VERIFICATION OF THE MATHEMATICAL MODEL OF THE ENERGY CONSUMPTION DRIVE FOR VIBRATING DISC CRUSHER
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ВЕРИФІКАЦІЯ МАТЕМАТИЧНОЇ МОДЕЛІ СПОЖИВАНИХ ЕНЕРГОВИТРАТ ПРИВОДА ВІБРАЦІЙНОЇ ДИСКОВОЇ ДРОБАРКИ

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Keywords: *vibrating disc crusher, energy consumption, grinding, cutting, impact*

ABSTRACT

One of the most energy-intensive operations used in feed technology for livestock is grinding. Therefore, scientific research aimed at minimizing the consumption of energy resources by technological machines – crushers and increasing the energy efficiency of the process in general is an important task. When grinding grain with a moisture content index above the basic condition, there is a low efficiency of the method of crushing by impact due to the increased plasticity of the material and an increase in the value of the marginal deformation that the grain can perceive before fracture. Partial solution of this problem is possible by combining the method of cutting and impact, which formed the basis of a technical solution implemented in the scientific laboratory of the Vinnytsia National Agrarian University.

In previous scientific research, the authors developed and analyzed a mathematical model of energy consumption by a crusher drive. As a result of theoretical studies, the analytical and graphical dependence of the energy consumption by the drive from the angular velocity of the crusher rotor shaft was obtained.

The aim of this article is to verify the reliability of a mathematical model by conducting experimental tests and comparing the results of experimental researches with theoretical researches.

The experimental part of the work was performed in the laboratories of the department of processes and processing equipment and food industries of the Vinnytsia National Agrarian University and the specialized laboratory of «Ovchatsky» MPD, SE «Ukrspirt», using the experimental-industrial model of the vibration disc-type crusher. The EMF-1:1 electronic wattmeter recorded energy consumption, as well as wireless tachometer UNI-T UT372 the rotation frequency of the drive shaft (rotor).

Verification of the mathematical model has shown a high level of its adequacy, and it can be used in the designing of a vibrating disc crusher of this type.

РЕЗЮМЕ

Однією із найбільш енергоємних операцій, що застосовуються в технології приготування кормів для тваринництва є подрібнення. Тому, наукові дослідження спрямовані на мінімізацію споживання енергетичних ресурсів технологічними машинами – дробарками та підвищення енергоефективності процесу в цілому є актуальною задачею. При подрібненні зерна із показником вологовмісту вище базисної кондиції спостерігається низька ефективність способу подрібнення ударом, що зумовлено підвищеною пластичністю матеріалу та збільшенням значення граничної деформації, яку зерно може сприймати до руйнування. Часткове вирішення даної проблеми можливе при комбінуванні способу різання та удару, що і лягло в основу технічного рішення реалізованого у науковій лабораторії Вінницького національного аграрного університету.

В попередніх дослідженнях авторами було розроблено та проаналізовано математичну модель споживання енергії приводом дробарки.

В результаті теоретичних досліджень було отримано аналітичну та графічну залежність споживання енергії приводом від кутової швидкості валу ротора дробарки. Метою даної статті є перевірка достовірності математичної моделі шляхом проведення експериментальних випробовувань та порівняння результатів експериментальних досліджень із теоретичними.

Експериментальну частину роботи виконано на базі лабораторій кафедри процесів та обладнання переробних та харчових виробництв Вінницького національного аграрного університету і спеціалізованої лабораторії «Овечацького МПД» ДП «Укрспирт» з використанням експериментально-промислового зразка вібродискової дробарки. Електронним ватметром EMF-1:1

фіксували показники споживання енергії, а безпровідним тахометром UNI-T UT372 частоту обертання приводного вала (ротора). Верифікація математичної моделі показала високий рівень її адекватності, та вона може бути використана при проектуванні вібраційної дискової дробарки даного типу.

