Crank.

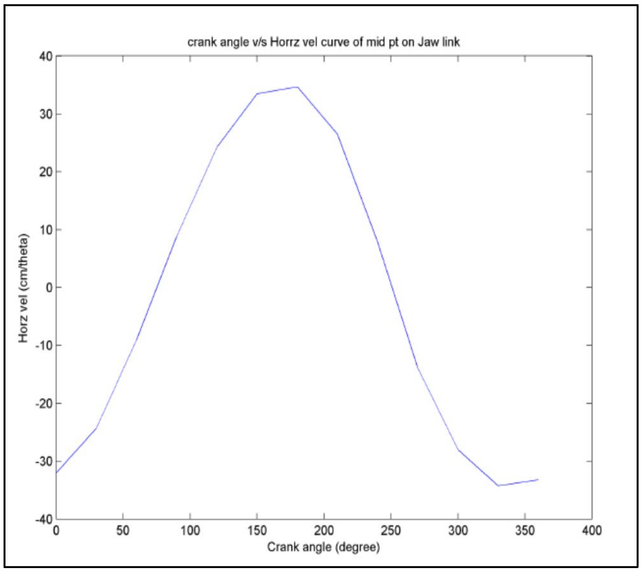

Fig 21: Horizontal Velocity Comp. of mid-point on Jaw v/s Angle of rotation of Crank.

Result 7: Resultant Velocity of mid-point on Jaw v/s Angle of rotation of Crank.

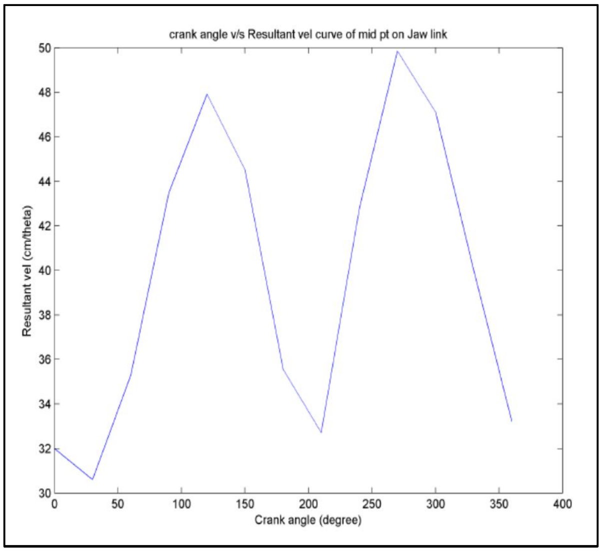

Fig 22: Resultant Velocity of mid-point on Jaw v/s Angle of rotation of Crank.

Result 8: Vertical Acceleration of mid-point on Jaw v/s Angle of rotation of Crank.

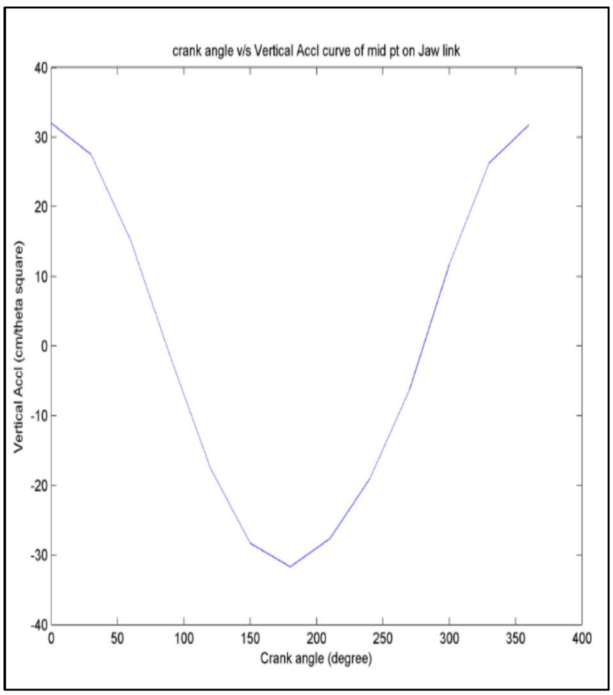

Fig 23: Vertical Acceleration of mid-point on Jaw v/s Angle of rotation of Crank.

