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A HYDRAULIC PLATFORM FOR THE UNLOADING OF CONTAINERED COMMUNAL WASTE

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Abstract. *This paper discusses a synthesis of the control systems for a hydraulic unloading ramp of containered communal waste. In the paper are given: the technical features of the ramp, the mathematical model of motion of the unloading ramp, the hydraulic system of the ramp which consists of two hydraulic telescopic actuators, a manifold for the decline speed regulation of the ramp, pressure transducers, a pressure regulator with speed and a position transformer and electric proportional valves, which are used for the control of the elevation speed of the ramp, a mathematical model of this system, a vector equation of the system status and control system of the hydraulic platform which consists of PLC and operates by speed and the position PID algorithm.*

Key Words: *Hydraulic platform, municipal solid waste, control system*

INTRODUCTION

An ever greater economic and ecological problem of every urban environment is communal waste, because every day about 1kg is yielded per resident is. The main part of municipal waste is organic in origin, which activated by micro-organisms, decomposes into methane and other gases, and water fills with heavy metals and other toxic elements.

Besides, such material becomes attractive to birds, insects or rodents, so the circle of the contamination of nature grows through soil, air and water. Exuded water goes to the rivers or other streams which might induce pollution of drinking water sources. On the other side, the composition of communal waste enables a utilization of the components in the manner of revitalization, biological or chemical treatment. It is expected that only 20% of communal waste material is not usable, namely, that it must be landfilled. That requires so-called sanitary landfill, where all of the disposed material is isolated from soil

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and air. Sanitary dumps are provided with a melioration system for receiving exuded water by a pipe drainage system, as well as with a plant for the refinement of exuded liquid waste.

The solid municipal waste processing itself comprises collecting, transporting, compacting, attrition and sorting, as physical process, and the aerobic and anaerobic decomposition of organic components, as biological process and thermal treatment including oxidation, pyrolysis and gas derivation.

The basic postulate of the adopted strategy for managing solid municipal waste in Serbia is the regional approach which presumes the forming of a transfer stations for every urban unit and processing centers and a waste disposal areas at the regional level. Such an approach could be realized by the application of appropriate means of transportation, and a system for compacting, namely, unloading. This paper shows only a part of this process, namely a short overview is given for the hydraulic unloading platform of containered municipal waste. The appearance and functionality of such platform is presented and a review of the elements for synthesis of the hydraulic system.

The Hydraulic Unloading Ramp for Containered Municipal Waste

Solid communal waste collected in an urban environment is carried by vehicles of public a communal company to the transfer station format in the neighborhood. There the communal vehicles are unloaded and the material is compacted in containers. The containers themselves are transported by low-chassis trucks all the way to the recycling centers. The unloading is realized by the hydraulic ramp. The truck, specifically, goes backwards and comes to the ramp limit stop. After that the cabin holder drops down loosens the front door of the container and the ramp goes up. At that moment, the communal waste is unloaded and the truck with empty container goes down in a horizontal position.

The technical features of the ramp

Load	50 t
Length	20 m
Width	4.5 m
Height	1.5 m
Elevation angle	43°
Elevation power of the hydraulic aggregate	35 kW
Time of elevation	240 sec
Drop time	60 sec
Hydraulic aggregate power for the cabin holder	3 kW
Hydraulic installation pressure	160 bar

The support structure is made of sheet metal box profile with a thickness of 12 mm.

On both sides of the platform there are two passageways with a width of 800 mm, which are provided with railings with handholds of 1100 mm. The elevation is realized with two telescope 4-motion cylinders 240, 212, 185 and 160 mm in diameter and an overall length of motion of 8000 mm. The force, by stages, is: 723.5, 564.5, 430 and 321.5 kN. The control of the unloading ramp is executed by command cabin.

