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

Prof. Anjali J. Joshi *, Dr. Jayant P. Modak **

*(Research Student, Department of Mechanical Engineering, Priyadarshini College of engineering, Nagpur (M.S) India)

** (Dean R&D, Department of Mechanical Engineering, Priyadarshini College of engineering, Nagpur (M.S) India)

ABSTRACT

In this paper alternate mechanism for design and analysis of small size stone crusher mechanism is discussed. 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. Presently for reducing sizes of stones from 10cm x 10cm to 2.5cm x 2.5cm in quarries is laborious job and is done manually our approach is to design a best optimum mechanism for said conditions.

Keywords – Dynamic, Kinematic synthesis and analysis, Sector gear, Static.

I. INTRODUCTION

The proposed research is interested in providing an altenative mechanism which includes the position synthesis and analysis of a mechanism with rigid bodies (Links) interconnected by kinematic pairs (Joints) i.e. kinematic chains. 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.

II. Kinematic Synthesis of Proposed (Alternative Mechanism) Stone Crusher

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

2.1) Crank and lever Mechanism
2.2) Rack and sector Gear Mechanism

2.1) 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.

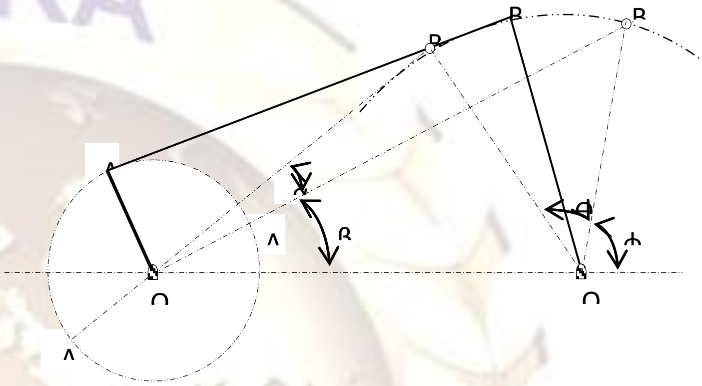


Fig -1 Synthesis of Crank and lever Mechanism

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. B_1 and B_2 are the two extreme positions of the pin at the end of the lever. A_1 and A_2 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

$$T.R = \frac{180+\alpha}{180-\alpha} \;\ldots\ldots\ldots\ldots\ldots (1)$$

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, $\Theta = 40^0$ 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 and optimum parameters are obtained as follows.

Crank Length – 32cm, Coupler – 110 cm, Lever – 100 cm, fixed Distance – 116 cm.

2.2) 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

3 Tons

oscillation of lever of a crank and rocker mechanism. As the lever oscillates in 40^0, 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 40^0.

$$L = \frac{\pi}{180}\Theta * R \quad \ldots\ldots\ldots\ldots\ldots (2)$$

$$20 = \frac{\pi}{180}(40) * R$$

R=28.64cm

Where,

L=Minimum Length of Rack=20cm

R=Radius of Pinion

Ɵ=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

$$\text{Circular Pitch} = \frac{\pi * D}{N} \ldots\ldots\ldots\ldots (3)$$

$$= \frac{\pi * 560}{56} = 31.4159 \text{ mm}$$

For designing sector gear which oscillates for 40^0,

Let,

N=Number of teeth on sector gear

$$N = \frac{\text{Angle of oscillation} * \text{No. of teeth on gear}}{\text{Angle of One revolution}} \ldots (4)$$

