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Article

Adaptive Balancing of Robots and Mechatronic Systems

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Abstract: Present paper is dealing with the adaptive static balancing of robot or other mechatronic arms that are moving in vertical plane and whose static loads are variable, by using counterweights and springs. Some simple passive and approximate solutions are proposed and an example is shown. The active and exact solutions by using adaptive real time control in the case of unknown variation of static loads are simulated on VIPRO platform developed at Institute of Solid Mechanics of Romanian Academy.

Keywords: adaptive, balancing, counterweight, mechatronic system, robot

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1. Introduction

Static balancing of a mechanical system is one of the first demanding steps in the design process of any mechanical system which is moving with relatively small accelerations and which is overcoming relatively large forces, in order to match first of all the need of energy consumption, and it is also an important aspect of the overall performance of it [1].

Static balancing can be regarded as the total or partial cancellation of the mechanical effects (force or moment) of static loads to the actuating system of mechanical system, in all configurations, respectively in a finite number of configurations, from functioning domain, under quasi-static conditions [1], [2]. The effect of this action is the maintaining of the mechanical system in a rest state at any configuration or at a finite number of configurations respectively, from working field, and its actuators are not required to overcome the static loads. The movement inside working field can be done with a power-less actuating system which consumes energy only for overcoming the friction forces and balancing errors. The friction forces are dependent on the motion sense and are opposed to the movement, contributing in this way to the maintaining of the mechanical system in a rest state.

The main static load is given by gravitational field of Earth, i.e. the weight forces of all bodies that compose the mechanical system. In the case that weight forces are the only static loads of static balancing operation then the mechanical system is called gravity compensate. Also the effect of these loads to the actuating system is present only in the case that the mechanical system is not working in horizontal plane with respect to gravity field. Consequently, the potential energy of mechanical system remains constant or approximately constant and the center of gravity of mechanical system remain fixed with respect to a referential frame or is moving along a horizontal direction or into a horizontal plane with respect to Earth. Another important observation and hypothesis is that due to the small displacements of the centers of gravity of mechanical system bodies, with respect to the distance from the center of the Earth to each body mass centers, then the weight forces are constant. In this case the actuators of mechatronic system are not required to sustain the weight of its moving elements.

But, in the case of a manipulation robot for example, as is also the case of cranes too, the manipulation weight could be variable in steps. As is presented in article [5] for the case of an industrial robot [9] which is designed to manipulate payloads of 16 kg maximum mass, balanced by springs for a middle weight mass of 8 kg, the forces induced in actuating system are amplified about 4 times when the weight is increasing or decreasing from the mean value. In fact, in terms of resistance moments (torques) at shafts of rotating actuators, as is shown in Figure 1.a for the most

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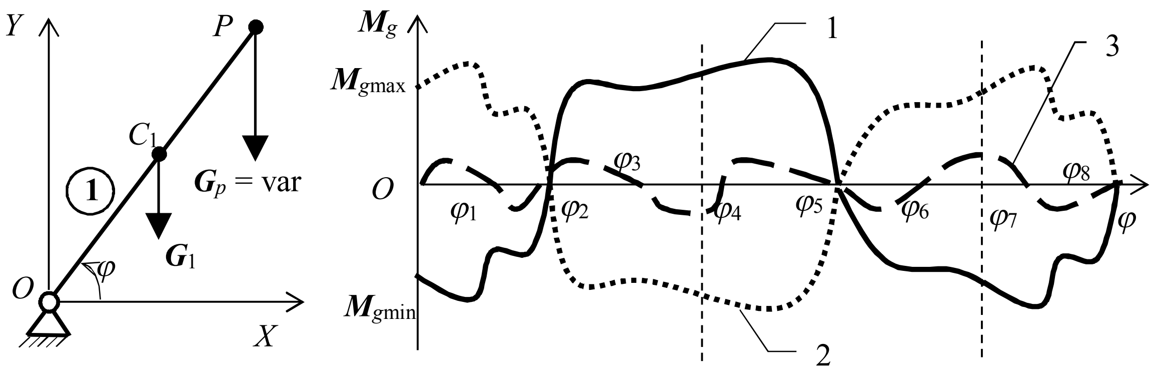
frequent case of an articulated arm, this variation occurs (and has a cosine variation) even the payload has constant weight G_p . In case the load has variable weight (as is the case of oil pump-jack systems for example [4]) then a more complex variation is possible (Figure 1.b – solid curve line 1). A special situation is the one when the variation is known and it is repeating during one cycle. In this case the adaptive solution could be a passive one (i.e. not actuated). Otherwise the balancing system should adapt in real time by using a local and supplementary actuation system and by aid of a controlling system and the required sensors and transducers [2].