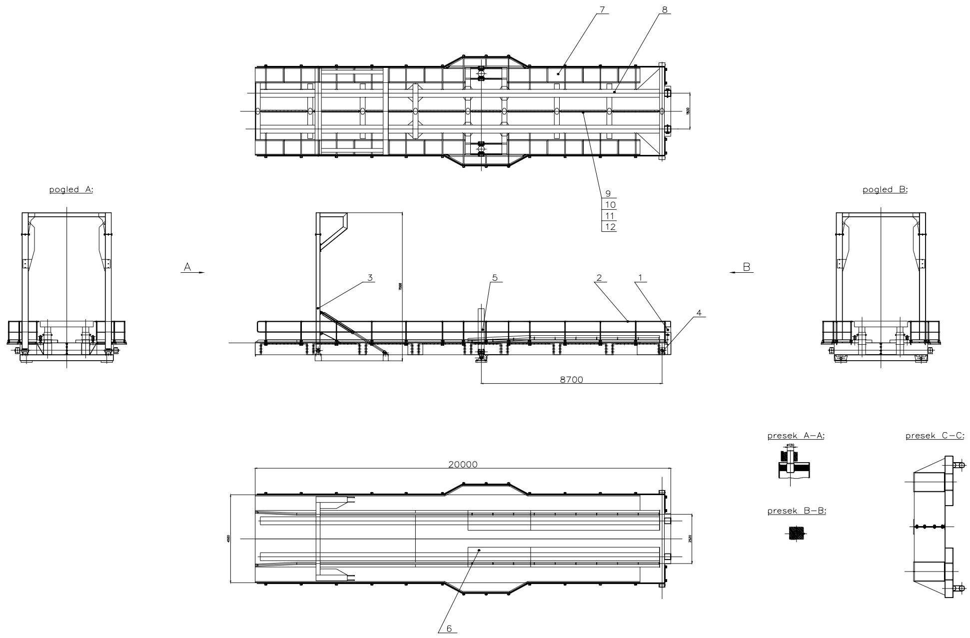

Fig. 1. Hydraulic ramp

THE MATHEMATICAL MODEL OF MOTION OF THE UNLOADING RAMP

The motion of the ramp is represented by a differential equation:

$$J_z \ddot{\varphi} = -G \cdot r \cdot \cos\varphi + M \tag{1}$$

where: J_z – is the moment of inertia of the ramp regarding the z- axis rotation axis, the bedding axis of the ramp,

φ – is the ramp elevation angle, referring to the general coordinate of the system,

$G = m \cdot g$ – is the force of gravity,

r – is the destination center of gravity to the ramp bedding and

M – is the moment of force of the hydraulic cylinders.

The magnitude of this moment is not constant because of an alteration in the diameter of the telescopic cylinder which means a change in force in the cylinder itself in one hand, and in the other hand, because of the angle of force alteration to the ramp, as is shown on Figure 2, which illustrates the ramp in elevation.

The relation of the length of cylinder and the system's general coordinate we can represent as:

$$b = \sqrt{a^2 + c^2 - 2a \cdot c \cdot \cos\varphi}\ , \tag{2}$$

where the lengths a and c are the constants of the ramp. Depending on the magnitude of b, namely the length of cylinder motion depends, and the force of the cylinder as is shown on Fig. 3.

