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Research Paper

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Kinematic Synthesis and Analysis of Alternate Mechanism for Stone Crusher Using Relative Velocity Method

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ABSTRACT

Stone crushers are developed for reducing the size of stones to a desired size. Till today for reducing sizes of stones from 10cm x 10cm to 2.5cm x 2.5cm in quarries is a laborious job and is done manually. So our approach is to develop a best optimum mechanism for said requirement. For this we are proposing different alternate mechanisms for same capacities, out of which best optimum design will be find out by using Multiple Regression Analysis using MATLAB software.
Purpose of this paper is to design and develop one alternate mechanism for a same capacity stone crusher mechanisms like we have already proposed in the different journals [ref.-1].The basic idea is to optimize the design of the crusher which would be best suited for stone which need crushing force of 3 Tons.

Introduction:
The proposed research is focused on designing an alternate stone crushing mechanism using relative velocity Method. This method, of completely geometrical nature, consists in finding the feasible configurations that a kinematic chain can adopt within the specified ranges for its degrees of freedom, a configuration is an assignment of positions and orientations to all links that satisfy the kinematic constraints imposed by all joints.

Kinematic Synthesis of Proposed Stone Crusher The Proposed stone crusher consists of two mechanisms, which needs to be synthesized separately.

1 Crank and lever Mechanism
2 Rack and sector Gear Mechanism

Kinematic Synthesis of Crank and Lever Mechanism
Basically this mechanism falls under class I of a four bar mechanism, in which the shortest link can make a full revolution relative to each of the others. The three longer links can only oscillate relative to each other.

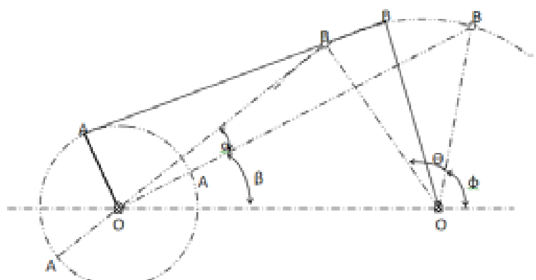

Fig -1 Synthesis of Crank and lever Mechanism

Fig 1 Crank – lever mechanism is shown with the notation to be used. As the crank (Link 1) rotates the lever i.e. link 3 oscillates through an angle ϴ. B1 and B2 are the two extreme positions of the pin at the end of the lever. A1 and A2 are the corresponding crank pin positions. Here it is important to note that the two swings of the lever do not take place during equal crank rotation angles.

The four bar function is a "Quick Return Mechanism". If the crank turns at a constant speed, the time ratio of two swings of the lever is

T.R= (180+ ∝)
 (180- ∝)

The most common design problem in which, the angle of oscillation ϴ and angle α (or the time ration, which determines α) are specified.

Considering the following input as Time ratio T.R= 1.15, ϴ = 40^0 and Length of lever = 100 cm.

Substituting the value of T.R. in equation 1, α = 12.558^0 is determined.

Detailed synthesis of the mechanism is carried out by Geometrical method .It is designers skill to choose the set of link lengths for developing mechanisms. Following is the set which we have selected for our design of stone crushing mechanism.

Crank Length – 27cm, Coupler – 195 cm, Lever – 100 cm, fixed Distance – 156 cm.

Kinematic Synthesis of Rack and sector Gear Mechanism
The rack and pinion is used in rotary to linear motion conversion or vice-versa. The center of rotation of sector gear is the fixed point of oscillation of lever of a crank and rocker mechanism. As the lever oscillates in 400, the rack meshes with the pinion and moves left and right in response to angular movement of oscillatory lever. The design of hopper of a stone crusher is such that at least two stones of size 10 cm X 10cm are to be placed at a time for crushing. So the minimum distance travelled by the rack should be at least equal to the 20cm, which is provided by sector gear which oscillates for 400.

