JJMIE

Volume 6, Number 1, Feb. 2012
ISSN 1995-6665
Pages 17 - 24

Jordan Journal of Mechanical and Industrial Engineering

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.Nehemiah and Dr.B.S.K.Sundara Siva 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,shak ing 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.

I.Esat,H.Bahai[11];Z.YE,M.R.Smith[12];V.H.Arakelia n and M.R.Smith[13] achieved complete moment balancing by geared inertia counterweights.More information on complete shaking moment balancing can be obtained in a critical review by I.S.Kochev[14],and Arakelian and Smith[15].D.Ilia,A.Cammarata,and R.Sinatra [16] proposed the kinematics and dynamics of a five-bar linkage using a novel and simplified approach where the dynamic balancing of mechanism is formulated and solved as an optimization problem under equality constraints.H.Chaudhary ,S.K.Saha[17] used the equimomental systems for balancing of shaking forces and shaking moments of planar mechanisms.BrianMoore,Josef,and Gosselin[18] presented a new method to determine the complete set of force and moment balanced planar four-bar linkages using complex variables to model the kinematics of the linkage,the force

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  - Volume 6, Number 1 (ISSN 1995-6665)

and moment balancing constraints are written as algebraic equations over complex variables and joint angular velocities.Using polynomial divison,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.Smitht[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.

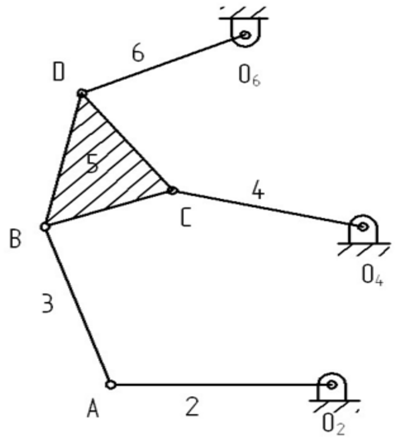

Figure 1: Mechanism with low degree of complexity.

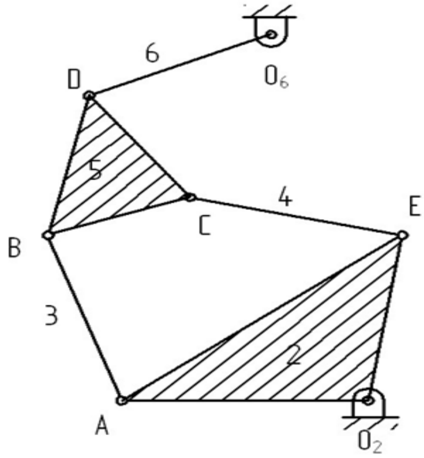

Figure 2: Mechanism with high degree of complexity.

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.

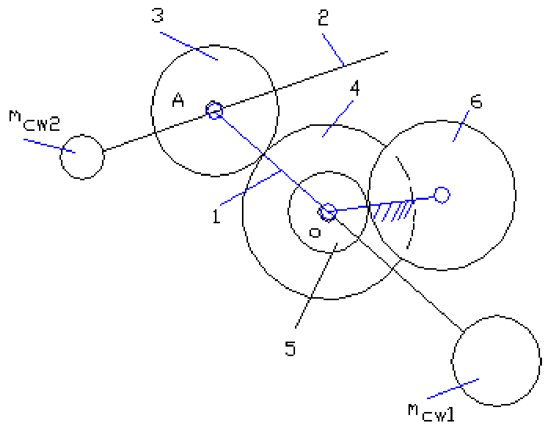

Figure 3: Complete shaking force and shaking moment balancing of an articulation dyad.

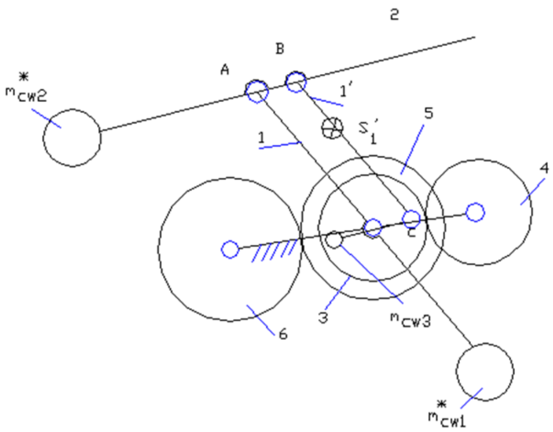

Figure 4: Complete shaking force and shaking moment balancing of an articulation dyad by gear nertia counterweights mounted on the base.