Many other mechanical systems, which are automatized more and more in these days, becoming in this way mechatronic systems, have to overcome variable payloads or resistant forces during the functioning. Beside the manipulation robots used in palletizing for example [5-9], articulated cranes [10-12] and pump-jack oil pumps [13-15], a large category of ergonomic manipulators [16-18] are facing the variable payload and have to adapt to this condition.



(a) **(b)**

Figure 1. (**a**) Rocker arm (**b**) Gravitational moment variation of the variable weight forces

By balancing, another moment which is opposing the load moment (Figure 1.b – dotted curve line 2) should be induced in order to compensate or eliminate the effect of load. If the difference between the load moment and the balancing moment is zero then the system is perfect (exact) balanced in all positions from its work field [19]. If there are only some positions where the difference is zero (Figure 1.b – discontinuous curve line 3) then an approximate balancing is obtained [20].

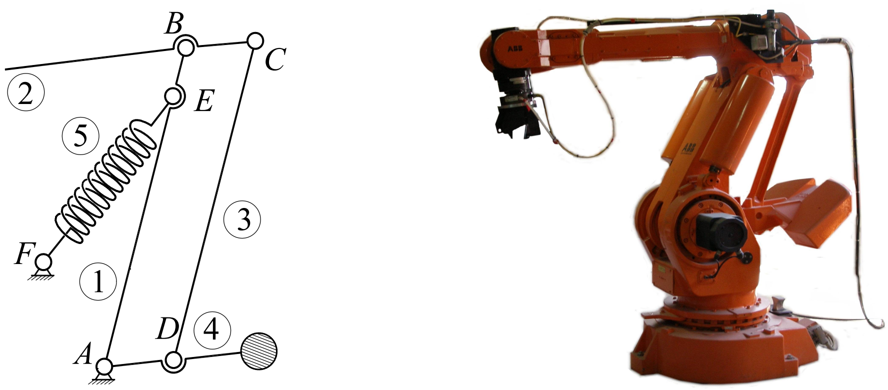


Figure 2. Industrial robot static balanced by counteweight and spring

In order to compensate the effects of static loads that depend to displacements then forces which depend also to displacements should be used. The main candidates are the weight forces represented by counterweights, and the elastic forces of springs or gases. Industrial robots are using both of these solutions (Figure 2) for example ABB industrial robot of IRB 6499 RF model [6].

Even in the case of static balancing by using counterweights the overall mass of the mechanical system is increased and from dynamics point of view the situation could become worst than in the case the mechanical system is even unbalanced, this solution is still useful and widely used in engineering because of the simplicity and for mechanical systems which are manipulating large loads and which are operating at low or moderate dynamics.

2. Adaptive Balancing by using Counterweights

The method of adding the counterweights involves the increasing of moving masses, overall size, inertia and the stresses of the mechanism links [20]. Some of the mechanical systems [1] accept this method because of operating at low or moderate dynamics, from security reasons or in cases where the right spring is difficult and costly to be obtained [2], or the spring balancing solution is too complicated to be fitted to [21]. Anyway, an internal mass redistribution so that parts of mechanical systems (actuators, electric motors, other mechanical transmission, either electric or electronic parts from controlling cabinet which could be relocated on the robot body) to act as counterweights like in the case of industrial robots [9], is first step when the static balancing problem starts [2].