L= π ϴ*R …………….. (2)
 180

20= π (40)*R
 180

R=28.64cm

Where,

L=Minimum Length of Rack=20cm, R=Radius of Pinion, ϴ=Angle of oscillation

Pitch circle diameter of sector gear=D=28x 2=56cm.

Considering standard module, m= 10 mm

No. of teeth on the circumference of gear = D/m =56

Circular Pitch = (π*D) ………… (3)
 N

= (π*560)= 31.4159 mm
 56

For designing sector gear which oscillates for 400,

Let,

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N=Number of teeth on sector gear

N= Angle of oscillation*No.of teeth on gear
Angle of One revolution ... (4)

= (40*56)/360 = 6.22

Therefore, Total number of teeth on sector gear = 7

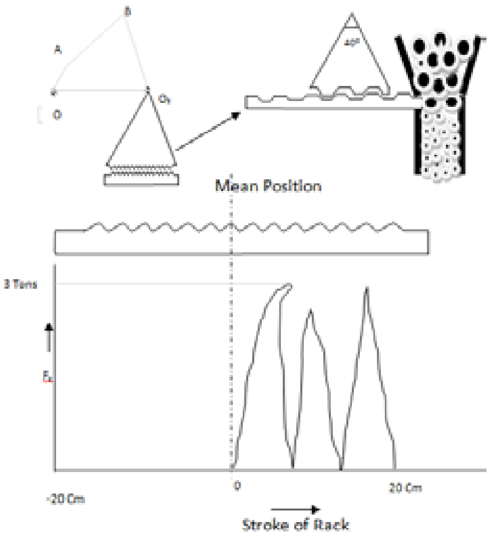

Fig -2 Synthesis of Rack and sector Gear Mechanism

STATIC FORCE ANALYSIS – Graphical Method
Analyses may be required for a number of mechanism positions; however, in many cases, critical maximum-force positions can be identified and graphical analyses performed for these positions only. An important advantage of the graphical approach is that it provides useful insight as to the nature of the forces in the physical system.

Figure below shows the angular position of crank $\Theta=60^0$

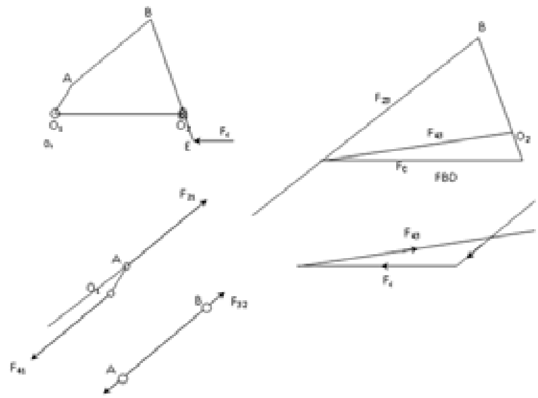

Fig-3 Static Force analysis – Graphical Method

The mechanism is analyzed graphically and ultimately the torque on the crank is computed. From the above Force polygon F_{23} =1.2 Tons in the direction shown.

Crushing force at the end of pinion is considered as 3 Tons. The crank is rotating at an angular speed of 120 rpm (Anticlockwise)

$F_{23} = -F_{32} = F_{12} = -F_{21} = F_{41}$ = 1.2Tons.

Summing Moments about point O1 gives torque required on the crank.

$\sum MO1 = T_{01A} + F_{21} \times h = 0$ .....(5)

For equilibrium, the torque TO1A must be equal to F21 X h. This is shown in Fig. Because the cross-product F_{21} x h is clockwise, the torque must be anticlockwise.

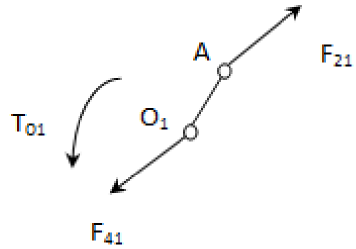

Fig: 4 Torque required on crank

So we have calculated a torque required on the crank which is given in a tabulated form.

