Shaking Force and Shaking Moment Balancing of Planar Mechanisms with High Degree of Complexity

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Abstract

Most of the research on the balancing of shaking force and shaking moment generated by planar linkages was limited to mechanisms with low degree of complexity. This paper attempts for complete shaking force and shaking moment balancing of planar mechanisms with high degree of complexity. Shaking force is balanced by the method of redistribution of mass and shaking moment by adding gear inertia counterweights. The method is illustrated for Stephenson’s linkage (Mechanism with high degree of complexity) and Atkinson engine mechanism and also for Self-balanced slider-crank mechanical systems. The conditions for shaking moment balancing are formulated by using the copying properties of the pantograph linkage and the method of dynamic substitution of distributed masses by concentrated point masses. These mechanical systems find a successful application in engines, agricultural machines and in various automatic machines.

Keywords: Shaking force; Shaking moment; Dynamic balancing

1. Introduction

Mechanisms particularly those that run at high speeds generate variable forces on their foundations. These forces cause noise, vibration, and unnecessary wear and fatigue. The balancing of a linkage would eliminate these undesirable qualities and maintains a peaceful and productive environment. Therefore the problems of shaking force and shaking moment balancing have attracted the attention of machine and mechanism designers for a long time.

One of the most effective methods for the reduction of these vibrations is the mass balancing of moving links of mechanism by Lowen and Berkof [1]. The effective method for balancing slider-crank mechanism was the method of duplicating mechanism [2, 3] by adding to the initial mechanism an identical mechanism which is a revolved mirror reflection of the initial mechanism. The disadvantages of such an approach are a partial balancing due to the shaking moment of inertia forces of the slider, as well as the greater friction losses due to the additional sliding pair. The method of adding idler loops can be used to entirely eliminate forces and moments of 4-bar 6-bar linkages [4], Kamenski [5] first used the cam mechanism for balancing of linkages. P.Nehemia and Dr. B.S.K.Sundara Rao [6] used a method to balance shaking moment by mounting gear inertia counterweights on the frame, the planetary gear trains mounted on the links that are not connected directly to the frame in earlier methods are mounted on base by kinematically linking the gears with the corresponding links by a link of known mass and center of mass and moment of inertia. A more referred method in the literature is the method of linearly independent vectors [7], which makes total center of mass of the mechanism stationary. I.S.Kochev [8] presented a general method using ordinary vector algebra instead of the complex number representation of the vector for full force balance of planar linkages. Elliott and Tesar [9] developed a theory of torque, shaking force, and shaking moment balancing by extending the method of linearly independent vectors. R.S.Berkof [10] proposed a method to balance shaking moment by inertia counterweight and physical pendulum.


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and moment balancing constraints are written as algebraic equations over complex variables and joint angular velocities. Using polynomial division, necessary and sufficient conditions for the balancing of planar four-bar are derived. The present work deals with the balancing of mechanisms with high degree of complexity, Atkinson engine mechanism and self-balanced slider-crank mechanical systems. The present work can be the extension of work contributed by V.H. Arakelian and M.R. Smith [13], where they did for single slider-crank mechanism, mechanism with low degree of complexity. In the present work, two identical slider-crank mechanism, mechanism with high degree of complexity, Atkinson engine mechanism, where slider-crank mechanism is an integral part of it are balanced.

1.1. Definition: Mechanisms with Low and High degree of complexity:

In complex mechanisms some radii of curvatures, required for the computation of normal acceleration components are not readily available and consequently, indirect or special methods of solution must be used. In a complex mechanism if only one radius of path curvature of one motion transfer point is not known such a mechanism is called a mechanism with low degree of complexity. In the mechanism shown in fig. 1 the radius of curvature of motion transfer point B is not known, so it is a mechanism with low degree of complexity.

In a complex mechanism if more than one radii of path curvature of motion transfer points are not known such a mechanism is called a mechanism with high degree of complexity. In the mechanism shown in fig. 2 the radii of curvature of motion transfer points B and C are not known, so it is a mechanism with high degree of complexity.

2. Complete Shaking Force and Shaking Moment Balancing of Sub Linkages

2.1. Articulation dyad:

An open kinematic chain of two binary links and one joint is called a dyad.

Figure 1: Mechanism with low degree of complexity.

