Imitation Model for Research of Conveyer Weigher

Valeriy Romanov, Leonid Tkachenko, Volodymyr Fesechko

Abstract - **In this paper the mathematical model to research a conveyer weigher work is suggested and the recommendations on the wegher's constructive parameters for the developers are given.**

Keywords – **Conveyer weigher, imitation system, mathematical model, vibroplatform.**

I. Introduction

Efficiency of exploitation of the known constructions of conveyer automatic weigher (CW) of continuous action and their metrological providing remains low because of the lack of exact and simple operations of testing and diagnostics. Attempts to check CW by using different mechanical imitators of the transported material or methods of the frequentative sampling of the material which is moved along the conveyer track through the weight-receiving device of weigher, do not allow achieving the considerable error reducing and improvement of conveyer weigher long-term stability.

Improving the weigher quality is possible to be achieved by reducing the dynamic influence of conveyer track, in particular, due to compensation of self-oscillations in the conveyer electric drive. By the help of mathematical modeling it is possible to detect the influence of conveyer track on the workload sensor*s* during one turn and to create the mathematical model of its influence on weight-measuring system. Compensation signal of certain value and sign, which is defined after this model and entered in control unit memory, allows reducing the calculated integral from the instantaneous value of conveyer track workload on weight-measuring system to the value defined after specification.

II. Imitation System Implementation

The conveyer transporting generates the great number of vibration sources. Vibration, by influencing on weigher, causes their self-oscillations, which causes the distortion of measuring result. In order to know how to improve the weigher construction, it is necessary to find out how these conveyer weigher react on different oscillations (different frequencies, amplitudes).

On Fig. 1, it is shown the generalized scheme of imitation system for conveyer weigher work design. On Fig. 2, the arrangement of measuring circuit sensors is presented.

Scheme of system measuring circuit includes three two-channel acceleration sensors which measure accelerations along the two perpendicular axes; six-channel ADC, which measures output voltages of acceleration sensors; MC, microcontroller which performs data exchange with ADC.

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System management is carried out through the user's interface. The user sets frequency and amplitude of the generated oscillations. Besides, he also sets damping and elasticity coefficients for X, Y, Z axes. These values are conducted to microcontroller which, in its turn, sets the defined frequency on oscillations generator, and sets the amplification coefficient on amplifier. The set oscillation is conducted to the weigher, and weigher oscillations are detected by sensors. Information for given 6 channels are conducted to ADC, and then to microcontroller. Data is conducted to computer on RS485.

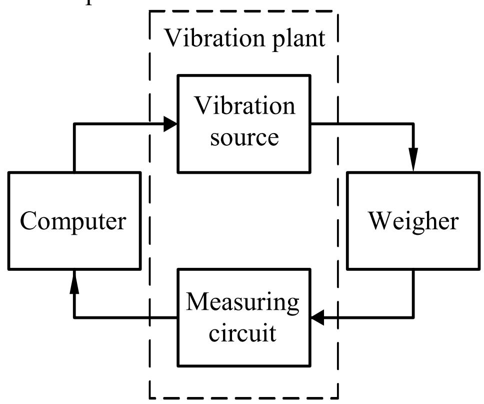

Fig.1 Generalized scheme of imitation system.

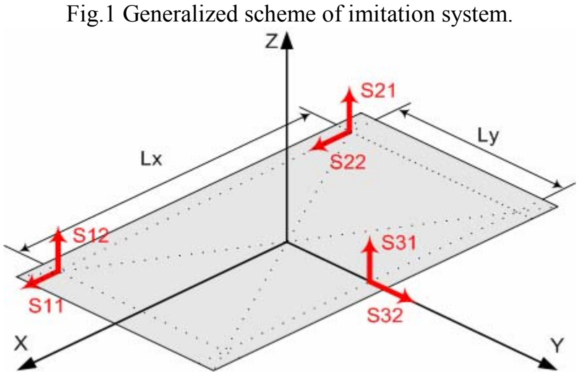

Fig.2 Arrangement of measuring circuit sensors.

