ballast; railway track; plate vibrator; compaction; actuating device

Sergey LEE*
Kazakh Automobile and Road Academy named after L.B. Goncharov (KazADA)
417 a, Raiymbek Avenue, 050061, Almaty, Kazakhstan
Mazhit AKHMETOV, Amanbai IBRAIMOV, Maxim TARAN
Kazakh Academy of Transport and Communication named after M. Tynyshpayev (KazATC)
97, Shevchenko street, 050012, Almaty, Kazakhstan
**Corresponding author.* E-mail: natali_11.13@mail.ru

UPGRADING OF VIBRATING COMPACTOR OF THE RAILWAY TRACK BALLAST OF VPO-3000 MACHINE

Summary. In the course of work a new vibrating compactor with single-direction inertial forces according to the authors' innovative license is applied. Examination of possible dependence on substantiation of the new vibrating device parameters is shown. The new single-direction vibrating device for road crushed rock compaction allows decreasing vibration to the machine frame and increasing its working capacity. The new vibrating device's efficiency to 11,7% increases the existing ones.

СОВЕРШЕНСТВОВАНИЕ ВИБРОУПЛОТНИТЕЛЯ БАЛЛАСТА ЖЕЛЕЗНОДОРОЖНОГО ПУТИ МАШИНЫ ВПО-3000

Аннотация. В работе приводится новый виброуплотнитель с односторонним действием инерционных сил по инновационному патенту авторов. Приведена проверка теоретических зависимостей по обоснованию параметров нового вибрационного устройства. Новое вибрационное устройство одностороннего действия для уплотнения путевого щебня позволяет снизить вибрацию на раму машины и увеличить производительность машины. Эффективность нового вибрационного устройства на 11,7% превышает существующие вибрационные устройства.

The ballast layer is the vulnerable part of the railway structure, and the stiffness of the railway structure depends on its condition. In the ballast layer, under the influence of loads of passing trains permanent deformations in the form of sagging, track shifting in plane and line are accumulated. Deformations are accumulated irregularly along the track length. Therefore, occasional track correction should be carried out by means of repair and works on the ballast compaction in sub-sleeper area, areas of arm slope and in crib works.

The main ballast layer's stabilization process is the force (mechanical) action to the ballast – its compaction. With a view to ballast section compaction in Kazakhstan and other CIS countries, line-tamper machines of VPO-3000 type are applied, that are equipped with plate vibrators for ensuring the ballast section compaction on the part of sleeper butt in horizontal plane. Compacting actuating devices of VPO-3000 machine apply the basic compacting method – horizontal vibrocompaction to the ballast with power feed – vibrating compression. Compacting actuating devices of continuous machines (VPO-3000 and others) - plate vibrators – are compacting the ballast in sub-sleeper area with wedges placed at an angle to the centre of track, on the part of sleeper butt [1]. The ballast is continuously being exposed to vibration compression under progressive motion of the machine (Fig. 1).

