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Optimum Design and Control of a Quick-Return Mechanism Used in a Jewelry Welding Powder Production Machine

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ABSTRACT

The movement of the machines used in the industry is generally driven by an electric motor. The rotational motion of the electric motor is transferred to the mechanism, and while the kinematic and dynamic analyzes of the mechanism are performed, it is considered that the input link (crank) of the mechanism to which the rotational motion of this motor is transferred rotates with a constant angular velocity. However, the inertia forces and moments created by all the other links of the mechanism prevent the crank from rotating with a constant angular velocity. To eliminate this problem, the angular velocity of the crank must be controlled using an appropriate control method. In this study, the optimum design of the quick return mechanism in a welding powder production machine to be used in the jewelry manufacturing industry has been made within the scope of some specified constraints. Then, the dynamic equations of the mechanism were obtained utilizing the Eksergian's Equations of Motion method and the sliding mode control method was used to ensure that the crank rotates with a constant angular velocity. In this way, it is aimed to obtain a more standard size jewelry welding powder.

1. INTRODUCTION

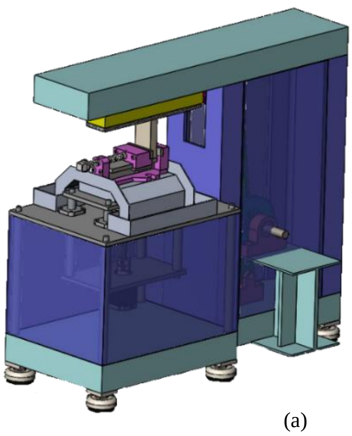
Quick-return mechanisms are widely used in mechanical engineering applications like cutting and shaping machines etc. These mechanisms are designed so that the forward movement with the intended work is slow (mostly cutting stroke) and the return movement without the work is faster (return stroke). In this way, efficiency can be achieved by shortening the idle time while the mechanism is running. The round trip ratio is known as TR (time ratio) in the mechanism and always has a value greater than 1 due to construction.

The movement of the mechanisms used in the industry is generally provided by using an electric motor and this motor is required to rotate the input link of the mechanism with a constant speed and all designs and analyzes of the mechanisms are made accordingly. However, due to the external forces and inertia forces that may occur during the operation of the mechanisms, it becomes difficult for the crank to rotate at a constant speed. In order to eliminate this problem, some machine elements that will create a flywheel effect can be used in mechanisms. However, as a result of advances in electronics and programming technology, by using different control methods, the motor can be provided to rotate the input link of the mechanism at a constant speed. Researchers are working to develop effective methods in this context. Fung and Chen [1], presented a study to control the crank constant velocity of a quick-return mechanism driven by a DC motor based on PID (Proportional Integral Derivative) controller. They used Hamilton's principle and Lagrange multiplier method to formulate the mechanism for dynamic analysis. They obtained good results for numerical simulations with or without applying cutting force to the system. El-Kribi et al. [2], made a multi-objective optimization study with continuous and discrete variables to minimize the motor torque and velocity fluctuation of a four-bar mechanism by using genetic algorithm. Tao and Sadler [3], made a control study on a four-bar mechanism for reducing the crank angular fluctuation by using PID control method. They used numerical optimization methods to obtain the control gains of the controller. Ha et al. [4], presented an identification method based on real-coded genetic algorithm for identifying the parameters of a slider-crank mechanism. Wai and Lin [5], controlled a slider-crank mechanism by using fuzzy neural network controller with adaptive learning rates utilizing analytical methods based on discrete type Lyapunov function. Lin and Wai [6], presented a fuzzy neural network sliding mode controller to obtain much less tracking error for the position control of motor-quick-return servomechanism. They showed the effectiveness of the presented method with the comparison of adaptive sliding mode controller with some simulations and experimental studies. Ouyang et al. [7], presented a study on a five bar hybrid machine system consisted of a constant speed motor and a servomotor for a constant velocity trajectory profile. They used extra mechanical flywheel to reduce the speed fluctuations for better performance. Gündoğdu and Erentürk [8], made a control simulation study of a four bar mechanism by using fuzzy logic controller and compared the results with an optimal PID controller. Affi et al. [9], presented a multi-objective optimization of a motor-driven four-bar mechanism considering the geometry and the dynamics of the system together. Yan and Yan [10], proposed an approach for designing the variable input speed servo four bar mechanism based on an integrated mechanism and controller design with dimensions of the links, the counterweights, input speed trajectory and controller parameters as design variables. Tanyıldızı and Çakar [11], used sliding mode control and moving sliding mode control algorithms to control the angular velocity of the crank of a slider-crank mechanism. They investigated the robustness of the control algorithms by applying a nonlinear force to the system. In another study, Çakar and Tanyıldızı [12], studied the same control algorithms to control the crank angular velocity fluctuations of a four bar mechanism with the presentation of simulation and experimental applications.