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_{cw_1} [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

3 is mounted on the base and is linked kinematically with link2 through link $1'$.Let us consider the complete shaking force and shaking moment balancing of 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

$$m_B = m_{1'} l_{CS_{1'}} / l_{BC}$$
$$m_C = m_{1'} l_{BS_{1'}} / l_{BC} \quad (1)$$

where, l_{BC} is the length of link 1, $l_{CS_{1'}}$ and $l_{BS_1'}$ 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

$$I^*_{S_1'} = I_{S_1'} - m_{1'} l_{BS_1'} l_{CS_1'} \quad (2)$$

where, $I_{S_1'}$ 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 $I^*_{S_1'}$. This fact allows for an easy determination of the parameters of the balancing elements as follows:

$$m_{CW_2} = (m_2 l_{AS_2} + m_B l_{AB}) / r_{CW_2} \quad (3)$$

where, m_2 is the mass of link 2, l_{AB} is the distance between the centers of the hinges A and B, l_{AS_2} is the distance of the center of hinge A from the center mass of S_2 of link 2, r_{CW_2} is the rotation radius of the center of mass of the counter weight with respect to A ,and

$$m_{CW_1} = [(m_2 + m_{CW_2} + m_B) l_{OA} + m_1 l_{OS_1}] / r_{CW_1} \quad (4)$$

where, m_1 is the mass of link 1, l_{OS_1} is the distance of the joint center O from the center of mass S_1 of link 1.
Also,

$$m_{CW_3} = m_C l_{OC} / r_{CW_3} \quad (5)$$

where, $l_{OC} = l_{AB}$, r_{CW_3} is the rotation radius of the center of mass of the counter weight.

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 by three rotational pairs(see fig.5) two replacement points A and B are considered . This results in the need to increase the mass of the counter weight. However, such a solution may be avoided by considering the problem of dynamic substitution of dynamic substitution of link masses by three point masses. Usually the center of mass of such an asymmetric link is located inside a triangle formed by these points.

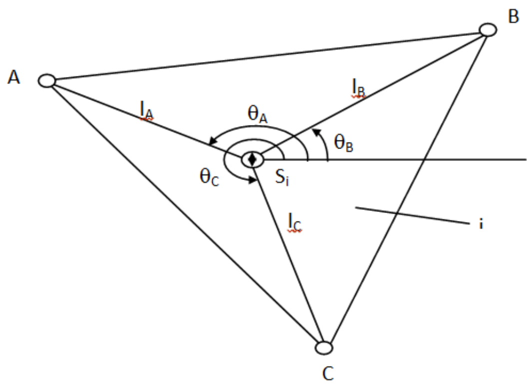

Figure 5: Dynamic substitution of the masses of the link by three rotational pairs.

The conditions for dynamic substitution of masses are the following:

$$\begin{bmatrix} 1 & 1 & 1 \\ l_A e^{i\theta_A} & l_B e^{i\theta_B} & l_C e^{i\theta_C} \\ l_A^2 & l_B^2 & l_C^2 \end{bmatrix} \begin{bmatrix} m_A \\ m_B \\ m_C \end{bmatrix} = \begin{bmatrix} m_i \\ 0 \\ I_{S_i} \end{bmatrix} \quad (6)$$

where, m_A, m_B and m_c are point masses, l_A, l_B and l_C are the moduli of radius vectors of corresponding points, θ_A, θ_B and θ_C are angular positions of radius vectors; m_i is the mass of link, I_{S_i} is the moment of inertia of the link about an axis through S_i (axial moment of inertia of link).From this system of equations the masses are obtained

$$m_A = D_A / D_i \,;\, m_B = D_B / D_i \,;\, m_C = D_C / D_i \quad (8)$$

where, D_A, D_B, D_C and D_i are determinants of the third order obtained from the above system of equations.

