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# Analysis of Shaking Moments in a Slider Crank Mechanism

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**Abstract** - This paper discuss about the Analysis of slider crank mechanism. Balancing of shaking forces and moments in high speed machinery has been challenging problem for mechanisms and machine designers. In recent years machines are being operated at higher and higher speeds. Smoothness of operation is frequently dominant consideration in the design of high speed machines, but most mechanisms are not naturally smooth in their operation. The objective of balancing a mechanism is to eliminate or reduce the effect of shaking force and shaking moment the mechanism exerts upon its frame and surroundings, in order that the mechanism will attain improved dynamic wear, noise, precision of operation properties and extended fatigue life. The result of this study will provide the designer enhanced control over dynamic properties of reciprocating machinery in the design stage

***Key Words***: shaking force and moments, mechanism, reciprocating machinery

## 1.INTRODUCTION

The dynamic balancing of machinery is essential for good high speed performance. A considerable amount of research on balancing of shaking force and shaking moment in spatial mechanisms has been carried out in recent years. The complete shaking force and shaking moment balancing of linkages is a difficult task.

When operated at high speeds, the mass distribution in the links of mechanism gives rise to forces and moments which are transmitted to the ground link of the machine. This forces and moments shake the foundation upon which the machine is mounted, causing vibration disturbing people and doing structural damage to the floor and often to the entire building.

An unbalanced linkage running at high speed transmits shaking force and shaking moment to its frame. The resultant inertia force exerted on the frame is equal to the vector sum of inertia forces associated with the moving links. These forces and moments cause vibrations, fluctuations in the input torque and stresses which decrease the performance of high speed machinery

**Generalized Slider Crank Mechanism**

The generalized slider crank is actually a spatial mechanism with multi-degree-of-freedom joints, whereas the conventional slider crank universally used in reciprocating engines and compressors is a planar mechanism with single-degree-of-freedom joints. In a planar slider crank, the joint between frame and crank is revolute and has one degree of freedom; i.e. rotation only. In generalized slider crank it is a cylindrical joint which has two degrees of freedom, one of translation and the other of rotation.

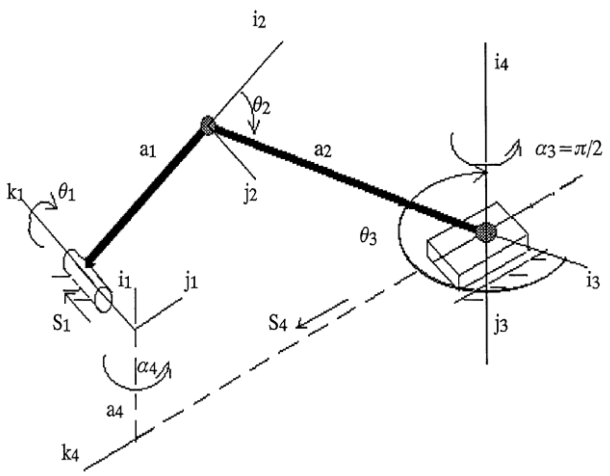


**Fig- 1:** Generalized Slider Crank with All Offsets

## 2. Body of Paper

**Need Identification**

In recent years machines are being operated at higher speeds. Smoothness of operation is frequently dominant consideration in the design of high speed machines, but most mechanisms are not naturally smooth in their operation. In the present work we are analysis a slider crank mechanism for balancing a shaking force and moments in high speed machinery

**Problem Definition**

It is required to balancing a mechanism is to eliminate or reduce the effect of shaking force and shaking moment.

- Improve the dynamic wear, precision of operation properties
- extended fatigue life

## LITERATURE REVIEW

**Himanshu Chaudhary & Subir Kumar Saha** [1]

have presented a general mathematical formulation of optimization problem for balancing of planar mechanisms. The inertia properties of mechanisms are represented by dynamically equivalent systems, referred as equimomental systems, of point-masses to identify design variables and formulate shaking forces and the shaking moments, the dynamic equations of motion for mechanisms are formulated systematically in the parameters related to the equimomental point-masses. The formulation leads to an optimization scheme for the mass distribution to improve the dynamic performances of mechanisms.

