Design of a New Bed Base Mechanism System

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Abstract

In daily life, most of the devices, machines and furniture, which we use in our homes and make our life easier, have many different opening and closing systems. Some criteria may be required for the opening/closing shape of systems. For example, in some systems, it may be desirable that opened part uses minimum volume, facilitates daily life, or the system is balanced in each opening position. Those type of opening and closing systems are widely used sofa beds, seats and bed bases in the furniture sector. Many opening/closing systems have various locking mechanisms which are used to ensure opened final position or in some cases it is held in the final position by the force of the human arm. In this study, a kinematic analysis was carried out to design a base mechanism which could be stabilized at every position instead of the final position, and a virtual prototype was made.

Key Words

Bed-base mechanism, Kinematic analysis, Virtual prototyping, Galerkin method

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1. INTRODUCTION

Weights, which rotate around an axis, are used in household goods, in the automotive sector and a lot of machines, tools, devices in industry life because of various reasons. The door of the dishwasher, oven's door, freezers whose doors are opened from the upper side, bonnet and trunk lid of the car can be shown as examples. In addition, it is also possible to encounter such caps in many applications. In some of the applications, the cap is ensured final position with the force of the human arm while in some of the applications caps remain the final position thanks to some mechanism. This process is done with the aid of a rod or a kind of guy wire after opened bonnet of the car. For some of the bed base mechanism, it is still used the human arm for the end position. It can be an accident which can cause injury or death in manual position balance applications which include heavy caps. In addition, lifting heavy caps with the human arm is not a desirable situation for human nowadays.

In this study, a new mechanical design has been improved to ensure that each position in the cap is statically balanced. The designed system is basically based on the principle that the amount of potential energy saved or lost by weight of the system is stored in the spring or returns to the system by the spring. In order to keep balance in every position of the system, change of the potential energy of the system must be equal to the change of the elasticity energy of the spring. The functional relationship between the changing of potential energy of the system and the changing of elastic energy in spring is not linear. That's why a mechanism is needed to ensure this relationship. In this work, crank and connecting rod mechanism, whose part count is less, are selected to obtain a linear function. The angular rotation of the system to be balanced is transferred to the crank arm. This angular rotation is transferred from the piston which is last part of slider-crank mechanism to spring as a linear motion. This transfer is carried out according to a transfer function depending on the kinematic dimensions of the slider-crank mechanism. If the transfer function is determined according to the relation between the weight of the system and energy of the spring, the system will be balanced. This is possible by the selection of the kinematic dimensions of the slider-crank mechanism properly. The determination of the mechanism dimensions according to the functional relationship between the input and output parts is called function synthesis in Andrew, Murray and Myszka (2017), Kafash and Nahvi (2017), Chavan, and Joshi (2010), Norton (2002).

There are two approaches to solving the problem of function generation. The first, investigation of the mechanisms that generate the desired function without errors. The second is to find the unknown parameters of the mechanism that can generate the desired function in an acceptable approximation. The number of mechanisms which generate the desired function is limited, this obstacle is mentioned in Artobolevskii (1975). It is difficult to produce accurate results because of the complexity of the algorithm. It is more logical to use approximate synthesis methods instead of exact synthesis because the investigation of the mechanism which generates function without error is both expensive and difficult.

Graphical approaches have been used to find solutions for the synthesis problems due to the fact that mathematical relations between the input and output part of the mechanics are very complicated. However, graphical method solutions, which can be examined in Darina, Peter and Gabriel (2016),are a long way to go for each new data. Computers have provided the opportunity to develop many analytical methods in the problem of function generation. The common point of many analytical methods developed is to minimize the difference between the desired function and the function generated by the mechanism. The methods have different names like galerkin, least square, accuracy points according to the mathematical approaches used when the error function is minimized. Bahmyari, Khedmati and Soares (2017) is example for galerkin method; Akcali and Dittrich (1989), Ramírez, Nogueir, Khelladi, Chassaing and Colominas (2014) can be given as example for the least squares method; for accuracy point method Diab and Smaili (2008), Jaiswal and Jawale (2017) can be shown.