Variation of gravitational moment given by the weight force of the rocking arm ① (Figure 1.a) $\boldsymbol{G}_1$ and by the variable payload $\boldsymbol{G}_p$ has the expression:

$$\boldsymbol{M}_g(t) = -\boldsymbol{G}_1\, OC_1 \cos\varphi(t) - \boldsymbol{G}_p(t)\, OP \cos\varphi(t) = f_1(t)\cos\varphi(t) \tag{1}$$

where:

$$f_1(t) = c_1 + c_2\, \boldsymbol{G}_p(t) \tag{2}$$

with:

$$c_1 = -\boldsymbol{G}_1\, OC_1 = \text{const. and } c_2 = -OP = \text{const.} \tag{3}$$

Then the balancing moment should be:

$$\boldsymbol{M}_b = \boldsymbol{M}_b(t) = f_2(t) \tag{4}$$

so that:

$$f_2(t) \cong -f_1(t)\cos\varphi(t) \tag{5}$$

Let suppose the case of the rocking arm ① which is gravity compensated for its weight $\boldsymbol{G}_1$ and for the weight of the constant part from the variation of payload $\boldsymbol{G}_{pc}$ (Figure 3) by a counterweight mounted fixed on the rocking arm ① at a proper distance on the opposite side then center of mass C_1 according to origin point O (not represented in the following). In this case:

$$c_1 = -\boldsymbol{G}_1\, OC_1 - \boldsymbol{G}_{pc}\, OP = \text{const.} \tag{6}$$

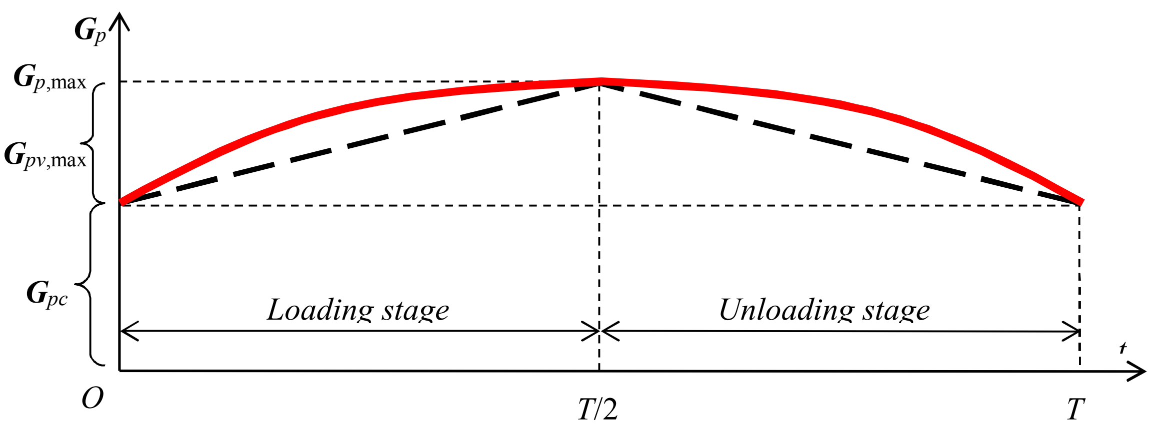


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Figure 3. Parabolic variation of a cyclic payload