**G. Alici, B. Shirinzadeh [2]**

have presented a methodology for optimum dynamic balancing of planar parallel manipulators typified with a variable speed 2 DOF parallel manipulator articulated with revolute joints. The dynamic balancing is formulated as an optimization problem such that a sum-squared values of bearing forces, driving torques, shaking moment, and the deviation of the angular momentum from its mean value are minimized throughout an operation range of the manipulator, provided that a set of balancing constraints consisting of the shaking force balancing conditions, the sizes of some inertial and geometric parameters are satisfied. Sets of optimization results corresponding to various combinations of the elements of the objective function are evaluated in order to quantify their influence on the resulting bearing forces, the driving torques, shaking moment and force. The results prove that the proposed optimization approach can be used to minimize any desired combination of the forces, moments, and torques involved in any parallel mechanism by choosing a suitable set of weighting factors.

## ANALYSIS OF MECHANISM

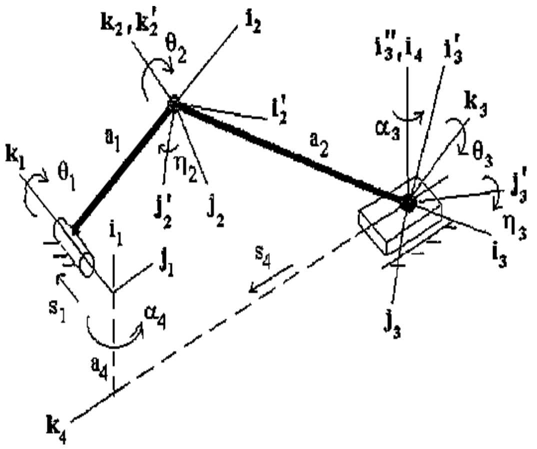

**Figure.2** Generalized Slider Crank Mechanism

**Fixed Coordinate Frame**

As seen in figure 2 coordinate frame {1} is the fixed coordinate frame, it does not move when the mechanism works. All forces and torques are expressed in terms of frame {1}.

**Moving Coordinate Frames**

The coordinate frame {2} is located at the distal end of the crank, frame {3} is located at the distal end of the connecting rod and frame {4} is located at the distal end of the slider. These are the moving coordinate frames. They move with respect to the fixed frame {1}.

**D-H Parameters**

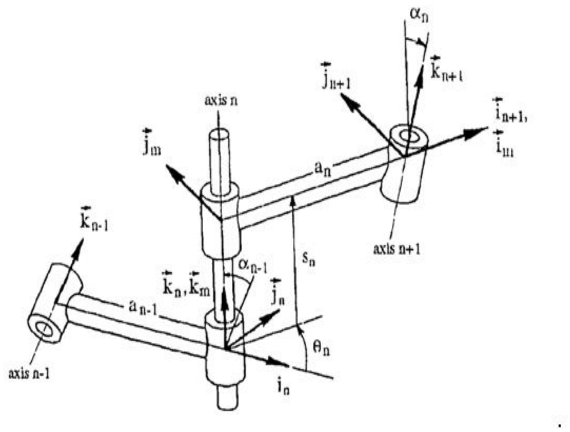

**Figure 3**: Generalized model of a link connecting two joints which are cylindrical, prismatic or revolute

The link-joint parameter table is shown in table 1. This table describes the complete geometry of the mechanism.

**Table No 1**: D-H parameters

| $i$ | $\theta_i$ | $S_i$ | $\eta_i$ | $\alpha_i$ | $a_i$ |
|---|---|---|---|---|---|
| 1 | $\theta_1$ | $S_1$ | 0 | 0 | $a_1$ |
| 2 | $\theta_2$ | 0 | $\eta_2$ | 0 | $a_2$ |
| 3 | $\theta_3$ | 0 | $\eta_3$ | $\pi/2$ | 0 |
| 4 | 0 | $S_4$ | 0 | $\alpha_4$ | $a_4$ |

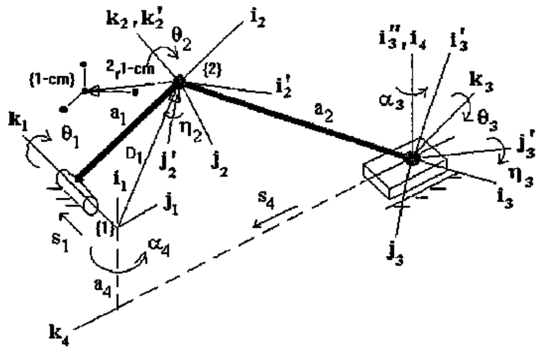


**Figure 4:** Crank is replaced by four point masses; $D_1$ is the position vector representing distance of distal frame {2} of the crank from frame {1}

