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Study on the Influence of Structural Parameters on Vibration Parameters for the Design of Vibrating Table Equipment

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Abstract **- This paper presents the scientific basis for the formulation and solution of the dynamic model of a vibrating table system under variations in structural parameters, including eccentric mass, spring stiffness, rotational speed of the eccentric shaft, and related factors. The objective is to determine an appropriate set of structural parameters serving as a foundation for the design and manufacture of vibrating table equipment in Vietnam.**

Keywords: Structural parameters, vibration parameters, vibrating table equipment, vibration system, Concrete compaction system.

I. Introduction

Concrete vibrators are used to compact concrete and reinforced concrete mixtures. At present, vibration methods include surface vibration, internal vibration, and volumetric vibration, among which surface vibration is the most commonly applied. Directional vibrating table compactors are widely used for compacting mass concrete components with complex geometries.

Worldwide studies have investigated the influence of vibration parameters on the compressive strength and impermeability of concrete mixtures. Surface vibration has been shown to achieve the highest efficiency when the thickness of the concrete layer does not exceed 15 mm, according to the results reported by B. Bhattacharjee. The flexural and tensile strengths of concrete products depend significantly on the void ratio within the concrete mixture, which is one of the key factors determining the service life of a structure.

In Vietnam, research has been conducted on the application of roller-compacted concrete technology, the development of equipment systems for finishing the surfaces of concrete canal linings, and the manufacturing technology of cement concrete paving machines. However, studies on the effects of variations in structural parameters on the vibration characteristics of vibrating tables remain limited.

Furthermore, the design of vibrating tables generally involves several stages, including the calculation of dynamic parameters, motor power, and steel structural components. This paper focuses only on the influence of structural parameters on vibration characteristics in order to support the determination of dynamic parameters, which constitutes the most important stage in the design process.

II. Determination of the Influence of Structural Parameters on the Vibration Characteristics of Vibrating Table Equipment

2.1 Structure and Operating Principle of the Directional Vibrating Table Equipment

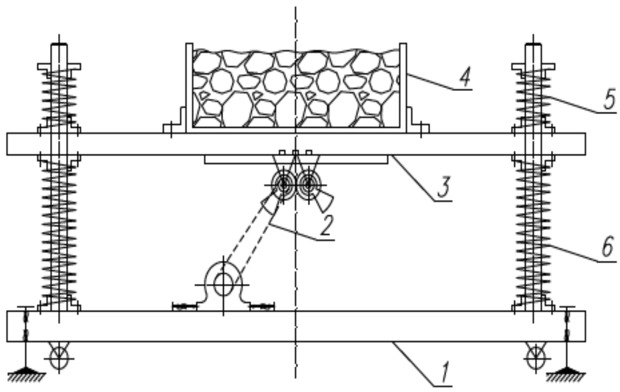

Figure 1: Schematic Diagram of the Experimental Directional Vibrating Table Compactor

The structural configuration (Figure 1) consists of: a fixed supporting frame (1), a vibration-generating unit (2) driven by a belt transmission system, a vibrating table (3), and a concrete molding chamber (4) supported by four upper springs (5) and four lower springs (6). Operating principle: Torque generated by the motor is transmitted through the belt drive to rotate the eccentric shaft, thereby creating an excitation force acting on the vibrating table. Through the action of the eight springs mounted above and below, the vibrating table undergoes vibration and transmits vibrational energy to the concrete block, resulting in the compaction of the concrete mixture.

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The excitation force acting from the vibrating table on the concrete mold depends on the structural parameters. The selection of these parameter values is determined by the mass of the vibrating component system (including the vibrating table and the concrete mixture), as presented in [3], [5], and [6].

For example, the experimental vibrating table system developed at the Dynamics Laboratory of the Military Technical Academy has a vibrating mass of 60.5 kg. The system allows the eccentric shaft rotational speed to be varied within the range of 0–2800 rpm by means of a frequency inverter; the eccentric mass can be adjusted from 0.4 to 3.2 kg using modular weights of 0.2 kg each; and the main spring stiffness is available in three configurations: $C_2 = 35{,}613$, 42,088, and 46,296 N/m.

2.2 Formulation of the Mathematical Model

For the above-mentioned equipment, the following assumptions are adopted in developing the computational model:

- The investigated center of mass of the vibrating table remains unchanged during operation;
- The damping forces of the supporting and compression springs are neglected;
- The stiffness and geometric dimensions of the main and auxiliary springs at the supports are assumed to be identical;
- The vibrating table is considered to be perfectly rigid, and the eccentric radius is assumed constant.