$$= \frac{40 * 56}{360} = 6.22$$

Therefore, Total number of teeth on sector gear = 7

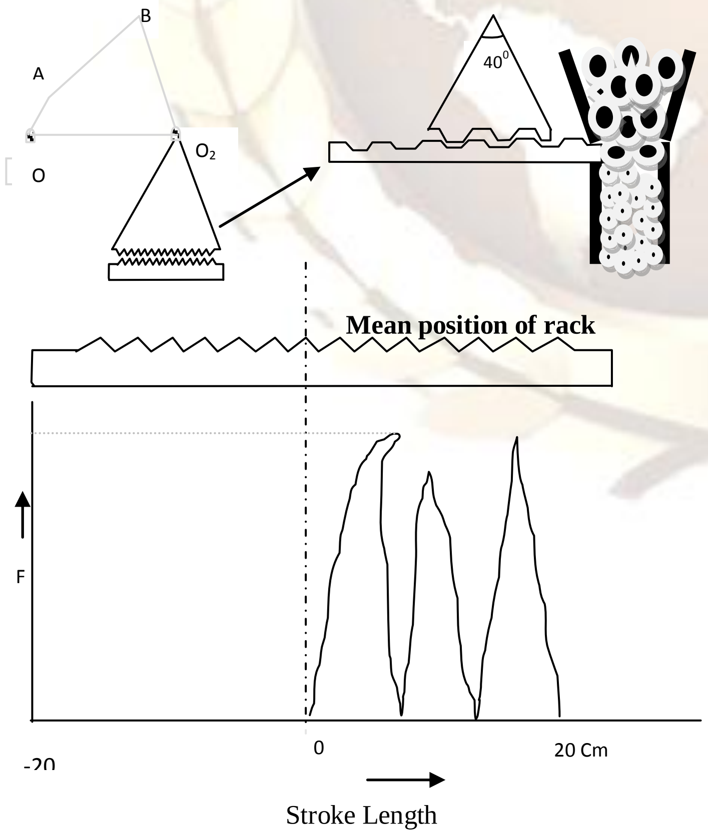


Fig -2 Synthesis of Rack and sector Gear Mechanism

III.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 $\Theta=60^0$

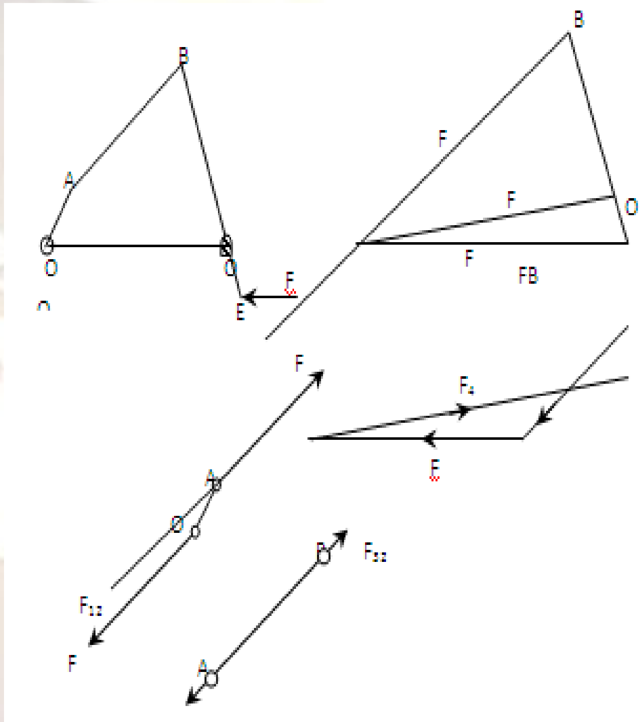


Fig-3 Static Force analysis – Graphical Method

The mechanism is analyzed graphically and ultimately the torque on the crank is computed. From the above Force polygon F_{23} =1.1 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)

$F_{23} = -F_{32} = F_{12} = -F_{21} = F_{41}$ = 1.1Tons.

Summing Moments about point O_1 gives torque required on the crank.

$\sum M_{O1} = T_{O1A} + F_{21} \times h = 0$ …..(5)

For equilibrium, the torque T_{O1A} must be equal to F_{21} X h. This is shown in Fig. Because the cross-product F_{21} x h is clockwise, the torque must be anticlockwise.

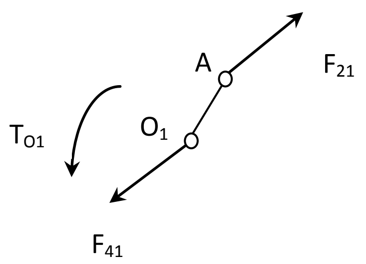


Fig: 4 Torque required on crank

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

Table: 1 Crank Angular Position with force and Torgue required on crank

Sr.No	Crank Angular Position	Coupler force on Link AB (in Tons)	Torque required on Crank O_1A (in Nm)
1	30^0	1	1200 Clockwise
2	60^0	**1.1**	1540 Anti-Clockwise
3	90^0	0.9	2340 Anti-Clockwise
4	120^0	0.9	2880 Anti-Clockwise
5	150^0	0.8	2240 Anti-Clockwise
6	180^0	0.8	1760 Anti-Clockwise
7	210^0	0.7	840 Anti-Clockwise
8	240^0	0.6	60 Anti-Clockwise
9	270^0	0.7	980 Clockwise
10	300^0	0.85	2040 Clockwise
11	330^0	1	3200 Clockwise
12	360^0	1	2600 Clockwise

Above table indicates that maximum force in a revolution of crank on coupler AB is 1.1 Tons.