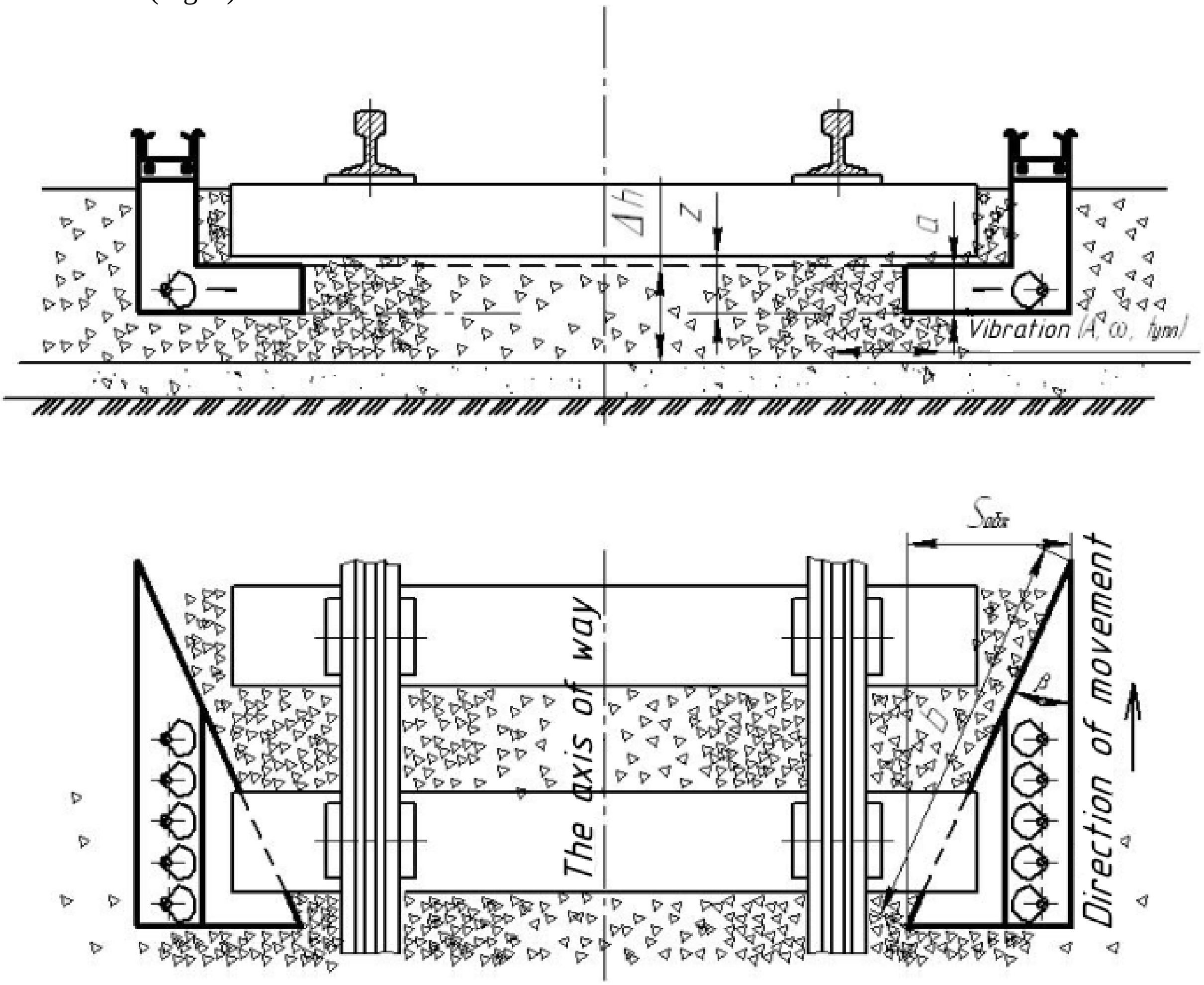

Fig. 1. The ballast layer compaction diagram on the part of sleeper butts: h – height of the ballast layer to compaction; Δh – medium lining track raising; c_m – spacing between sleeper axis; a, b – height and length of tamping pick (packing wedge); β – an angle of attack of a packing wedge to the track axis; z – embedding of tamping pick (plate vibrator)

Рис. 1. Схема уплотнения балластного слоя со стороны торцов шпал: h – высота балластного слоя до уплотнения; Δh – средняя выправочная подъемка пути; $c_ш$ – расстояние между осями шпал; a, b – высота и длина подбойки (уплотнительного клина); β – угол атаки уплотнительного клина к оси пути; z – заглубление подбойки (виброплиты)

The research analysis of the issues concerning upgrading of track machines for the railway track ballast compaction allowed determining that the existing machines (including VPO-3000 and VPO-3000M) in fact have depleted reserves of the following improving efficiency. This is related to the fact, that increase of production capacity of the machines is limited with actuating devices' speed, as well as depends on design and especially engineering working characteristics of actuating devices of the machine.

For the time being, VPO-3000 machine is the most high-production machine in the world with technical output 3 km/h. Working capacity of the machine is 1.2 – 2.2 km/h under the satisfactory quality of the ballast section. Increase of the machine's operating speed to the design one (3 km/h) results in performance quality deterioration due to increased vibration of the machine, and increase in energy consumption for compaction. In order to prevent the deleterious vibration effect on the machine's operating personnel, it is remotely-operated. Due to insufficient visibility of the ballast tamping pick by the operators, tamping pick quality is deteriorated and working capacity is decreased, which results in increased energy consumption. Besides, an idle direction of vibration forces (200 kN) affects on elastic suspension of vibrating block significantly decreasing its durability. The increased machine vibration, occurring due to vibration forces transfer from the machine [1].

Intensification of compaction and stabilization process of the ballast section is one of the methods of further capacity increase and decrease in machine vibration under work quality maintenance and improvement. This task's solution shall allow significant decreasing in labour intensity and a scope of line-tampering works (the ballast compaction) in the course of railway track operation.

In this connection, the authors of this article propose a vibrating device with single-direction effect of inertial forces (the innovative license of the Republic of Kazakhstan No.22585 "Vibrating Compactor", certificate No. 6 dated 15.06.2010). Technical results of the invention are decrease of vibration being transmitted to machine frame, and improvement of working conditions, as a result of which, finally, machine's working capacity shall be increased and the quality of the compacted crushed rock ballast shall be improved.

The proposed single-direction vibrating device (Fig. 2) consists of frame1with platform vibrator 2. Inside frame 1 shaft 3 and two parallel shafts 4 and 5 are positioned. Shafts 4 и 5 are placed at bearings 17 and on them synchronizing gears 6 and 7 joining the shafts are installed. On either side of gear-type synchronizing wheels 6 and 7 balance weights 8 are positioned on shafts. Shaft 3 is attributable to an electric motor (it is not shown conditionally) and on it gear 9 being linked to gear-type synchronizing wheel 6 is installed. Each balance weight contains a fixed element 10 being fixed on shaft, and moving element 11. Fixed element 10 of each balance weight is arranged in the section as a full circle and in it notch 12 is arranged. On fixed element 10 of each balance weight moving element 11 is hingedly fixed on axis in such a manner, that its end falls outside the range of balance weight's fixed element. Mass of moving element 11 of balance weight is equal to that one of the material extracted from notch 12 of fixed element 10 of balance weights. Moving element 11 of balance weight can be cut of fixed element 10 of balance weight, forming a notch 12 and, in this case, its form is similar to that one of the notches. Moving element 11 of balance weight can have a form of the sector and an end rounded off radially. On fixed elements 10 being placed adjacent to balance weights of two shafts, fixed elements 11 of balance weights are fixed symmetrically. End of moving element 11 of each balance weight is connected with one end of lever14 by means of axis 15. The second end of the lever 14 is hingedly fixed to side wall of frame 1 on axis 16. The levers are fixed out of center towards longitudinal axis of shafts 4 and 5 to the side, in the direction of which vibration generation is required.