Figure 2: Mechanism with high degree of complexity.

Figure 3: Complete shaking force and shaking moment balancing of an articulation dyad by gear inertia counterweights mounted on the base.

Figure 4: Complete shaking force and shaking moment balancing of an articulation dyad by gear inertia counterweights mounted on the base.

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2. Complete Shaking Force and Shaking Moment Balancing of Sub Linkages

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To link 2 is added a counterweight which permits the displacement of the center of mass of link 2 to joint A, then, by means of a counterweight with mass \( m_{cw1} \) [fig.3] a complete balancing of shaking force is achieved. A complete shaking moment balance is realized through four gear inertia counter weights 3-6, one of them being of the planetary type and mounted on link 2 (Gao Feng, 1990[19]).

The scheme used in the present paper [fig.4] is distinguished from the earlier scheme by the fact that gear
Let us consider the complete shaking force and shaking moment. The articulation dyad with the mass and inertia of link 1' taken into account. For this purpose initially, we shall statically replace mass $m'_1$ of link 1' by two point masses $m_B$ and $m_c$ at the centers of the hinges B and C.

\[ m_a = m_1 l_{CS1} / l_{BC} \]
\[ m_c = m_1 l_{BS1} / l_{BC} \]  

(1)

where, \( l_{BC} \) is the length of link 1, \( l_{CS1} \) and \( l_{BS1} \) are the distances between the centers of joints C and B and the center of mass \( S'_1 \) of link 1', respectively. After such an arrangement of masses, the moment of inertia of link 1' will be equal to

\[ I'_S = I_S - m_1 l_{BS1} l_{CS1} \]  

(2)

where, \( I_S \) is the moment of inertia of link 1 about the center of mass \( S'_1 \) of the link.

Thus we obtain a new dynamic model of the system where the link 1' is represented by two point masses \( m_B, m_C \) and has a moment of inertia \( I'_S \). This fact allows for an easy determination of the parameters of the balancing elements as follows:

\[ m_{cw} = (m_2 l_{AB} + m_a l_{AB}) / r_{cw} \]  

(3)

where, \( m_2 \) is the mass of link 2, \( l_{AB} \) is the distance between the centers of the hinges A and B, \( l_{AS2} \) is the distance of the center of hinge A from the center mass of \( S_2 \) of link 2, \( r_{cw} \) is the rotation radius of the center of mass of the counterweight with respect to A, and

\[ m_{cw} = (m_2 + m_{cw} + m_a l_{OS} + m_1 l_{OS}) / r_{cw} \]  

(4)

where, \( m_1 \) is the mass of link 1, \( l_{OS} \) is the distance of the joint center O from the center mass \( S'_1 \) of link 1.

Also,

\[ m_{cw} = m_1 l_{OC} / r_{cw} \]  

(5)

where, \( l_{OC} = l_{AB} \cdot r_{cw} \) is the rotation radius of the center of mass of the counterweight.

2.2. Asymmetric link with three rotational pairs:

In previous work relating to balancing of linkages with a dynamic substitution of the masses of the link, three rotational pairs (see fig. 5) two replacement points A and B are considered. This results in the need to increase the

3 is mounted on the base and is linked kinematically with link 2 through link 1'. Let us consider the complete shaking force and shaking moment. The articulation dyad with the mass and inertia of link 1' taken into account. For this purpose initially, we shall statically replace mass \( m'_1 \) of link 1' by two point masses \( m_B \) and \( m_c \) at the centers of the hinges B and C.

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3.1. Stephenson’s link motion (Mechanism with high degree of complexity):

The method has been applied to a mechanism with high degree of complexity shown in fig.6.

Figure 6: Mechanism with high degree of complexity (Stephenson’s link motion).

3.1.1. Shaking force balancing of the mechanism:

Link 5 has been replaced by dynamic substitution of link masses by three point masses and attached a counterweight . For link 6 to be dynamically replaced by two point masses the condition to be satisfied is , where, is the radius of gyration of link 6 about its center of mass, is arbitrarily fixed and is obtained from the above condition. Similarly other links can be dynamically replaced and force counterweights can be added to balance shaking force.