IV.DYNAMIC FORCE ANALYSYS – Graphical Method

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

$$F + (-ma_G) = 0 \quad \text{...............} (6)$$

$$T_{eG} + (-I_G \alpha) = 0 \quad \text{.................} (7)$$

The terms in parentheses in Eq. (6) and (7) 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

$F_i = -ma_G$ …………………(8)

Where $\sum F$ refers here to the summation of external forces and $\sum T_{eG}$ is the summation of external moments, or resultant external moment, about the center of mass *G*

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: $C_i = -I_G \alpha$ … *(9)*

As indicated, the inertia torque is a pure torque or couple. From Equations (8) & (9), their directions are opposite to that of the accelerations. Substitution of Equation (8) and (9) into Equation (6) and (7) leads to equations that are similar to those used for static-force analysis:

$$\sum F = \sum F_e + F_i = 0 \quad \text{.................} (10)$$

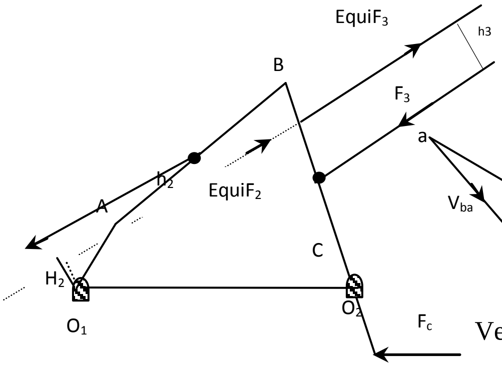
$$\sum T_G = \sum T_{eG} + C_i = 0 \quad \text{...............} (11)$$

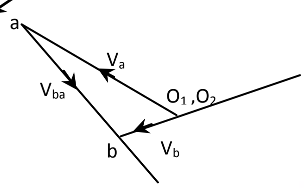
V. CALCULATIONS:

$\alpha_2 = a^t_{ba}/AB = 4700/110 = 42.72 rad/s^2$………from acceleration diagram

$a_{G2} = 5500 N/cm^2$……………………..From acceleration diagram

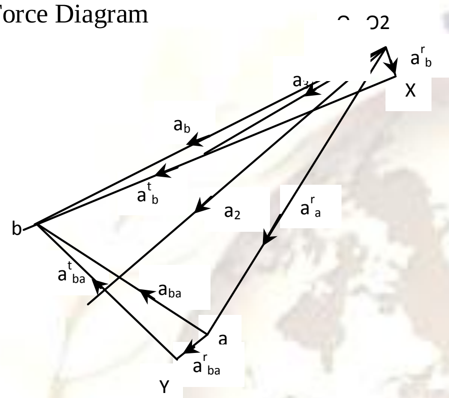
AllowableStrength





Velocity Diagram

Force Diagram



Acceleration Diagram

Fig: 5 Dynamic Force Analyses – Graphical Method

$$= \frac{S\ X\ Size\ Factor\ \ X\ Ignorance\ \ F}{\text{Reliability Factor X Stress concentration Factor}} \ldots(12)$$

$$= \frac{16800\ X\ 0.5\ \ X\ 0.5}{0.85\ X\ 2.2}$$

$= 2245.989\ N/cm^2$

But,

Allowable Strength $= \frac{Max.Force\ on\ link\ AB}{\text{Cross section of link AB}} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (13)$

$2245.989 = \frac{1.1\ x\ 10000}{\text{Cross section of link AB}}$

Cross section of link AB = 4.91 cm^2

Selecting cross section of link as 3cm x 3 cm ,

m_2 = Mass of Link AB = Mass density of Link x Volume of Link…………………… (14)

$=7.8 \times 10^{-3} \times 3 \times 3 \times 110 = 7.72$ Kg.

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

= 7786.35Kg-cm^2

Similarly we have calculated,

$\alpha_3 = a^t_{o2b}/O_2B = 6250/100 = 62.5\ rad/s^2$…………………from acceleration diagram

a_{G3}= 3150 N/cm^2……………………..From acceleration diagram

m_3 = Mass of Link O_2B = 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 F_2 is equal and parallel to inertia force $F_{12.}$ Therefore F_2= 42460 N.