INTRODUCTION

Today the hammer crushers *(Yanovych V.P., Honcharuk T.V., Honcharuk I.V., Kovalova K.V., 2018)* are usually used in livestock farms and feed mills for grain grinding. In this technological machines, the destruction of the material occurs as a result of the successive stages of the process: applying a distributed load to the area of the hinged-hanging hammer (*Kudinov Ye.S., Bojko I.G., 2010*), the appearance of deformations in the material and the internal tension, the achievement of the limit values of internal tension and deformations, the breaking of the bonds of atoms and molecules among themselves *(Toneva P., Epple P., Breuer M., 2011; Nanka O.V., 2015)*.

In the process of crushing mainly fragile and plastic fractures occur. For a fragile fracture, a slight deformation of the material is characteristic, and after the destruction there is no residual deformation. The impact energy is used to overcome the forces of adhesion of body particles, that is, the formation of a new surface. During the destruction of plastic materials energy is spent on the break of structural bonds and on significant plastic deformation. The energy expended on deformation turns into heat.

The material strength and its extreme deformation is determined by the structural and mechanical characteristics of the grain and depends on the variety, size, density, moisture content, temperature, etc. *(Hvozdiev O.V., Shpyhanovych T.O., Yalpachyk O.V., 2011)*. If moisture content of the material increases, then the fragility and strength of the material decrease, while the plasticity and absolute deformation, which the grain can withstand before the beginning of destruction, increase.

Grinding brittle materials requires significantly less energy than plastic. The fragility and plasticity of a material depends on its physical condition, therefore it is advisable to grind the material in a fragile state.

As experience shows, fodder grain with moisture content, which exceeds the basic conditions, is used mainly for the production of feed. This is due to the material aspects (market value of raw grain is much lower) and the production capacity of a particular enterprise.

It therefore becomes apparent that the profitability and competitiveness of the livestock sector depend on the energy-efficient nature of this technological operation implementation, and the reduction of energy costs in the process is an important task.

In order to reduce costs, it is very promising to use machines for grinding grain proposed by Sergeev N.S. *(Sergeev S.N., 2008)*, Yanovych V.P. *(Yanovych V.P., 2017),* Nanka O.V. *(Nanka O.V, Boyko I.G., 2012)*. The principle of these machines is based on the combination of cutting and impact action on the material. The advantage of such a combination is the local excessive stress of the surface microwaves in the places of loads application. In the cutting process, the knife blade is wedged into the product and at the contact surface a specific pressure is created that is sufficient for the destruction of the body.

Using the infrastructure of the laboratory of the process and processing equipment and food industry of Vinnytsia National Agrarian University, a vibration disc crusher was developed (Fig. 1) *(Palamarchuk I.P., Yanovych V.P., Kupchuk I.M., Solomko I.V., 2013)*. In this crusher, the electric motor *5* transmits to the eccentric shaft *7* (through the coupling *6*) a rotary movement. The counterweight *8* is set to the eccentric shaft *7*. The eccentric shaft *7* is equipped with a rotor *9* with discs *10*.

The rotational movement of the eccentric shaft and counterweight leads to the imbalance of the rotor *9* and the discs *10 (Palamarchuk I.P., Yanovych V.P., Kupchuk I.M., Solomko I.V., 2013; Palamarchuk I.P., Yanovych V.P., Kupchuk I.M., 2015)*.

The material is continuously fed through the feeding throat *2* and is crushed as a result of the rotating and oscillating motion of the discs *10*. With reduced particle size, the crushed material under the action of centrifugal forces and oscillatory movement of the screen is sieved. The particles equal to or smaller than the diameter of the sieve holes *4* are discharged through the neck for unloading *3*; the residue should be re-grinded *(Palamarchuk I.P., Yanovych V.P., Kupchuk I.M., Solomko I.V., 2013; Palamarchuk I.P., Yanovych V.P., Kupchuk I.M., 2015)*.

This combination of methods of action (impact and cutting) makes it possible to grind raw materials with high moisture content and reduce energy costs for this technological operation *(Palamarchuk I.P., Yanovych V.P., Kupchuk I.M., 2015; Yanovych V.P., Kupchuk I.M., Kovalchuk O.S., 2016)*.