2. MATERIALS AND METHOD

In this study, analytical solution method is used to design the slider-crank mechanism according to the function synthesis. Then simulation of the bed base mechanism is performed.

2.1. Mathematic Model of Equilibrium Mechanism

The kinematic diagram and parameters of the mechanism that rotates around a certain axis and self-balancing at each position are shown in Figure 1. In order to the weight of the system passing through from θ_0 angular position to any θ angular position can be statically balanced by means of a linear spring, the sum of the elastic energy in the spring and the potential energy of the weight at two positions as θ_0 and θ should be equal to each other in Equation 1.

$$Wl \sin \theta_0 + \frac{1}{2} k x_0^2 = Wl \sin \theta + \frac{1}{2} k x^2 \tag{1}$$

Here, k is the spring constant, x_0 is the amount of elongation at θ_0 angular displacement, x is the amount of elongation at θ angular displacement. Equation 1 can be written as Equation 2.

$$Wl(\sin\theta_0 - \sin\theta) = \frac{1}{2}k(x^2 - x_0^2)$$ (2)

If x is left alone in Equation 2, in Equation 3 can be obtain.

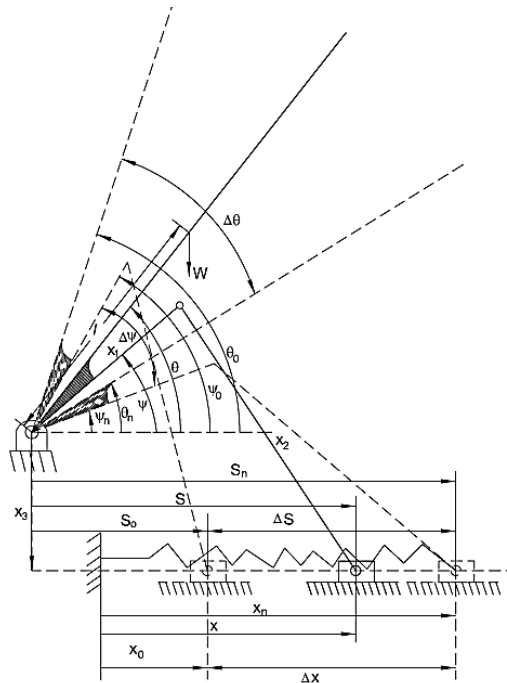$$x = \sqrt{x_0^2 + \frac{2Wl}{k}(\sin\theta_0 - \sin\theta)}$$ (3)

Figure 1. Kinematic parameters of equilibrium mechanism

Slider-crank mechanism design in Figure 1 have been carried out using the following inputs:

x_0: The amount of elongation in the spring when the object to be balanced is in the initial position (mm).
W: Weight of object to be balanced (Newton).
l: Distance from the axis of rotation of the object to the center of gravity (mm).
k: Linear spring coefficient (Newton/mm).
θ_0: The angle between the center of gravity and the axis passing through the axis of rotation, (degree).
$\Delta\theta$: Angular motion amount of object to be balanced (degree).

Kinematic dimensions of the slider-crank mechanism are x_1, x_2, x_3 and s_0 are shown in Figure 1. These parameters definition are as follows:

x_1: Crank length of slider-crank mechanism (mm).
x_2: Connecting rod length of slider-crank mechanism (mm).
x_3: Vertical distance between crankshaft joint point and piston (mm).
ψ_0: Initial angle of the crank rod (degree).
s_0: Horizontal distance between crankshaft joint point and piston joint point (mm).