3. Application of the Method for Complete Shaking Force and Shaking Moment Balancing of Multiple Linkages.

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.

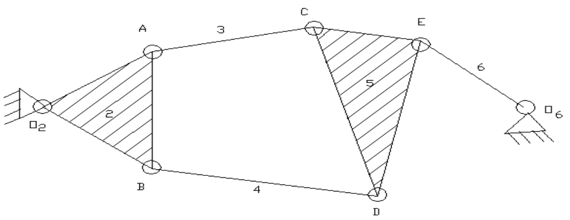

Figure 6: Mechanism with high degree of complexity (Stephenson's link motion).

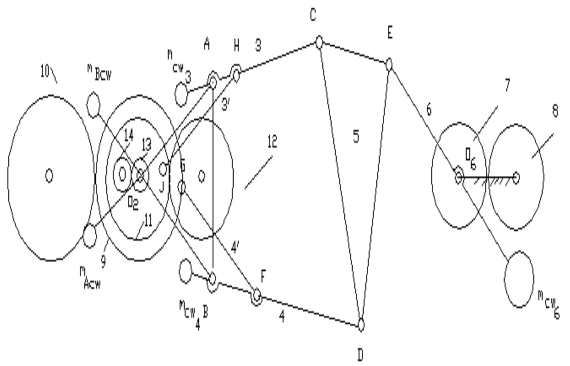

Figure 7: Balanced mechanism with high degree of complexity (stephenson's link motion).

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 .Link 6 has been dynamically replaced by two point masses and 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

$$M^{\text{int}} = M_6^{\text{int}} + M_2^{\text{int}} + M_A^{\text{int}} + M_{O_2}^{\text{int}} + M_B^{\text{int}} \quad (8)$$

Where

$$M_6^{\text{int}} = (I_{S6} + m_6 l_{O_6S_6}^2 + m_{E5} l_{O_6E}^2 + m_{CW_6} r_{CW_6}^2)\alpha_6$$

$$M_2^{\text{int}} = (I_{S_2} + I_{S_3}'^{*} + I_{S_4}'^{*} + (m_4 + m_{D5} + m_{CW_4} + m_F + m_{B2}) l_{O_2B}^2$$
$$+ (m_3 + m_{CW_3} + m_{C5} + m_H + m_{A2}) l_{O_2A}^2)\alpha_2$$

$$M_A^{\text{int}} = (I_{S_3} + m_3 l_{AS_3}^2 + m_{C5} l_{AC}^2 + m_H l_{AH}^2 + m_{CW_3} r_{CW_3}^2)\alpha_3$$

$$M_{O_2}^{\text{int}} = (2m_J l_{O_2J}^2 + 2m_G l_{O_2G}^2)\alpha_2$$

$$M_B^{\text{int}} = (I_{S_4} + m_4 l_{BS_4}^2 + m_{D5} l_{BD}^2 + m_F l_{BF}^2 + m_{CW_4} r_{CW_4}^2)\alpha_4$$

3.2. Atkinson engine mechanism:

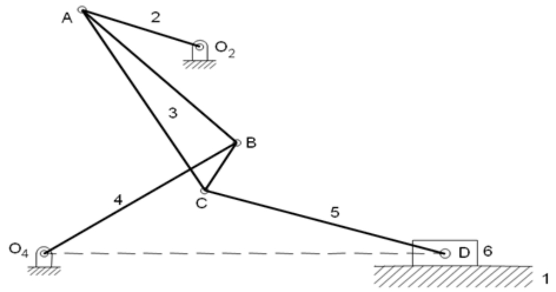

Figure 8: Atkinson engine mechanism.

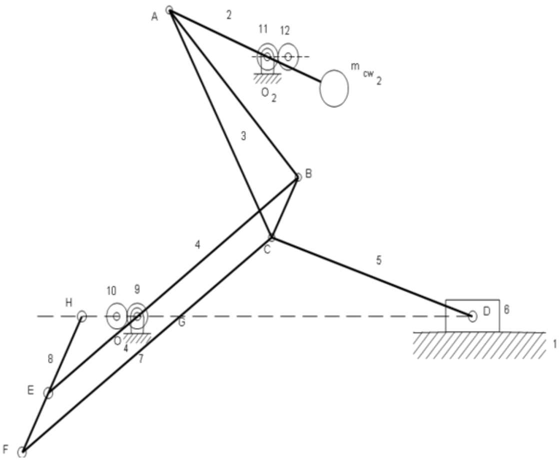

Figure 9: Balanced Atkinson engine mechanism.