The results of theoretical research of energy consumption on the crusher drive showed a reduction in consumption compared to existing crushers *(Yanovych V.P., Kupchuk I.M., Kovalchuk O.S., 2016)*. However, the use of these mathematical models for the design of crushers requires their verification.

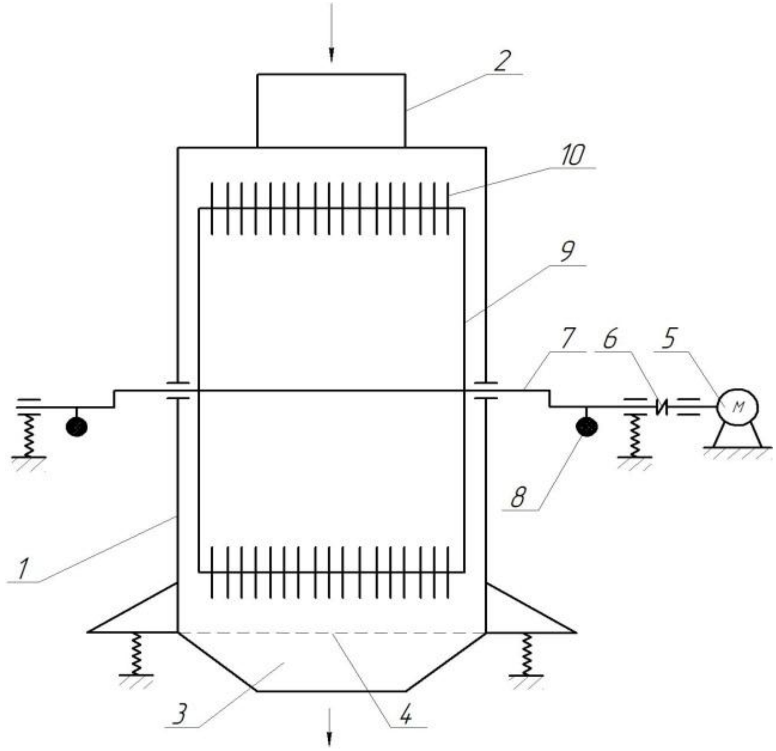

Fig 1 - Vibration disc crusher (kinematic scheme):
1 – frame; 2, 3 - neck for loading and unloading; 4 - sieve; 5 – electric motor; 6 – coupling; 7 – eccentric shaft; 8 - counterweight; 9 - rotor; 10 – discs.

The aim of the article is to check the adequacy of the mathematical model of energy consumption for driving a crusher by comparing the results of theoretical and experimental research.

MATERIALS AND METHODS

The results of theoretical research on energy consumption by crusher drive are presented in the previous article *(Yanovych V.P., Kupchuk I.M., Kovalchuk O.S., 2016)* in the form of analytical (1) and graphical dependence (Fig. 2).