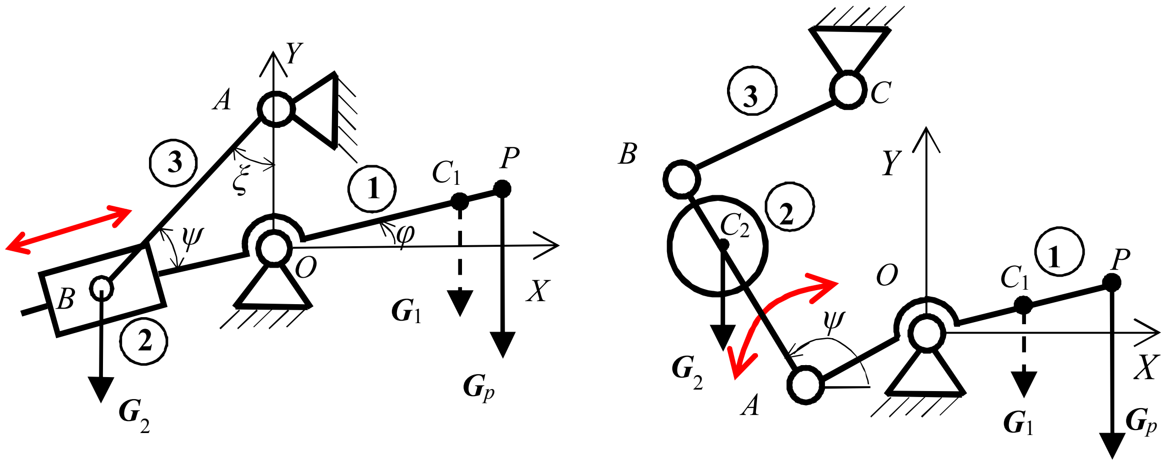
A variation of static load in linear form (as is represented in Figure 3 - dashed line) was studied in [22] and will be taken as comparison in Example section. By taking into consideration some frictions in the mechanical system of payload let suppose the variation of payload is known and cyclic with a symmetric variation of second degree evolution during one period of time *T* (Figure 2):

$$G_p(t) = G_{pc} + 4\frac{G_{pv,\max}}{T}t - 4\frac{G_{pv,\max}}{T^2}t^2 \quad \text{where} \quad t \in [0, T] \tag{7}$$

In order to gravity compensate the variable component G_{pv} by using also a supplementary counterweight then 2 possibilities could be taken into consideration: a variable weight of the additional counterweight or a movable counterweight with a fixed weight.

To make a variable weight for the counterweight is not impossible but is complicated and in order to compensate a continuous variation then liquid weights are needed, which are complicating much more the system and the dynamics became also very important. From practical point of view the changing of the location of the additional counterweight on the balanced element (as is the studied rocking arm ① in Figure 1.a) is a feasible solution when the speeds and accelerations are not very high.

There are also 2 possible ways of moving the additional counterweight ② relatively to the balanced element: by translating onto it (Figure 4.a without bar ③) or by rotating around a point which is becoming a joint on it by using an additional bar (Figure 4.b without bar ③).



(a) **(b)**

Figure 4. Movable counterweight in order to compensate variable payload

Despite of the pretentious prismatic joint the solution with translating counterweight became very popular [4, 15] due to the better dynamics of the multi-body system and due to the simplicity of the transmission of the supplementary actuator.

In case of a known cyclic variation of payload, as it is represented in Figure 3, then a passive adaptive solution is possible to be used. The simplest solution is presented in [15] by linking the counterweight ② to the mechanism base through a simple bar denoted by ③ and connected by 2 joints as is shown in Figure 4.a.

In Figure 3.a is presented the symmetric solution which is leading to a reduced number of exact balancing positions (maximum three). In this case the gravitational moment which has to be compensated is:

$$M_g(t) = -G_{pv}(t)\, OP \cos\varphi(t) = c\, f_3(t), \tag{8}$$

where:

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$$\boldsymbol{G}_{pv}(t) = \boldsymbol{G}_{p}(t) - \boldsymbol{G}_{pc} \tag{9}$$

$$c = -4\, OP \frac{G_{pv,\max}}{T} \tag{10}$$

and:

$$f_3(t) = t\,(1 - \frac{t}{T})\cos\varphi(t) \text{ where } t \in [0, T], \tag{11}$$

The balancing moment of counterweight ② has the expression:

$$\boldsymbol{M}_b(t) = \boldsymbol{G}_2\, OB(t) \cos\varphi(t), \tag{12}$$

where in the weight $\boldsymbol{G}_2$ could be count as added the part of the weight of the connecting bar ③ concentrated in point *B* because is fixed (Figure 3.a).