Table: 1 Crank Angular Position Vs force on various links

Sr.No	Crank Angular Position	Coupler force on Link AB (in Tons)	Torque required on Crank O_1A (in Nm)
1	30^0	1	400 Clockwise
2	60^0	1	1600 Anti-Clockwise
3	90^0	0.9	2160 Anti-Clockwise
4	120^0	1	2700 Anti-Clockwise
5	150^0	0.9	1980 Anti-Clockwise
6	180^0	0.8	1120 Anti-Clockwise
7	210^0	0.78	390 Anti-Clockwise
8	240^0	0.9	900 Anti-Clockwise
9	270^0	1.1	1980 Clockwise
10	300^0	1.2	3120 Clockwise
11	330^0	1.2	2880 Clockwise
12	360^0	1.1	1320 Clockwise

Above table indicates that maximum force in a revolution of crank on coupler AB is 1.2 Tons

Dynamic Force analysis – Graphical Method
Dynamic force Analysis for Proposed Stone crusher uses d'Alembert's principle can be derived from Newton's second law.

$$F + (-ma_G) = 0 \tag{1}$$

$$T_{eG} + (-I_G \alpha) = 0 \tag{2}$$

The terms in parentheses in Eq. (1) and (2) are called the reverse-effective force and the reverse-effective torque, respectively. These quantities are also referred to as inertia force and inertia torque. Thus, we define the inertia force F, as

Fi = -maG (3)

This reflects the fact that a body resists any change in its velocity by an inertia force proportional to the mass of the body and its acceleration. The inertia force acts through the center of mass G of the body. The inertia torque or inertia couple C, is given by:

$$C_i = -I_G \alpha \tag{4}$$

As indicated, the inertia torque is a pure torque or couple. From Eq.(3) and (4), their directions are opposite to that of the accelerations. Substitution of Eq. (3) and (4) into Eq. (1) and (2) leads to equations that are similar to those used for static-force analysis:

$$\sum F = \sum F_e + F_i = 0 \tag{5}$$

$$\sum T_G = \sum T_{eG} + C_i = 0 \tag{6}$$

Where refers here to the summation of external forces and is the summation of external moments, or resultant external moment, about the center of mass G

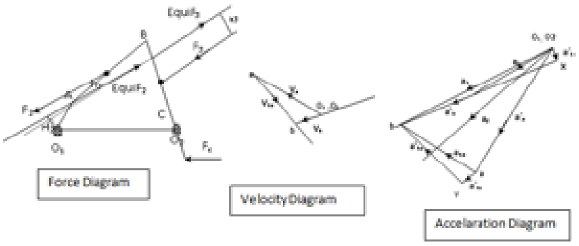

Fig: 5 Dynamic Force Analysis-Graphical Method

V. CALCULATIONS:

α2=a^{t}ba/AB=4500/195=23.07rad/s^2………from acceleration diagram

a_{G2}=3200N/cm^2……………………..From acceleration diagram

Allowable Strength = S X Size Factor X Ignorance F
Reliability Factor X Stress concentration Factor…(12)

= (16800 X 0.5 X 0.5
0.85 X 2.2)

= 2245.989 N/cm^2

But,

Allowable Strength = (Max.Force on link AB)/(Cross section of link AB)…………………………. (13)

2245.989 =1.2 x 10000
Cross section of link AB

Cross section of link AB = 5.35 cm^2

Selecting cross section of link as 3cm x 3 cm

m2 = Mass of Link AB = Mass density of Link x Volume of Link…………………… (14)

=7.8 x 10-3 x 3 x 3 x195=13.68 Kg.

I_{G2}=Mass moment of Inertia of Link AB @ C.G. = 1/12 x Mass of Link AB x(Link AB) 2………… (15)

= 43377 Kg-cm^2

Similarly we have calculated,

α_3 = ato2b/O2B = 3600/100 = 36 rad/s^2…………………from acceleration diagram

a_{G3}= 2000 N/cm2……………………..From acceleration diagram

m_3 = Mass of Link O2B = 7.02 Kg.