The link parameters are:

$a_1$ = crank length

$a_2$ = connecting rod length

$\alpha_3$ = twist angle between two joint axes-the axis of the joint between connecting rod and slider i.e. axis $k_3$(not shown) and the axis of the joint between slider and frame i.e, axis $k_4$; the angle is measured counterclockwise according to right- hand rule and equal to 90 degrees.

$a_4$ = offset distance between two joint axes, one is the axis of the joint between frame and crank i.e axis $k_1$ and the another is the axis of the joint between the slider and the frame i.e axis axes $k_4$. Axis $k_1$ and $k_4$ are mutually perpendicular axes and the distance $a_4$ is measured from $k_4$ to $k_1$.

$A_4$= twist angle between two joint axes $k_1$ and $k_4$; the angle is measured counter clockwise According to right-hand rule. For the ideal (planar) slider crank this angle is 270 degrees.

The joint variables are:

$\theta_1$, $\theta_2$ and $\theta_3$ = angular displacements about joint axes at joints 1,2 and 3 respectively.

$\eta_2$ = angular displacement about the axis $j_2$, perpendicular to the joint axis as well as the link axis at joint 2.

$\eta_3$= angular displacement about the axis $j_3$, perpendicular to the joint axis as well as the link axis at joint 3 i.e. axis $j_3$.

$S_1$=displacement of the crank along the axis of joint 1 i.e. axis $k_1$.

$S_4$=slider displacement which is measured about the axis $k_4$.

The acceleration of half point masses $m_{11}/2, m_{12}/2, m_{13}/2$ and point mass $m_{14}$ can respectively be written as follows. Since they are stationary with respect to distal coordinate frame {2}, first and second time derivatives of their distances from {2} are zero.

$$a_{11,P} = \omega_1 \times (\omega_1 \times {}^2d_{11,P}) + \frac{d\omega_1}{dt} \times {}^2d_{11,P} + \ddot{D}_1$$

$$a_{11,N} = \omega_1 \times (\omega_1 \times {}^2d_{11,N}) + \frac{d\omega_1}{dt} \times {}^2d_{11,N} + \ddot{D}_1$$

$$a_{12,P} = \omega_1 \times (\omega_1 \times {}^2d_{12,P}) + \frac{d\omega_1}{dt} \times {}^2d_{12,P} + \ddot{D}_1$$

$$a_{12,N} = \omega_1 \times (\omega_1 \times {}^2d_{12,N}) + \frac{d\omega_1}{dt} \times {}^2d_{12,N} + \ddot{D}_1$$

$$a_{13,P} = \omega_1 \times (\omega_1 \times {}^2d_{13,P}) + \frac{d\omega_1}{dt} \times {}^2d_{13,P} + \ddot{D}_1$$

$$a_{13,N} = \omega_1 \times (\omega_1 \times {}^2d_{13,N}) + \frac{d\omega_1}{dt} \times {}^2d_{13,N} + \ddot{D}_1$$

$$a_{14} = \omega_1 \times (\omega_1 \times {}^2d_{14}) + \frac{d\omega_1}{dt} \times {}^2d_{14} + \ddot{D}_1$$

The acceleration of half point masses $m_{21}/2, m_{22}/2, m_{23}/2$ and point mass $m_{24}$ can respectively be written as follows. Since they are stationary with respect to distal coordinate frame {3}, first and second time derivatives of their distances from {3} are zero.

$$a_{21,P} = \omega_2 \times (\omega_2 \times {}^3d_{21,P}) + \frac{d\omega_2}{dt} \times {}^3d_{21,P} + \ddot{D}_2$$