Data transfer to microcontroller, and also to generator and amplifier, sensor data transfer back is designed programmatically. In software data is counted to forward and angular accelerations in relation to X, Y, Z axes.

There is possibility to display all the received signals, and recounted accelerations in the graphical window by the help of graphs.

III. Mathematical Model Design

Oscillations accompany the work of conveyer and are undesirable. The task is to isolate payload from the source of oscillations or, vice versa, to isolate the source of oscillations. In spite of all of construction distinctions, essence of the oscillations extinguishing system is identical. The passive system consists of spring and damper. A spring is designed to soften oscillations and shoves, and damper has to liquidate oscillations appearing in the system. The active system uses an additional pair, consisting of accelerometer and electromagnetic drive also, that allows achieving high degree of vibroisolation.

The real stimulating influence on vibroisolating system consists of many frequencies. Oscillations with frequencies which are higher than system resonance frequency undergo the considerable extinguishing, and frequencies which are lower than resonance frequency, remain practically unchanging. It can be clearly seen, that it is required to provide minimum possible resonance frequency of the system, and at this, the damping coefficient must be high enough, in order to avoid the considerable increase of oscillations amplitude in resonance. Remaining oscillations occur mainly on low frequencies.

In the active vibroisolation system, along with spring shock absorbers, accelerometer – electric drive pairs are used. Accelerometer allows detecting remaining oscillations. Its signal is sent to electric drive through electronic circuit and accelerometer. Such system allows achieving the considerable reducing of oscillations amplitude, especially its high-frequency components.

The ideal weighed platform can be represented as an oscillation-extinguishing structure. It has typical extinguishing perfect transient characteristics, and is described by a differential equation of weigher motion:

$$m \cdot a = -D \cdot S - k \cdot V + F_0 \cdot \cos(\omega \cdot t),$$

where: **m** is mass; **a** is acceleration; **k** is elasticity module (coefficient); **S** is moving; **D** is friction force coefficient (extinguishing constant), damping; **V** is speed; $\mathbf{F_0}$ is oscillation amplitude; **ω** is oscillation frequency (design); **t** is time.

The decision of this differential equation will be:

$$X(t) = \frac{F_0}{m\sqrt{\left(\frac{D}{m} - \omega_E^2\right)^2 + \frac{K^2}{m^2}\omega_E^2}} \cdot \cos(\omega_E t);$$

$$\omega_E = (\omega_0^2 - k^2/4m^2)^{1/2},$$

where $\boldsymbol{\omega}_E$ is resonance frequency of damped ($\boldsymbol{\omega}_0$ **- not damped**) system.

User in the developed system, by setting D, K, ω, F_0, gets the weigher oscillation function in the output.

In the process of design it is necessary to find out those values of damping and elasticity coefficients, at which ratio of vibroplatform and amortized system oscillations amplitudes would be increased insignificantly (< 3 dB) near-by resonance frequency.

Algorithm of selection of damping and elasticity coefficients is the following:

1. If we need just a review ratio of vibroplatform and amortized system oscillations amplitudes, it is necessary to set mass, damping and elasticity coefficient, and also the range of frequencies. Further, the graph of amplitudes ratio will be brought to a screen, on which it will be possible to define by sight: maximum of the given graph; - how much is the system sensible to the oscillation (it is especially important on resonance frequency). If the function maximum of function will be more than 3 (it means triple strengthening of oscillation, that is impermissible), then it is necessary to select coefficients at which the function maximum will be less than 3, manually.

2. At automatic selection, it is necessary to set one of coefficients (damping or elasticity), and also accuracy with which it is necessary to select the second coefficient. Then it will be selected coefficient at which the ratio of amplitudes will be equal to three, and limits for these two coefficients will be brought to the screen. Within these limits it is possible to change these coefficients at any combination; the ratio of amplitudes will not exceed 3.