3.1.2. Shaking moment balancing of the mechanism:

The shaking moment of the mechanism is determined by the sum

\[ M_{m} = M_{m}^b + M_{m}^c + M_{m}^d + M_{m}^e + M_{m}^f \]  

Where

\[ M_{m}^b = (I_{b,k} + m_{b,k}l_{b,k}^2 + m_{c,b}l_{c,b}^2 + m_{w_b}l_{w_b}^2)\alpha_k \]
\[ M_{m}^c = (I_{c,k} + m_{c,k}l_{c,k}^2 + m_{c,\alpha}l_{c,\alpha}^2 + m_{c,\beta}l_{c,\beta}^2 + m_{c,\gamma}l_{c,\gamma}^2)\alpha_k \]
\[ M_{m}^d = (I_{d,k} + m_{d,k}l_{d,k}^2 + m_{d,\alpha}l_{d,\alpha}^2 + m_{d,\beta}l_{d,\beta}^2 + m_{d,\gamma}l_{d,\gamma}^2)\alpha_k \]
\[ M_{m}^e = (I_{e,k} + m_{e,k}l_{e,k}^2 + m_{e,\alpha}l_{e,\alpha}^2 + m_{e,\beta}l_{e,\beta}^2 + m_{e,\gamma}l_{e,\gamma}^2)\alpha_k \]
\[ M_{m}^f = (2m_f l_{f,k}^2 + 2m_f l_{f,\alpha}^2)\alpha_k \]

3.2. Atkinson engine mechanism:

To the mechanism an articulation dyad CFE is connected, which forms a pantograph with the initial mechanism OBCD. By selecting, for constructional reasons, the similarity factor of formed pantograph

\[ K = \frac{l_{c,E}}{l_{b,B}} = \frac{l_{b,B}}{l_{h,B}} = \frac{l_{h,B}}{l_{h,D}} \]  

(9)

The length of the articulation dyad is determined as:

\[ l_{cF} = l_{b,B} + l_{h,D} \]
\[ l_{HM} = l_{H} + l_{HM} = l_{HC} + kl_{HD} \]

3.2.1. Shaking force balance:

Link 3 is dynamically replaced by 3 point masses \( m_{A3}, m_{B3}, m_{C3} \). Link 4 is dynamically replaced by two point masses \( m_{B4}, m_{E4} \) and a force counterweight \( m_{CW_4} \), is added to balance the shaking force. Link 5 is statically replaced by two point masses \( m_{Cx}, & m_{Dx} \). An articulation dyad CFE is added. Now link 4 (\( m_B \ & m_F \)) is to be balanced about point 04. Link 7 (\( m_C \ & m_F \)) about point G. Finally the masses
The center of mass of link 7, with the point masses \( m_7 \), \( m_8 \), \( m_{11} \), \( m_{12} \) are taken into account, coincides with the working point \( G \) of the pantograph.

Where \( l_{0,B} \), \( l_{0,E} \) and \( l_{0,S4} \) are the distances of joint centers \( B \), \( E \), and of the center of mass \( S4 \) of the link 4 from the point \( O4 \). \( l_{CG} \), \( l_{FG} \) are the distances of the centers of the joints \( C \) \( F \) from the working point \( G \) of the pantograph.

\( I_{CS7} \) is the distance of the center of the joint \( C \) from the center of mass \( S7 \) of Link 7. \( l_{BD} \), \( l_{IBC} \) are the distances of the centers of the joints \( D \), \( C \) from the center of joint \( B \). \( l_{FF} \), \( l_{FH} \) are the distances of the centers of the joints \( E \), \( H \) from the center of joint \( F \). \( m_7 \) is the mass of link 7, \( m_{F} \), \( m_{E} \) are point masses obtained after dynamic substitution. \( m_x \) is the mass of link 8. \( l_{FS8} \) is the distance of the center of the joint \( F \) from the center of mass \( S8 \) of link 8. \( l_{S8} \) is the axial moment of inertia of link 8.

