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BALANCING OF PLANAR SIX-BAR MECHANISM WITH GENETIC ALGORITHM

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Abstract: In the present work, the optimal balancing of the planar six-bar mechanism is investigated to minimize the fluctuations of shaking force and shaking moment. An optimization problem is formulated for balancing the planar six-bar mechanism by developing an objective function. The genetic algorithm and MINITAB software were used to solve the optimization problem. The selection of weighting factors has a crucial role to obtain the optimum values of design parameters. Two sets of weighting factors were considered as per the contribution of X and Y components of the shaking force and shaking moments. Shaking force and shaking moments were minimized drastically and were compared with the original values.

Keywords: shaking force, shaking moment, dynamic balancing, six-bar mechanism, genetic algorithm.

1. INTRODUCTION

Six bar mechanism is a one degree of freedom mechanism which is constructed from six links. Klann linkage used to drive the legs of a walking machine. Six-bar mechanism is used in Watt mechanism, Stephenson mechanism, missile launcher and bellow valves etc [1].

Shaking force, shaking moment, and input-torque are the dynamic performance characteristics which depend on the inertia of each moving link and its mass center location. It is essential to optimally distribute the link masses to reduce shaking force and shaking moments. Minimization of both shaking force and shaking moment fluctuations is essential for dynamic balancing which improves the mechanism fatigue life by reducing vibration, noise and wear. Cheng-HO Li and Pei-Lum TSO [2] proposed the concept to reduce the shaking force and shaking moment using both linkage balance and counterweight disks. S. Balasubramanian *et al.* [3] presented the design equations for complete shaking force balancing of planar Stephenson's and Watt's type 6R 6-bar slider-crank regular force transmission mechanisms using the method of linearly independent mass vectors. Gao Feng *et al.* [4] derived the design equations and techniques for complete balancing of shaking force and shaking moments of linear and rotary inertia of different types of six-bar linkages without applying external loads. Jianguo Hu *et al.* [5] proposed the two-phase design scheme of Stephenson six-bar working mechanisms for servo mechanical presses with high mechanical advantage. The transmission characteristics of the optimized working mechanism with that of the slide-crank mechanism and symmetrical toggle mechanism were compared with the help of simulations based on the software ADAMS. Dewen Jin *et al.* [6] used Computer simulation and experimental method to investigate the advantages of the mechanism as used in the prosthetic knee from the kinematic and dynamic points of view. The results of the expected trajectory of the ankle joint in the swing phase were compared for six-bar and four-bar mechanisms. Kailash Choudhary *et al.* [7] were determined the shaking force and shaking moment for a complete cycle of motion using hyper works. MBD simulation was carried out for Stephenson six-bar mechanism using Motion Solve [7]. Sebastian Briot *et al.* [8] obtained the complete shaking force and shaking moment balancing by using a coupler link by adding a class-two assur group with prescribed geometrical and mass parameters. P.Nehemiah *et al.* [9] presented the method for complete balancing of shaking force and shaking moments of 3 types of four-bar linkages without external loads only with revolute pairs due to rotary inertia. Basayya K. Belleri and Shravankumar B. Kerur [10] presented a computer-oriented procedure for solving the dynamic force analysis problem for general planar mechanisms and that was extended to a six-bar planar mechanism with variable topology. F C Chen *et al.* [11] were used Taguchi method to investigate the influence of manufacturing tolerance and joint clearance on the quality of the dual-purpose six-bar mechanism. Erkaya *et al.* [12] investigated 2D articulated mechanism to minimize the shaking force and shaking moment fluctuations with a Genetic algorithm by selecting weighing factors.

In the present work, MATLAB (Version: R2018a, The MathWorks Inc.) tool was used to determine the position, velocity, acceleration and forces of four-bar mechanism and extended to six-bar mechanism by variable topology approach.