The program works in accordance with algorithm shown above, i.e., it enables data entry (damping and elasticity coefficients, mass, amplitude, coefficients determination accuracy, range of frequencies) for the test of weigher mathematical model, designs and brings the graph of ratio of vibroplatform and amortized system oscillations amplitudes, also gives possibility of automatic calculation of coefficients at different set value.

Software is divided to 3 parts:

1. Determination of limit values for damping and elasticity coefficients (separate program, intended only for these aims);
2. Setting and automatic control of all of values, which are necessary for research conducting (main program, basic window);
3. Graphic representation of weigher oscillation analysis after the graphs (main program, graphic window).

The program is written in Delphi language.

IV. User Interface Description

The basic window of the program and graph are represented on Figure 2. For the simple graph displaying it is necessary to enter the following parameters: m, D, K, f. For example: m=5, D=1, K=1, f=1÷5.

Leaving all he other parameters by default and pushing the button TO MODEL, the graph of amplitudes ratio is brought to the screen.

Pair of coefficients K=1 and D=1 meet a condition, that amplitude must not be increased more than in 3 times. Analogically, selection of the other coefficients is performed.

In order to selects coefficients automatically it is necessary to enter only one of these coefficients. For example, m=5, K=1, f=1÷5**.** After pressure of the button TO FIND LIMIT, a value at which the maximal amplitudes ratio is close to 3, is automatically put in the cell of damping coefficient (as there is a basic condition – amplitudes ratio must not exceed 3)

Pushing the button TO FIND LIMIT, the following information was brought to the screen: D>=0,594**.** It means that at elasticity coefficient value of K=1, possible range of the damping coefficient use is more than 0,594. If we take the damping coefficient less than 0,594, in resonance frequency we will get more than triple oscillation strengthening (Fig. 3) that is undesirable.

V. Weigher Mathematical Model Research

In tables 1 and 2 are presented data on given research determination of ranges of damping and elasticity coefficients, which meet a condition: ratio of vibroplatform and damped system amplitudes is no more than 3.