Equation 3 is valid provided that $\theta_0 > \theta > \theta_n$. Slider-crank mechanism is settled on the purpose of elongation of linear spring, it is shown in Figure 1. The relationship between the angular rotation ψ of the input arm of the mechanism and the linear displacement s of the output arm is as follows:

$$s = f(\psi), \quad \psi_0 \geq \psi \geq \psi_n$$ (4)

The following equations always apply for Equation 3 and Equation 4.

$$\Delta x = \Delta s; \quad \Delta \theta = \Delta \psi$$

$$\Delta x = x_n - x_0; \quad \Delta s = s_n - s_0 = x(\theta_n) - x(\theta_0) \tag{5}$$

If the kinematic dimensions of the slider-crank mechanism are determined in such a way that the difference of the functional changes between Equation 3 and Equation 4 is minimized, self-balance will be possible to in every position.

2.2. Designed Slider-Crank Mechanism

Function synthesis of slider-crank mechanism are performed according to functional relation between rod angle (ψ) with displacement of piston (s). Design parameters of slider-crank mechanism are represented in Figure 2. Here, x_1 is the length of the crank, x_2 is the length of the rod, x3 is the vertical distance from of piston to the crank pin center, ψ is the crank arm's motion start angle and s0 is the perpendicular distance from the piston to the $A0_1$ line. Two different functions can be mentioned in the design of the slider-crank mechanism. The firstly, it is the functional relationship between the arm's rotate angle (ψ) and linear displacement of the piston (s). The other is a relation between the arm's rotate angle (ψ) and horizontal angle of connecting rod (δ). In this study, ψ-s functional relation is concerned. As shown in Figure 2, the squares of the vectorial expression are written from the AO_1BC loop and horizontal angle of the connecting rod is eliminated as a result the following equation;

$$x_1^2 - 2sx_1 Cos\psi + 2x_1 x_3 Sin\psi + x_3^2 + s^2 - x_2^2 = 0 \tag{6}$$

Relation between connecting rod angle with function variable and relation between displacement of the piston with function dependent variable can be written as;

$$\psi = \psi_0 + \psi'; \quad \psi' = R_x(x - x_0); \quad R_x = \frac{\Delta\psi}{\Delta x}$$

$$s = s_0 + s'; \quad s' = R_y(y - y_0); \quad R_y = \frac{\Delta s}{\Delta y} \tag{7}$$

In Equation 7, $\Delta\psi = \psi_n + \psi_0$, $\Delta s = s_n + s_0$ are crankshaft and piston operating range, respectively, $\Delta x = x_n + x_0$, $\Delta y = y_n + y_0$ are amount of changes of dependent and independent variable.
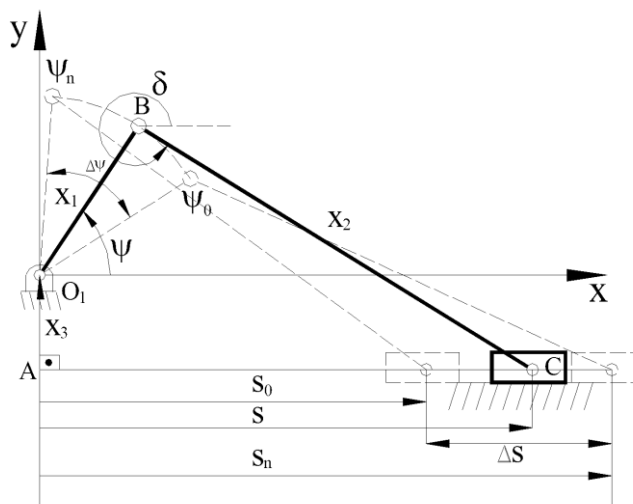

Figure 2. Slider-crank mechanism parameters

If Equation 7 is substituted in Equation 6, equation of motion is obtained such as G($s_0, \psi_0, x_1, x_2, x_3, x, y$).

$$2s's_0 + s'^2 + Z_3(Sin\psi_0 Cos\psi' + Cos\psi_0 Sin\psi') + Z_1(-s_0 Cos\psi_0 Cos\psi' + s_0 Sin\psi_0 Sin\psi'$$
$$- s' Cos\psi_0 Cos\psi' + s' Sin\psi_0 Sin\psi') + Z_2 = 0 \tag{8}$$