Vibrating compactor is operating in the following manner. Under an electric motor switching on the rotation from shaft 3 through gears 9 and gear-type wheels 6 and 7 is transmitted to shafts 4 and 5, which are rotating synchronously towards opposite sides. In one rotation of shafts during rotation of moving element 11 of balance weight on axis 13, its end falls outside the range of fixed element 10 of balance weight. At that, the radius of balance weight 8 is increased and an exciting force directed to one side only (platform vibrator side) is generated. Further on, the moving element 11 of each balance weight resets by means of lever 14, rotating about the axis 16. At that, exciting forces are reducing at the expense of decrease in the balance weight radius, and they are also balanced by rotating symmetrically, but towards opposite direction of moving element 11 that is adjacent to the second shaft of balance weight. As a consequence of this, vibrations occur only in one direction of the moving element's end 11 of balance weight out of bounds of the fixed element 10 of balance weight. The device's vibrations are perceived by platform vibrator of the frame and are transmitted to the vibrating object. Creation of single-directed vibrations allows decreasing that ones that are transmitted to the frame of vibrating compactor and accordingly increasing its durability.

Fig. 3 shows an operational scheme of the proposed vibrating block structure of VPO-3000 machine, position I, when all balance weights are completely opened and placed in one position, then peak vibrating forces directed by the arrows indicated in the figure are occurred.

In quarter turn of balance weights (position II), vibrating forces occurring in each pair of balance weights shall be directed to the opposite sides; as these forces are equal in value the vibration does not occur. The next turn of balance weights to another quarter turn shows the position, when all moving balance weights are closed and are integral with the fixed element, then vibrating forces do not occur (position III). In turn the balance weights to another quarter turn that is similar to position II, the vibrating forces in each pair of balance weights are directed to the opposite sides and are balanced (position IV). Having analyzed the circular turn of balance weights (one turn) it is obvious that the vibrating forces are occurring only once (position I). In a new vibration device the constraining force is equal to the value of that one of separate balance weights, and operates perpendicularly to longitudinal axis of the railway track.

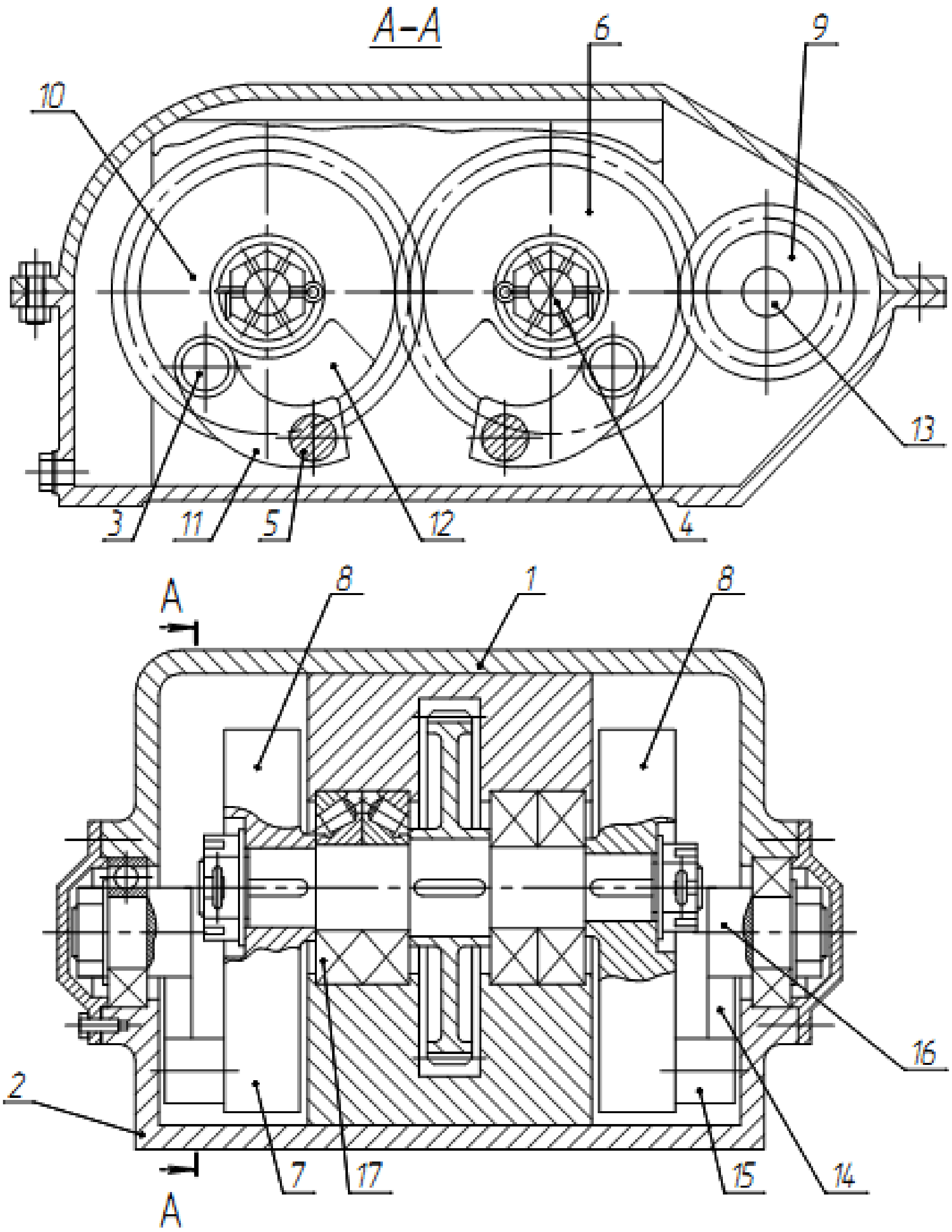

Fig. 2. Single-direction vibrating compactor: 1 – frame; 2 – platform vibrator; 3, 4, 5 – shafts; 6, 7 – synchronizing device gears; 8 – balance weight; 9 – gear; 10 – fixed element of balance weight; 11 – moving element of balance weight; 12 – notch; 13,15,16 – axis; 14 – lever; 17 – bearings

