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Design And Development Of A Small Capacity Stone Crusher Mechanism.

KEYWORDS: Mechanism, kinematic synthesis and analysis

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ABSTRACT *Existing Stone crushers available in the market are designed for bigger sized stones and using same for crushing smaller sized stones is not an optimal solution. Presently for reducing sizes of stones from 4 X 4 to 1 X1 in quarries is laborious job and is done manually. So development of smaller sized crusher is today's requirement. | In this paper detailed design and analysis of proposed 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.*

Introduction:

The proposed research is interested in 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. In mechanical engineering the solution of this problem is a fundamental issue, however its relevance is even greater because important problems of other research areas can be reduced to it.

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. Double Rocker 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.

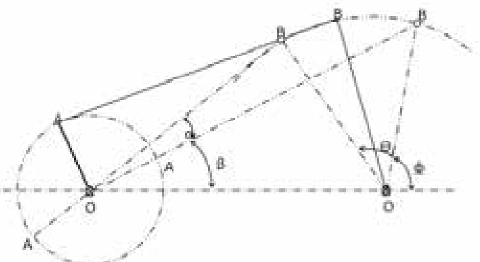

Fig -1 Synthesis of Crank and lever Mechanism

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 ω 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 The most common design problem in which, the angle of oscillation ϕ 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 and optimum parameters are obtained as follows.

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

Kinematic Synthesis of Double Rocker Mechanism

This mechanism falls under Class II of four Bar mechanism, In which all members of this class no link can make a full revolution relative to any another.

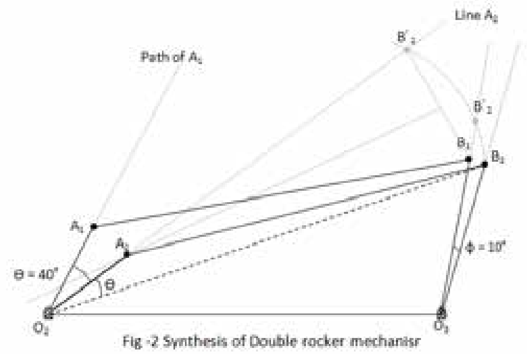

Fig -2 Synthesis of Double rocker mechanisr

Fig -2 Synthesis of Double rocker mechanism

Fig 2 Double rocker mechanism is shown with the notation to be used. As the Input rocker O_2C (Link 3) oscillates, the other output rocker i.e. O_3D (link5) oscillates through an angle ϕ.

Considering the following input as Output Rocker Length O_3D =50cm then fixed distance O_3O_2 = 80cm

Input angle of oscillation = 40^0

Detailed synthesis of the mechanism is carried out by Geometrical method for various output angle of oscillation ϕ. (10^0, 15^0, 20^0)

Since mechanism with output angle of oscillation ϕ =10^0 satisfies the Grashoff's law, following parameters are obtained.

Input Rocker Length - 12 cm, Coupler – 104cm

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 θ=600

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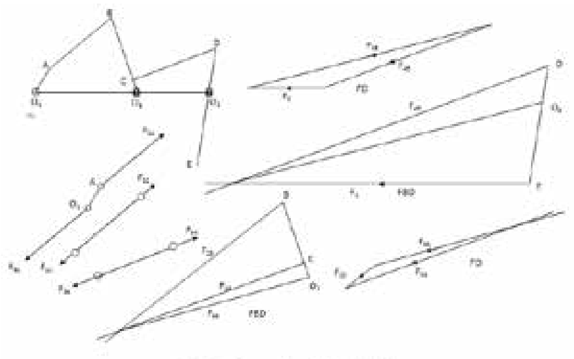

Fig-3 Static force analysis – Graphical Method

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_{45} =7.3 tons in the direction shown.

Crushing force at the extension of output rocker is considered approx. 3 Tons. And Crank is rotating at an angular speed of 120 rpm (Anticlockwise)

$F_{45} = -F_{54} = F_{34} = -F_{43}$ and F_{23} = 1.3 tons.

$F_{32} = -F_{23} = F_{12} = -F_{21} = F_{61}$

Summing Moments about point O_1 gives torque required on the crank.
$\sum M_{O1} = T_{O1A} + F_{21} \times h = 0$
For equilibrium, the torque T_{O1A} must be equal to $F_{21} \times h$. This is shown in Fig. Because the cross-product $F_{21} \times h$ is clockwise, the torque must be anticlockwise.

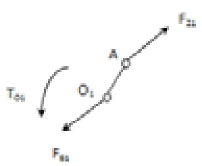

Fig:4 Torque required on crank

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)	Coupler force on Link CD (in Tons)	Torque required on Crank O_1A (in Nm)
1	30^0	1.3	6.2	1560 Clockwise
2	60^0	1.3	7.3	1690 Anti-Clockwise
3	90^0	1	7	2400 Anti-Clockwise
4	120^0	0.7	7	2240 Anti-Clockwise
5	150^0	0.7	6.6	2030 Anti-Clockwise
6	180^0	1.1	7.1	2310 Anti-Clockwise
7	210^0	0.8	6.4	960 Anti-Clockwise
8	240^0	1	7.5	1000 Anti-Clockwise
9	270^0	1.1	7.2	1320 Clockwise
10	300^0	1.2	7.4	3120 Clockwise
11	330^0	0.8	6.7	2880 Clockwise
12	360^0	0.8	7.8	4200 Clockwise