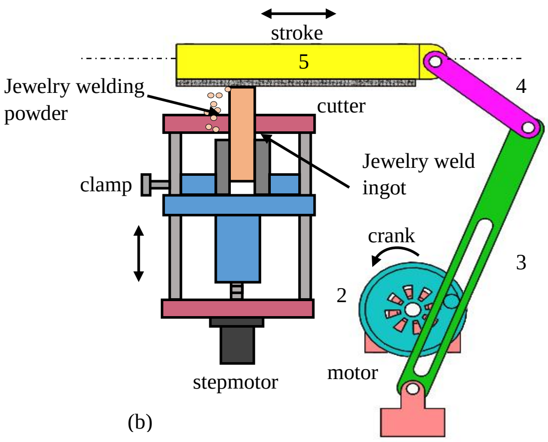
In this study, the optimum design of a quick-return mechanism in a welding powder production machine used in the jewelry industry has been made and the dynamic equations of the mechanism have been obtained by the Eksergian's Equations of Motion method. In addition, the motor to which the crank is connected was controlled with the sliding mode control method in order for the crank, on which the mechanism is driven, to rotate with a constant speed to provide more standard size jewelry welding powder.

2. OPTIMUM DESIGN OF THE QUICK RETURN MECHANISM

While designing any machine, it is desired that the designed machine should have optimum features in terms of efficiency and cost while fulfilling the specified functions. The features such as the maximum work of the machine in minimum volume, high mechanical advantage and being economical in terms of cost must be taken into consideration. In this context, an optimization study has been carried out to ensure that the mechanism designed in this study is in optimum dimensions, taking into account some of the constraints determined. The representation of the designed machine and the schematic of the mechanism are given in Figure 1.



(a)



(b)

Figure 1. a) Jewelry welding powder production machine design, b) Quick-return mechanism of the machine

The mechanism, schematically illustrated in Figure 1.b, is designed to produce welding powder used in the jewelry manufacturing industry. The mechanism is driven by the crank (link 2) connected to the shaft of an electric motor. A cutter unit that forms welding powder is attached to the output link (link 5) of the mechanism, where linear motion is obtained. In this way, while welding powder is produced in the forward movement of the link 5 of the mechanism, no work is done on the return cycle and this period is completed quickly.

The produced welding powder and it's usage in a filigree application are depicted in Figure 2.a and Figure 2.b respectively.

Figure 2. a) Jewelry welding powder, b) jewelry welding powder operation

Jewelry welding powder is used especially in filigree type designed jewelries or crafted object applications. Some examples of crafted objects using this type of welding powder are shown in Figure 3.

Figure 3. Filigree silver objects in which jewelry welding powder is used

The design parameters of the mechanism are r_2 (x_1), r_3 (x_2), r_4 (x_3), a (x_4) and b (x_5) as seen in Figure 4.

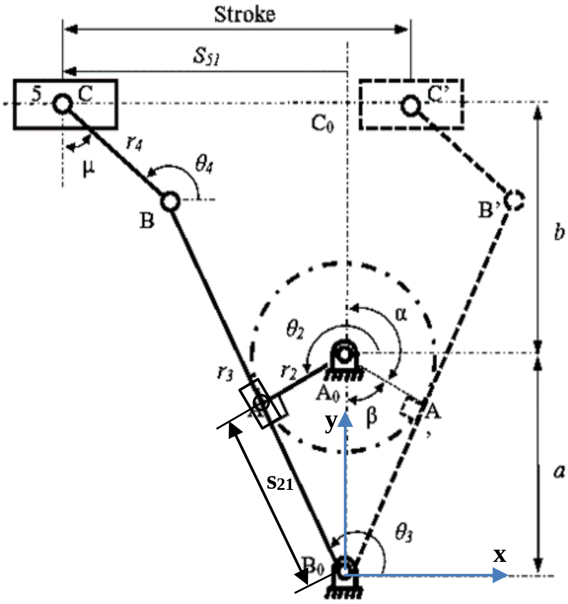


Figure 4. The kinematic diagram of quick-return mechanism

The objective function is:

$$f_{obj} = min(x_1 + x_2 + x_3 + x_4 + x_5) \tag{1}$$

The design constraints of the mechanism are given below, where all lengths are in (mm).

$$\begin{aligned}
&c_1 ;\ x_1 < x_4 \\
&c_2 ;\ 200 < x_4 + x_5 < 500 \\
&c_3 ;\ 250 < 2x_2 \sin\theta_3 < 350 \\
&c_4 ;\ 250 < 2x_2 \sin\theta_3 + x_3 \sin\theta_4 < 500 \\
&c_5 ;\ 45^\circ < \mu < 55^\circ \\
&c_6 ;\ 1.5 < TR < 2
\end{aligned} \tag{2}$$

Here, μ is the transmission angle and TR is time ratio of the mechanism. For the optimization of the mechanism Matlab *fmincon* algorithm was used. The determined optimum design parameters and related physical parameters i.e. lengths (r), masses (m) and moment of inertias (I) of the links are given in Table 1.

TABLE I
DETERMINED DESIGN PARAMETERS AND THE PHYSICAL PARAMETERS OF THE MECHANISM

	r (mm)	m (kg)	I (kgm^2)*10^{-4}
r_2	54	0.35	8.95
r_3	324	0.32	53
r_4	78	0.36	8.5
s_{51}	-	1	-
a	140	-	-
b	207	-	-

3. EQUATIONS OF MOTION OF THE MECHANISM

The kinematic equations can be obtained by using vector loops for the mechanism given in Figure 4 as below:

$$\begin{aligned}
&\mathbf{B_0A} = \mathbf{B_0A_0} + \mathbf{A_0A} \\
&s_{21}e^{i\theta_3} = ia + r_2 e^{i\theta_2}
\end{aligned} \tag{3}$$

$$\begin{aligned}
&\mathbf{B_0B} + \mathbf{BC} = \mathbf{B_0C_0} + \mathbf{C_0C} \\
&r_3 e^{i\theta_3} + r_4 e^{i\theta_4} = i(a + b) - s_{51}
\end{aligned} \tag{4}$$

By using Equations (3-4) one can obtain the following equations in complex plane.

$$\begin{aligned} & s_{21}\left(\cos\theta_3 + i\sin\theta_3\right) = ia + r_2\left(\cos\theta_2 + i\sin\theta_2\right) \\ & \text{real}: s_{21}\cos\theta_3 - r_2\cos\theta_2 = 0 \\ & \text{imag}: s_{21}\sin\theta_3 - a - r_2\sin\theta_2 = 0 \end{aligned} \tag{5}$$

$$\begin{aligned} & r_3\left(\cos\theta_3 + i\sin\theta_3\right) + r_4\left(\cos\theta_4 + i\sin\theta_4\right) \\ & = i\left(a+b\right) - s_{51} \\ & \text{real}: r_3\cos\theta_3 + r_4\cos\theta_4 + s_{51} = 0 \\ & \text{imag}: r_3\sin\theta_3 + r_4\sin\theta_4 - a - b = 0 \end{aligned} \tag{6}$$

Here, s_{21} in Equation (5) can be cancelled as:

$$\frac{s_{21}\sin\theta_3}{s_{21}\cos\theta_3} = \tan\theta_3 = \frac{a + r_2\sin\theta_2}{r_2\cos\theta_2} \tag{7}$$

By rearranging Equations (6-7), the constraint equations of the mechanism can be written as follows:

$$\begin{aligned} & f_1\left(\theta_2,\theta_3,\theta_4,s_{51}\right) = r_2\sin\theta_3\cos\theta_2 \\ & \qquad - \cos\theta_3\left(a + r_2\sin\theta_2\right) = 0 \\ & f_2\left(\theta_2,\theta_3,\theta_4,s_{51}\right) = r_3\cos\theta_3 + r_4\cos\theta_4 + s_{51} = 0 \\ & f_3\left(\theta_2,\theta_3,\theta_4,s_{51}\right) = r_3\sin\theta_3 + r_4\sin\theta_4 - (a+b) = 0 \end{aligned} \tag{8}$$