I_{G3} = Mass moment of Inertia of Link O_2B @ C.G. = 5850Kg-cm^2

In graphical force analysis we will account for inertia torques by introducing equivalent inertia forces. These forces are shown in figure, and their placement is determined. For link 2 offset forces F2 is equal and parallel to inertia force F12. Therefore

F2= 43776 N.

It is offset from the centre of mass G2 by a perpendicular amount equal to

h_2= $(I_{G2}\alpha_2)/(m_2 a_{G2})$……………………..(16)

h_2 = (43377 * 23.07)/ (13.68*3200) = 22.85 cm

And this offset is measured to the left to produce the required clockwise direction for the inertia moment about point G2.

In a similar manner the equivalent offset inertia force for link 3 is F3 = 14040 N at an offset distance

h_3 = $(I_{G3}\alpha_3)$ / $(m_3 a_{G3})$……………………………..(17)

h_3 = (5850*36) / (7.02* 2000) = 15 cm

And this offset is measured to the right to produce the required clockwise direction for the inertia moment about point G3.Values of H2 and H3 are taken from acceleration diagram.

Taking moment @ O1

= (F_2H_2) + (F_3H_3)………………………………..(18)

= (43776 x 10) + (14040 x 6)

= 5220000 N-cm= 5220 N-m.

Similarly Velocity, accelerations and corresponding Torque at various positions are calculated.

Shown in Tabulated form.

Table: 2 Crank Angular Position Vs Torque on Crank Considering Dynamic Loading

Sr. No	Crank Angular Position	Torque required on Crank O_1A Considering Dynamic Loading (in Nm)
1	30^0	51516 Clockwise
2	60^0	5220 Clockwise
3	90^0	27499.68 Anti-clockwise
4	120^0	49680 Anti-clockwise
5	150^0	44435.5 Anti-clockwise
6	180^0	30827.52 Anti-clockwise
7	210^0	13347.36 Anti-clockwise
8	240^0	6663.6 Anti-clockwise
9	270^0	9575.28 Clockwise
10	300^0	10416.96 Clockwise
11	330^0	20478.24 Clockwise
12	360^0	31972.85 Clockwise

Total torque = Static Torque + Dynamic Torque Below table shows Total torque required at various Crank Positions.

Table: 3 Crank Angular Position Vs Net Torque on Crank

Sr. No	Crank Angular Position (Θ)	Torque required on Crank O_1A Considering Static Loading (in Nm)	Torque required on Crank O_1A Considering Dynamic Loading (in Nm)	Net Torque on Crank O_1A (T) in Nm
1	30^0(Loaded Condition)	400 Clockwise	51516 Clockwise	51916 Clockwise
2	60^0(Loaded Condition)	1600 Anti-Clockwise	5220 Clockwise	3620 Clockwise
3	90^0(Loaded Condition)	2160 Anti-Clockwise	27499.68 Anti-clockwise	29659.68 Anti-clockwise
4	120^0(Loaded Condition)	2700 Anti-Clockwise	49680 Anti-clockwise	52380 Anti-clockwise

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5	150^0(Loaded Condition)	1980 Anti-Clockwise	44435.5 Anti-clockwise	46415.5 Anti-clockwise
6	180^0(Loaded Condition)	1120 Anti-Clockwise	30827.52 Anti-clockwise	31947.52 Anti-clockwise
7	210^0(No Load Condition)	0	13347.36 Anti-clockwise	13347.36 Anti-clockwise
8	240^0(No Load Condition)	0	6663.6 Anti-clockwise	6663.6 Anti-clockwise
9	270^0(No Load Condition)	0	9575.28 Clockwise	9575.28 Clockwise
10	300^0(No Load Condition)	0	10416.96 Clockwise	10416.96 Clockwise
11	330^0(No Load Condition)	0	20478.24 Clockwise	20478.24 Clockwise
12	360^0(No Load Condition)	0	31972.85 Clockwise	31972.85 Clockwise

Net Torgue on the crank is calculated and the actual torque required in one revolution of crank is tabulated in the following graph.