The position of the counterweight on the balanced arm ① has the expression:

$$OB(t) = \sqrt{AB^2 - OA^2 \cos^2 \varphi(t)} - OA \sin \varphi(t) \tag{13}$$

or:

$$OB^2 = OA^2 + AB^2 - 2\, OA\, AB \cos\xi \tag{14}$$

where:

$$\xi = \frac{\pi}{2} - \varphi - \psi \quad \text{and} \quad \sin\psi = \frac{OA}{AB}\cos\varphi \tag{15}$$

Unbalancing moment is given by relation:

$$\boldsymbol{M}_u = \boldsymbol{M}_e + \boldsymbol{M}_g, \tag{16}$$

and by comparing relations (12) and (13) with (8)-(11) is obvious that unbalancing can not be zero which is anyway shown in Example section. But the unbalancing is better than in the case of linear variation of payload in same condition.

As for the solution from Figure 4.b, with the rocking counterweight, the balancing is also approximate. The position of bar *BC* with respect to reference system *XOY* has a more complicated form (resulted by the solving of positional kinematics of *RRR* dyad composed by elements *BC* and *CA*) because it depends to:

- the position of points *A* and *B*;
- the length of bars *BC* and *CA*.

Analytic solving (and numerical one too [23]) leads to two mathematical solutions from kinematics but only one is correct from balancing point of view, the one when $\pi/2<|\psi|<\pi$.

3. Adaptive Balancing by using Springs

There are many papers and patents [1, 2] which studied during the time the problem of static balancing by using springs. Most of them consider the problem when the static load is constant and more of that do not take into consideration the spring mass.

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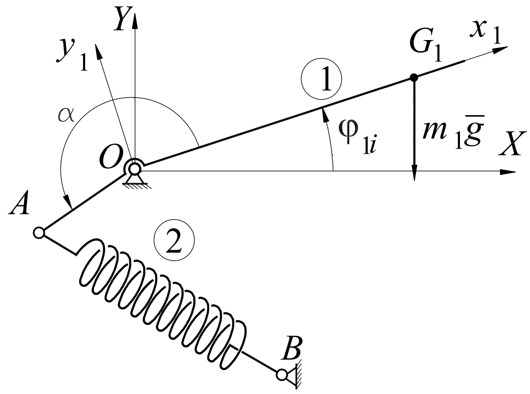
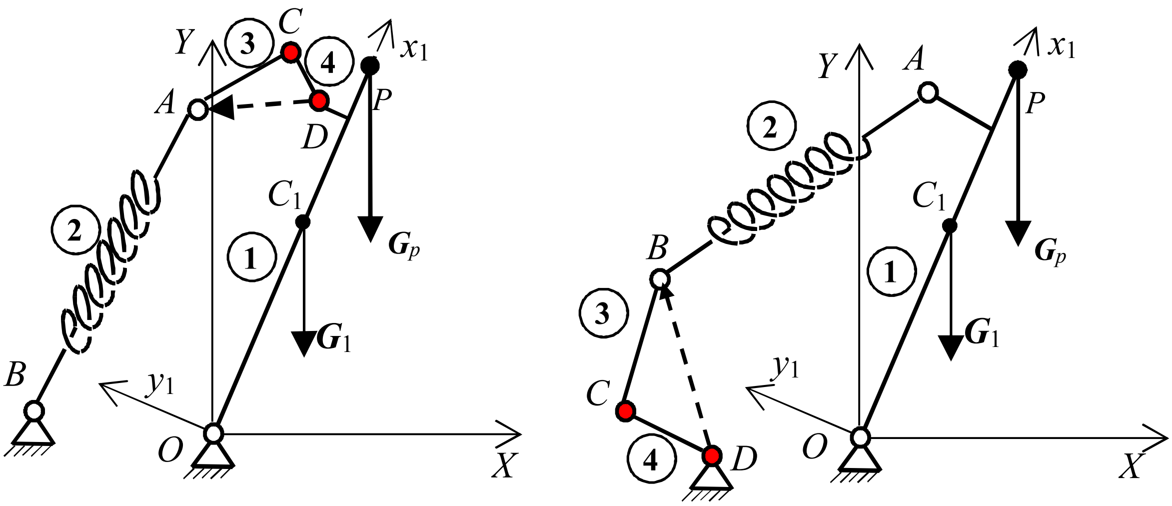