2. METHODOLOGY

2.1. Dynamic force analysis of a six -bar mechanism

The six-bar mechanism as shown in Figure 1, it consists of two loops four-bar mechanisms, one ie, ABCD (Loop-1) and DCEF (Loop-2). The joint forces and input torque required on the crank were calculated [10]. The outputs of the first four-bar mechanism were used as an input parameter for the second four-bar mechanism and position, velocity, acceleration and forces were analyzed. With the output parameters of the second four-bar mechanism, the force analysis of the first four-bar mechanism was carried out. After combining the two loops of four-bar mechanisms, the resultant joint forces were determined at joints C and D.

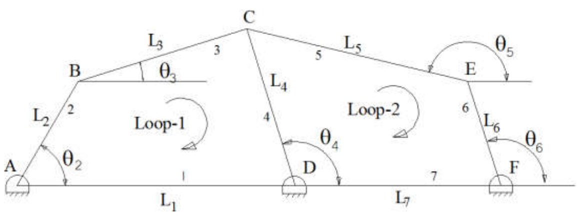

Fig. 1. Six-bar mechanism

2.2. Shaking force and shaking moment

The sum of all the forces acting on the ground plane is called the shaking force and reaction moment felt by the ground plane is called the shaking moment. It is essential to know the net effect of the dynamic forces as felt on the ground plane as this can set up vibrations in the structure. In six-bar linkage (Fig. 1), there are three points (A, D and F) at which the dynamic forces can be transferred to ground (link 1 and link 7). Shaking force and shaking moment for six-bar linkage were calculated by using equations as follows:

$$F_{sh} = F_{21} + F_{41} + F_{47} + F_{67}, \quad (1)$$

$$M_{sh} = T_{21} + (L_1 * F_{41}) + T_{47} + (L_7 * F_{67}). \quad (2)$$

2.3. Optimization process

In order to balance the mechanism completely, it is essential to eliminate or reduce both the shaking force and shaking moment. By attaching the counterweights to the moving links of the mechanism, the shaking force can be eliminated. But, this increases the total mass and inertia of the mechanism which increases the shaking moment, requires more driving torque, and forces at the joints. Another approach [11] to minimize shaking force and shaking moments along with other dynamic parameters such as driving torque and bearing reactions is to optimize all design parameters was used in the present work.

The regression equation was obtained for the twenty-seven design variable. The obtained regression equation is used as a Fitness function and all twenty-seven design parameters were optimized by using a Genetic Algorithm. The following objective function (minimize) was considered for optimization:

$$F(X) = \sum_{n=1}^{s} \begin{matrix} [(W_1(F_{21x_n}) + W_2(F_{21y_n}) + \\ +W_3(F_{41x_n} + F_{47x_n}) + \\ +W_4(F_{41y_n} + F_{47y_n}) + \\ +W_5(F_{67x_n}) + W_6(F_{67y_n}) + \\ +W_7(M_{sh_n})] \end{matrix} ,(3)$$

$$\text{subject to } g_k(X) \le 0, \quad (4)$$

$$x_r^{min} \le x_r \le x_r^{max}, \quad (5)$$

$$x_r \epsilon X, \quad (6)$$

where: W_h are weighting factors; s is the number of the points considered during the one rotation of the crank; gk are the constraints arising from the condition satisfying the crank rocker motion. The objective function minimized the related shaking force and moment provided that the generated solution satisfies a set of constraints. X is the vector consisting of the 27 independent design variables (xr).