Formulation of the computational model: Based on Figure 1, the dynamic analysis model of the directional vibrating table is illustrated in Figure 2.

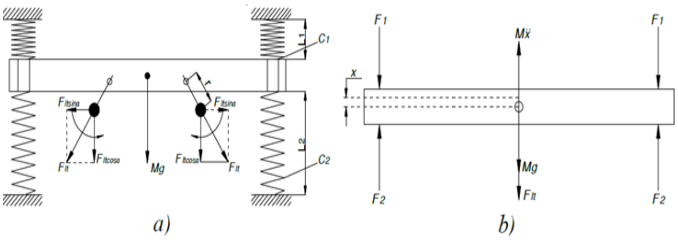

Figure 2: Vibration Analysis Model of the Directional Vibrating Table

The symbols used in the model are defined as follows:

- C_1, C_2: stiffness coefficients of the auxiliary and main springs, respectively;
- L_1, L_2: initial lengths of the auxiliary and main springs, respectively;
- x: displacement of the center of mass of the vibrating table relative to the equilibrium position;
- x_{01}, x_{02}: initial compressions of the auxiliary and main springs, respectively;
- x_{11}, x_{12}: compressions of the auxiliary and main springs during operation, respectively;
- $M\ddot{x}$: inertial force of the vibrating system;
- Mg: gravitational force of the vibrating system;
- F_1, F_2: elastic restoring forces of the auxiliary and main springs, respectively;
- F_{lt}: centrifugal force generated by the eccentric shaft.

By applying Jean le Rond d'Alembert's principle to the computational model (Figure 2b), according to [1] and [3], the differential equation governing the motion of the vibrating table can be expressed as follows:

$$M\ddot{x} = Mg - 4F_1 + 4F_2 + F_{lt} \qquad (1)$$

Where $M$is the total mass of the vibrating system. According to [2] and [3]:

$$F_1 = C_1 x_{11} - C_1 x_{01}$$
$$F_2 = C_2 x_{12} + C_2 x_{02}$$

and according to [1]:

$$F_{lt} = m\omega^2 r\cos\omega t$$

Where m is the eccentric mass (kg), r is the eccentric radius (m), and ω is the angular velocity of the eccentric shaft (rad/s).

Substituting these expressions into Equation (1), the following equation is obtained:

$$M\ddot{x} + 4C_2 x + 4C_1 x + 4C_2 x_{02} - 4C_1 x_{01} = Mg + m\omega^2 r\cos\omega t \quad (2)$$

or equivalently,

$$M\ddot{x} + 4x(C_1 + C_2) = -4C_2 x_{02} + 4C_1 x_{01} + Mg + m\omega^2 r \cos\omega t \quad (3)$$

From Equation (3), it can be observed that if M, r, x_{01}, and x_{02}remain constant, and since the auxiliary springs mounted on the upper side are mainly used to generate the initial compression and have negligible influence on the vibration behavior, C_1may be considered constant. Consequently, Equation (3) depends only on C_2, m, and ω.

Solve equation (3) using MATLAB with the following initial conditions:

$$z_1(0) = 0, z_2(0) = 0, \omega(0) = 0$$

2.3 Computed results

For convenience in the calculation process, the data used were selected from the experimental vibrating table system in

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the Laboratory of the Department of Dynamics, Military Technical Academy, specifically:

$$C_1 = 27090 \text{ N/m}; r = 40 \text{ mm}; x_{01} = 6 \text{ mm}; x_{02} = 4 \text{ mm}$$

a) Determination of the effect of the eccentric mass m

In addition to the above data, the following parameters were selected:

$$M = 60.5 \text{ kg}; m = 0.8;\ 1.2;\ 1.6 \text{ kg}; C_2 = 42088 \text{ N/m}$$

According to references [1] and [6], the optimal rotational speed of the eccentric shaft is usually chosen as

$$\omega = 261 \text{ rad/s}$$

corresponding to

$$n = 2800 \text{ rpm}$$

The results were investigated when the motor operated under steady-state conditions, as shown in Figure 3.