It is offset from the centre of mass G_2 by a perpendicular amount equal to

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

$h_2 = (7786.35 * 42.72)/ (7.72*5500) = 7.83$ cm

And this offset is measured to the left to produce the required clockwise direction for the inertia moment about point $G_{2.}$

In a similar manner the equivalent offset inertia force for link 3 is F_3 = 22113 N at an offset distance $h_3 = (I_{G3}\alpha_3) / (m_3 a_{G3})$……………………………..(17)

$h_3 = (5850*62.5) / (7.02* 3150) = 16.53$ cm

And this offset is measured to the right to produce the required clockwise direction for the inertia moment about point G_3

Taking moment @ O_1

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

= (42460 x 17) + (22113 x 10)

= 942950 N-cm= 9429.5 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	27253.54 Clockwise
2	60^0	9429.5 Clockwise
3	90^0	13109.08 Anti-Clockwise
4	120^0	23115.36 Anti-Clockwise
5	150^0	16869.08 Anti-Clockwise
6	180^0	5273.08 Anti-Clockwise
7	210^0	1225.88 Anti-Clockwise
8	240^0	5190.88 Clockwise
9	270^0	8678.8 Clockwise
10	300^0	22751.28 Clockwise
11	330^0	19385.01 Anti-Clockwise
12	360^0	14389.2 Anti-Clockwise

Total torque =Static torque +Dynamic Torque….(19)

Table shows Total torque 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 (in Nm)
1	30^0(Loaded Condition)	1200 Clockwise	27253.54 Clockwise	28453.54 Clockwise
2	60^0(Loaded Condition)	1540 Anti-Clockwise	9429.5 Clockwise	7889.5 Clockwise
3	90^0(Loaded Condition)	2340 Anti-Clockwise	13109.08 Anti-Clockwise	15449.08 Anti-Clockwise
4	120^0(Loaded Condition)	2880 Anti-Clockwise	23115.36 Anti-Clockwise	25995.36 Anti-Clockwise
5	150^0(Loaded Condition)	2240 Anti-Clockwise	16869.08 Anti-Clockwise	19136.08 Anti-Clockwise
6	180^0(Loaded Condition)	1760 Anti-Clockwise	5273.08 Anti-Clockwise	7033.08 Anti-Clockwise
7	210^0(No Load Condition)	0	1225.88 Anti-Clockwise	1225.88 Anti-Clockwise
8	240^0(No Load Condition)	0	5190.88 Clockwise	5190.88 Clockwise
9	270^0(No Load Condition)	0	8678.8 Clockwise	8678.8 Clockwise
10	300^0(No Load Condition)	0	22751.28 Clockwise	22751.28 Clockwise
11	330^0(No Load Condition)	0	19385.01 Anti-Clockwise	19385.01 Anti-Clockwise
12	360^0(No Load Condition)	0	14389.2 Anti-Clockwise	14389.2 Anti-Clockwise

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

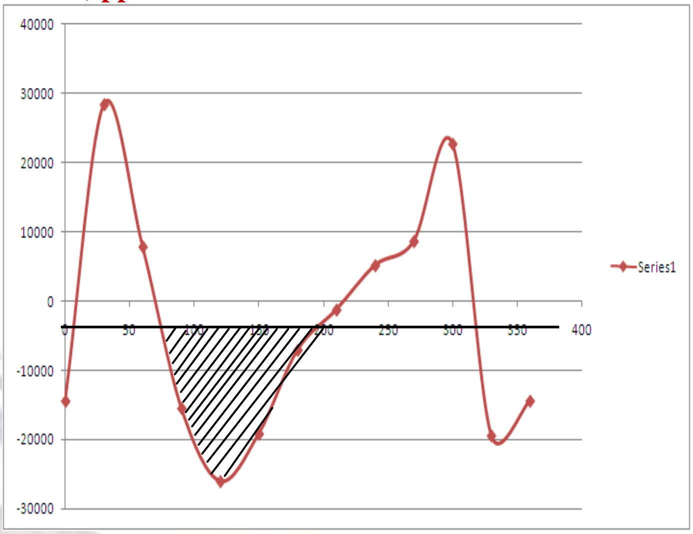


Fig: 6 Graph of Total Torque On crank Vs One revolution of crank

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= $\int T\, d\theta$

$= \frac{\Delta\Theta}{2}[T_1 + T_N + 2(T_2 + T_3 + \ldots\ldots\ldots T_{N-1})]$…….. (21)