$$X=[\ Li\ \delta i\ mi\ Igi\ rgi\ ]T, \quad (4)$$

where: L_i, δ_i, m_i, Ig_i and rg_i are link lengths, structural angles, masses, moment of inertia of moving links and the position vectors respectively. x_r^{min} and x_r^{max} are the lower and upper limits of design parameters. The lower and upper limits of link lengths are set as L_i-0.1xL_i and L_i+0.1xL_i respectively. Lower and upper limits of structural angles (δ_i) are considered as 0° and 360° respectively. Lower and upper bounds of m_i, Ig_i, rg_2, rg_3 and rg_4 were arranged by taking into account the link geometries. The value of weighting factors has significant effect on the optimum of design variables. Each weighting factor must assure the following condition [12]:

$$0 \le W_h \le 1 \text{ and } \sum_{h=1}^{7} W_h = 1. \quad (5)$$

As per the contribution of x and y components of the forces and shaking moments, the two sets of weighting factors were selected. Case I: W_1=0.206, W_2=0.013, W_3=0.507, W_4=0.154, W_5=0.087, W_6=0.0005, W_7=0.0325. Case II: W_1=0.15, W_2=0.1, W_3=0.3, W_4=0.1, W_5=0.1, W_6=0.2, W_7=0.05.

3. RESULTS AND DISCUSSION

The pin forces and input driving torque for the six-bar mechanisms as explained in methodology was demonstrated with the following numerical. The operating speed of the mechanism was constant, and it was considered as 300 rpm. The original values of the 27 parameters were shown in Table 1. The two sets of weighting factors were selected as per the X and Y components of the forces acting on the frame and moment. The optimum design variables were calculated using MINITAB and Genetic Algorithm and obtained optimum values were shown in the Table 1.

Tab. 1. Original and optimized parameters of six-bar mechanism

Sl. No.	Parameter	Description	Original Value	Optimized Value	
				Case I	Case II
1	L_1, mm	Length of fixed link	600	450.00	524.37
2	L_2, mm	Length of crank	100	104.43	90.02
3	L_3, mm	Length of coupler	400	310.04	340.73
4	L_4, mm	Length of follower	320	348.76	329.99
5	L_7, mm	Length of fixed link	600	570.88	570.57
6	L_5, mm	Length of coupler	400	365.50	399.98
7	L_6, mm	Length of follower	320	339.98	329.84
8	m_2, (kg)	Mass of crank	0.36	0.36	0.38
9	m_3, (kg)	Mass of coupler	1.30	0.84	0.81
10	m_4, kg	Mass of follower	1.05	0.81	1.04
11	m_5, kg	Mass of coupler	1.30	0.04	0.82
12	m_6, kg	Mass of follower	1.05	1.05	1.47
13	δ_2, degree	Structural angle of crank	0	2.23	0.88
14	δ_3, degree	Structural angle of coupler	0	2.92	0.42
15	δ_4, degree	Structural angle of follower	0	4.54	0.75
16	δ_5, degree	Structural angle of coupler	0	0.25	0.45
17	δ_6, degree	Structural angle of follower	0	220.16	138.23
18	I_{g2}, kg m^2	Inertia moment of crank	4.13×10^{-4}	0.45×10^{-3}	0.11×10^{-2}
19	I_{g3}, kg m^2	Inertia moment of coupler	1.87×10^{-2}	0.6×10^{-3}	1.132×10^{-2}
20	I_{g4}, kg m^2	Inertia moment of follower	9.87×10^{-3}	10.37×10^{-2}	8.4×10^{-2}
21	I_{g5}, kg m^2	Inertia moment of coupler	1.87×10^{-2}	0.30	0.50
22	I_{g6}, kg m^2	Inertia moment of follower	9.87×10^{-3}	4×10^{-2}	2.416×10^{-2}
23	r_{g2}, mm	Position vector of crank	50	23.85	51.13
24	r_{g3}, mm	Position vector of coupler	200	171.66	172.01
25	r_{g4}, mm	Position vector of follower	160	115.29	103.07
26	r_{g5}, mm	Position vector of coupler	200	165.91	198.75
27	r_{g6}, mm	Position vector of follower	160	158.96	153.56