Result 9: Horizontal Acceleration of mid-point on Jaw v/s Angle of rotation of Crank.

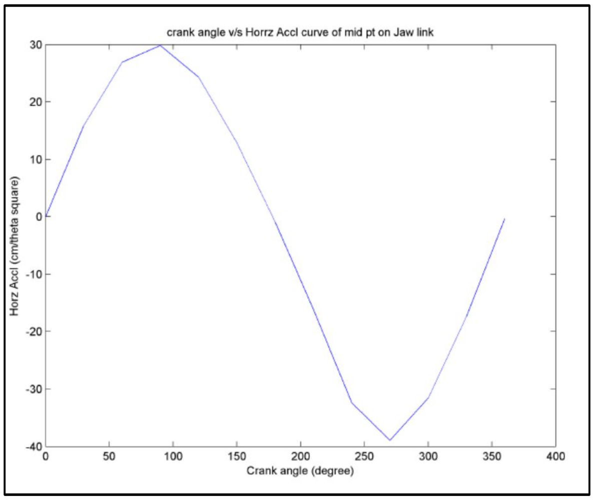

Fig 24: Horizontal Acceleration of mid-point on Jaw v/s Angle of rotation of Crank.

Result 10: Resultant Acceleration of mid-point on Jaw v/s Angle of rotation of Crank.

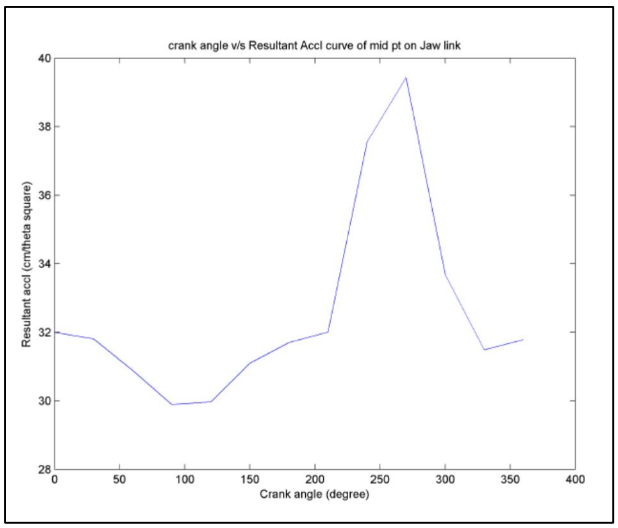

Fig 25: Resultant Acceleration of mid-point on Jaw v/s Angle of rotation of Crank.

IV. CONCLUSIONS AND FUTURE SCOPE

In the present work the fact which has been intended to establish is that, for the synthesis of any mechanism it is very much required to first do the kinematic and dynamic analysis of the mechanism. Synthesis of any mechanism is not a shot job. Trial and error method type of activity is involved. To rationalize the trial and error type of activity Optimization methods are employed in mechanism design. So it is needed to vary the input parameters for optimization. Therefore graphical method is not at all employable where synthesis of a mechanism is concern. In reference [7] Dr. Modok et al have investigated a crusher mechanism with help of graphical method. In the present work a code has been generated by virtue of which one can do the kinematic and dynamic analysis of a mechanism generically. So an effort has been made to make the kinematic analysis of a mechanism applicable for any type of configuration.

In the present work first a mathematical model have been developed to find out linear as well as angular displacement of any point on the crushing link which is call Jaw as a function of angular position of crank rotation. Then this mathematical model has been converted into a MATLAB program. The output results or parameters from this program for a given set of input parameter (as per reff [7]) have been validated with the output generated by graphical method using AutoCAD.

Next the mathematical model for displacement has been converted into mathematical model for Velocity and Acceleration by differentiating once and twice respectively.