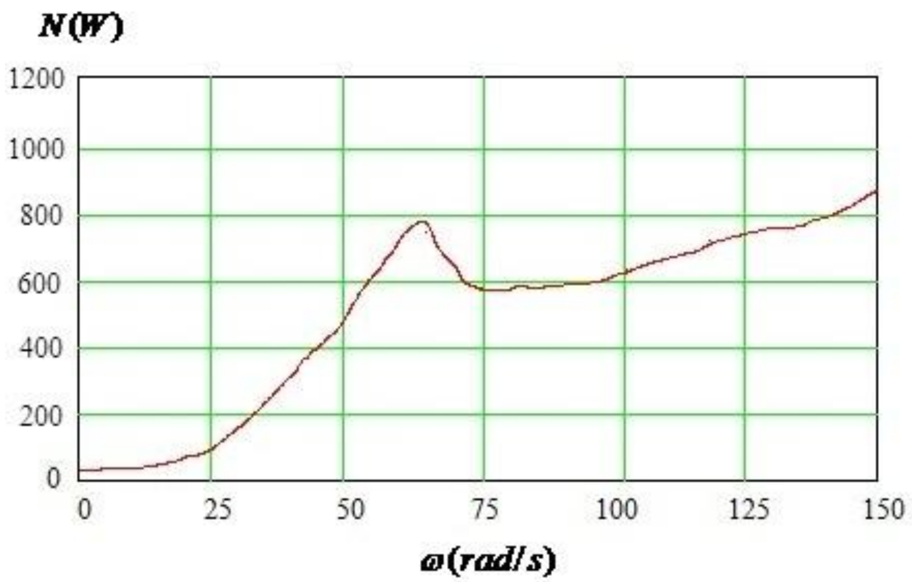

Fig. 2 - Energy consumption by crusher drive (theoretical study)

$$N=\left[\left[\begin{gathered}\sqrt{\begin{gathered}\left(\left(\frac{(m_2+m_3)\omega_{rotor}{}^2e-m_4\omega_{rotor}{}^2l}{m_1}\right)+\frac{m_2\cdot\omega_{rotor}\cdot\mathrm{e}+m_3\cdot r\cdot\omega_{rotor}-m_3\cdot 2ku\cdot r_{disk}\cdot\omega_{disk}+m_4\cdot\omega_{rotor}\cdot l}{cos(\omega_{rotor}\cdot t)m_1}\right)^2+\\ \left(\frac{(m_2+m_3)\omega_{rotor}{}^2e-m_4\omega_{rotor}{}^2l}{m_1}-\frac{(m_1+m_2+m_3+m_4)g}{sin(\omega_{rotor}\cdot t)\cdot m_1}\right)^2\end{gathered}}\times\\ \sqrt{\begin{gathered}\left(\frac{\left(\left(\frac{(m_2+m_3)\omega_{rotor}{}^2e-m_4\omega_{rotor}{}^2l}{m_1}\right)+\frac{m_2\cdot\omega_{rotor}\cdot\mathrm{e}+m_3\cdot r\cdot\omega_{rotor}-m_3\cdot 2ku\cdot r_{disk}\cdot\omega_{disk}+m_4\cdot\omega_{rotor}\cdot l}{cos(\omega_{rotor}\cdot t)m_1}\right)\cdot\alpha_x\omega_{rotor}}{\left(k_x^2-\omega^2{}_{rotor}\right)^2+\alpha_x^2\omega^2{}_{rotor}}\cdot\right.\\ \left.\cdot(\alpha_x\omega_{rotor}\cos\omega_{rotor}t-\left(k_x^2-\omega^2{}_{rotor}\right)\sin\omega_{rotor}t)\right)^2+\\ \left(\frac{\left(\frac{(m_2+m_3)\omega^2{}_{rotor}e-m_4\omega^2{}_{rotor}l}{m_1}-\frac{(m_1+m_2+m_3+m_4)g}{sin(\omega_{rotor}\cdot t)\cdot m_1}\right)\cdot\alpha_z\omega_{rotor}}{\left(k_z^2-\omega^2{}_{rotor}\right)^2+\alpha_z^2\omega^2{}_{rotor}}\cdot\right.\\ \left.\left(k_z^2-\omega^2{}_{rotor}\right)\cos\omega_{rotor}t+\alpha_z\omega_{rotor}\sin\omega_{rotor}t)\right)^2\end{gathered}}\end{gathered}\right]+\\ \left[\begin{gathered}0{,}5\cdot\mu\cdot d_{bearing}\cdot\omega_{rotor}{}^2\cdot\\ \sqrt{\begin{gathered}\left(\left(\frac{(m_2+m_3)\omega_{rotor}{}^2e-m_4\omega_{rotor}{}^2l}{m_1}\right)+\frac{m_2\cdot\omega_{rotor}\cdot\mathrm{e}+m_3\cdot r\cdot\omega_{rotor}-m_3\cdot 2ku\cdot r_{disk}\cdot\omega_{disk}+m_4\cdot\omega_{rotor}\cdot l}{cos(\omega_{rotor}\cdot t)m_1}\right)^2+\\ \left(\frac{(m_2+m_3)\omega_{rotor}{}^2e-m_4\omega_{rotor}{}^2l}{m_1}-\frac{(m_1+m_2+m_3+m_4)g}{sin(\omega_{rotor}\cdot t)\cdot m_1}\right)^2\end{gathered}}\end{gathered}\right]\right]\times\gamma_{eem}^{-1}\tag{1}$$