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

$$K = \frac{l_{0_4E}}{l_{0_4B}} = \frac{l_{0_4H}}{l_{0_4H}} = \frac{l_{EH}}{l_{BD}} \quad (9)$$

The length of the articulation dyad is determined as:

$$l_{CF} = l_{0_4B} + l_{0_4E}$$
$$l_{FH} = l_{EF} + l_{EH} = l_{BC} + kl_{BD}$$

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_{C_5}, & m_{D_5} .An articulation dyad CFE is added.Now link 4 (m_B & m_E) is to be balanced about point 0_4. Link 7 (m_C & m_F) about point G. Finally the masses

m_{G}, m_D , m_6 & m_H about point 0_4.The necessary conditions are as follows:

$$\begin{aligned}
&m_B\, l_{0_4 B} + m_4\, l_{0S_4} - m_E\, l_{0_4 E} = 0\\
&m_F\, l_{FG} - m_7\left(l_{CG} - l_{CS_7}\right) - m_c l_{CG} = 0\\
&m_H = \left[m_D + m_6 + \left(m_C + m_F + m_7\right)\frac{l_{BC}}{l_{BD}}\right]/k\\
&m_F + m_E + m_H = m_8\\
&m_F l_{FS_8} + m_E\left(l_{FS_8} - l_{EF}\right) - m_H\left(l_{FH} - l_{FS_8}\right) = 0\\
&m_F l_{FS_8}^2 + m_E\left(l_{FS_8} - l_{EF}\right)^2 - m_H\left(l_{FH} - l_{FS_8}\right)^2 = Is_8
\end{aligned} \quad (10)$$

where $l_{0_4 B}, l_{0_4 E}$ and $l_{0_4 S_4}$ are the distances of joint centers B, E and of the center of mass S_4 of the link 4 from the point O_4. l_{CG}, l_{FG} are the distances of the centers of the joints C, F from the working point G of the pantograph.

l_{CS_7} is the distance of the center of the joint C from the center of mass S_7 of Link 7. l_{BD}, l_{BC} are the distances of the centers of the joints D, C from the center of joint B. l_{FE}, 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_g is the mass of link 8, l_{FS_8} is the distance of the center of the joint F from the center of mass S_8 of link 8. , I_{S_8} is the axial moment of inertia of link 8.

The desired parameters are obtained as follows:

$$\begin{aligned}
&m_8 = m_F + m_E + m_H\\
&l_{FS_8} = \left(m_E l_{EF} + m_H l_{EH}\right)/m_8\\
&l_{S_8} = m_F l_{FS_8}^2 + m_E\left(l_{FS_8} - l_{EF}\right)^2 - m_H\left(l_{FH} - l_{FS_8}\right)^2 x
\end{aligned} \quad (11)$$

Where

$$m_F = \left[m_c l_{c_G} + m_7\left(l_{CG} - l_{CS_7}\right)\right]/l_{FG}$$

$$m_E = \left(m_B l_{0_4 B} + m_4 l_{0_4 S_4}\right)/l_{0_4 E}$$

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_F 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 $m_6 + m_D$, m_F, m_H and m_G 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:

$$\bar{F}_H^{-int} = \bar{F}_D^{-int} + \bar{F}_6^{-int} + \left(\bar{F}_G^{-int} = \bar{F}_C^{-int} + \bar{F}_F^{-int} + \bar{F}_4^{-int}\right) \quad (12)$$

where F^{int} (i, = C, D, F, G, H, 4, 6) – inertia forces from corresponding mosses).