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$$a_{21,N} = \omega_2 \times (\omega_2 \times {}^3d_{21,N}) + \frac{d\omega_2}{dt} \times {}^3d_{21,N} + \ddot{D}_2$$

$$a_{22,P} = \omega_2 \times (\omega_2 \times {}^3d_{22,P}) + \frac{d\omega_2}{dt} \times {}^3d_{22,P} + \ddot{D}_2$$

$$a_{22,N} = \omega_2 \times (\omega_2 \times {}^3d_{22,N}) + \frac{d\omega_2}{dt} \times {}^3d_{22,N} + \ddot{D}_2$$

$$a_{23,P} = \omega_2 \times (\omega_2 \times {}^3d_{23,P}) + \frac{d\omega_2}{dt} \times {}^3d_{23,P} + \ddot{D}_2$$

$$a_{23,N} = \omega_2 \times (\omega_2 \times {}^3d_{23,N}) + \frac{d\omega_2}{dt} \times {}^3d_{23,N} + \ddot{D}_2$$

$$a_{24} = \omega_2 \times (\omega_2 \times {}^3d_{24}) + \frac{d\omega_2}{dt} \times {}^3d_{24} + \ddot{D}_2$$

***Concept of mechanism***

. The inertia forces and torques exerted on the frame link by the moving links of the mechanism will be determined. By considering the inertia forces and external forces as applied forces acting on the system it is possible to apply D'Alembert's principle and reduce the analysis to the application of static equilibrium conditions. The mass distribution of the moving links will be replaced by a dynamically equivalent system of point masses. After calculation of their vector coordinates and accelerations, the inertia forces and torques will be obtained as well.

**ANALYTICAL CALCULATIONS**

Determination of Shaking Force and Shaking Moment

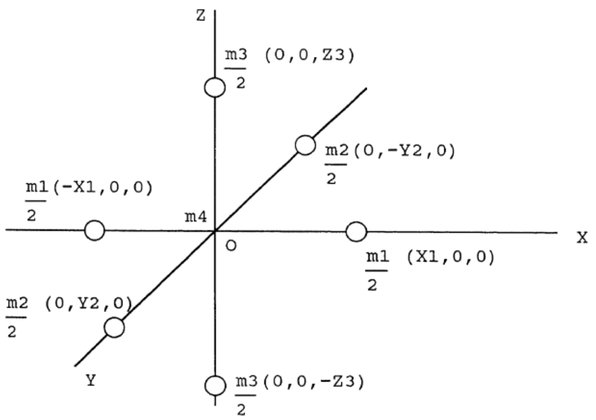



**Figure 5:** Co-ordinate system of point masses

The shaking force is the sum of the forces exerted upon the frame by each moving link

$$F = P_1 + P_2 + P_3$$

$$= \frac{m_{11}(a_{11,P} + a_{11,N}) + m_{12}(a_{12,P} + a_{12,N}) + m_{13}(a_{13,P} + a_{13,N})}{2} + m_{14}a_{14}$$

$$+ \frac{m_{21}(a_{21,P} + a_{21,N}) + m_{22}(a_{22,P} + a_{22,N}) + m_{23}(a_{23,P} + a_{23,N})}{2} + m_{24}a_{24}$$

$$+ m_3\ddot{D}_3 \qquad (4.$$

The shaking moment is the sum of the torque exerted upon the frame by each moving link

$$T = T_1 + T_2 + T_3$$

$$= \frac{(D_{11,P} \times m_{11}a_{11,P}) + (D_{12,P} \times m_{12}a_{12,P}) + (D_{13,P} \times m_{13}a_{13,P})}{2}$$

$$+ \frac{(D_{11,N} \times m_{11}a_{11,N}) + (D_{12,N} \times m_{12}a_{12,N}) + (D_{13,P} \times m_{13}a_{13,N})}{2}$$

$$+ (D_{14} \times m_{14}a_{14})$$

$$+ \frac{(D_{21,P} \times m_{21}a_{21,P}) + (D_{22,P} \times m_{22}a_{22,P}) + (D_{23,P} \times m_{23}a_{23,P})}{2}$$

$$+ \frac{(D_{21,N} \times m_{21}a_{21,N}) + (D_{22,N} \times m_{22}a_{22,N}) + (D_{23,N} \times m_{23}a_{23,N})}{2}$$

$$+ (D_{24} \times m_{24}a_{24})$$

$$+ D_3 \times m_3\ddot{D}_3$$

where all the variables except point masses $m_{11}, m_{12}, m_{13}, m_{14}, m_{21}, m_{22}, m_{23}, m_{24}$ and $m_3$ are known from kinematic analysis.