To generate a mathematical model and code for dynamic analysis and synthesis of a mechanism one can use the codes for velocity and acceleration analysis as developed in the present work.

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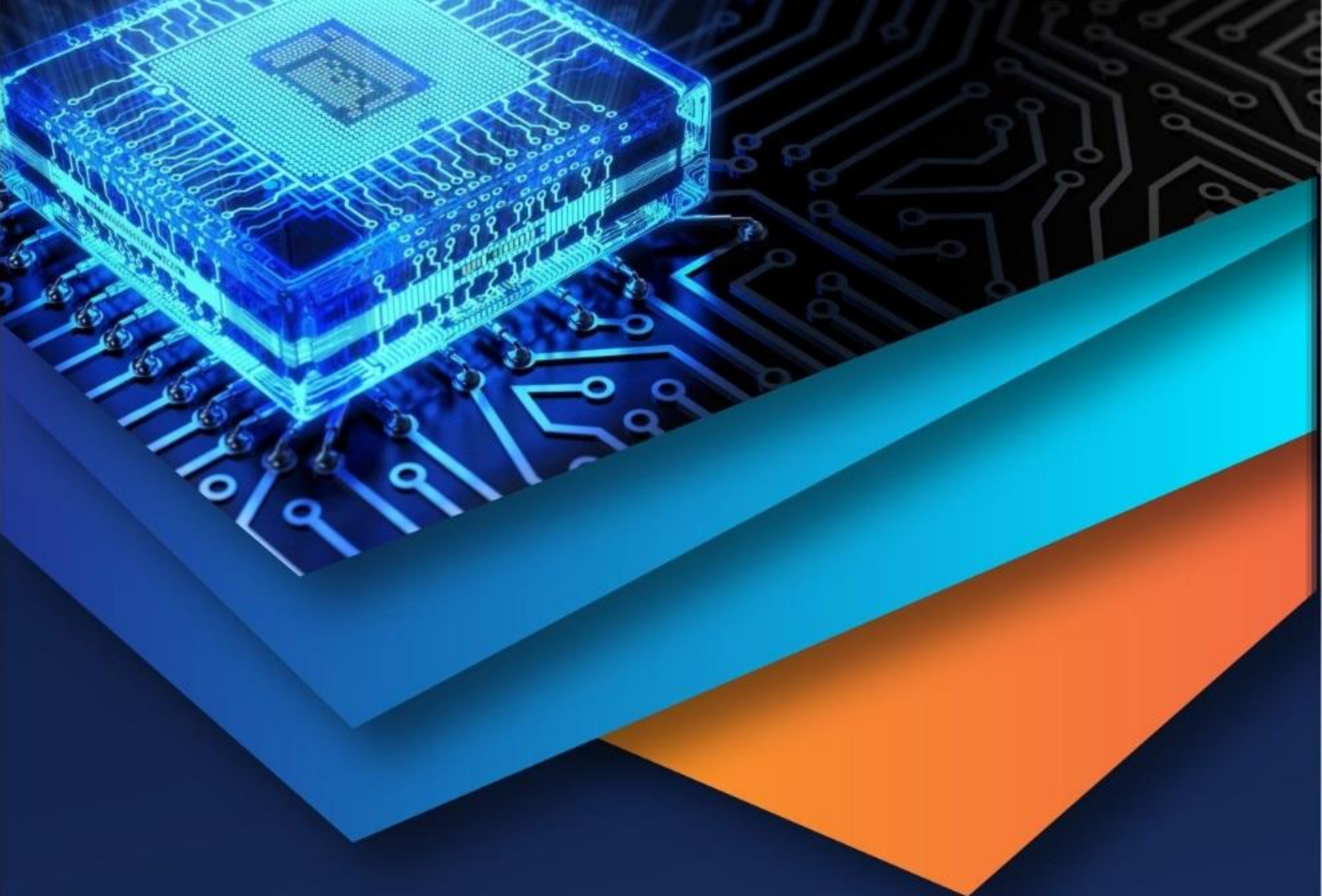

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