Рис. 2. Виброуплотнитель одностороннего действия: 1 – корпус; 2 – виброплощадка; 3,4,5 – валы; 6,7 – шестерни синхронизатора; 8 – дебалансы; 9 – шестерня; 10 – неподвижная часть дебаланса; 11 – подвижная часть дебаланса; 12 – паз; 13,15,16 – оси; 14 – рычаг; 17 – подшипники

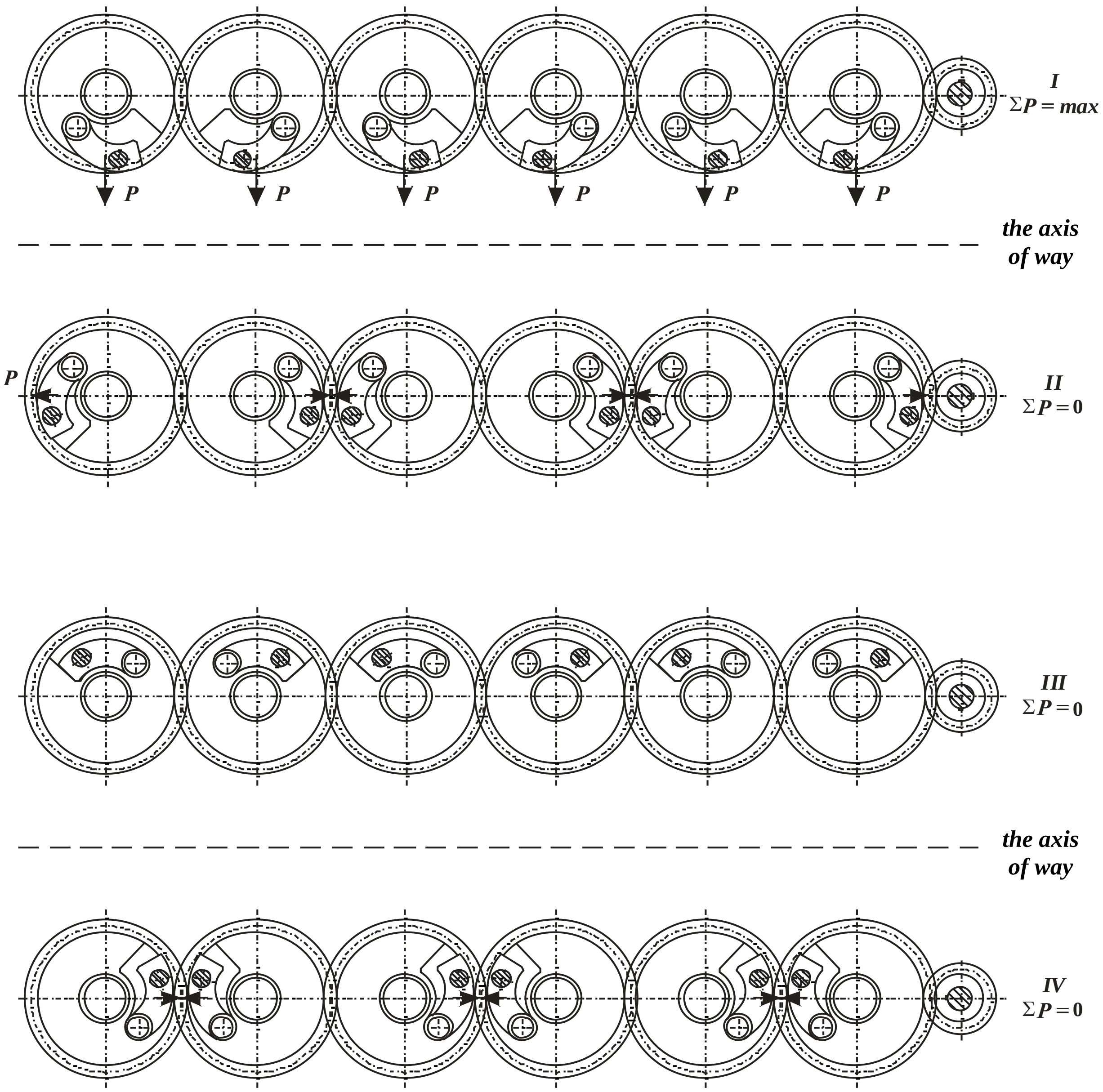

Fig. 3. Operational scheme of the proposed vibrating block structure of VPO-3000 machine
Рис. 3. Схема работы предлагаемой конструкции виброблока машины ВПО-3000

Two tamping units mounted to the machine frame by springs were installed at VPO-3000 machine, they are compacting the ballast in sub-sleeper area using wedges installed at an angle β to axis of the railway track on the part of sleeper butts in horizontal plane (Fig. 1). The ballast is continuously being exposed to vibration compression under progressive motion of the machine.

The required degree and uniformity of compaction are the required indicators of the ballast's compaction quality, ensuring train movements along the repaired area of the railway track without speed limitation.