$$m_1=m_\kappa+m_m+m_{sf}+m_b;\tag{2}$$

$$m_2=m_r+m_c;\tag{3}$$

$$m_3=m_{id};\tag{4}$$

$$m_4=m_{cw};\tag{5}$$

$$m_r=m_{esh}+m_{cd}+m_{var}+m_{sup}+m_{axles},\tag{6}$$

where m_κ – mass of frame, kg; m_m – mass of material, kg; m_{sf} – mass of support frame, kg; m_b – mass of bearing units, kg; m_r – rotor weight, kg; m_c – mass of couplings, kg; m_{id} – mass of impact discs, kg; m_{cw} – weight of counterweight, kg; m_{esh} – mass of eccentric shaft, kg; m_{cd} – mass of intermediate discs, kg; m_{var} – a mass of eccentric variation mechanisms, kg; m_{sup} – mass of support discs, kg; m_{axles} – mass of disc axles, kg; $d_{bearing}$ – bearing diameter, m; γ_{eem} – efficiency of the electric motor; ω_{rotor} – rotor angular velocity, s^{-1}; ω_{disk} – discs angular velocity, s^{-1}; ku – torque transmission ratio; *e* – eccentricity of the shaft, m; *l* – length from the counterweight mass centre to the rotation axis of the rotor, m; r – length from disc to rotor, m; r_{disk} – disc radius, m; g – acceleration of gravity, m/s^2; α_x, α_z – dissipation coefficients relative to the axes OX and OZ; k_x^2, k_z^2 – frequency of system free oscillations relative to the axis OX and OZ*, Hz*; t – time, s; μ – coefficient of friction in bearings.

Taking into account the permissible errors of the measuring equipment, a critical value of the discrepancy between the experimental and theoretical research was taken by 15% *(Yanovych V.P., Kupchuk I.M., 2017)*. Exceeding this boundary indicates the unreliability of the mathematical model and the inability to use it when designing a crusher of this type. Processing and analysis of the research results were carried out in the Microsoft Excel software environment *(Kupchuk I.M., 2017)*.

The experimental research was carried out in the laboratories of the department of processes and equipment for the processing and production of food products in Vinnytsia National Agrarian University and the specialized laboratory "OVECHATSKY MTD" of the SE «Ukrspirt» *(Yanovych V.P., Kupchuk I.M., 2017)* using the experimental model of a vibrating disc crusher *(Palamarchuk I.P., Yanovych V.P., Kupchuk I.M., Solomko I.V., 2013)* (Fig. 3).

To manage and change rotation frequencies of the motor shaft, the AOSN-20-220-75 autotransformer was used (Figure 4) *(Kupchuk I.M., 2017; Yanovych V.P., Kupchuk I.M., 2017)*. It contains mobile current-collecting contact in the form of graphite roller and allows you to smoothly change voltage from zero to maximum. Also, the winding of the above mentioned autotransformer has several terminals, which can generate various characteristics of the current at the output.