The independent generalized coordinate (q) and the dependent coordinates (ϕ_1, ϕ_2, and ϕ_3) of the mechanism are presented as follows:

$$\theta_2 = q;\quad \theta_3 = \phi_1;\quad \theta_4 = \phi_2;\quad s_{51} = \phi_3 \tag{9}$$

The first order influence coefficients can be calculated by using Equation (8), where J is the Jacobian matrix and f is constraint equations vector which are calculated as shown below.

$$g = -J^{-1}f' \tag{10}$$

$$f' = \left\{ \frac{\partial f_1}{\partial q} \quad \frac{\partial f_2}{\partial q} \quad \frac{\partial f_3}{\partial q} \right\}^T = \left\{ r_2\cos(q-\phi_1) \quad 0 \quad 0 \right\}^T \tag{11}$$

$$J = \begin{bmatrix} \frac{\partial f_1}{\partial \phi_1} & \frac{\partial f_1}{\partial \phi_2} & \frac{\partial f_1}{\partial \phi_3} \\ \frac{\partial f_2}{\partial \phi_1} & \frac{\partial f_2}{\partial \phi_2} & \frac{\partial f_2}{\partial \phi_3} \\ \frac{\partial f_3}{\partial \phi_1} & \frac{\partial f_3}{\partial \phi_2} & \frac{\partial f_3}{\partial \phi_3} \end{bmatrix} = \begin{bmatrix} \frac{r_2\cos q}{\cos^2\phi_1} & 0 & 0 \\ -r_3\sin\phi_1 & -r_4\sin\phi_2 & 1 \\ r_3\cos\phi_1 & r_4\cos\phi_2 & 0 \end{bmatrix} \tag{12}$$

The first order influence coefficients then can be obtained as follows:

$$g = \begin{Bmatrix} g_1 \\ g_2 \\ g_3 \end{Bmatrix} = \begin{Bmatrix} \dfrac{r_2\cos(q-\phi_1)}{D} \\ -\dfrac{r_2 r_3\cos(q-\phi_1)\cos\phi_1}{r_4\cos\phi_2 D} \\ \dfrac{r_2 r_3\cos(q-\phi_1)\sin(\phi_1-\phi_2)}{\cos\phi_2 D} \end{Bmatrix} \tag{13}$$

Here, $D = a\sin(\phi_1) + r_2\cos(q-\phi_1)$. By using first order influence coefficients the velocities of the dependent links can be calculated with following equation.

$$\dot{\phi}_i = g_i\dot{q} \qquad \left(i = 1,2,3\right) \tag{14}$$

The velocity of the center of gravity for the links can be expressed with the first order influence coefficient;

$$\begin{aligned} & v_{G_i} = \left(\sqrt{v_{G_i}^{x\,2} + v_{G_i}^{y\,2}}\right)\dot{q}; \\ & v_{G_i}^x = \frac{\partial x_i}{\partial q} + \sum_{j=1}^{3}\frac{\partial x_i}{\partial \phi_j}g_j; \quad v_{G_i}^y = \frac{\partial y_i}{\partial q} + \sum_{j=1}^{3}\frac{\partial y_i}{\partial \phi_j}g_j \end{aligned} \tag{15}$$

$$\begin{aligned} & v_{G_2}^x = -c_2\sin q \\ & v_{G_2}^y = c_2\cos q \\ & v_{G_3}^x = \left(-c_3\sin\phi_1\right)g_1 \\ & v_{G_3}^y = \left(c_3\cos\phi_1\right)g_1 \\ & v_{G_4}^x = -\left(r_3\sin\phi_1\right)g_1 - \left(c_4\sin\phi_2\right)g_2 \\ & v_{G_4}^y = \left(r_3\cos\phi_1\right)g_1 + \left(c_4\cos\phi_2\right)g_2 \\ & v_{G_5}^x = -\left(r_3\sin\phi_1\right)g_1 - \left(r_4\sin\phi_2\right)g_2 \\ & v_{G_5}^y = 0 \end{aligned} \tag{16}$$