Above table indicates that maximum force in a revolution of crank on coupler AB is 1.3 Tons and on Coupler CD is 7.8 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.

$$F + (-ma_G) = 0 \quad (1)$$

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

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:

$$C_i = -I_G \alpha \quad (4)$$

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:

$$\sum F = \sum F_e + F_i = 0$$

(5)

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

(6)

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

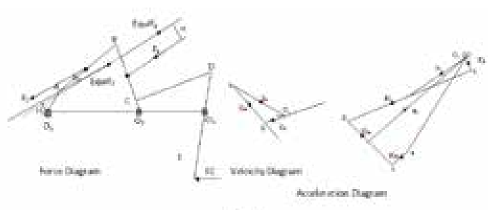

Fig: 5
In graphical force analysis we will account for inertia torques by introducing equivalent inertia forces. These forces are shown in figure, and their placement is determined. For link 2 offset forces F2 is equal and parallel to inertia force F12. Therefore
F2= 78825 N.
It is offset from the centre of mass G2 by a perpendicular amount equal to
h2IG2α2m2aG2
h2 = (10598.08 * 28.10)/ (10.51*7500)
h2 = 3.77cm.
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 = 39155 N at an offset distance
h3 = (IG2α2) / (m2aG2)

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h3 = (7692.5*82.5) / (9.55 * 4100)
h3 = 16.67cm
And this offset is measured to the right to produce the required clockwise direction for the inertia moment about point G3
Taking moment @ O1
= (F1H1) + (F2H2)
= (78825 x 25) + (39155 x 6)
= 2205555 N-cm
= 22055.5 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	22055.5
2	60^0	8655.09
3	90^0	17820.98
4	120^0	32328.84
5	150^0	29472.28
6	180^0	7270.08
7	210^0	4879.62
8	240^0	7219.01
9	270^0	11404.5
10	300^0	34764.0
11	330^0	26493.5
12	360^0	24138.78

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 (□)	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)
1	30^0(Loaded Condition)	+1560	+22055.5	23615.5
2	60^0(Loaded Condition)	-1690	+8655.09	6965.09
3	90^0(Loaded Condition)	-2400	-17820.98	-20220.98
4	120^0(Loaded Condition)	-2240	-32328.84	-34568.84
5	150^0(Loaded Condition)	-2030	-29472.28	-31502.08
6	180^0(Loaded Condition)	-2310	-7270.08	-9580.08
7	210^0(No Load Condition)	0	+4879.62	+4879.62
8	240^0(No Load Condition)	0	+7219.01	+7219.01
9	270^0(No Load Condition)	0	+11404.5	+11404.5
10	300^0(No Load Condition)	0	+34764.0	+34764.0
11	330^0(No Load Condition)	0	-26493.5	-26493.5
12	360^0(No Load Condition)	0	-24138.78	-24138.78

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

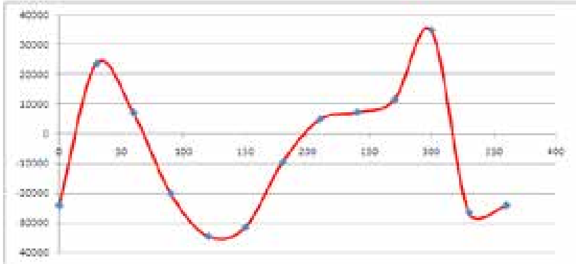


Fig:6 Graph of Total Torgue On crank Vs One revolution of crank Applying Newton-Cotes quadrature formula and composite Trapezoidal rule, Total area under this Curve is calculated which represents work done per revolutions.

Work done per revolution= $\int T\, d\theta$

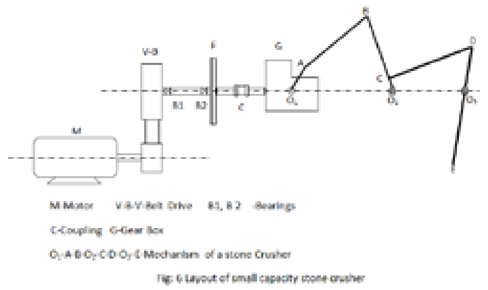
$$= \frac{\Delta\theta}{2}\,[T_1 + T_N + 2(T_2 + T_3 + \ldots\ldots\ldots\ldots\ldots\ T_{N-1})]$$

= -30051.42 Nm.rad

$$T_{mean} = \frac{\int T d\theta}{\int d\theta}$$

=-4782.83 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



Flywheel Parameters:
Type of Flywheel: Solid disc geometry with inside and outside radius
Type and Density of Material: 7200 Kg/m^2
Speed: 900 r. p. m.
Inside radius: 387 mm
Outside radius: 484 mm
Thickness: 48mm
Total mass of flywheel: 92.49Kg.

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

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

Motor:
75 KW, 4Pole, 1500rpm

Conclusion: -
Here we discussed about graphical method to synthesize and analyzed the mechanism and derived a related parameters used for proposed stone crusher. With the same logic we can develop software which will be useful for manufacturing industries to design the small capacity stone crusher.

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