In Equation 8, $Z_1 = 2x_1$; $Z_2 = x_1^2 + x_3^2 + s_0^2 - x_2^2$; $Z_3 = 2x_1 x_3$. Equation 8 can be converted five unknown nonlinear equations that can be calculated according to Subspace, Galarkin and the other methods are as follows:

$$(a_i s_0 + g_i) + Z_3(c_i Sin\psi_0 + b_i Cos\psi_0) + Z_1(-c_i s_0 Cos\psi_0 + b_i s_0 Sin\psi_0$$
$$- d_i Cos\psi_0 + e_i Sin\psi_0) + f_i Z_2 = 0 \quad i = 1, 2, 3, 4, 5 \tag{9}$$

In this work, Galarkin method is prefered. (ai, bi, ci, di, ei, fi, gi) coefficients are calculated according to Galerkin method as follows.

$$a_i = 2\int_{x_0}^{x_n} s' w_i(x) dx; \quad b_i = \int_{x_0}^{x_n} Sin\psi' w_i(x) dx; \quad c_i = \int_{x_0}^{x_n} Cos\psi' w_i(x) dx; \quad d_i = \int_{x_0}^{x_n} s' Cos\psi' w_i(x) dx$$

$$e_i = \int_{x_0}^{x_n} s' Sin\psi' w_i(x) dx; \quad f_i = \int_{x_0}^{x_n} w_i(x) dx; \quad g_i = \int_{x_0}^{x_n} s'^2 w_i(x) dx \qquad i = 1, 2, 3, 4, 5 \tag{10}$$

Equation 9 is rearranged and Z_1, Z_2, Z_3 are eliminated thus Equation 9 obtain depend on ($\tan\psi_0$) and (s_0).

$$a_{pk} s_0^2 + (b_{pk} + c_{pk} \cos 2\psi_0 + d_{pk} \sin 2\psi_0)s_0 + e_{pk} + f_{pk} \cos 2\psi_0 + g_{pk} \sin 2\psi_0) = 0 \quad k = 1, 2 \tag{11}$$

Coefficients in Equation 11 can be calculated by computer-aided mathematics programs. If s_0 is eliminated and ψ_0 is applied mathematical manipulation which is performed in Akcali and Mutlu (2006), 12 th-degree polynomial can be obtained as follows:

$$w_0 + w_1 t + w_2 t^2 + w_3 t^3 + w_4 t^4 + w_5 t^5 + w_6 t^6 - w_5 t^7 + w_4 t^8 - w_3 t^9 + w_2 t^{10} - w_1 t^{11} + w_0 t^{12} = 0 \tag{12}$$

wi i=1,2.,...,6 in Equation 12 can be calculated computer aided mathematics. Here, t= $tan\frac{\psi_0}{2}$. There are twelve solutions of the ψ_0.

$$\psi_{oi} = 2 \arctan t_i; \quad i = 1, 2...., 12 \tag{13}$$

If s_0^2 is eliminated in Equation 11, s_0, x_1, x_2, x_3 can be written as follows for i=1,2,....,12;

$$s_{oi} = -\frac{a_{p1}e_{p2} - a_{p2}e_{p1} + (a_{p1}f_{p2} - a_{p2}f_{p1})\cos 2\psi_{0i} + (a_{p1}g_{p2} - a_{p2}g_{p1})\sin 2\psi_{0i}}{a_{p1}b_{p2} - a_{p2}b_{p1} + (a_{p1}c_{p2} - a_{p2}c_{p1})\cos 2\psi_{0i} + (a_{p1}d_{p2} - a_{p2}d_{p1})\sin 2\psi_{0i}}; i = 1,..,12 \tag{14}$$

$$x_{1i} = \frac{Z_{1i}}{2} \quad i = 1, 2, ...12 \tag{15}$$

$$x_{3i} = \frac{Z_{3i}}{2x_{1i}} \quad i = 1, 2, ...12 \tag{16}$$

$$x_{2i} = \sqrt{x_{1i}^2 + x_{3i}^2 + s_{0i}^2 - Z_{2i}} \quad i = 1, 2, ..12 \tag{17}$$

Z_{1i}, Z_{2i}, Z_{3i} are any term depend on a,b,c,d,e,f,g coefficients. Those values can be calculated automatically by created software program.