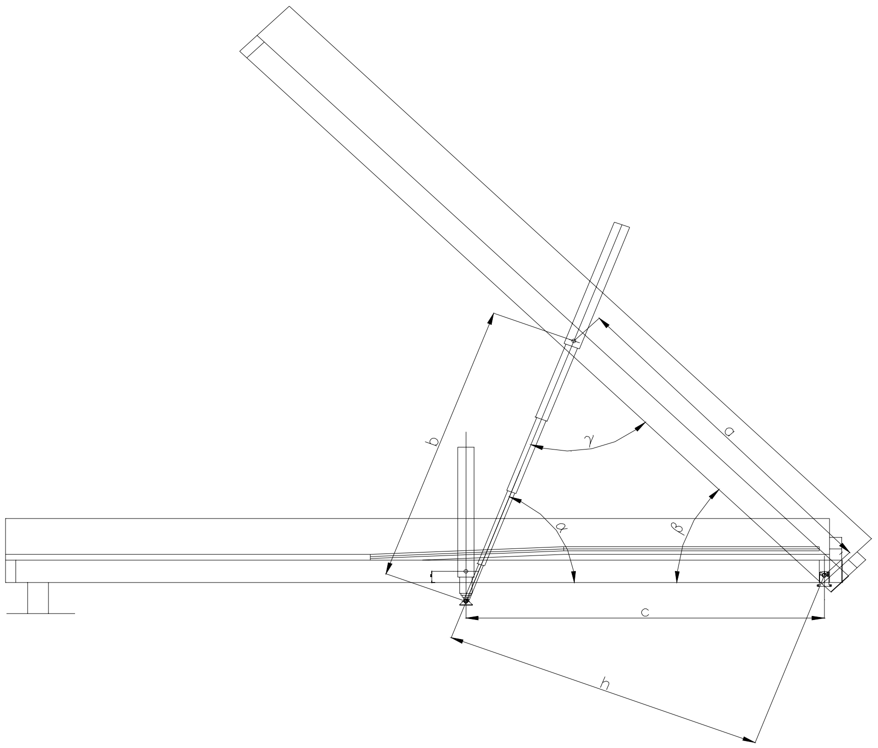

Fig. 2. Ramp in elevation

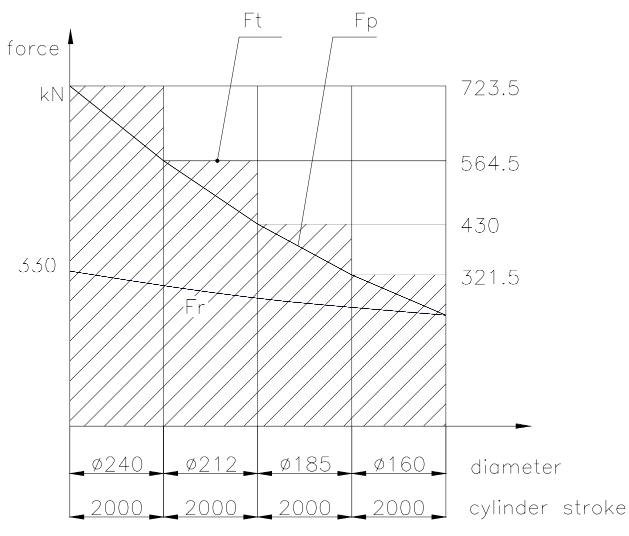

Fig. 3. Diagram of cylinder force (Ft- theoretical, Fp- permissive and Fr- real force)

The amount of momentum we deduced based on the relation:

$$M = A_i \cdot p \cdot h\,, \tag{3}$$

where:

A_i – is the effective area of the cylinder,

p – is cylinder pressure and

h – is the destination based on Fig 2. calculated by the relation:

$$h = c \cdot \sin\alpha\,, \tag{4}$$

$$\sin\alpha = \frac{a \cdot \sin\varphi}{b}\,. \tag{5}$$

What remains is to identify the magnitude of pressure in the cylinders. This magnitude we could identify based on a real pipeline configuration and an effectuated rating of the system.

If we use a model of a pipeline as illustrated on Figure 4, the magnitude of pressure loss

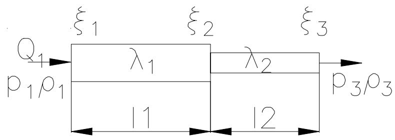

Fig. 4. Pipeline model

is represented as follows:

$$p_1 - p_3 = p_s = \frac{\rho}{2}[v_1^2(\xi_{\lambda 1} + \xi_1) + v_2^2(\xi_{\lambda 2} + \xi_2 + \xi_3)] \tag{6}$$

where: ξ_1, ξ_2, ξ_3, are coefficients of local loses, which correspond to the velocity of flux v_1 for ξ_1, namely v_2 for ξ_2 and ξ_3, and

$$\xi_\lambda = \frac{\lambda \cdot l}{d} \tag{7}$$

represents the coefficient of loses for length l for real pipeline with an inner diameter d and a resistance coefficient λ.

RAMP HYDRAULIC SYSTEM

The hydraulic system of the ramp consists of two hydraulic telescopic actuators, manifolds for decline speed regulation of the ramp, pressure transducers, a pressure regulator with speed and position transformer and electric proportional valves which are used for the control of the elevation speed of the ramp.

Figure 5 illustrates one of the variants of the hydraulic system. Instead of double pipe proportional valves, it is possible to use 4/2 proportional manifolds where the second output is connected with feedback of the system which contributes to the soft operation of the pump in a blocked status.

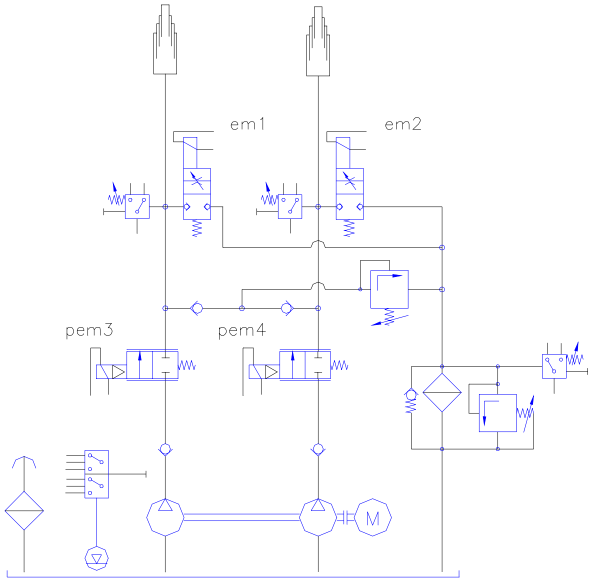

Fig. 5. Hydraulic drive elevation schematic

A drive aggregate is made with two mechanical coupled pumps, which assures the synchronization of the actuator. The motor of the drive aggregate could have two speeds, or a motor with frequency converter for soft start of system could be used.

