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DESIGN AND DEVELOPMENT OF A ALTERNATE MECHANISM FOR STONE CRUSHER USING RELATIVE VELOCITY METHOD

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Abstract- In this paper alternate mechanism for design and analysis of small size stone crusher mechanism is proposed. 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, d'Alembert's principle, Grashoff's law, Force Analysis, Kinematic synthesis and analysis, Newton-Cotes quadrature formula, composite Trapezoidal rule, Sector gear, Static, Transmission Ratio.

I. INTRODUCTION

The proposed research is interested in providing an alternative 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 determining the feasible configuration that a kinematic chain can adopt within the given 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) Double Rocker 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 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 $\Theta$. $B_1$ 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

$$T.R = \frac{180 + \Theta}{180 - \Theta} \quad \text{…………… (1)}$$

The most common design problem in which, the angle of oscillation $\Theta$ and angle $\alpha$ (or the time ration, which determines $\alpha$) 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, $\alpha = 12.558^0$ is determined.

Detailed synthesis of the mechanism is carried out by Geometrical method and optimum parameters are obtained as follows.

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

2.2) 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.
The mechanism is analyzed graphically and ultimately the torque on the crank is computed. From the above force polygon $F_{45} = 7.9$ 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)

$$F_{45} = -F_{54} = F_{43} = F_{43} = 1.1 	ext{ tons.}$$

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

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

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

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.

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

<table>
<thead>
<tr>
<th>Srno.</th>
<th>Crank Angular Position</th>
<th>Coupler force on Link AB (in Tons)</th>
<th>Coupler force on Link CD (in Tons)</th>
<th>Torque required on Crank $O_1A$ (in Nm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>40°</td>
<td>0.9</td>
<td>6.4</td>
<td>800 Clockwise</td>
</tr>
<tr>
<td>2</td>
<td>60°</td>
<td>1.1</td>
<td>7.9</td>
<td>1042 Anti-Clockwise</td>
</tr>
<tr>
<td>3</td>
<td>90°</td>
<td>1.0</td>
<td>7.4</td>
<td>1430 Anti-Clockwise</td>
</tr>
<tr>
<td>4</td>
<td>120°</td>
<td>0.9</td>
<td>6.8</td>
<td>2700 Anti-Clockwise</td>
</tr>
<tr>
<td>5</td>
<td>150°</td>
<td>1.3</td>
<td>8.0</td>
<td>2600 Anti-Clockwise</td>
</tr>
</tbody>
</table>
Above table indicates that maximum force in a revolution of crank on coupler AB is 1.8 Tons and on Coupler CD is 8 Tons.

IV. 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.

\[ F + (-ma_G) = 0 \]  
\[ T_{eg} + (-I_G \alpha) = 0 \]

Where \( e \) 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 \( F_i \), as

\[ F_i = -ma_G \]

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 \( C_i \), is given by:

\[ C_i = -I_G \alpha \]

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:

\[ \sum F = \sum F_e + F_i = 0 \]

\[ \sum T_G = \sum T_{eg} + C_i = 0 \]

V. CALCULATIONS

\[ \alpha_2 = \frac{a_{10}^2}{t^2 \cdot a_{20}} = \frac{4900}{195} = 25.128 \text{rad/s}^2 \]

\[ a_{G2} = 3300 \text{N/cm}^2 \]

Allowable Strength = \[ S \times \text{Factor X Ignorance F} \times \text{Reliability Factor} \times \text{Stress concentration Factor} \]

\[ = \frac{16800 \times 0.5 \times 0.5 \times 0.85 \times 2.2}{2245.989} \text{N/cm}^2 \]

But,

\[ \text{Allowable Strength} = \frac{\text{Max Force on link AB}}{\text{Cross section of link AB}} \]

\[ = \frac{24.34}{8.03} \text{cm} \]

Selecting cross section of link AB as 4cm x 4cm ,

Similarly cross section of link CD as 8cm x 8cm

\[ I_{G2} = \frac{1}{12} \times \text{Mass of Link AB} \times \text{Volume of Link} \]

\[ = 7.8 \times 10^3 \times 3 \times 3 \times 195 = 24.34 \text{ Kg} \]

\[ I_{G2} = \frac{1}{12} \times \text{Mass of Link AB} \times \text{Volume of Link} \]

\[ = 77114.7 \text{Kg-cm} \]

Similarly we have calculated,

\[ \alpha_3 = \frac{a_{20}}{20^2} \]

\[ O_2B = 4000/100 = 62.5 \text{rad/s}^2 \]
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\( a_{G3} = 2000 \text{ N/cm}^2 \) ……………………From acceleration diagram

\( m_3 = \text{Mass of Link O}_2\text{B} = 12.48 \text{ Kg.} \)

\( I_{G3} = \text{Mass moment of Inertia of Link O}_2\text{B @ C.G.} = 10400 \text{ Kg-cm}^2 \)

Similarly we have calculated

\( m_3 = \text{Mass of Link CD} = 51.9 \text{ Kg.} \)

\( m_3 = \text{Mass of Link O}_3\text{D} = 24.96 \text{ Kg.} \)

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 \( F_2 = 80322 \text{ N} \).