2.3. Design of Balancing Mechanism

Various kinematic dimensions of the slider-crank mechanism have been obtained with the help of input parameters. Among the results obtained, the best in terms of functioning error and size ratios were researched. In addition, the dimensions of the slider-crank mechanism must be compatible with the volume constraint of bed base sizes to be installed. For this purpose, kinematic dimensions of the slider-crank mechanism can be obtained by computer program after entering the input parameter. The parameters that are easiest to change by the designer are the spring coefficient (k) and the initial elongation amount (x_0) at the spring. In computer aided numerical works, as the spring constant (k) and the initial elongation amount (x_0) were increased, the arm lengths of the slider-crank mechanism decreased. As a result, the most suitable design dimensions were determined with the help of computer program.

The design results obtained in this study were determined by taking into consideration the physical dimensions and mass of the bed base. According to this, the physical parameters of the bed base input to the computer program are determined as follows:

Cap weight = 40 kg
Cap weight (W) = 392.4 N
Place of weight center (l) = 1000 mm
Initial angle of cap (θ_0) = 70^0
Required minimum angle for balance (θ_n) = 10^0
Cap work range ($\Delta\theta$) = 60^0
Selected spring coefficient (k) = 8.240 N/mm
Selected first extension amount (x_0) = 100 mm

After using input parameter, most appropriate slider-crank mechanism dimensions in Figure 1 are obtained as follows:

x_1 = -117.7 mm; x_2 = 103.8 mm; x_3 = -112.4 mm; ψ_0= -53.70; s_0 = -172.0 mm; Δs =115.6 mm

Changing of function error of slider-crank mechanism according to the angle range of cap is shown in Figure 3. This graph shows the difference between the theoretical spring elongation in Equation 3 and the spring elongation amount performed by the mechanism. As shown in Figure 3, the functional error varies uniformly from about 0.025 mm to 0.030 mm.
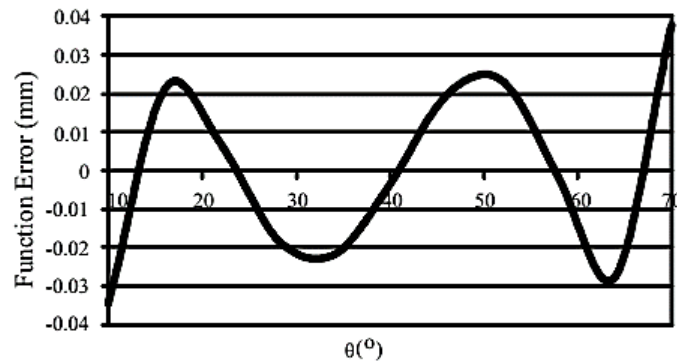

Figure 3. Functional error changing

The difference between the theoretical F force that balances the designed mechanism and the actual force (F_y) produced by the spring shows the success of the balancing. For this calculation, the external forces acting on the mechanism at any position in the working range of the slider-crank mechanism in Figure 1 can be written as:

$$W\delta h + F\delta s = 0 \tag{18}$$

To obtained the relation between the force F and the kinematic parameters of the mechanism, the following expression can be written from the loop equations of the slider-crank mechanism.

$$\frac{\delta s}{\delta \psi} = \frac{2x_1 x_3 \cos\psi + 2x_1 s \sin\psi}{2x_1 \cos\psi - 2s} \tag{19}$$

In here, $h = l\sin\theta$, $\delta h = l\cos\theta\,\delta\theta$, $\delta\psi = \delta\theta$. F force can be written as following:

$$F = \frac{Wl\cos\theta(x_1\cos\psi - s)}{x_1 x_3 \cos\psi + x_1 s \sin\psi} \tag{20}$$