The second order influence coefficient can be obtained by using Equation (17) as:

$$g_i' = \frac{\partial g_i}{\partial q} + \sum_{j=1}^{3}\frac{\partial g_i}{\partial \phi_j}g_j \qquad \left(i = 1,2,3\right) \tag{17}$$

$$g_1' = -\frac{ar_2\left[g_1\cos q + \sin(q-\phi_1)\sin\phi_1\right]}{D^2} \tag{18}$$

$$\begin{aligned} g_2' = & \frac{r_2 r_3\begin{bmatrix} a\cos(q-\phi_1) - a\sin(q-\phi_1)\sin\phi_1\cos\phi_1 \\ +r_2\cos^2(q-\phi_1)\sin\phi_1 \end{bmatrix}}{r_4\cos\phi_2 D^2}g_1 \\ & - \frac{r_2 r_3\cos(q-\phi_1)\cos\phi_1\sin\phi_2}{r_4\cos^2\phi_2 D}g_2 \\ & + \frac{ar_2 r_3\sin(q-\phi_1)\cos\phi_1\sin\phi_1}{r_4\cos\phi_2 D^2} \end{aligned} \tag{19}$$

$$g_3' = -\frac{r_2 r_3 \begin{bmatrix} r_2 \cos^2(q-\phi_1)\cos(\phi_1-\phi_2) \\ +a\cos(q-\phi_1)\cos(\phi_1-\phi_2)\sin\phi_1 \end{bmatrix}}{\cos\phi_2 D^2} g_1 - \frac{a r_2 r_3 \begin{bmatrix} \cos(q-\phi_1)\sin(\phi_1-\phi_2)\cos\phi_1 \\ +a\sin(q-\phi_1)\sin(\phi_1-\phi_2)\sin\phi_1 \end{bmatrix}}{\cos\phi_2 D^2} g_1 - \frac{r_2 r_3(\cos(q-2\phi_1)+\cos\phi_1)}{(\cos(2\phi_2)+1)D} g_2 - \frac{a r_2 r_3 \sin(q-\phi_1)\sin(\phi_1-\phi_2)\sin\phi_1}{\cos\phi_2 D^2} \quad (20)$$

By using second order influence coefficients the accelerations of the dependent links can be calculated as follows:

$$\ddot{\phi}_i = g_i'\dot{q}^2 + g_i\ddot{q} \qquad (i=1,2,3) \quad (21)$$

The acceleration of the center of gravity for the links can be expressed with the influence coefficients as:

$$a_{G_i} = \left(\sqrt{a_{G_i}^{x\,2} + a_{G_i}^{y\,2}}\right)\dot{q}^2 + \left(\sqrt{v_{G_i}^{x\,2} + v_{G_i}^{y\,2}}\right)\ddot{q}$$
$$a_{G_i}^x = \frac{\partial v_{G_i}^x}{\partial q} + \sum_{j=1}^{3}\frac{\partial v_{G_i}^x}{\partial \phi_j} g_j\,;\ a_{G_i}^y = \frac{\partial v_{G_i}^y}{\partial q} + \sum_{j=1}^{3}\frac{\partial v_{G_i}^y}{\partial \phi_j} g_j \quad (22)$$

$$a_{G_2}^x = -c_2\cos q$$
$$a_{G_2}^y = -c_2\sin q$$
$$a_{G_3}^x = -c_3\cos\phi_1 g_1^2 + \sin\phi_1 g_1'$$
$$a_{G_3}^y = -c_3\sin\phi_1 g_1^2 - \cos\phi_1 g_1'$$
$$a_{G_4}^x = -r_3\cos\phi_1 g_1^2 - c_4\cos\phi_2 g_2^2 - c_4\sin\phi_2 g_2' - r_3\sin\phi_1 g_1' \quad (23)$$
$$a_{G_4}^y = -r_3\sin\phi_1 g_1^2 - c_4\sin\phi_2 g_2^2 + c_4\cos\phi_2 g_2' + r_3\cos\phi_1 g_1'$$
$$a_{G_5}^x = -r_3\sin\phi_1 g_1^2 - r_4\cos\phi_2 g_2^2 - r_3\sin\phi_1 g_1' - r_4\sin\phi_2 g_2'$$
$$a_{G_5}^y = 0$$

