Annals of Warsaw University of Life Sciences – SGGW
Agriculture No 66 (Agricultural and Forest Engineering) 2015: 37–47
(Ann. Warsaw Univ. of Life Sci. – SGGW, Agricult. 66, 2015)

Impact of working parameters on loading of stone rake rotor

JAROSŁAW CHLEBOWSKI[1], ALEKSANDER LISOWSKI[1], ADAM STRUŻYK[1], JACEK KLONOWSKI[1], MICHAŁ SYPUŁA[1], TOMASZ NOWAKOWSKI[1], KRZYSZTOF KOSTYRA[1], JAN KAMIŃSKI[1], ADAM ŚWIĘTOCHOWSKI[1], OLE GREEN[2], DANIEL LAURYN[1], JAROSŁAW MARGIELSKI[1]
[1] Department of Agricultural and Forest Engineering, Warsaw University of Life Sciences – SGGW
[2] Kongskilde Industries A/S, Skælskørvej, Denmark

Abstract: *Impact of working parameters on loading of stone rake rotor*. The objective of this study was to determine the impact of the kinematic coefficient and the angle of positioning of the stone rake rotor upon its torque. Research was conducted in a soil bin. In order to achieve the objective, a stone rake model was developed and fixed to the frame of the soil bin tool trolley, using supports. The external diameter of the rotor was 400 mm. The study was performed at the rotational speed of 80 rpm, at two speeds of movement of the picking unit and five angles of positioning of the rotor in relation to perpendicular direction to tool movement within the range of 15 to 35°. The impact of the technical parameters examined on torque driving the stone rake rotor was found to be statistically significant. It was determined that the angle of positioning of the rotor shaft equal to 35° was beneficial due to the energy loads generated by the rotating unit. However, taking into account the working width of the stone rake, for practical purposes, the rotor positioning angle can be recommended to range between 25 and 30°.

Key words: rock, stone rake rotor, torque

INTRODUCTION

Stones found on fields in the topsoil layer hinder mechanization of the plant production process. They reduce the quality of agrotechnical operations, as well as quality of agricultural produce, in particular, root vegetables and crops, including potatoes. Stones are the main factor causing mechanical damages in the tubers and dark spots caused by impact, which may, in fact, completely eliminate the raw material to be used for production of fries and potato chips. These damages are associated with the soil conditions. The stone mass should not exceed 20% in the tuber mass [Gruczek 2002, Czerko et al. 2014]. On the basis of the available literature, it can be stated that the quantity of stones found in the topsoil layer (30–35 cm) on a field of the area of 1 ha in the soils found within the territory of Poland is diversified, ranging from 5 to 500 $t \cdot ha^{-1}$ [Mitrus 1979]. While the stone content in the topsoil layer ranges between 200 and 500 $t \cdot ha^{-1}$, Gruczek [2000] and Czerko et al. [2014] point to the need for stone removal from soils designated for cultivation of ware potatoes and plants to be used for food processing purposes.

Stones lying on the field surface or buried at the depth of 3–5 cm hinder many agrotechnical tasks, including surface cultivation, sowing, caring for spaces between rows, harvesting of cereals, greens and roots, as well as post-harvest processing of crops, harvested with stones. According to research, these constitute only 8–12% of all stones found in the topsoil layer. Stones at the depth of 13–15 cm hinder the tasks, listed above, as well as planting of potatoes and harvesting of root plants [Mitrus 1980].

Rock removal is performed using separate machines as stone rakes, arranging stones in rows, and stone pickers that pick stones from the rows. There are machines, which pick the rocks and transport them to containers in a single pass. These can also be used for construction purposes and in recreational areas to pick rocks, rubbish, roots, etc. They can be used in grasslands, golf courses, sports facilities and during preparatory works in road construction.

One of the most significant aspects of rock picking is raking. In most solutions, the stone rake units are rotors with tines, which turn in the direction opposite to movement of the machine along the field. In the available literature, there is no information on energy loads of such units. Therefore, research in this regard seems reasonable. In particular, laboratory experiments that limit the impact of various random events on the research results are recommended [Powałka and Buliński 2014a, b]. Research conducted under such conditions allows for appropriate preparation of the experiment and correct reasoning [Piotrowska and Klonowski 1996, Sahu and Raheman 2006, Javadi et al. 2012, Mandal et al. 2014, Nowakowski et al. 2015]. On the basis of examination of machines with active working components, it can be said that energy loads will depend on many factors that can be grouped as associated with the soil conditions, working parameters of the working units and configuration of working components of machines [Majewski et al. 1982, Marenya 2009, Libin et al. 2010]. Tupkarii et al. [2014]. The main technical parameters of units with rotating working components that exert impact on movement resistance of teh working units include: the direction of rotation of rotors or drums, forward speed of the machine, the kinematic coefficient or the ratio of peripheral speed of the working components to forward speed of the machine, working depth [Gach et al. 1989, Marenya et al. Musonda 2003, Salokhe and Ramalingam 2003]. In the available literature on the subject, very little has been said on stone rake rotors with regard to the quality and energy-related aspects of their work.

The objective of this work was to determine the impact of the kinematic coefficient and the stone rake rotor positioning angle on its torque.