There is need an expression such as $s = f(\psi)$ in Equation 20. From the loop equation of mechanism, s can be written as follows:

$$s = \frac{-B \pm \sqrt{B^2 - 4AC}}{2A} \tag{21}$$

$$A = 1; \quad B = -2x_1\cos\psi; \quad C = x_1^2 + x_3^2 - x_2^2 + 2x_1 x_3 \sin\psi; \tag{22}$$

The real force F_y, as opposed to the theoretical force F which provides the equilibrium in Equation 20, as follows:

$$F_y = -k(x_0 + s - s_0) \tag{23}$$

The error E of the balancing force is given below:

$$E = F - F_y$$

<div align="right">(24)</div>

Chancing of the error according to θ is shown in Figure 4.
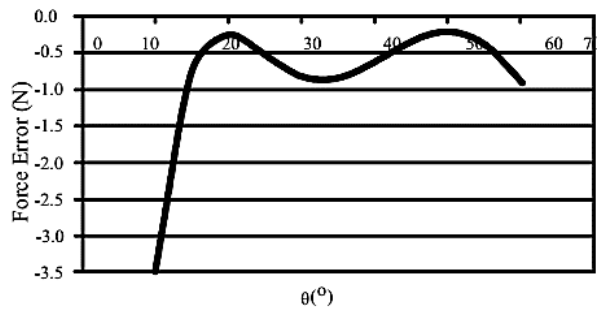

Figure 4. Balancing force error graph

As it is seen in Figure 4, the force error is maximum at the first and last position of the mechanism. Nevertheless, it does not exceed ± 0.5 N in a very wide range. It is seen on the virtual model that these differences, which are quite small, are damped by the friction forces in the bearings. The balancing mechanism obtained from the equations above has been assembled as in Figure 5. Number 1 represent to bed base, number 2 is the cap of the bed base, number 3 is designed the mechanism and number 4 is linear spring used in mechanism.
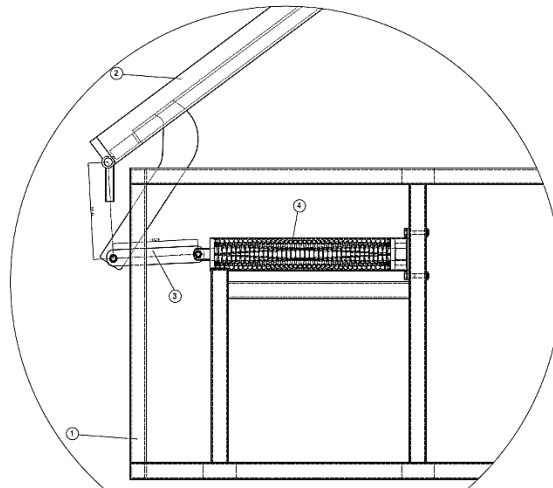

Figure 5. Inside view of the balancing mechanism

Figures 6 and 7 show the change in angular velocity and angular position of the cap when the cap is in the closed position when a person who wants to open the cap applies a force of about 36 N for 1 second to the cap. As it can be seen in Figure 6, when the cap is subjected to a force of 36 N for 1 second, the cap reaches a maximum angular velocity of 30 deg/s after 1 second. When the force is removed, the angular velocity of the cap reaches zero at about 1.1 seconds, it stabilizes itself. Meanwhile, it can be seen in Figure 7 which the cap makes an angular displacement of about 40°. The simulated images of the bed base mechanism in Figure 8 are shown.
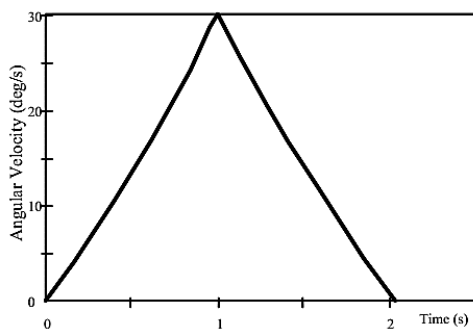