= - 19207.06 Nm.rad

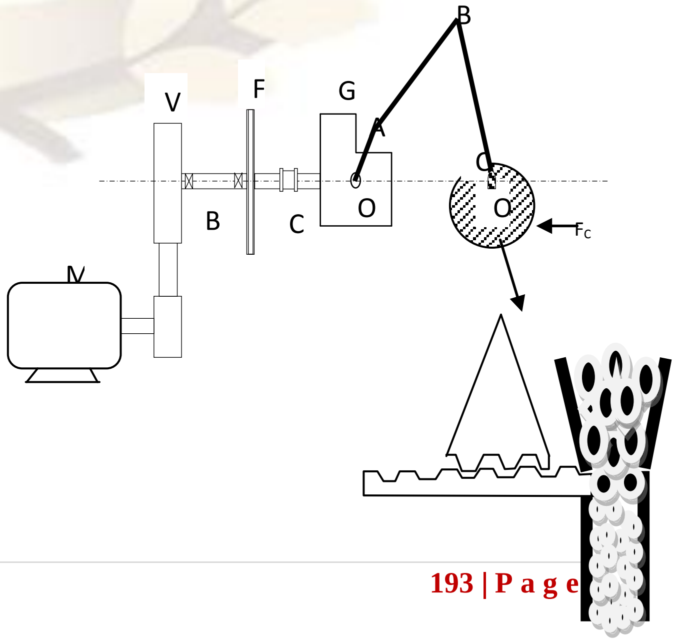
$T_{mean} = \frac{\int T d\Theta}{\int d\Theta}$ ………………………………….(22)

= **-** 3056.90 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



M-Motor V-B-V-Belt Drive B1, B 2 - Bearings C-Coupling G-Gear Box
O_1-A-B-O_2-C-D-O_3-E-Mechanism of a stone Crusher

Fig: 6 Layout of small capacity stone crusher

Outcomes of calculations

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/m^3

Speed: 900 r. p. m.

Inside radius: 37.8 cm

Outside radius: 47.3 cm

Thickness: 4.73cm

Total mass of flywheel: 86 Kg

Worm Gear Box:

Specifications: Worm Gear Box, Speed Reduction ratio 7.5:1, Power: 55KW

V-Belt Drive

For speed reduction from 1500 rpm to 910 rpm, Number of belts: 8 of type 5V1250, Type of Pulley 8V Grooves Sheaves of D Type.

Motor: 52 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; whose required parameters are as follows:

Flywheel Parameters: Solid disc geometry with inside and outside radius, made up of cast iron with density=7200 Kg/$m^2$of 92.49Kg.

Worm Gear Box: Power: 109 KW, Speed Reduction ratio 7.5:1,

V-Belt Drive: Number of belts: 8 of type 8V1250, Type of Pulley 8V Grooves Sheaves of D Type.

Motor: Power: 75 KW, 4Pole, 1500rpm

VII. COMPARISION OF TWO DESIGNS FOR OPTIMUM SOLUTION:

An Advantages of rack and sector gear are as follows:

1) Power required for same capacity of stone crusher is less.

2) Worm gear box of smaller size is required.

3) Overall construction cost of a stone crushing plant is less.

An Advantages of double rocker mechanism [ref.1], are as follows

1) Manufacturing cost is less.

2) Easy and cost effective maintenance.

3) Overall efficient mechanism.

VIII. CONCLUSION:

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.

H

REFERENCES:

Journal Papers:

1) Anjali J. Joshi, Dr. Jayant P. Modak, Design and Development of a small capacity stone crusher mechanism, *Indian Journal of Applied Research, Volume : 3,Issue : 2* February 2013 ,ISSN - 2249-555X
2) Dr. Louis G. Reifschneider, Teaching Kinematic Synthesis of Linkages without Complex Mathematics, *journal of Industrial Technology, Volume 21*, Number 4-October 2005 through December.

Books:

3) Dr. Habib ,MEG 373"*Kinematic and Dynamics of Machinery",* Chapter 5 Force Analysis.
4) Jack A. Collins, Henery Busby, George Stoab, "*Mechanical Design of Machine Elements & Machines"* Chapter 18 Flywheel and High –Speed Rotors.
5) Shriniwas S. Bali, Satish Chand, Transmission Angle in Mechanism (Triangle in mech), Pergamon *Mechanism and Machine Theory* 37(2002)175-195.
6) Kennth J. Waldron / Gary L. Kinzel, Kinematice , *Dynamics,& Design of Machinery*, edition 2007.
7) R.S. Khurmi and J.K. Gupta, *Theory of Machines*, edition 2002
8) Robert L.Norton,Machine Design,pearson Second edition.
9) Amitabha Ghosh and Ashok Kumar Mallik,*Theory of Mechanisms and Machines*,Third Edition Reprint 2008.

Theses:

10) James G. Donovan Ph. D Thesis in Minning and Mineral Engineering, "Fracture toughness based models for the prediction of power consumption,product size, and capacity of jaw crushers" July 2003.
11) Robert L.Norton,Machine Design,pearson Second edition.