The kinetic energy of the mechanism can be written as:

$$E_k = \frac{1}{2}\sum_i\left(m_i g_i'^2 + I_{G_i} g_{\theta_i}'^{\,2}\right)\dot{q}^2 \qquad (i=1,2,3) \quad (24)$$

Equivalent inertia is

$$\Im = \sum_i\left(m_i g_i'^2 + I_{G_i} g_{\theta_i}'^{\,2}\right) \qquad (i=1,2,3) \quad (25)$$

Equivalent force including gravity, cutting and friction forces and torques can be expressed as:

$$\mathbf{Q} = \sum_i\left(\mathbf{F}_i g_i' + \mathbf{T}_i g_{\theta_i}'^{\,2}\right) \qquad (i=1,2,3) \quad (26)$$

For any mechanism, reducing the inertia of all links to the input link provides great convenience in terms of controllability of the mechanism. The generalized equation of motion of any reduced mechanism is given by Paul [13],

$$\Im\ddot{q} + C\dot{q}^2 = Q \quad (27)$$

Where C is called centripetal coefficient and calculated as:

$$C = \frac{1}{2}\frac{d\Im}{dq} \quad (28)$$

The mathematical model and the output torque of a DC motor with a reducer can be written as:

$$\frac{di_m}{dt} = \frac{1}{L_m}\left(V_m - R_m i_m - nK_g\dot{q}\right) \quad (29)$$

$$T_o = n\left(-nJ\ddot{q} - nB\dot{q} + K_m i_m - T_f\right) \quad (30)$$

Where i_m, L_m, V_m, R_m, K_g, J, B, K_m, T_f and n are the current, inductance, input voltage, resistance, motor voltage constant, mass moment of inertia, viscous damping, motor torque constant motor torque losses and reducer speed ratio respectively.

The only torque acting on the mechanism is T_0 applied to the crank. With calling all the available external forces as Q_0, then Q becomes,

$$Q = Q_0 + T_0 \quad (31)$$

By rearranging the Equations (26-31), one can obtain the following equation

$$\ddot{q} = \frac{-n^2 B_m\dot{q} - C\dot{q}^2 + nK_m i_m - nT_f + Q_0}{\Im + n^2 J} \quad (32)$$

The state space equations are expressed as:

$$z_1 = q;\ z_2 = \dot{q};\ z_3 = i_m \quad (33)$$

$$\dot{z}_1 = z_2$$
$$\dot{z}_2 = \frac{-n^2 B_m z_2 - Cz_2^{\,2} + nK_m z_3 - nT_f + Q_0}{\Im + n^2 J} \quad (34)$$
$$\dot{z}_3 = \frac{1}{L_m}\left(V_m - R_m z_3 - nK_g z_2\right)$$

4. CONTROLLER DESIGN

Sliding mode control is an extremely robust control technique that can provide the desired dynamic behavior despite the uncertainties, parameter changes and disturbances in the physical systems. Since it is an effective control method and easy to implement, it is widely used in many control studies. With the development of switching technology, its popularity and usage area has developed considerably.

In this study, the sliding mode control technique was used to rotate the crank of the presented mechanism, which was designed optimally, at a constant speed. The block diagram and equations of this control method are given below (Figure 5).

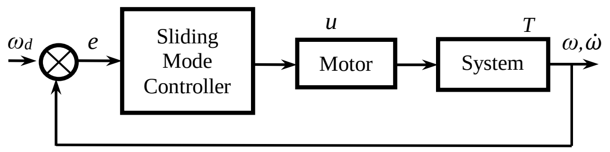


Figure 5. Sliding mode controller block diagram

In Fig. 5 ω_d, u, T, ω and $\dot{\omega}$ are the desired angular velocity of the crank, control signal, torque, output velocity and acceleration of the crank respectively.