Coupling parameters of a plate vibrator and the ballast shall depend on the ballast's vibration compression modes and are changing under variation of amplitude, vibrational frequency and crimping speed of the ballast. Vibration compression mode is defined by the proportion of crimping speed and the peak speed of vibrations $C = \upsilon_{crimp.}/A\omega$ [1]. It is found that the best effect of the ballast crushed rock compaction is reached under the vibration speed $A\omega$ = 1.2- 1.5 m/s.

Change of basic coupling parameters of plate vibrator under vibration crimping of crushed rock depends on the proportion $\upsilon_{mov}/\upsilon_n$ ($\upsilon_n = A\omega$ – peak vibration velocity of plate vibrator wavering, m/s). They are used under the selection of vibration compaction parameters of actuating devices of VPO-3000 machines. In VPO-3000 machine the vibration devices with six balance weights are forming the aggregate constraining force that is equal to 200 kN. In a new vibration device with single-direction of force effect in position I of balance weights (Fig. 3), the constraining force is equal to the value of that one of separate balance weights, and operates perpendicularly to longitudinal axis of the railway track.

In positions II, III and IV of balance weights the constraining force of the vibrator is equal to zero, as the constraining forces of separate balance weights are mutually balanced. For exciting of plate vibrator was used the balance weight vibrator in-built to the frame, with constraining force directed transverse to railway track.

We have carried out the checking calculation of basic parameters of vibrating compactor using materials and according to track machines calculation procedure of the Central Designing Department of Heavy Track Machines of the Ministry of Communication Lines of Russia [2]. For determination of amplitude of oscillation A of plate vibrator the following formula is used:

$$A = \frac{v_m \cdot tg\beta}{0{,}12\,\omega}, \tag{1}$$

where: υ_{m} – operational speed of the machine, n/s; β – angle of attack of packing wedge to axis of the track (see Fig. 1).

Grade radius of balance weights R = 200 mm – accordingly radii of 1st, 2nd, 3rd, 4th, 5-th and 6th of synchronizing gears. Rotational frequency n = 1000 rot./min, then angular speed of balance weights $\omega = \frac{n\pi}{30} = 104 rad/\sec$.

Centrifugal force of balance weights depends on the three factors: radius, angle speed and mass. All these factors have effect on value of quantity of centrifugal forces, and values of angle speeds – on their directions. Therefore, the values of angle speed were taken in such a manner, that in one full turn of balance weights all centrifugal forces have one direction and are directed to one side (to the ballast compaction side). The combined values of masses and angle speeds shall influence not only to the values of centrifugal forces, in case when they are directed to one side, but also shall balance these forces' influence in other directions, significantly decreasing the action force towards the opposite side, i.e., in the opposite direction.

As an initial position the one is taken, when all balance weights are directed downward, i.e., under, $t = 0$, $\varphi = 0$ (position I on Fig. 3). Now, let us suppose that the balance weights are turned around 90 degrees $\left(\varphi = \frac{\pi}{2}\right)$ (position II on Fig. 3).

In position III: $\varphi = \pi$.

In position IV: $\varphi = \frac{3\pi}{2}$.

In position V: $\varphi = 0$.

In position V, all balance weights are directed downwards, as in position I. All balance weights are turned to angles that are equal to (2 π).

Mass of each balance weight is taken as m = 15 kg.

Centrifugal force of one balance weight is equal to:

$$F = m\omega^2 R. \tag{2}$$

System of axis is taken as shown in Figure 1. Centrifugal forces of balance weights are changed according to their time and projection at axis x and y and are indicated (for all balance weights) as follows:

$$F_y = 6m\omega^2 R \cos \omega t \tag{3}$$

$$F_x = -6m\omega^2 R \sin \omega t \tag{4}$$

Total equivalent force of vibrations shall be equal to:

$$F = \sqrt{F_x^{\ 2} + F_y^{\ 2}}\ . \tag{5}$$

Formula (4) indicates the existing constraining force for the plate vibrator proposed for this kind of work, where F_y and F_x are indicated by formulas (2) and (3) accordingly.
The existing constraining force for the plate vibrator of the VPO-3000 machine is indicated as follows:

$$F = F_0 \sin \omega t\ , \tag{6}$$

where: F_0 – is peak constraining force.

$$F_0 = \sum_{i=1}^{k} m_i r_i \omega^2\ , \tag{7}$$

where: m_i – mass of balance weights; r_i - eccentricity (radii); ω – angle speed; κ – number of balance weights.

Figures 4 - 6 shows the computer-calculated force diagrams F_x, F_y, F depending on an angle of rotation $(\varphi = \omega t)$ in one complete rotation.