**RESULTS AND DISCUSSION**

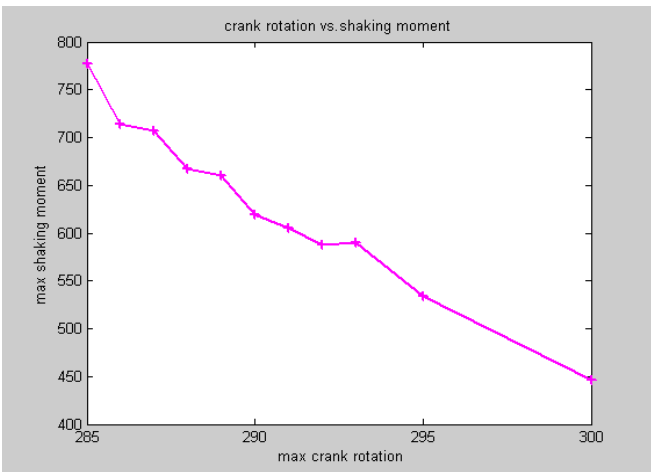
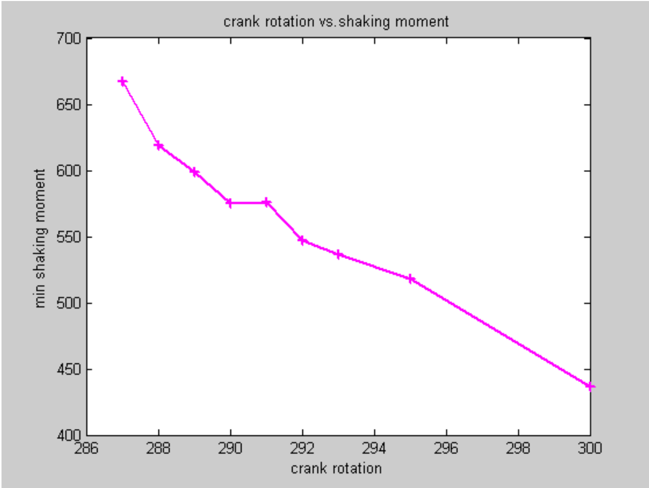


**Table 2:** Mass Set No. Vs Crank Angle

| MASS SET NO | CRANK ANGLE | | | |
|---|---|---|---|---|
| | **286** | **287** | **288** | **289** |
| 1 | SM =538.742 | SM =538.885 | SM =539.007 | SM =539.109 |
| 2 | SM =517.322 | SM =517.489 | SM =517.638 | SM =517.768 |
| 3 | SM =556.326 | SM =556.474 | SM =556.599 | SM =556.703 |
| 4 | SM =444.549 | SM =444.784 | SM =445.006 | SM =445.216 |
| 5 | SM =551.676 | SM =551.819 | SM =551.941 | SM =552.041 |
| 6 | SM =622.588 | SM =622.666 | SM =622.717 | **SM =622.739** |
| 7 | SM =554.461 | SM =554.590 | SM =554.697 | SM =554.783 |
| 8 | SM =604.451 | SM =604.563 | SM =604.649 | SM =604.708 |
| 9 | SM =598.336 | SM =598.401 | SM =598.441 | **SM =598.455** |
| 10 | SM =434.032 | SM =434.268 | SM =434.493 | SM =434.706 |
| 11 | SM =620.632 | SM =620.713 | SM =620.766 | **SM =620.792** |
| 12 | SM =698.012 | **SM =698.040** | SM =698.034 | SM =697.993 |
| 13 | **SM =776.941** | SM =776.896 | SM =776.810 | SM =776.683 |
| 14 | SM =660.569 | SM =660.637 | SM =660.675 | **SM =660.681** |
| 15 | SM =535.719 | SM =535.860 | SM =535.9800 | SM =536.080 |




## CONCLUSION

1. In the recent studies on parallelopiped model by Himanshu Chaudhary and Subir Kumar Saha (2008) negative masses are also allowed. But we have taken all positive masses and compared the shaking moment values for best 15 sets among 1000 sets of point masses and mass set no 13 is found to be giving a maximum value of shaking moment.
2. The maximum & minimum shaking moments at which corresponding set no'(12,13,14& 4,10,02).
3. The variation of shaking moment with crank angle from 281° to 300° for these sets is graphically represented

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