The sliding surface s of the controller is defined as:

$$s(x,t) = \lambda e_1(t) + e_2(t) \tag{35}$$

Here, λ is the slope of the sliding surface, e_1 and e_2 represent the tracking errors which are given below.

$$\begin{aligned} e(t) &= \begin{bmatrix} e_1(t) & e_2(t) \end{bmatrix} \\ &= \begin{bmatrix} \omega_1(t) - \omega_{1d}(t) & \dot{\omega}_1(t) - \dot{\omega}_{1d}(t) \end{bmatrix} \end{aligned} \tag{36}$$

The control signal (voltage supplied to DC motor) of the sliding mode controller system is obtained as:

$$u(t) = \begin{bmatrix} \left(-\lambda(\Im + J)\dot{e}_1 + B_m z_2 + T_f - Q_0 + C z_2^2\right)\dfrac{R_m}{K_m} \\ +K_g z_2 + L_m \dot{z}_3 \end{bmatrix} - Ksign(s) \tag{37}$$

Here, K is a positive constant and usually determined with trial error method. The control simulations in this study were made by using Matlab software. The physical parameters of the DC motor used in simulation were taken from ref. [1] as given in Table 2.

TABLE II
THE PHYSICAL PARAMETERS OF THE DC MOTOR

Parameter	K_g (Vs/rad)	R_m (Ω)	B_m (Nms/rad)	K_m (Nms/A)	Lm (H)	J(kgm²)
Value	0.678	0.4	0.226	0.678	0.05	0.0565

The crank of the mechanism is supposed to rotate with a constant angular velocity of 5 rad/s. While applying constant input voltage of 48 V with a transmission ratio of 10 the output angular velocity of the crank fluctuates between 6.1-6.4 rad/s (Fig. 6). The small peaks after settling time are due to cutting force interactions. After applying sliding mode control its fluctuations are reduced significantly. In control applications, K and λ are determined as 1000 and 50 respectively. The cutting force is supposed to be 50 N. The steady state error of crank velocity is calculated as 0.2% (zoomed area of Figure 6). Input voltage is bounded between -48 V and +48 V. A section of input voltage is given in Figure 7.

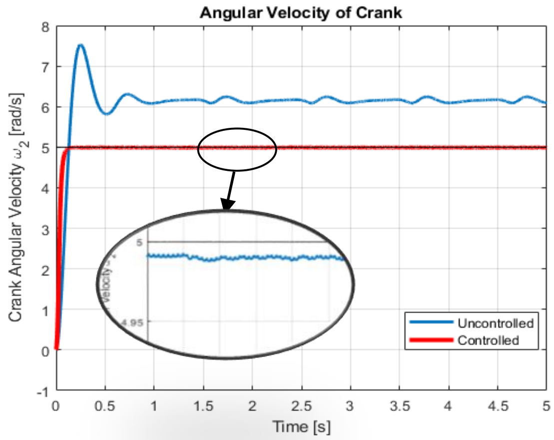


Figure 6. Crank angular velocity

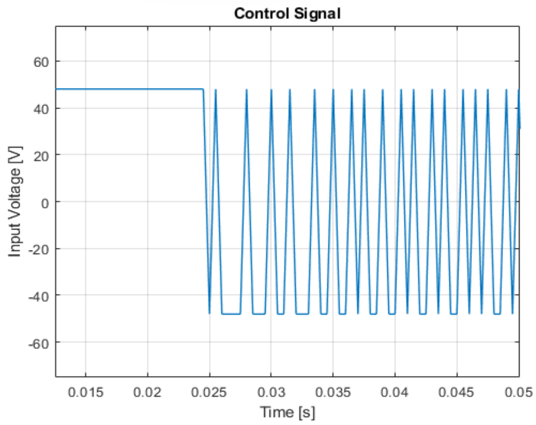


Figure 7. Input signal

5. CONCLUSION

In this study, an optimum design and control of a quick return mechanism used in a jewelry welding powder production machine are presented. For optimum design of the mechanism with a cost function, some constraints as transmission angle, time ratio and lengths of the links of the mechanism were defined with design variables. Matlab fmincon algorithm was used for the optimization and minimum lengths of the mechanism were determined. For obtaining the equation of motion of the mechanism Eksergian's Equation of Motion method was used and all the inertias of the mechanism were reduced on the crank. For controlling the crank of the mechanism in constant angular velocity sliding mode control method was used. After the controlling, the fluctuation of the angular velocity of the crank was reduced to admissible level.

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