Figure 5: Static balancing by spring

In this way a better idea is to join the spring to the balanced arm so that its weight concentrated in the joint to act as a counterweight too (Figure 5). But as is wrote in many papers, and even from the beginning started by Carwardin [24], the solution from Figure 5 requires zero-free-length springs [21] or elastic systems [25]. One of the solution is to remove one of the spring joints and to intercalate some linkages with zero degrees of freedom. In case of variable load this solution requires to intercalate linkages with active joints in order to obtain the required adaptation. In [26] is proposed a solution with active prismatic joints. Prismatic joints are always more complicated from maintenance point of view and not only. So revolute joint are more proper and in Figure 6 are represented solutions to relocate spring joints by using active controlled joints.



(a) **(b)**

Figure 6. (**a**) Controlled relocation of joint *A* of spring; (**b**) Controlled relocation of fixed joint *B* of spring

Joint *C* and *D* are only controlling active joints. Once the adaptation to the variable load $\boldsymbol{G}_p$ is done then joint *A*, and joint *B* respectively, are fixed to the arm and to the ground respectively.

Mixed solution with prismatic and revolute joints as active control joints are presented in Figure 7.

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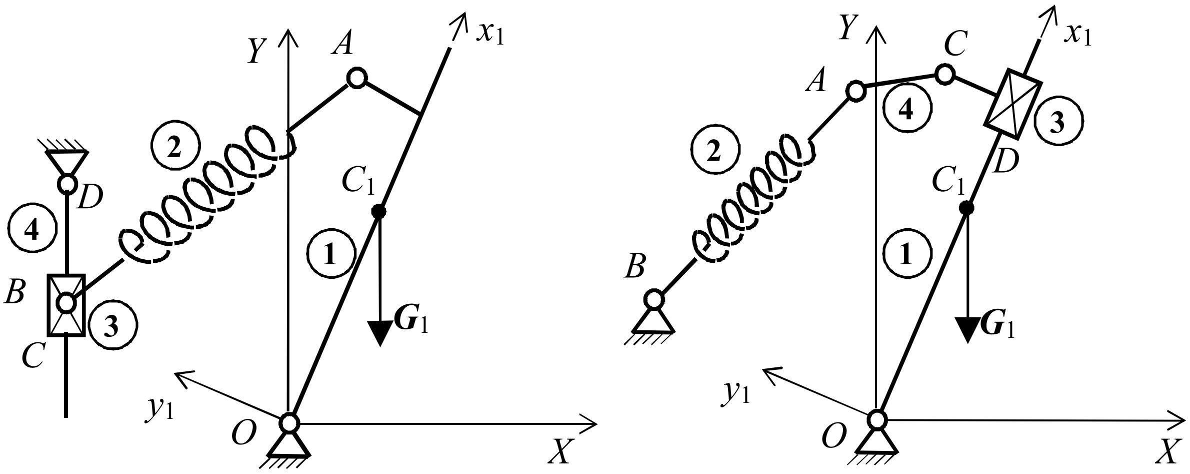

(a) (b)

Figure 7. (**a**) Controlled relocation of joint *A* of spring; (**b**) Controlled relocation of fixed joint *B* of spring

Let take as example the simple one degree of freedom relocation of fixed joint *B* by a prismatic joint presented in Figure 9.