Figure 6 illustrates the system spring response, Figure 7 the frequency characteristic, and Figure 8 the linear flux characteristic of the proportional valve.

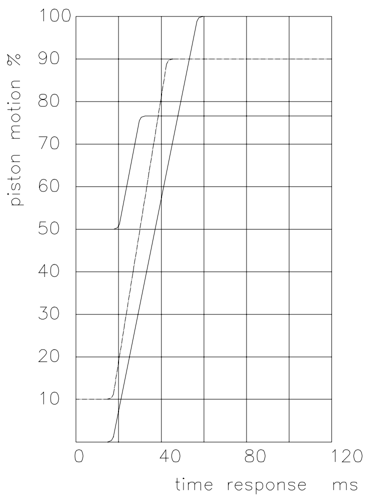

Fig. 6. Spring response of proportional valve

The mathematical model of this system form from the continuity equation of the hydraulic system:

The continuity equation is:

$$Q = Q_m + Q_c + Q_t \tag{8}$$

Where:

Q – is the flux of the servo manifold,
Q_m – is the necessary flux hydro-cylinder motion,
Q_c – is the inner drip flux and
Q_t – is the compressible fluid flux.

The magnitude of this fluxes are:

$$Q = K_q x\,, \tag{9}$$

$$Q_m = A_i \cdot \dot{x}_i\,, \tag{10}$$

$$Q_t = \frac{V}{B}\frac{dp}{dt}\,. \tag{11}$$

Where: K_q – is the flux gain coefficient,
x – is the piston movement of the servo manifold,
x_i – is the magnitude of the movement of the telescopic cylinder, corresponding to the length b of the telescopic cylinder,
A_i – is the cylinder working area according to the distinct length of movement (Fig. 3),
p – is the pressure in the system,
V – is the overall volume of fluid and
B – is the compressible volume modulus.

The balance of forces equation which impacts the working area of the cylinder is:

$$A_i \cdot p = m\ddot{x}_i + \mu\dot{x}_i \tag{12}$$

By means of the equation (8) we obtain:

$$A_i\dot{x}_i + C_c[\frac{m}{A_i}\ddot{x}_i + \frac{\mu}{A_i}\dot{x}_i] + \frac{V}{B}[\frac{m}{A_i}\dddot{x}_i + \frac{\mu}{A_i}\ddot{x}_i] = K_q x \tag{13}$$

Arranging by derivations x_i, we obtain:

$$\frac{V}{B}\frac{m}{A_i}\dddot{x}_i + (\frac{V}{B}\frac{\mu}{A_i} + \frac{C_c m}{A_i})\ddot{x}_i + (A_i + \frac{\mu}{A_i})\dot{x}_i = K_q x \tag{14}$$

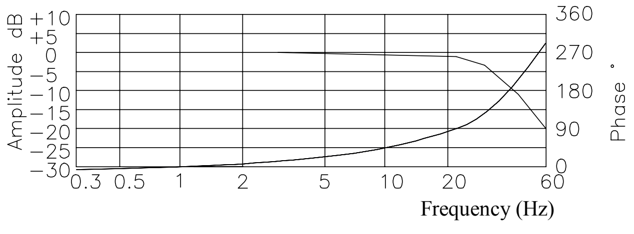

Fig. 7. Frequency response of proportional valve

If we substitute:

$$\frac{C_c}{A_i} = \frac{1}{C_2}, \frac{1}{A_i} = C_1 \tag{15}$$

we obtain a following relation:

$$\frac{Vm}{BA_i^2}\dddot{x}_i + (\frac{V\mu}{BA_i^2} + \frac{C_1}{C_2}m)\ddot{x}_i + (1 + \mu\frac{C_1}{C_2})\dot{x}_i = K_q C_1 x \tag{16}$$

Where: C_c – is the inner drip coefficient,
μ – is the viscous friction coefficient and
m – is the equivalent mass (load).

The magnitude of equivalent mass we calculate based on relation:

$$m = m_c + (m_r + m_k + m_m)\cdot\cos\alpha \tag{17}$$

where: m_c – is the mass of the second, third and forth extent of the telescopic cylinder,
m_r – is the mass of a ramp,
m_k – is the mass of a truck with container and
m_m – is the mass of the material in the container.