3.2.2. Shaking moment:

The shaking moment of the mechanism is determined by the sum

$$M^{int} = M_4^{int} + M_7^{int} + M_{0_4}\left(F_i^{int}\right) \quad (13)$$

where M_4^{int} and M_7^{int} are the shaking moments of the rotating links 4 and 7 with the inertia of the replaced point masses taken into account.

$$M_4^{int} = \left(Is_4 + m_4 l_{0_4 S_4}^2 + m_B l_{0_4 B}^2 + m_E l_{0_4 E}\right)\alpha$$

$$M_7^{int} = \left(Is_7 + m_7 l_{GS_7}^2 + m_c l_{CG_4}^2 + m_F l_{FG_4}\right)\alpha$$

Where Is_4 and Is_7 are the axial moments of inertia of links 4 and 7, $\alpha = \alpha_4 = \alpha_7$ is the angular acceleration of links 4 and 7, $Mo_4\left(F_i^{int}\right)$ is the moment resulting from the force of inertia of the masses and m_H performing a transactional rectilinear motion relative to pivot $0_4 . m_6 + m_D, m_G$

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:

$$\begin{aligned}
I_{gear} &= Is_4 + Is_7 + m_4 l_{o_4 S_4}^2 + m_B l_{o_4 B}^2 + m_7 l_{GS_7}^2\\
&+ m_C l_{CG}^2 + m_E l_{0_4 E}^2 + m_F l_{FG}^2
\end{aligned} \quad (14)$$

In most constructions of the mechanisms the moment $m_{0_4}\left(F_i^{int}\right)$ is very small that in many balancing problems this moment may be neglected.

To balance this shaking moment gears 9 and 10 are mounted on the pivot point 0_4.

Link 2 is dynamically replaced by the point masses m_{A_2} and m_{F_2} and a counterweight m_{CW_2} is added to balance shaking force

$$m_{CW_2} = \left(m_{A_3} l_{Ao_2} + m_2 lo_2 s_2\right)/r_{CW_2}$$

Shaking moment: The shaking moment at point 0_2 is determined by the sum.

$$M_2^{int} = \left(Is_2 + m_2 l_{o_2 s_2}^2 + \left(m_{A_2} + m_{A_3}\right)l_{Ao_2}^2 + m_{CW_2} r_{CW_2}^2\right)\alpha_2 \quad (15)$$

To balance this shaking moment gears 11 and 12 are mounted on the point 0_2.

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.

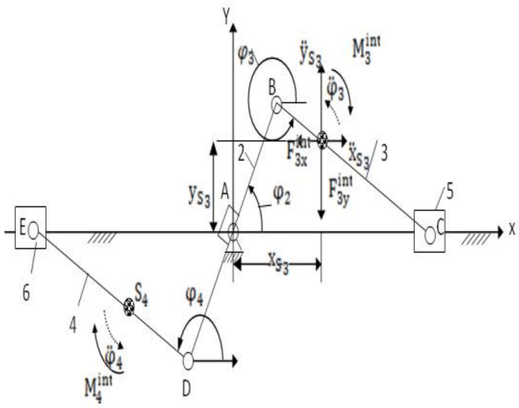

Figure 10: Self balanced slider – crank system.

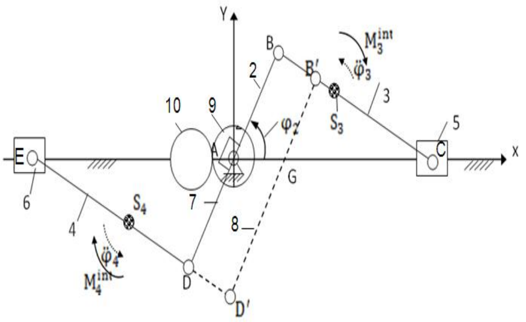

Figure 11: Self-balanced slider-crank system with an imagined articulation dyad $B'D'E$.