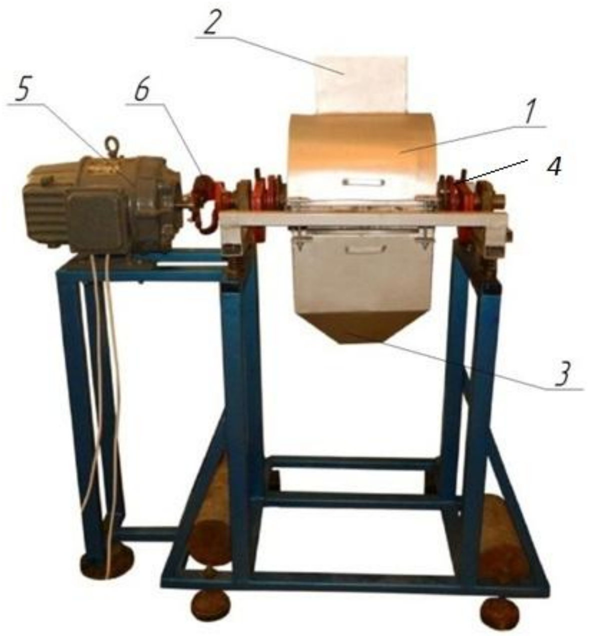

Fig. 3 - Vibrating disc crusher (experimental model):
1 – frame; 2, 3 – neck for loading and unloading; 4 – rotor; 5 – electric motor; 6 – coupling.

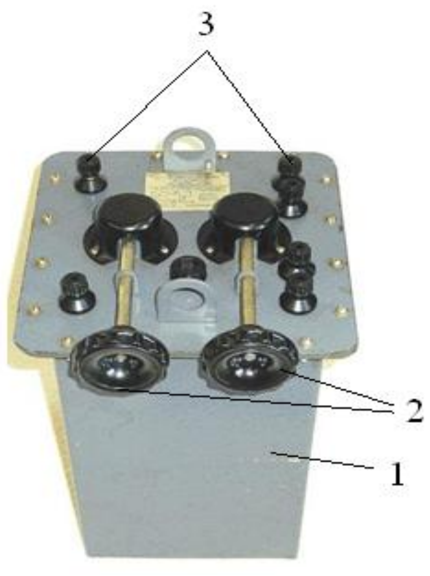

Fig. 4 - Laboratory autotransformer AOSN-20-220-75:
1 - external casing; 2 - voltage regulators; 3 -cells.

To record the angular velocity values of the drive shaft, the UNI-T UT372 wireless tachometer was used (Fig. 5). The operating principle and operating rules for the tachometer are described in the technical documentation *(Kupchuk I.M., 2017; Yanovych V.P., Kupchuk I.M., 2017)*.

Fig. 5 - Tachometer UNI-TUT372:
1 - laser indication; 2 - digital indicator; 3 - control panel

Fig.6 - Electronic wattmeter EMF-1
1 - wattmeter housing; 2 - control panel; 3 - display

To determine the energy consumption to drive the crusher, the EMF-1 electronic wattmeter was used (Figure 6).

RESULTS

Given the technological condition of material destruction by crusher discs (linear velocity of the edge of the impact disc) *(Yanovych V.P., Honcharuk T.V., Honcharuk I.V., Kovalova K.V., 2018; Sergeev S.N., 2008)*, the experiment was carried out at an angular velocity of the rotor ω = 100...150 s^{-1}.