The desired parameters are obtained as follows:

\[
m_h = m_x + m_F + m_H,
I_{FS8} = \frac{(m_E l_{FS8}^2 + m_{FG} l_{FG}^2)}{m_h}
I_{S8} = m_F l_{S8}^2 + m_E (l_{S8} - l_{FD}) - m_h (l_{FG} - l_{S8})
\]

Where

\[
m_F = \left[ m_B l_{0,B} + m_D l_{0,D} \right] / l_{0,E}
m_E = \left[ m_A l_{0,A} + m_C l_{0,C} \right] / l_{0,E}
\]

Thus, a dynamic model of the mechanism fully equivalent to the real mechanism involving the rotating link 4 and 7 [The parameters of link 8 are selected so that the center of mass of link 7, with the point masses \( m_C \), \( m_B \) taken into account, coincides with the working point \( G \) of the pantograph, due to which the motion of this link is represented as a translational rectilinear motion of its center of mass and a rotary motion relative to point \( G \) and four point masses \( m_0 \), \( m_3 \), \( m_4 \), \( m_5 \) three of which perform a translational rectilinear motion in horizontal sense is obtained. As may be seen from this equivalent model, a complete shaking force balancing of the movable links of the mechanism has been achieved.

The shaking moment at point 0 is determined by the sum

\[
M_0 = M_4 + M_7 + m_0 (F_{int})
\]

The moments of rotating links may be balanced by means of the gears mounted on the base of the mechanism. The moment of inertia of such a gear is given by the following equation:

\[
I_{gear} = I_S + m_F l_{n,F}^2 + m_{DG} l_{DG}^2 + m_E l_{NS}^2
\]

where \( \alpha = \alpha_4 = \alpha_7 \) is the angular acceleration of links 4 and 7. \( M_{OA}(F_{int}) \) is the moment resulting from the force of inertia of the masses and \( m_H \) performing a translational rectilinear motion relative to pivot \( O4 \). \( m_6, m_9, m_6 \) are the shaking moments of the rotating links 4 and 7 with the inertia of the replaced point masses taken into account.

The shaking moment of the mechanism is determined by the sum

\[
M_{int} = M_4 + M_7 + m_0 (F_{int})
\]

Where \( M_4 \) and \( M_7 \) are the shaking moments of the rotating links 4 and 7 with the inertia of the replaced point masses taken into account.

The shaking moment at point 0 is determined by the sum

\[
M_0 = M_4 + M_7 + m_0 (F_{int})
\]

where \( F_{int} \) is the equivalent inertia forces from the center of mass and a rotary motion relative to point \( G \) and

\[
F_H = F_D + F_6 + \left( F_C + F_F + F_4 \right)
\]

where \( F_{int} \) is the equivalent inertia forces from the center of mass and a rotary motion relative to point \( G \) and

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where \( F_{int} \) is the equivalent inertia forces from the center of mass and a rotary motion relative to point \( G \) and

\[
M_0 = M_4 + M_7 + m_0 (F_{int})
\]
3.3. Self-balanced slider- crank mechanism:

In the two identical slider-crank mechanism shown in fig.10 shaking forces are balanced by two similar but opposite movements.

![Image](https://via.placeholder.com/150)

**Figure 10:** Self balanced slider – crank system.

![Image](https://via.placeholder.com/150)

**Figure 11:** Self-balanced slider-crank system with an imagined articulation dyad \( B'D'E \).

Fig.11 shows a self-balanced slider-crank system with an imagined articulation dyad \( B'D'E \), which forms a pantograph with the initial system. The similarity factor of the formed pantograph is \( k = l_{AD}/l_{AB} = 1 \) and \( l_{BF} = l_{DE}/l_{EF} = l_{AD} + l_{AB} \).

By substituting dynamically the mass \( m_3 \) of the connecting coupler 3 by point masses at the centers \( B, B' \) and \( C \) and using the following condition

\[
\begin{bmatrix}
1 & 1 & 1 \\
-l_{BS_2} & -l_{CS_2} & -l_{BS_3} \\
l_{BS_3} & l_{CS_3} & l_{BS_2}
\end{bmatrix}
\begin{bmatrix}
m_B \\
m_C \\
m_{B'}
\end{bmatrix}
= \begin{bmatrix}
m_3 \\
0 \\
-I_{S_3}
\end{bmatrix}
\]

(16)

where \( l_{BS_2}, l_{CS_3}, l_{BS_3} \) are the distances of joint centers B, C and \( B' \) from the centers of masses \( S_3 \) of the link 3; \( I_{S_3} \) is the axial moment of inertia of link 3, we determine the value of the point masses \( m_B = D_B / D_1; m_C = D_C / D_2; m_{B'} = D_{B'} / D_3 \) (17)

where \( D_B, D_1, D_{B'}, D_C \) are determinants of the third order obtained from the system of equations.