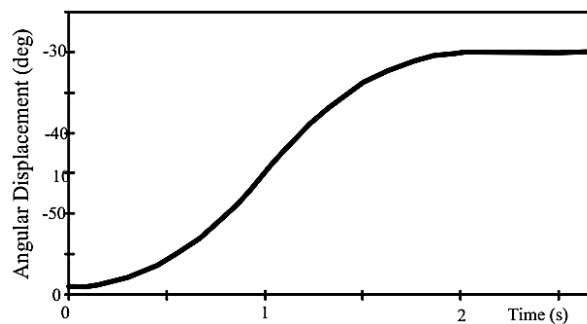**Figure 6. Change of Angular Velocity of Cap by Time**

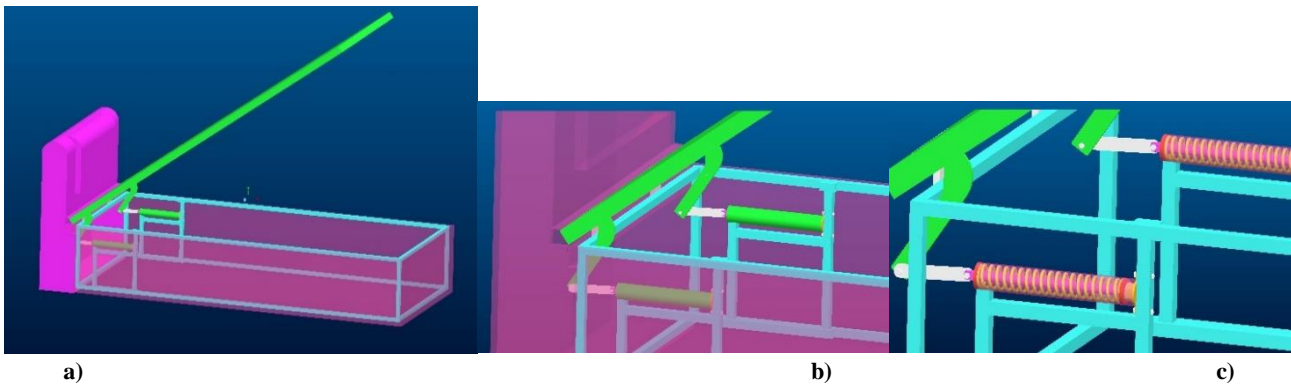**Figure 7. Change of Angular Position of Cap by Time**

Figure 8. a) bed base image b) balancing mechanism c) inside view of the balancing mechanism

3. RESULTS

The main feature of bed base that the moving the bed part and using the other part of the bed base efficiently. However, it is necessary the human arm power because of the weight of the cap of the bed base when it is opening/closing. In addition, stabilization is a very important topic for this systems.

In this work, the new bed base mechanism is designed according to balance every position. The designed system has consisted of spring and slider-crank mechanism. If the potential energy changing of the cap of the bed base is equal to elastic energy of spring, it is possible to stabilization of the cap of the bed base in every position. The balancing system for the bed bases is analytically created. Conformity of the solid model of the designed mechanism to actual operating conditions was carried out by using the SimWise 4D software. In the model, spring, the mechanism parts, the coefficient of friction between the rotary joints, the rotary and sliding joints between the members are defined in accordance with the actual working conditions. In the experiments made on the model, it was seen that the cap bed base performance has a big harmony according to the theoretical results. Moreover, if the cap is manually reach to the desired position and then release, it is observed the cap balances itself in the computer-based experiments.

In current systems, it needs to very much human power for the opening of cap and closing it when it opened. This phenomenon is getting hard to use current system in daily life. Thanks to the newly designed system, more than 40 kg weight of the bed can be opening/closing easily with less than 1 kg human force owing to weight balancing. Therefore, using bed-base system is more easy and safe via the new system.

ACKNOWLEDGEMENTS

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