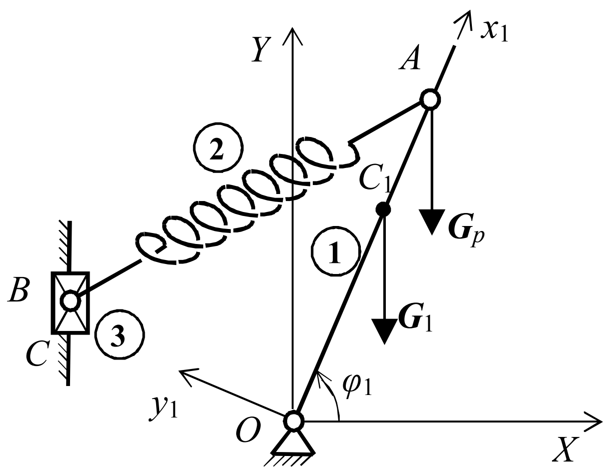

Figure 8: Controlled relocation of fixed joint *B* of spring by active prismatic joint

Without any reduction of the generality of the study let consider joint *A* on Ox_1 axis and also the point of action of payload in same point *A*. In this case the equilibrium equation of rocking arm ① is given by equation:

$$\boldsymbol{F}_a \, OA \sin(\theta - \varphi_1) - \boldsymbol{M}_{g1} = 0 \tag{17}$$

where:

$$\boldsymbol{M}_{g1} = (m_1 \, OC_1 + m_p \, OA) \, g \cos\varphi_1 \tag{18}$$

$$\boldsymbol{F}_a = \boldsymbol{F}_{a0} + k \, (l_a - l_{a0}), \tag{19}$$

$$\theta = \operatorname{atan}\frac{Y_A - Y_B}{X_A - X_B}, \quad \begin{pmatrix} X_A \\ Y_A \end{pmatrix} = \begin{pmatrix} \cos\varphi_1 & -\sin\varphi_1 \\ \sin\varphi_1 & \cos\varphi_1 \end{pmatrix} \begin{pmatrix} x_{1A} \\ 0 \end{pmatrix}, \tag{20}$$

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$$OA = \sqrt{X_A^2 + Y_A^2}\ ,\ l_a = AB = \sqrt{(X_A - X_B)^2 + (Y_A - Y_B)^2}\ , \tag{21}$$

and where are known or considered known: $\boldsymbol{F}_{a_0}$, l_{a_0}, x_{1A}, X_B and Y_B .

When a modification of payload occur then:

$$\boldsymbol{G}_p' = \boldsymbol{G}_p + \Delta\boldsymbol{G}_p \text{ or } m_p' = m_p + \Delta m_p \tag{22}$$

According with this modification the Y-coordinate of point B should be changed by controlling system i.e.:

$$Y_B' = Y_B + \Delta Y_B \tag{23}$$

Accordingly Relations (19)-(21) will became:

$$\boldsymbol{F}_a' = \boldsymbol{F}_{a_0} + k\,(l_a' - l_{a_0}) = \boldsymbol{F}_a + k\,\Delta l_a, \tag{19'}$$

$$l_a' = l_a + \Delta l_a = AB' = \sqrt{(X_A - X_B)^2 + (Y_A - Y_B')^2}\ , \tag{20'}$$

$$\theta' = \text{atan}\frac{Y_A - Y_B'}{X_A - X_B}, \tag{21'}$$

and new balancing equation:

$$\boldsymbol{F}_a'\ OA\ \sin(\theta' - \varphi_1) - \boldsymbol{M}_{g1} - \Delta\boldsymbol{M}_g = 0 \tag{17'}$$

where:

$$\Delta\boldsymbol{M}_g = \Delta m_s\ g\ OA\ \cos\varphi_1 = \Delta\boldsymbol{G}_s\ OA\ \cos\varphi_1 \tag{24}$$

Due to nonlinearity of Equation (17′), comes from Relations (19′), (20′) and (21′), it is impossible to get an explicite relation like:

$$Y_B' = Y_B'(m_s(t)) \tag{25}$$

or

$$\Delta Y_B = \Delta Y_B\,(\Delta m_s(t)) \tag{25'}$$

which is necessary to the control. Only by using a numerical method could solve this problem.