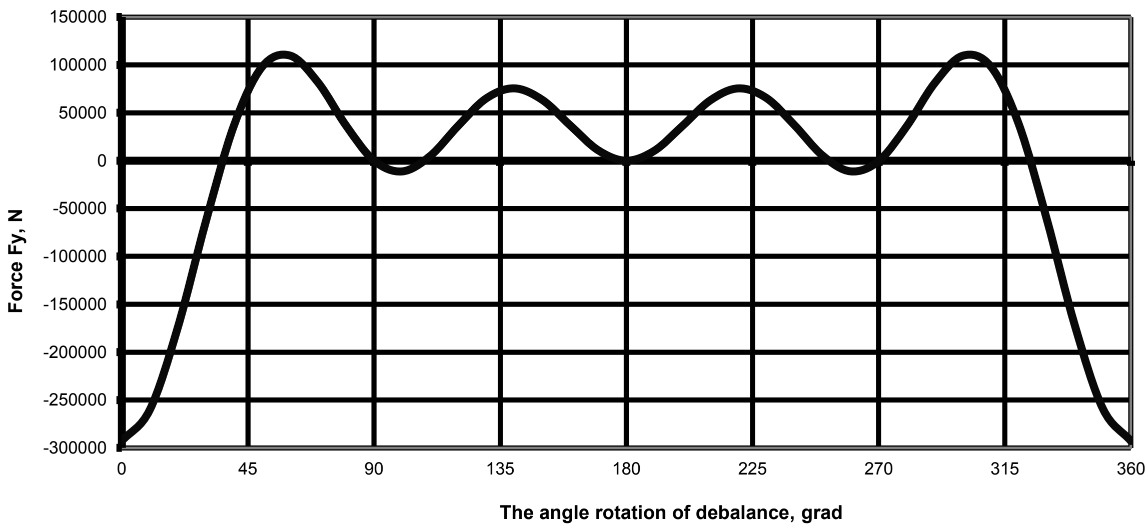

Fig. 4. Dependence of force projection change to axle y
Рис. 4. Зависимость изменения проекции силы на ось *y*

Out of force change diagrams it is obvious, that under angles $\varphi=0^0$ and $\varphi=2\pi$, the resulting constraining force F and its projection to axle y (F_y) have maximum values and according to absolute value are equal to 292 kN, as under these angles the constraining force projection to axle x (F_x) is equal to zero.

According to diagrams, under the angles $\varphi=\pi/2$, $\varphi=\pi$, $\varphi=2\pi$, the values of all the three forces are equal to zero, it is confirmed by disposition of balance weights in positions II, III, IV on Figure 3. Out of the reviewed diagrams one essential factor can be noticed, which confirms the very important technical characteristic of the proposed plate vibrator, that between the angles of rotation from $\varphi=50^0$ to $\varphi=300^0$, all the three forces F_x, F_y, F have insignificant vibrations in comparison with their values at the end of angle section under $0^0 \leq \varphi < 50^0$ and $300^0 < \varphi \leq 360^0$. Under these angles all the three forces have maximum values according to absolute value, and the resulting force is directed to the ballast's side.

Direction of the resulting force can be determined by the below formulas. If the resulting force direction makes an angle α with axle y and an angle β with axle x, then the directional cosines shall be equal to:

$$\left.\begin{aligned} \cos\alpha = \frac{F_y}{F} \\ \cos\beta = \frac{F_x}{F} \end{aligned}\right\}, \tag{8}$$

from here, angles α and β can be found

$$\left.\begin{aligned} \alpha = \arccos(\frac{F_y}{F}) \\ \beta = \arccos(\frac{F_x}{F}) \end{aligned}\right\}. \tag{9}$$

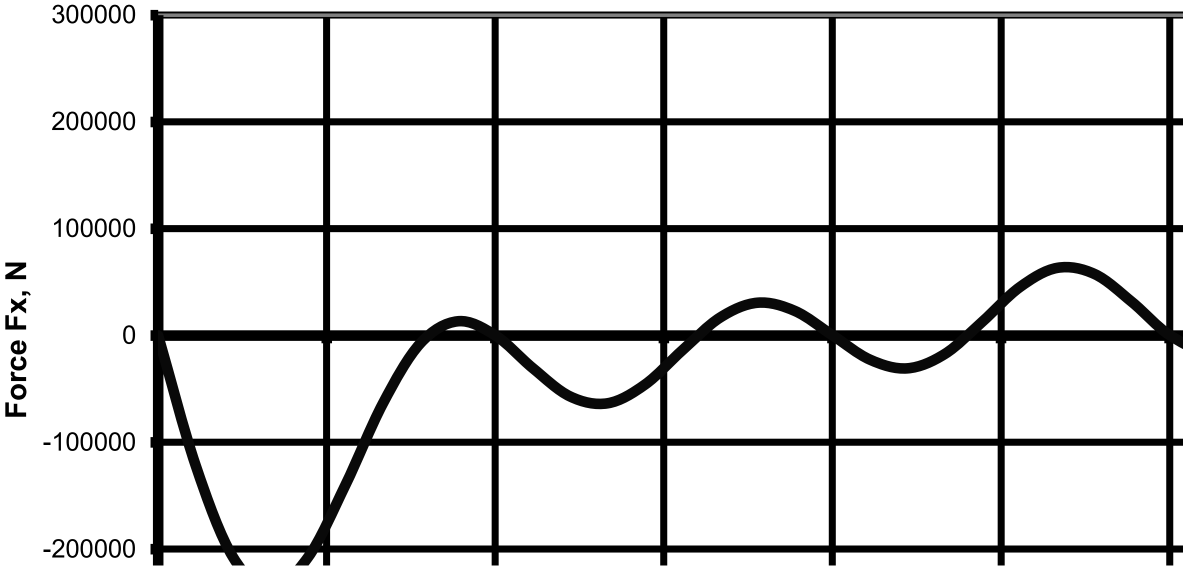

Fig. 5. Dependence of force projection change to axle x
Рис. 5. Зависимость изменения проекции силы на ось *x*

Let us review an example when the resulting constraining force *F* has peak value. The resulting constraining force is always positive therefore a question arises, where this force is directed to.
Out of diagrams on Figures 4, 5, 6 it is seen, that under angles *φ= 0°* and *φ = 360°*.

$$F_{\max} = 292{,}830 \quad N$$
$$F_y = -292{,}830 \quad N$$
$$F_x = 0 \quad N$$

Then according to formula (9) we have:

$$\begin{aligned} \alpha = \arccos(-1) = 180^0 \\ \beta = \arccos(0) = 90^0 \end{aligned} \tag{10}$$

By this means, the forces F_y and F_x are directed mutually perpendicularly, and the resulting constraining force is directed to the ballast's side, as well as the force F_y.