We now require imagined link \( B'D' \) to be balanced about point \( G \) of the pantograph, i.e.,

\[ m_{EY} = m_B l_{BF} / l_{DE} \]

The concentrated point masses \( m_B, m_C, m_E \) to be balanced about center \( A \), i.e.,

\[ m_E = (m_B l_{BF} + m_C l_{BC}) / l_{DE} \]

where \( l_{BF}, l_{BC} \) are the distances of joint centers \( B, C \) from the joint center \( B, l_{DE} \) is the distance of joint center \( D \) from the joint center \( E \).

\[ m_G = m_B + m_D \]

Finally the concentrated point masses \( m_B, m_D \) are also to be balanced about center \( A \), i.e.,

\[ m_D = m_B l_{AB} / l_{AD} \]

Thus we obtain the values of three concentrated point masses \( m_B, m_D, m_E \), which allow the determination of the mass and inertia parameters of the connecting coupler 4:

\[ m_4 = m_B + m_D + m_E \]

\[ l_{ES_4} = (m_B l_{BS_4} + m_D l_{DS_4}) / m_4 \]

(19)

\[ I_{S_4} = m_B l_{BS_4}^2 + m_D l_{DS_4}^2 + m_E l_{ES_4}^2 \]

Where

\[ I_{DS_4} = l_{DE} - l_{ES_4}; l_{DS_4} = l_{DE} - l_{ES_4} \]

Shaking moment balancing:

\[ M^\text{int} + M^\text{int} = (I_{S_4} + m_B l_{BS_4}^2 + m_D l_{DS_4}^2 + m_E l_{ES_4}^2) \alpha_2 \]

(20)

\[ M^\text{int} = (I_{S_5} + m_B l_{BS_5}^2 + m_D l_{DS_5}^2 + m_E l_{ES_5}^2) \alpha_6 \]

(21)

Total Shaking moment generated by the mechanism:

\[ M^\text{int} = M^\text{int} + M^\text{int} + M^\text{int} \]

(22)

The shaking moment generated by the mechanism is balanced by addition of gear inertia counter weights 9 and 10.
4. Numerical Example

The parameters of the self-balanced slider-crank system are the following:

\[ I_{ab} = I_{de} = 0.05m; I_{dc} = I_{ed} = 0.2m; I_{cs} = I_{sc} = 1.0m; \]
\[ m_s = m_s = 0.35kg; m_3 = m_4 = 2kg; \]
\[ I_{b} = I_{s} = 0.005kg - m^2; \alpha_{ab} = 3\pi / s; \alpha_{dc} = 450 / s^2; \]
\[ m_1 = m_1 = 0.3kg; \]
\[ I_{bc} = I_{dc} = 0.025m; I_{ba} = 0.006kg - m^2; m_6 = 0.6kg \]

Figure 12 shows the variations of the shaking moment of the initial mechanical system (curve “a”). For cancellation of the shaking moment it is necessary to redistribute the masses of the second connecting coupler. By dynamically substituting the mass of the connecting coupler 3 by point masses at centres B, B’, C and taking into account conditions, we calculate the mass and inertia parameters of the connecting coupler. Fig.12 illustrates the obtained results, by optimal redistribution of the masses of the connecting coupler 4, the shaking moment is cancelled (curve ‘b’).

![Figure 12: Shaking moment Vs Time.](image)

Figure 12: Shaking moment Vs Time.

5. Conclusions

The advantage of the schemes presented here is the fact that all the gear inertia counterweights needed for balancing shaking moment of mechanism with high degree of complexity are mounted on the frame of the mechanism, which is constructively more efficient. The method can be applied to any complex planar mechanism. The paper also presents a solution for improving the balancing of double slider-crank systems. In these systems the shaking force balancing is achieved by two identical slider-crank mechanisms, which execute similar but opposite movements. However, the shaking moments are not balanced and can be a source of vibrations. By modifying the parameters of the second connecting coupler of the system the complete shaking moment balancing is achieved. The conditions for shaking moment balancing are formulated by using the copying properties of the pantograph linkage and the method of dynamic substitution of connecting rod by the concentrated point masses. A numerical example illustrates the application of the suggested solution. The method can be applied to any complex mechanism.

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