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

\[
h_2 = \frac{(1)7114.7 \times 25.128}{(24.34 \times 3300)} = 24.124 \text{ cm}
\]

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 \( F_3 = 22113 \text{ N} \) at an offset distance

\[
h_3 = \frac{(10400\times 40)}{(12.48 \times 2000)} = 16.67 \text{ cm}
\]

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

Taking moment @ \( O_1 \)

\[
= (F_2 h_2) + (F_3 h_3) \quad \Rightarrow \quad (80322 \times 8) + (22113 \times 41) = 380784 \text{ N-cm} = 3807.84 \text{ N-m. (Clock wise )}
\]

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

Shown in Tabulated form

<table>
<thead>
<tr>
<th>Sr. No</th>
<th>Crank Angular Position (( \theta ))</th>
<th>Torque required on Crank ( O_1A ) Considering Static Loading (in Nm)</th>
<th>Torque required on Crank ( O_1A ) Considering Dynamic Loading (in Nm)</th>
<th>Net Torque on Crank ( O_1A ) (in Nm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>40°(Loaded Condition)</td>
<td>800 Clockwise</td>
<td>66858.56 Clockwise</td>
<td>66058.56 Clockwise</td>
</tr>
<tr>
<td>2</td>
<td>60°(Loaded Condition)</td>
<td>1042 Anti-Clockwise</td>
<td>3807.84 Clockwise</td>
<td>2765.84 Clockwise</td>
</tr>
<tr>
<td>3</td>
<td>90°(Loaded Condition)</td>
<td>1430 Anti-Clockwise</td>
<td>36576 Anti-Clockwise</td>
<td>38006 Anti-Clockwise</td>
</tr>
<tr>
<td>4</td>
<td>120°(Loaded Condition)</td>
<td>2700 Anti-Clockwise</td>
<td>90914.4 Anti-Clockwise</td>
<td>93614.4 Anti-Clockwise</td>
</tr>
<tr>
<td>5</td>
<td>150°(Loaded Condition)</td>
<td>2600 Anti-Clockwise</td>
<td>79200.54 Anti-Clockwise</td>
<td>81800.54 Anti-Clockwise</td>
</tr>
<tr>
<td>6</td>
<td>180°(Loaded Condition)</td>
<td>960 Anti-Clockwise</td>
<td>46380.72 Anti-Clockwise</td>
<td>47340.72 Anti-Clockwise</td>
</tr>
<tr>
<td>7</td>
<td>225°(No Load Condition)</td>
<td>0 Anti-Clockwise</td>
<td>8192.94 Anti-Clockwise</td>
<td>8192.94 Anti-Clockwise</td>
</tr>
<tr>
<td>8</td>
<td>270°(No Load Condition)</td>
<td>0 Anti-Clockwise</td>
<td>63422.92 Anti-Clockwise</td>
<td>63422.92 Anti-Clockwise</td>
</tr>
<tr>
<td>9</td>
<td>315°(No Load Condition)</td>
<td>0 Anti-Clockwise</td>
<td>55730.28 Anti-Clockwise</td>
<td>55730.28 Anti-Clockwise</td>
</tr>
<tr>
<td>10</td>
<td>360°(No Load Condition)</td>
<td>0 Anti-Clockwise</td>
<td>155355.6 Clockwise</td>
<td>155355.6 Clockwise</td>
</tr>
</tbody>
</table>

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

Table shows Total torque at various Crank Positions.

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

<table>
<thead>
<tr>
<th>Sr. No</th>
<th>Crank Position</th>
<th>Angular Torque required on Crank ( O_1A ) Considering Dynamic Loading (in Nm)</th>
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<td>10</td>
<td>360°</td>
<td>155355.6 Clockwise</td>
</tr>
</tbody>
</table>

Net Torque on the crank is calculated and the actual torque required in one revolution of crank is tabulated in the following graph.
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Applying Newton-Cotes quadrature formula and composite Trapezoidal rule, Total area under this Curve is calculated which represents work done per revolutions.

\[
\text{Work done per revolution} = \int T \, d\theta = \frac{\Delta \theta}{2} \left[ T_1 + T_N + 2(T_2 + T_3 + \ldots + T_{N-1}) \right] \quad (21)
\]

\[
\Delta \theta = 174071.768 \text{ Nm.rad (Anti-clockwise)}
\]

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

**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/m³
- Speed: 900 r.p.m.
- Inside radius: 46.7 cm
- Outside radius: 58.4 cm
- Thickness: 5.84cm
- Total mass of flywheel: 162.43 Kg

**Worm Gear Box:**
- Specifications: Helical Gear Box, Speed Reduction ratio 7.5:1, Power: 498 KW
- 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: 425 KW, 4Pole, 1500rpm

**VI. PREVIOUS WORK**

We have proposed one more design for same capacity [ref.1], in which we proposed the similar mechanism in which selection of crank and lever mechanism was different resulting which design parameters calculated was different. Like this we have design various mechanisms for same capacities by replacing double rocker mechanism with rack and sector gear mechanism.
VII. 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 can be designed. Further by multiple regressions method using MATLAB software the best and optimum model can be obtained.

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