For checking the theoretical dependencies according to parameters substantiation of single-action vibrating device and estimating its working capacity, a computer experiment was carried out on force determination when operated at vibrating block.

Fig. 7 shows the dependency of change of plate vibrator's forced vibrations, maximum deflections of springs of plate vibrator f1 and f2 are shown on Fig. 8.

From the graphs analysis of Figure 7 it is visible that the value of amplitude of oscillation about the X axis (red line with small squares) is defined by result of vibrational force action at the same time as the oscillations about the Y axis (green line with triangles) are absent, because the vibrational forces generated in each pair of unbalances are aimed directly at the opposite side, i.e. they balance each other.

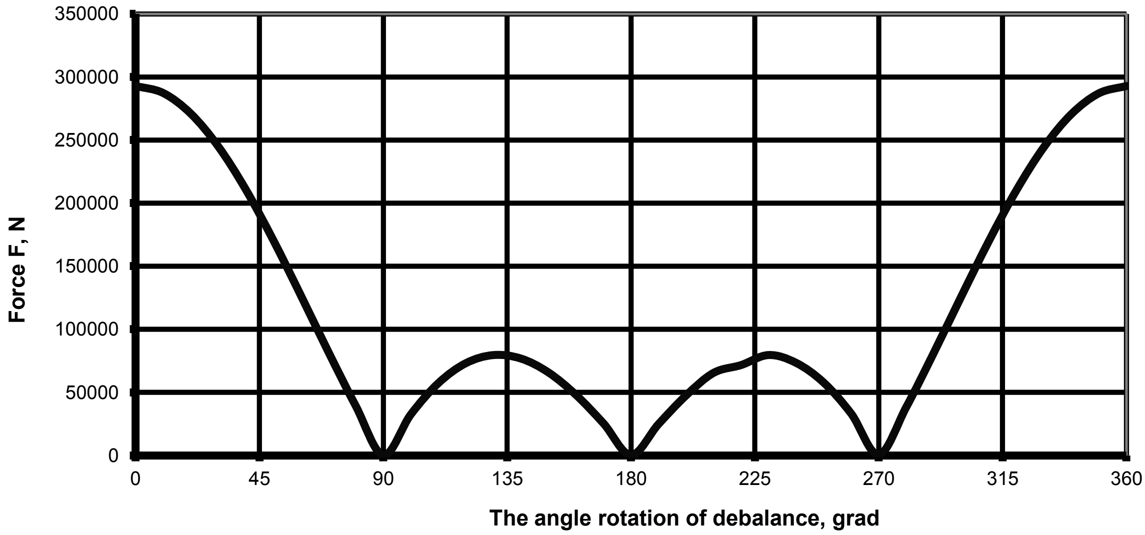

Fig. 6. Dependence of change of the resulting constraining force F
Рис. 6. Зависимость изменения результирующей вынуждающей силы F

The analysis of the figure 8 shows the deviation of spring plates under the influence of vibrational force (blue line with small rhombus), as well as the inertial forces that arise due to elastic spring plates (red line with small squares).

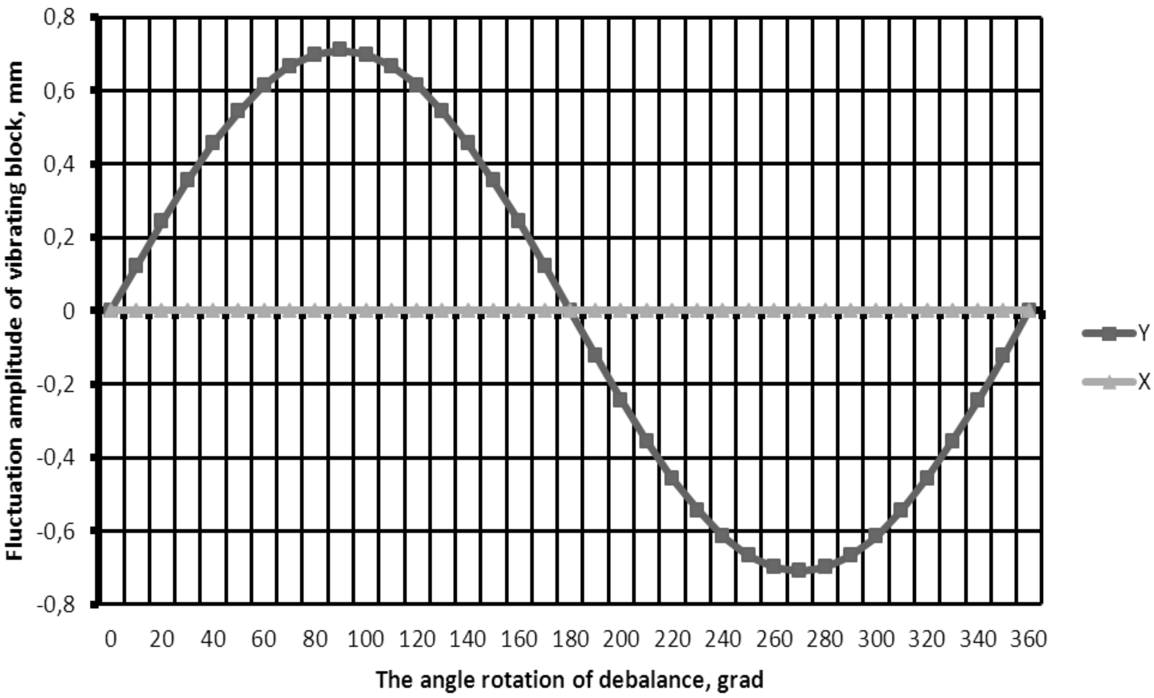

Fig. 7. Dependence of vibrational amplitude on the angle of rotation of vibrating block
Рис. 7. Зависимость амплитуды колебания от угла поворота виброблока