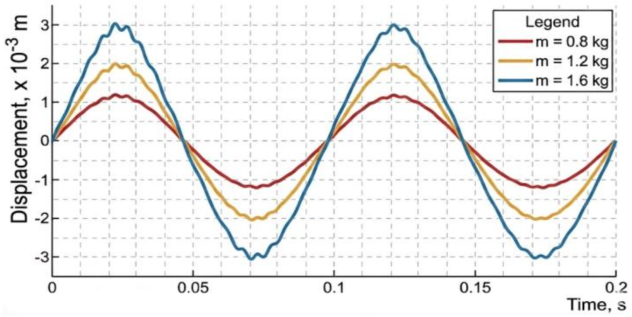

Figure 3

b) Determination of the effect of the main spring stiffness C_2

In addition to the above data, the following parameters were selected:

$$m = 0.8 \text{ kg}; \omega = 261 \text{ rad/s}$$

According to reference [6], when

$$M = 60.5 \text{ kg}$$

the spring stiffness is in the range

$$C_2 = 32000 \div 48000 \text{ N/m}$$

Using standard springs, the selected values are:

$$C_2 = 35613;\ 42088;\ 46296 \text{ N/m}$$

The results were investigated when the motor operated under steady-state conditions, as shown in Figure 4.

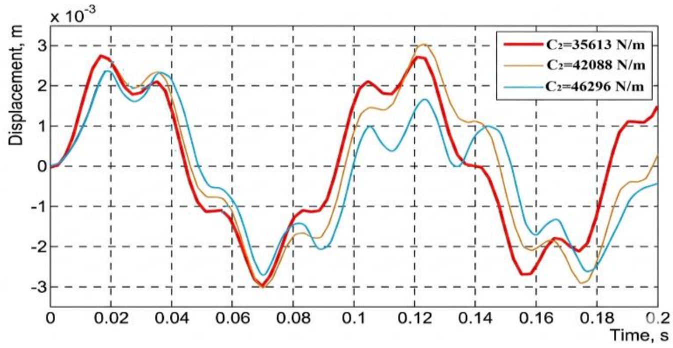

Figure 4

c) Determination of the effect of the angular vibration frequency ω

In addition to the above data, the following parameters were selected:

$$M = 60.5 \text{ kg}; m = 0.8 \text{ kg}; C_2 = 42088 \text{ N/m}$$

According to references [1] and [6], the optimal rotational speed of the eccentric shaft of the equipment is usually taken as

$$\omega = 261 \text{ rad/s}$$

corresponding to

$$n = 2800 \text{ rpm}$$

According to reference [4], a standard motor with a rated rotational speed of

$$2800 \text{ rpm}$$

was selected. Therefore, the operating range was chosen as:

$$\omega = 157;\ 209;\ 261 \text{ rad/s}$$

corresponding to

$$n = 1500;\ 2000;\ 2800 \text{ rpm}$$

The results were investigated when the motor operated under steady-state conditions, as shown in Figure 5.

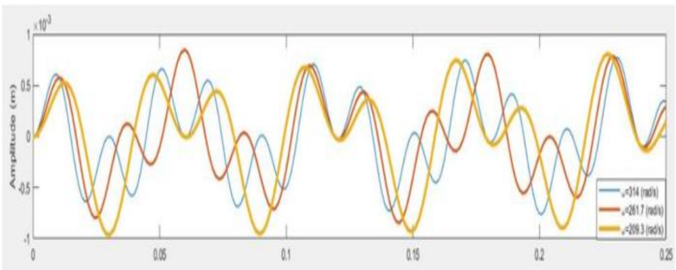

Figure 5

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d) Determination of the effect of the vibrating mass Mon the selection of the eccentric mass m

The following initial values were selected:

$$C_2 = 42088 \text{ N/m}; \omega = 261 \text{ rad/s}$$

$$M = 60.5;\ 90.5;\ 121 \text{ kg}$$

and

$$m = 0.8;\ 1.2;\ 1.6 \text{ kg}$$

The calculated results of the maximum vibration parameters are presented in Table 1.

Table 1: Maximum values of vibration parameters corresponding to $M$and m

Vibrating mass (kg)	Eccentric mass (kg)	Maximum displacement (m)	Maximum velocity (m/s)	Maximum acceleration (m/s²)
$M = 60.5$	$m = 0.8$	3×10^{-3}	0.32	60
	$m = 1.2$	4.5×10^{-3}	0.48	88
	$m = 1.6$	5.9×10^{-3}	0.65	117
$M = 90.5$	$m = 0.8$	2.3×10^{-3}	0.21	35
	$m = 1.2$	3.4×10^{-3}	0.32	52
	$m = 1.6$	4.8×10^{-3}	0.42	72
$M = 121$	$m = 0.8$	1.8×10^{-3}	0.15	27
	$m = 1.2$	2.7×10^{-3}	0.23	39
	$m = 1.6$	3.8×10^{-3}	0.32	54