Figure.2 shows the forces at joint ‘A’ (F_{12}) which are the sub-component of shaking force for three cases (original, case-I and case-II) for one complete rotation of the crank. The optimized values for case-I and case-II were decreased by 50.75% and 41.4% respectively. Before optimization, the maximum force of 440.13 N was acting at a crank angle of 180°, that was reduced to 78.74 N and 134.01 N in case-I and case-II respectively. The forces at joint ‘B’ (F_{23}) for three cases were shown in Fig. 3. The optimized values for case-I and case-II were decreased by 49.99% and decreased by 46.6% respectively. 422.96 N was the maximum value of force acting at a crank angle of 180°, that was decreased to 71.32 N for case-I and decreased to 115.84 N for case-II. The forces at joint ‘C’ (F_{43} + F_{45}) for three cases are shown in Figure.4. After optimization, the forces at joint ‘C’ for case-I and case-II were decreased by 65.5% and 69.8% respectively. The maximum value of 430.73 N force was acting at an angle of 180°, and that was decreased to 41.46 N and 66.97 N in case-I and case-II respectively.

For one complete rotation of the crank, the forces at joint ‘D’ (*F14 + F74*) for three cases were shown in Fig. 5. After optimization, the values for case-I and case-II were decreased by 58.2% and 73.83% respectively. The maximum 515.07 N force was acting at an angle of 1800, after optimization for case-I and case-II were reduced to 88.75 N and 39.4 N respectively. Figure 6 shows the force at joint ‘E’ (F_{65}) for three cases. The optimized values were decreased by 24.43% in case-I and 18.94% in case-II. The maximum force of 45.25 N was acting at an angle of 60° and was increased to 60.48 N and increased to 50.88 N for case-I and case-II respectively. The forces at joint ‘F’ (F_{76}) for three cases were shown in Figure.7. The optimized values for case-I increased by 2.01% and decreased by 1.98% in case–II. The maximum force of 47.95 N acting at a crank angle of 60°, which was increased to 67.4 N in case-I and decreased to 27.48 N in case-II. The shaking forces of original, case-I and case-II were shown in Figure.8. In case-I, and case-II the shaking forces were reduced by 48.5% and 51.51% respectively. At a crank angle of 180°, the maximum force acting was 554.8 N and it reduced to 158.7 N in case-I and 190.3 N in case-II.
Shaking moments of original, case-I and case-II were shown in Figure 9. The shaking moments were reduced in case-I and case-II by 32.35% and 92.42%. The shaking moment of 101.38 Nm was developed at a crank angle of 180° were reduced to 22.5 Nm and 0.98 Nm in case-I and case-II respectively.

The driving torques required on the crank as shown in Figure 10. It was observed that 84.33% of torque was reduced in case-I and 29.09% in case-II. The objective function values were shown in Figure 11. The objective function values for case-I and case-II were reduced to 91.01% and 91.23%.

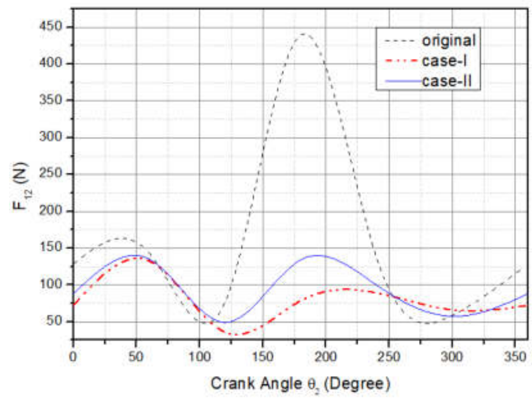

Fig. 2. Forces at joint A (F_{12}) vs. crank angle (θ_2)

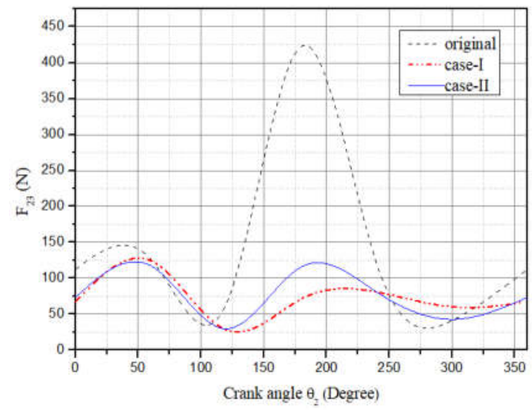

Fig. 3. Forces at joint B (F_{23}) vs. crank angle (θ_2)