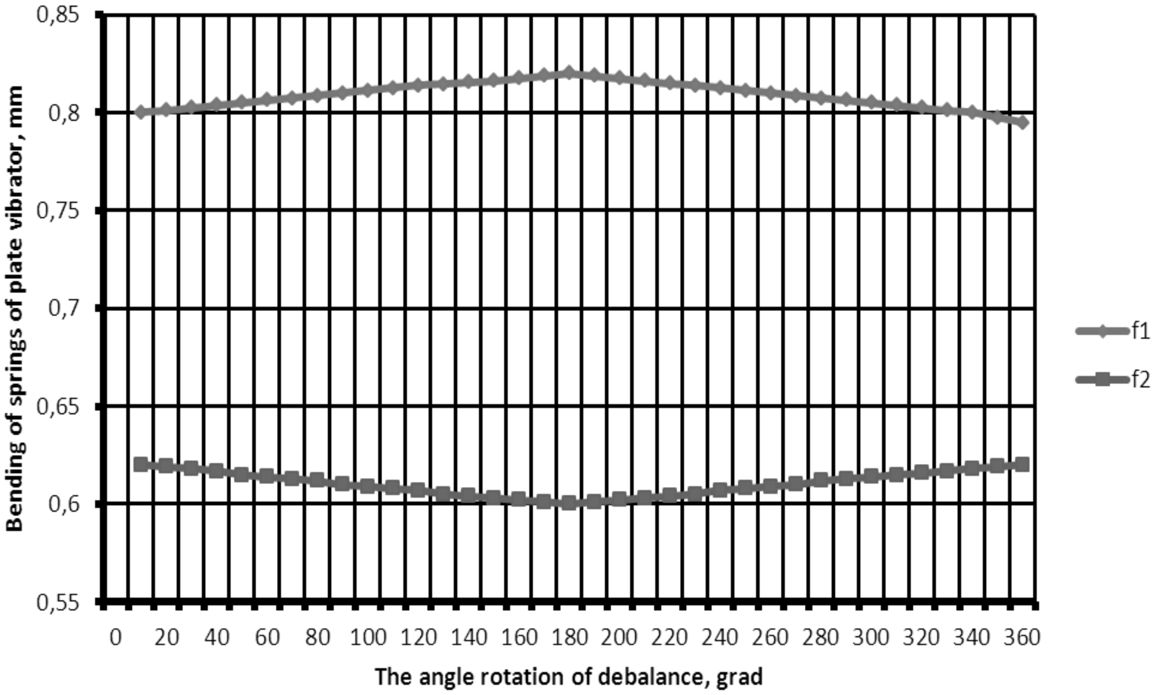

Fig. 8. Dependence of spring vibrational amplitude of plate vibrator
Рис. 8. Зависимость амплитуды колебаний рессор виброплиты

Performance evaluation of single-action vibrating device for the railway track ballast compaction was carried out in comparison with the existing vibrating compactor of VPO-3000 machine according to parameter [3]:

$$\sigma = \frac{120N_{cp}}{m(x_c \cdot n)^2} = \frac{60A_0}{mx_c n^2}, \tag{11}$$

where: σ – compaction ability, n – rotational frequency, A_0 – amplitude, N_{av}. – capacity, x_d – displacement

According to the comparison, $\sigma^{H} > \sigma^{A}$ to 11.7% the efficiency of the new single-action vibrating device exceeds the existing ones to 11.7%.

CONCLUSIONS

Basically new vibrating compactor design with single-action of inertial forces was developed [6]. Examination of theoretical dependencies according to parameter substantiation of the new vibrating device was carried out. The efficiency of the new vibrating device exceeds the existing ones to 11.7%. The new single-action vibrating device for crushed rock compaction allows decreasing of vibration to the machine frame, and improving the machine's working capacity.

Bibliography

1. Соломонов С.А., Бугаенко В.М., Бураков А.А. и др.: *Путевые машины*. Желдориздат, Москва 2000.
2. *Машина выправочно-подвивочная-рихтовочная ВПР-2. Техническое описание и инструкция по эксплуатации.* Центральное конструкторское бюро тяжелых путевых машин. Транспорт, Москва 1995.

3. Сырейщиков Ю.П., Дмитриев Е.С., Лукин Е.А., Солнцев А.К.: *Новые путевые машины (Подбивочно-выправочные и рихтовочная ВПР-1200, ВПРС-500, Р-200)*. Транспорт, Москва 1984.
4. Ли С.В., Ахметов М.Ф., Ускембаева Б.О., Ибраимов А.К.*: Вибрационное устройство однонаправленного действия.* Промышленный транспорт Казахстана No. 3, Алматы 2010, с. 98-99.
5. Forssblad L.: *Vibrations of soils and foundations*, Dynapac Maskin AB, Solna, Sweden 1981.
6. Муратов А.М., Ахметов М.Ф., Ускембаева Б.O.S.: *Виброуплотнитель*. Инновационный патент Республики Казахстан No. 22585, 15.06.2010. Бюл.№6.

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