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 similarty factor of the formed pantograph is $k = l_{AD}/l_{AB}$ =1 and

$l_{BB'} = l_{DD'}, l_{B'D'} = l_{AD} + l_{AB}.$

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

$$\begin{bmatrix} 1 & 1 & 1 \\ l_{BS_3} & -l_{CS_3} & l_{B'S_3} \\ l_{BS_3}{}^2 & l_{CS_3}{}^2 & l_{B''S_3}{}^2 \end{bmatrix} \begin{bmatrix} m_B \\ m_C \\ m_{B'} \end{bmatrix} = \begin{bmatrix} m_3 \\ 0 \\ I_{S_3} \end{bmatrix} \quad (16)$$

where $l_{BS_3,}$, $l_{CS_3}, l_{B'S_3}$ 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

$$m_B = D_B / D_3; m_C = D_C / D_3; m_{B'} = D_{B'} / D_3 \quad (17)$$

where $D_{B'}, D_3, D_B, D_C$ 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,

$$m_{D'} = m_{B'} l_{B'G} / l_{D'G}$$

The concentrated point masses $m_{G,} m_C, m_E$ to be balanced about center A, i.e,

$$m_E = (m_G l_{BB'} + m_C l_{BC}) / l_{DE}$$

where $l_{BB',} l_{BC}$ 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 $m_{D'}, m_D, m_E$ which allow the determination of the mass and inertia parameters of the connecting coupler 4;

$$\begin{aligned} m_4^* &= m_{D'} + m_D + m_E \\ l_{ES_4}^* &= (m_D l_D s_4 + m_{D'} l_{D'S_4}) / m_4^* \\ I_{S_4}^* &= m_D l^2{}_{DS_4} + m_{D'} l^2{}_{D'S_4} + m_E l^2{}_{ES_4} \end{aligned} \quad (19)$$

Where

$$l_{DS_4}^* = l_{DE} - l_{ES_4}; l_{D'S_4} = l_{D'E} - l_{ES_4},$$

Shaking moment balancing:

$$\begin{aligned} M_2^{int} + M_7^{int} = (I_{S_2} + m_2 l_{AS_2}^2 + m_B l_{AB}^2 + \\ m_D l_{AD}^2 + m_7 l_{AS_7}^2 + I_{S_7}) \alpha_2 \end{aligned} \quad (20)$$

$$M_8^{int} = (I_{S_8} + m_8 l_{GS_8}^2 + m_{D'} l_{D'G}^2 + m_{B'} l_{B'G}^2) \alpha_8 \quad (21)$$

Total Shaking moment generated by the mechanism:

$$M^{int} = M_2^{int} + M_7^{int} + M_8^{int} \quad (22)$$

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:

$$l_{AB} = l_{AD} = 0.05m; l_{BC} = l_{DE} = 0.2m; l_{CS_3} = l_{ES_4} = 0.1m;$$
$$m_3 = m_4 = 0.35kg; m_5 = m_6 = 2kg;$$
$$I_{S_3} = I_{S_4} = 0.005kg - m^2; \omega_{AB} = 30\pi / s; \alpha_{AB} = 450\pi / s^2;$$
$$m_2 = m_7 = 0.3kg; I_{S_2} = I_{S_7} = 0.003kg;$$
$$l_{AS_2} = l_{AS_7} = 0.025m; I_{S_8} = 0.006kg - m^2; m_8 = 0.6kg$$

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 coupler4. Fig.12 illustrates the obtained results.so by optimal redistribution of the masses of the connecting coupler 4, the shaking moment is cancelled (curve 'b).

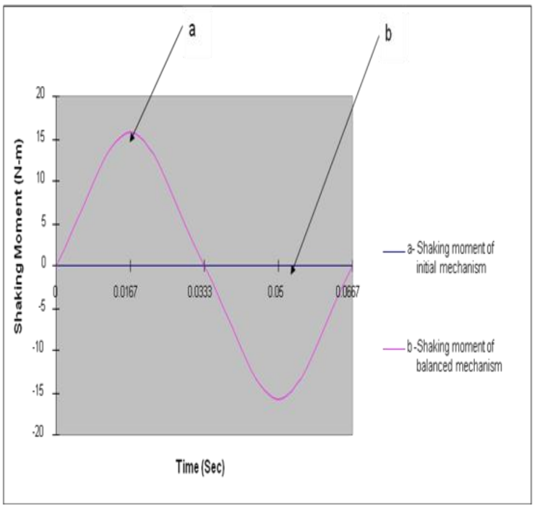

Figure 12: Shaking moment Vs Time.

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 mechanical 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 mass 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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