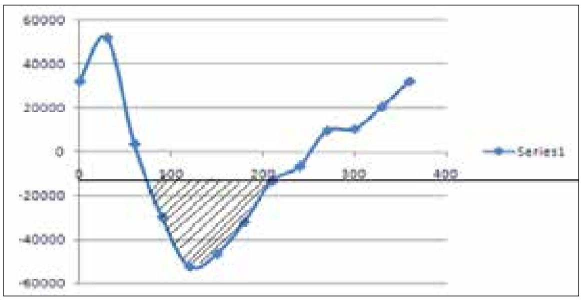

Fig:6 Graph of Total Torgue On crank Vs One revolution of crank

Applying Newton-Cotes quadrature formula and composite Trapezoidal rule, Total area under this Curve is calculated which represents work done per revolutions.

Work done per revolution=∫T dθ

= ΔΘ
2 T1 +TN +2(T2 +T3+................. TN-1)]

= 94222.87 Nm.rad (Anticlockwise)

T_{mean} = ∫TdΘ
∫dΘ

= 14996.03 Nm (i.e. Anticlockwise)

Further mean torque is calculated which decides the other input parameters like drive rating, flywheel etc.

Area under shaded portion gives a maximum fluctuation of energy based on which flywheel is designed.

Based on design of complete stone crusher mechanism the other parameters of a stone crusher like design of Motor, belt drive, Flywheel design, gear box etc are decided. Figure given below shows a complete layout of small capacity stone crusher

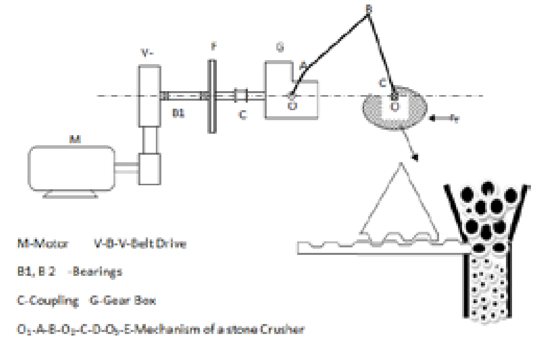

Fig: 6 Layout of small capacity stone crusher

Flywheel Parameters:

Type of Flywheel: Solid disc geometry with inside and outside radius

Type and Density of Material: Cast Iron with density as 7200 Kg/m3

Speed: 900 r. p. m.

Inside radius: 44.5 cm

Outside radius: 55.6 cm

Thickness: 5.56cm

Total mass of flywheel: 139.73 Kg

Worm Gear Box:
Specifications: Worm Gear Box, Speed Reduction ratio 7.5:1, Power: 148KW

V-Belt Drive
For speed reduction from 1500 rpm to 900 rpm, Number of belts: 12 of type 8V1250, Type of Pulley 8V Grooves Sheaves of D Type.

Motor: 125 KW, 4Pole, 1500rpm

VI. PREVIOUS WORK:
We have proposed one more design for same capacity [ref.1], in which we proposed double rocker mechanism instead of Rack and sector gear mechanism.

In our recent paper we have selected a different combinations of link lengths for designing crank and lever mechanism and Rack and sector gear mechanism.

Conclusion: -
Here we discussed about graphical method to synthesize and analyzed an alternate mechanism and derived a related parameters used for proposed stone crusher. With the same logic similar machines can be designed which are of same capacities. Various designs may have advantages and disadvantages over one another. Based on generated data for various designs, mathematical model based on dimensional analysis is designed. Further by multiple regressions method using MATLAB software the best and optimum model can be obtained.

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