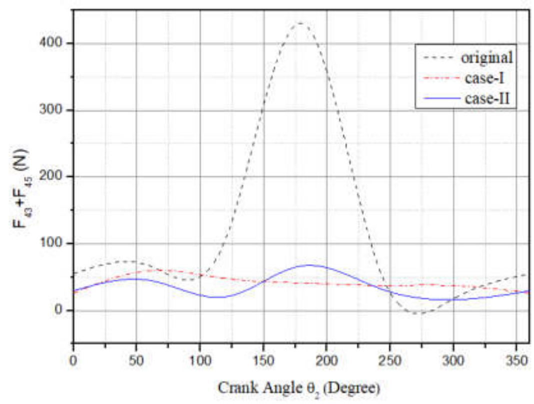

Fig. 4. Forces at joint C (F_{43}+F_{45}) vs. crank angle (θ_2)

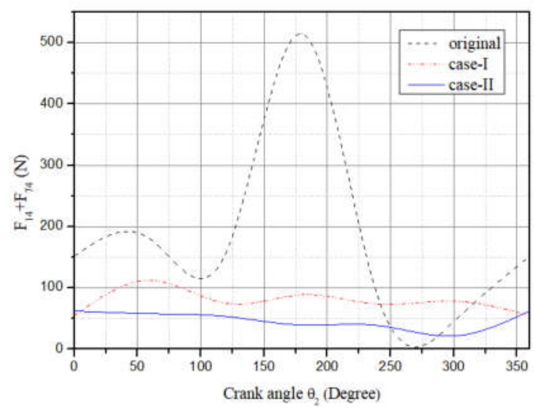

Fig. 5. Forces at joint D (F_{14}+F_{74}) vs. crank angle (θ_2)

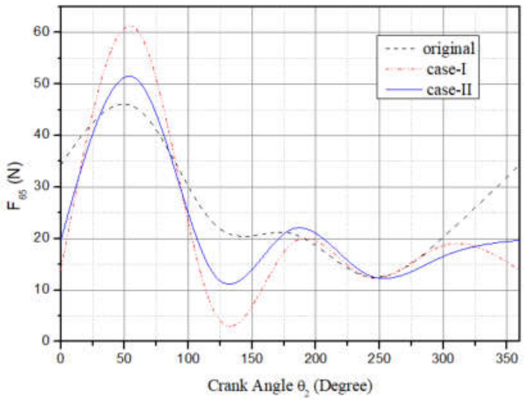

Fig. 6. Forces at joint E (F_{65}) vs. crank angle (θ_2)

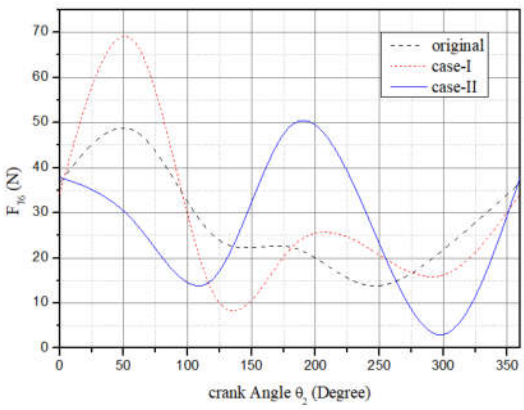

Fig. 7. Forces at joint F (F_{76}) vs. crank angle (θ_2)