4. Results

In the case of solution from Fig. 4.a let suppose that the variable part of payload has the maximum value $\boldsymbol{G}_{pv,\max}$ = 4 N and is acting at distance OP = 2 m while the workfield of balanced arm ① is symmetric with respect to the horizontal axis: $\varphi \in [-\pi/2, \pi/2]$. Suppose that the counterweight ② has the weight $\boldsymbol{G}_2$ = 3 N and the connecting road ③ has the length AB = 2 m and is articulated on vertical direction at distance OA = 1 m.

By taking into consideration a variation of payload as is represented in Figure 3 the maximum unbalancing moment is when the position of balanced arm ① is near the horizontal (φ = 0.095[rad]) and has the value 0.828 Nm (represented by function $h_2(x)$ plotted in red color in graph from Figure 9).

The plotted red dashed curve - represented by function $h_1(x)$ in Figure 9 – show the variation of unbalancing moment in case o linear variation of static load [27] which has the maximum value about double than in case of parabolic variation (about 1.4 Nm at position φ = – 0.5 rad).

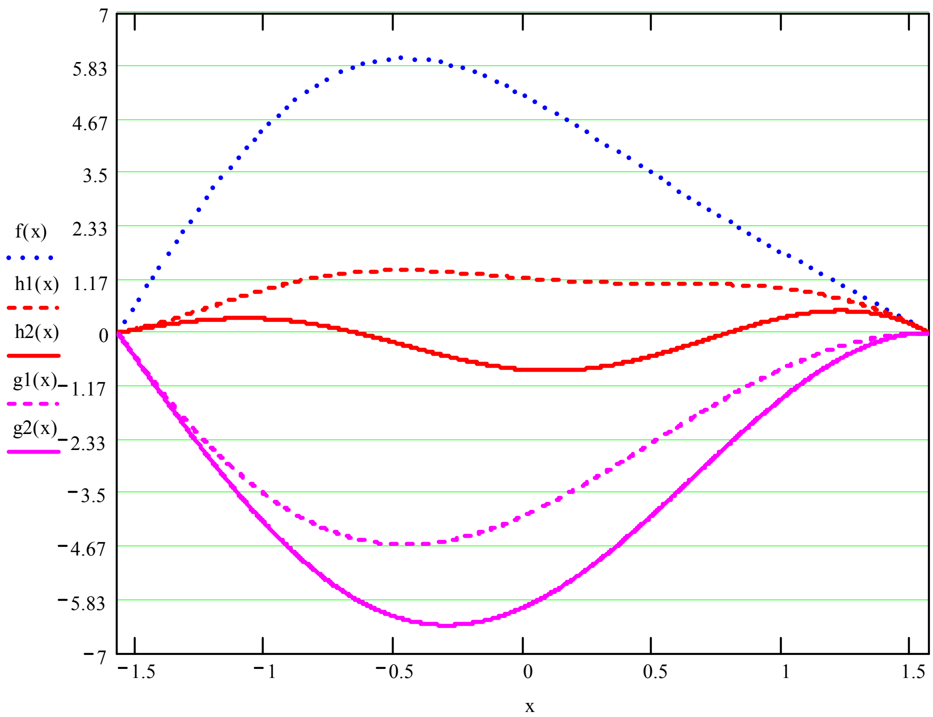

Figure 9: Gravitational moment, unbalanced moments and counterweight balancing moments

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