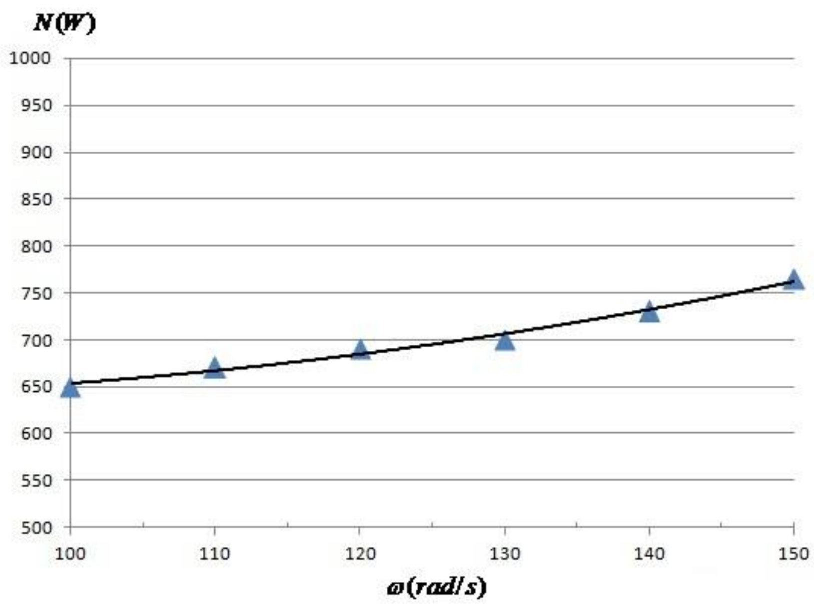

Fig. 7 - Energy consumption by crusher drive (experimental research)

As it can be seen from the results of experimental studies of electric motor consumed power (Figure 7), with increasing rotor angular velocity ω=100...150 rad/s, power consumption N increases almost proportionally N= 650...765 W. To compare the results of experimental and theoretical research (Table 1), we show them in the form of graphical dependencies (Fig. 8).

Table 1

Energy consumption by crusher drive (experimental and theoretical research results)

ω [s^{-1}]	N_T [W]	N_E [W]	Divergence	
			+/-	%
100	635	650	15	2,31
110	685	670	-15	-2,24
120	710	690	-20	-2,90
130	780	700	-80	-11,43
140	800	730	-70	-9,59
150	840	765	-75	-9,80

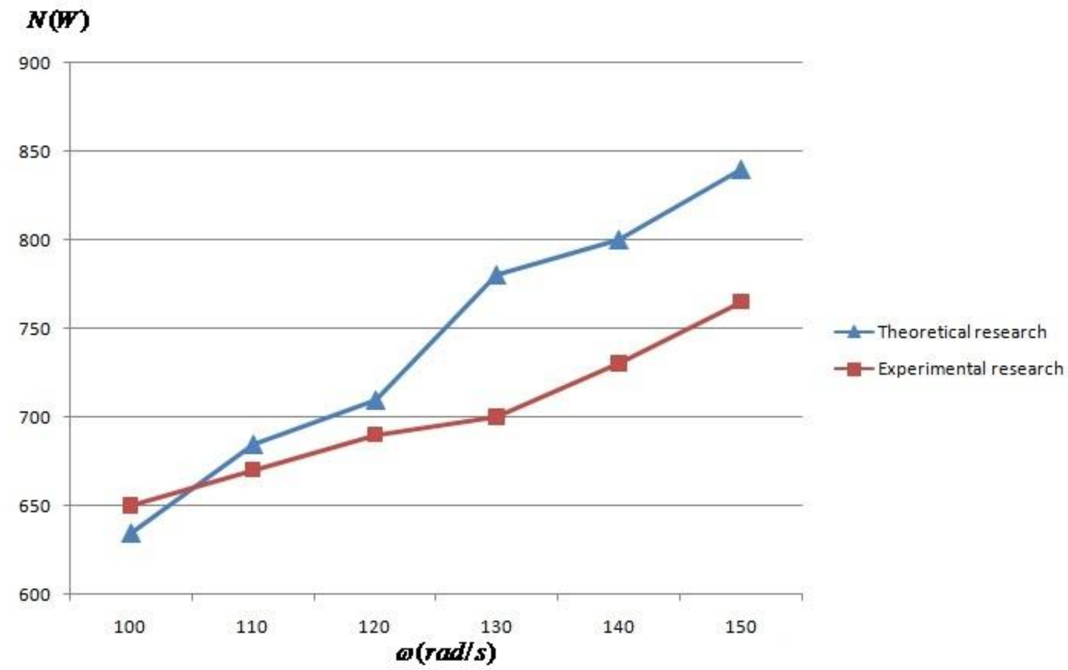

Fig. 8 - Comparison of theoretical and experimental research results

CONCLUSIONS

Comparing the theoretical and experimental research revealed a discrepancy, which is 2.24...11.43 % for the operational regime. Since this discrepancy is lower than the critical one (15%), the mathematical model (1) can be considered verified and it can be used in the construction of vibrating disc crushers of this type.

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