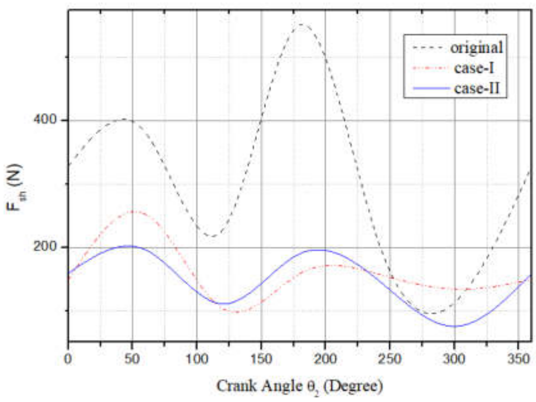

Fig. 8. Shaking forces (F_{sh}) vs. crank angle (θ_2)

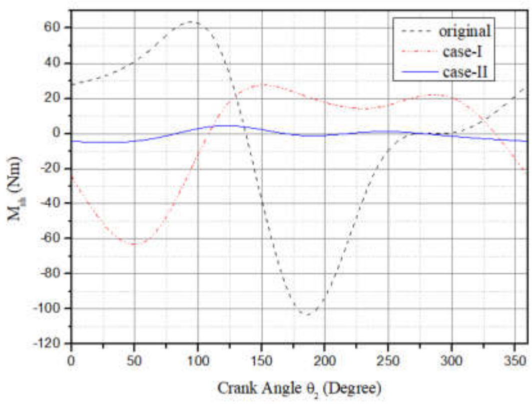

Fig. 9. Shaking moments (M_{sh}) vs. crank angle (θ_2)

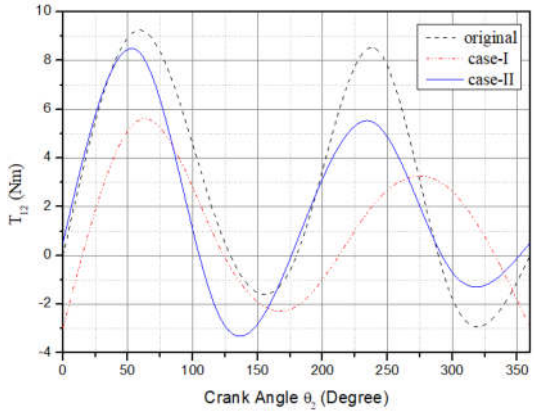

Fig. 10. In-put Torque (t_{12}) vs. crank angle (θ_2)

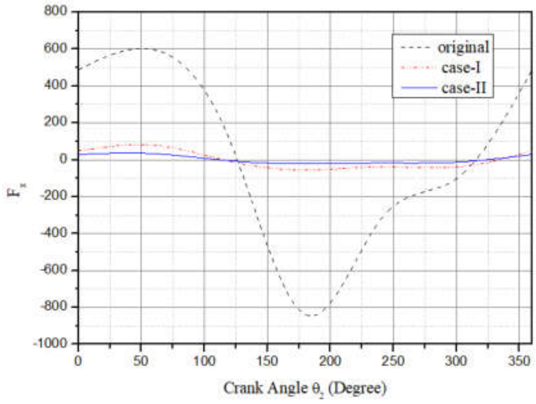

Fig. 11. Objective function value (F_x) vs. crank angle (θ_2)

4. CONCLUSIONS

The balancing of planar six-bar mechanism by using the Genetic Algorithm and MINITAB software was studied by considering two sets of weighting factors. Results presented in this investigation reveal the optimal design variables by adjusting weighting factors. The shaking forces were reduced by 48.5% and 51.51% in case-I and case-II and shaking moments were drastically reduced to 32.35% and 92.42% in case-I and case-II respectively. It was concluded that the set of weighting factors of case-II gives the optimum values of design variables.

Nomenclature

Symbols

F_{ij} – force exerted by member *i* on member *j*
F_{sh} – shaking force
I_{gi} – moment of Inertia of link
L_i – length of link
m_i – mass of link
M_{sh} – shaking moment
r_{gi} – position vector of link
W_h – weighting factors

Greek letters

θ_i – angle of inclination of link
δ_i – structural angle of link

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