If we substitute:

$$\omega_n = \sqrt{\frac{BA^2}{Vm}(1 + \mu\frac{C_1}{C_2})}\ , \tag{18}$$

$$\xi = \omega_n \frac{C_1 m}{C_2} \frac{1 + \mu\frac{VC_2}{C_1 BA^2 m}}{1 + \mu\frac{C_1}{C_2}}\ , \tag{19}$$

where: ω_n – is the resonance frequency and
ξ – is the damping coefficient,

we get a transition function:

$$\frac{x_i(s)}{x(s)} = W(s) = \frac{K_H}{s(s^2 + 2\xi\omega_n s + \omega_n^2)} \tag{20}$$

with:

$$K_H = \frac{K_q C_1}{1 + \mu\frac{C_1}{C_2}}\omega_n^2 = \frac{\omega_n^2}{\tau_1} \tag{21}$$

The vector equation of the system status is:

$$\ddot{x} = \begin{vmatrix} 0 & 1 & 0 \\ 0 & 0 & 1 \\ 0 & -\omega_n^2 & -2\xi\omega_n \end{vmatrix} x + \begin{vmatrix} 0 \\ 0 \\ K \end{vmatrix} u \tag{22}$$

where:

$$K = K_i K_M K_{ME} \frac{\omega_n^2}{\tau_1},\tag{23}$$

$$x = (x_1, x_2, x_3)^T,\tag{24}$$

K_i – is the voltage/current transducer coefficient,
K_M – is the force gain coefficient,
K_{ME} – is the measure element coefficient.

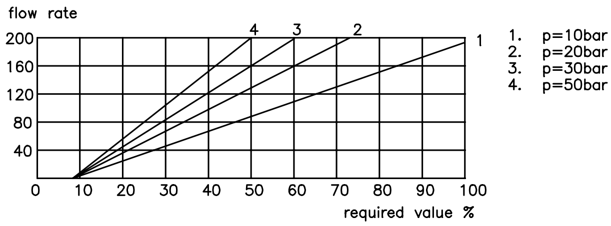

Fig. 8. Linear characteristic of flux of proportional valve

The output equation of system is:

$$x_i = x_1\tag{25}$$

The control system of the hydraulic platform consists of a PLC which operates by speed and the position PID algorithm:

$$u(k) = K[e(k) + \frac{T_0}{T_i}\sum_{i=0}^{k} e(i) + \frac{T_D}{T_0}(e(k) - e(k-1))]\tag{26}$$

with:

$$e(k) = x_i - x_i(k)\tag{27}$$

for positioning and

$$e(k) = x_i - \frac{x(k) - x(k-1)}{T_0}\tag{28}$$

for the speed PID algorithm.

CONCLUSIVE REMARKS

The configuration of the hydraulic system of the unloading ramp is made in such way as to assure the synchronization of both telescopic actuators, the maximal moving speed control of elevation, as well as operational safety. It is selected by electric control proportional manifolds in two pipe structures with the following basic characteristics:

– an adjustable electric damping valve with an edge in the main stage,
– a piston movement of the damping diaphragm proportional to the electric input signal,
– a compensation for drawback with a regulatory circuit with an inductive position sensor,
– high dynamics of manifold with a short time for the plug in,
– and automatic closure by timely shutdown or pipeline breakage.

This paper provides all the elements of the system synthesis of the hydraulic as well as control elements based on a PID algorithm of regulation. The control system is realized with PLC, which effectuates a control of other performing organs of the ramp, like the cabin holder, the signal system of the working stages and the main lubrication system.

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HIDRAULIČKA PLATFORMA ZA ISTOVARIVANJE KONTEJNIRANOG KOMUNALNOG OTPADA

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Članak prikazuje sintezu kontrolnog sistema hidrauličke istovarne rampe kontejniranog komunalnog otpada. U članku su dati: tehnički parametri rampe, matematički model kretanja istovarivanja rampe, hidraulički sistem rampe koji se sastoji od dva teleskopska aktuatora, sistema za regulaciju brzine spuštanja rampe, senzora pritiska, regulatora pritiska sa pretvaračima brzine i pozicije i elektro-proporcionalnim ventilima koji se koriste za kontrolu brzine elevacije rampe, matematički model tog sistema, vektorska jednačina stanja sistema i kontrolni sistem hidrauličke platforme koji sadrži PLC i radi pri brzinskom i pozicionom algoritmu.

Ključne reči: *Hidraulička platforma, kontejner komunalnog otpada, kontrolni sistem*