III. Results and Discussions

a) Regarding the variation law

When the structural parameters are changed, the calculated results presented in Figures 3, 4, and 5 allow several conclusions to be drawn as follows:

- The variations of displacement, velocity, and acceleration parameters are harmonic oscillations with periodic cycles. However, the variation laws of these parameters under the influence of each factor are different. Specifically, when $m$changes, the vibration parameters are in phase and directly proportional to m. When $C_2$and ωvary, these vibration parameters are no longer directly proportional; moreover, the variations of velocity and acceleration with respect to changes in ωexhibit phase differences.

From these observations, when designing similar vibrating tables, it is preferable to adjust m or C_2, whereas the rotational speed of the eccentric shaft should be predetermined (commonly selected as $\omega = 261$ rad/s or $n = 2800$ rpm).

b) Regarding the parameter values

According to the results presented in Figures 3, 4, and 5, the maximum values of the vibration parameters of the vibrating table system are given in Table 2.

Table 2: Maximum vibration parameters when varying m, C_2, and ω

Parameter	When varying m(kg), $C_2 = 42088$ N/m, $= 261$ rad/s			When varying C_2(N/m), $m = 0.8$ kg, $\omega = 261$ rad/s			When varying ω(rad/s), $m = 0.8$ kg, $C_2 = 42088$ N/m		
	m1=0.8	m2=1.2	m3=1.6	C21=35613	C22=42088	C23=46296	ω1=157	ω2=209	ω3=261
Displacement (m)	2.6×10^{-3}	4.2×10^{-3}	6.4×10^{-3}	3.0×10^{-3}	3.1×10^{-3}	3.2×10^{-3}	2.2×10^{-3}	2.8×10^{-3}	3.1×10^{-3}
Velocity (m/s)	0.32	0.48	0.62	0.32	0.33	0.34	0.20	0.25	0.32
Acceleration (m/s²)	55	82	112	64	65	66	23	34	64

- According to Table 1: With $C_2$and ωkept constant, when Mremains unchanged, increasing $m$leads to an increase in the vibration parameters. Conversely, when Mincreases, the vibration parameters decrease.
- According to Table 2: When varying $m$and ω, the vibration parameters of the vibrating table change significantly. However, when varying C_2, these vibration values change insignificantly. This indicates that the vibration parameters depend strongly on the variations of $m$and ω.

Therefore, in designing vibrating table equipment, the main spring stiffness C_2should first be selected, after which the eccentric mass and the rotational speed of the eccentric shaft should be calculated appropriately to satisfy the technical requirements of the equipment

c) Selection of parameter values

According to references [1], [5], and [6], in order to design a directional vibrating compactor with a reasonable structure and to ensure the compaction quality of the concrete mixture, the theoretically optimal vibration parameters are:

- displacement: 2.5 ÷ 5mm,
- velocity: 0.3 ÷ 0.4m/s,
- acceleration: 50 ÷ 70m/s ^{2}.

From the obtained results, it can be observed that:

- For $M = 60.5$kg, $m = 0.8$kg is selected;
- For $M = 90.5$kg, $m = 1.2$kg is selected;

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- For $M = 121$kg, $m = 1.6$kg is selected.

Thus, the eccentric mass $m$is selected proportionally to the increase in the vibrating mass M.

- The vibration parameters of the equipment depend on the structural parameters. In other words, for a given vibrating mass, it is necessary to select an appropriate set of structural parameters such that the vibration parameters generated during operation remain within the theoretically optimal range mentioned above.

For example, when designing a vibrating table compactor with a vibrating mass of

$$M = 60.5 \text{ kg},$$

the appropriate structural parameter set should be selected as:

$$m = 0.8 \text{ kg}, C_2 = 42088 \text{ N/m}, \omega = 209 \text{ rad/s}$$

So that the concrete mixture compaction process achieves the highest efficiency.

IV. Conclusion

The structural parameters of the compaction equipment directly affect the vibration parameters during operation and indirectly influence the compaction quality of the concrete mixture. In order to obtain vibration parameters of the directional vibrating table that are consistent with the theoretically optimal values, it is necessary to select an appropriate set of structural parameters.

The quantitative assessment of these effects has been rarely mentioned in previous studies and technical documents. Based on the research method and results presented in this paper, it is possible to apply these findings to the calculation, design, and manufacture of vibrating tables with similar models in Vietnam.

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