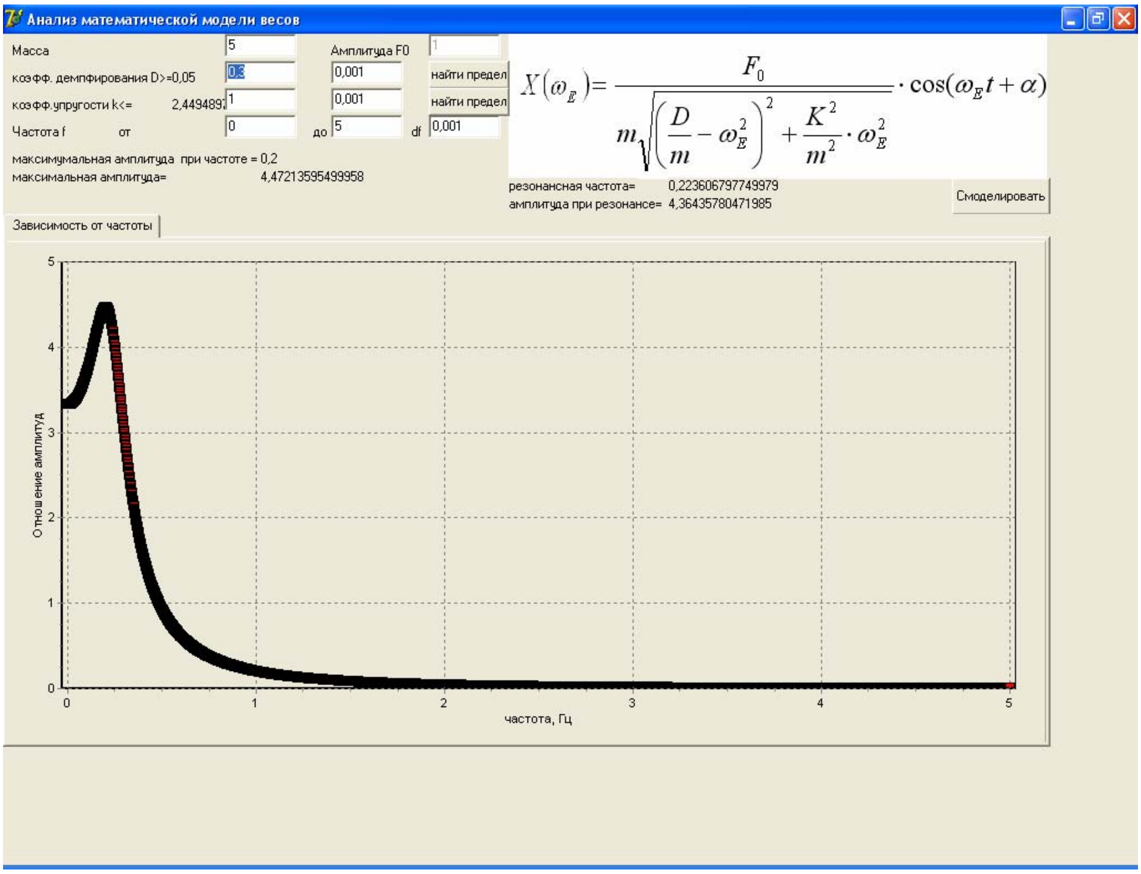

Fig. 3 Graph of ratio of vibroplatform and damping system oscillation amplitudes.

After setting the damping coefficient it is possible to find the range of elasticity coefficient change for it (Table 1). After setting the elasticity coefficient it is possible to find the range of damping coefficient change for it (Table 2).

TABLE 1

RESEARCH DATA

Damping coefficient	Elasticity coefficient	Resonance frequency
0,4	1,281 – 2,828	0,252
0,7	0,911 – 3,741	0,362
0,9	0,796 – 4,243	0,417
1	0,753 – 4,472	0,441
2	0,528 – 6,325	0,63
5	0,333 – 10	0,999
10	0,235 – 14,42	1,414

TABLE 2

RESEARCH DATA

Damping coefficient	Elasticity coefficient	Resonance frequency
>=55,556	0,1	3,333
>=13,891	0,2	1,66
>=6,177	0,3	1,111
>=3,479	0,4	0,833
>=2,232	0,5	0,666
>=0,594	1	0,329
>=0,289	2	0,133

REFERENCES

[1] Назаров В.Н. Существующие конструкции и способы поверки конвейерных весов и их совершенствование / В.Н.Назаров, О.В.Круг // Метрологическое обеспечение весоизмерительной техники. “Весы-2006”: всероссийская начн.-практ. конф., 4-8 сент. 2006 г.: сб. докл. – Самара: ВНИИМС, 2006.-с.55-67.

[2] Кошевой Н.Д., Черепащук Г.А., Калашников Е.Е. Способ модернизации существующих весоизмерительных систем на базе ленточного конвейера. Вісник СевДТУ. Вип.95: Автоматизація процесів та управління: збю наук.пр. – Севастополь: Вид-во СевНТУ, 2009, с.36-39.

VI. CONCLUSION

The mathematical model of conveyer weigher is developed. Experiments are conducted, that show that for every damping coefficient there is a range of elasticity coefficients values, at which the ratio of vibroplatform amplitudes and amortized system (weigher) will not exceed 3, and, vice versa, the range of damping coefficients values corresponds to every elasticity coefficient. Besides, it can be seen from the resulted calculations, that oscillation frequency must not be less than 2 Hz. These results allow decreasing the error of measuring on conveyer band weigher and giving certain recommendations on structural parameters to the developers of such weigher.