MATERIALS AND METHODS

Tests of torque on the stone rake rotor shaft were conducted in a soil bin. A soil bin of dimensions of 10 × 2 m (length × width) allows for performance of tests using full-size working components of cultivation tools and machines [Buliński and Sergiel 2014]. A track is fixed to the frame of the soil bin, along which the tool trolley moves. The units to be examined were fixed to the tool trolley, moving on a set of rollers gripping the rod guide tubes located along the two sides of the soil bin. The trolley was powered by a double rope system through a gear rack from a 22 kW electric motor, controlled by a V2500 power inverter. The trolley drive allowed for measurement passes in both directions at the speed within the range of 0.1–7.2 km·h^{-1} [Buliński et al. 2010].

In order to achieve the defined objective, a frame with a turntable was built, as well as a model of stone rake rotor (Fig. 1) at Kongskilde Polska Sp. z o.o. Fixing of the rotor frame to the turntable

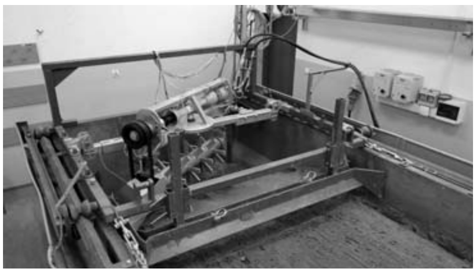

FIGURE 1. The working unit with the shaft of the stone rake rotor fixed to the tool trolley of the soil bin

allowed for a gradual (every 5°) change of the angle of positioning of the rotor in relation to the line perpendicular to the direction of movement within the range of 15–35° (Fig. 2). The external diameter of the rotor amounted to 400 mm. 32 tines, with a rectangular cross-section, sized 20 × 40 mm, were welded radially into a pipe of diameter of 160 mm. The spacing between the tines was 30 mm, and the shifting angle was 45°. The frame of the rotor unit was fixed to a tool trolley of the soil bin using supports.

FIGURE 2. The soil bin with arranged stones of diameter of 150 mm

The stone rake shaft was driven by a hydraulic motor through an articulated telescopic shaft and a V-belt with the transmission ratio of 1 : 1. The hydraulic motor was connected with a hydraulic unit AG 5089. The rotational speed of the hydraulic motor was set by reduction of hydraulic oil flow rate.

On the drive shaft, torque sensor MTR 500 was mounted with the nominal measurement range of 500 Nm with an in-built rotational speed sensor. The torquemeter with a rotational speed indicator was calibrated in accordance with the procedure described by Lisowski et al. [2009]. The sensors cooperated with meter MW2006-4. Measurement signals were recorded in the computer using a digital measurement and recording system DMCplus.

For monitoring of movement speed of the working unit with the tool trolley, a measurement set was used, equipped with an optical sensor installed on the roller with indicators, rolling along the soil bin frame guide.

The working depth of the rotor tines was constant, amounting to 70 mm. It was set using the adjustable supports, connecting the rigid frame of the unit with the frame of the soil bin tool trolley. The soil was levelled using a sweeper attached to the tool trolley frame.

Before every measurement, the soil surface was levelled with the sweeper and stones were arranged in accordance with the plan (Fig. 2). Stones of the average substitutive diameter of 150 mm (27 pieces) were used. The stones were placed using a template making small holes, in which the stones were placed. The number and mass of stones distributed on the surface of the soil in the soil bin was equivalent to the average number of stones found within a land plot of the area of 30 t·ha^{-1}.

Tests were conducted for the established rotational speed of the stone rake rotor n = 80 rpm^{-1} and for two working speeds set v_1 = 0.7 and v_2 = 1.0 m·s^{-1}. Soil humidity was established using the dry oven test, in accordance with the requirements of PN-ISO 11465:1999, and it was maintained within the range of 8–10%. All measurements were made in three repetitions.

The kinematic coefficient [Hendrick and Gill 1971a, b, c, Gach et al. 1991, Libin et al. 2010] was established on the basis of the following formula:

$$\lambda = \frac{u}{v} = \frac{\pi \frac{dn}{60}}{v} = \frac{\pi dn}{60v}$$

where:
λ – kinematic coefficient;
u – peripheral speed of the rotor tine ends [m·s^{-1}];
v – movement speed [m·s^{-1}];
d – diameter of the stone rake rotor [m]; according to measurement d = 0.4 m;
n – rotational speed of the shaft [rpm].

Correctness of the assumptions made in relation to maintaining of the pre-defined rotational speed of the stone rake rotor and speed of movement of the unit was checked using variance analysis with the aid of statistical package Statistica v.12.

RESULTS AND DISCUSSION

On the basis of the two-factor analysis of variance conducted, it was found that the rotational speed of the stone rake rotor was maintained in the subsequent trials ($F_{v1=2, v2=4079}$ = 0.1, at p = 0.856) and for both working speeds set ($F_{v1=1, v2=4079}$ = = 1.0, at p = 0.287). On the basis of test results, it can be concluded that the rotational speed of the stone rake rotor during movement of the working unit along the soil bin at the speed of v_1 and v_2 did not differ significantly, amounting to 80.7 and 80.8 rpm (Table 1), respectively, and the rotational speed variability coefficient for the entire test range was only 0.2%.

TABLE 1. Average values ($n_{śr}$), standard deviation (SD) and 95% ranges for rotational speed of the stone rake rotor shaft (*n*) for two speed levels of movement of the unit v_1, v_2

Movement speed level (v)	Trial number	$n_{śr}$ [rpm]	SD n [rpm]	–95% n [rpm]	+95% n [rpm]	Number (N)
v_1	1	80.79	0.16	80.49	81.10	820
	2	80.70	0.17	80.37	81.02	703
	3	80.53	0.16	80.20	80.85	732
v_2	1	80.76	0.19	80.39	81.13	607
	2	80.86	0.19	80.49	81.23	609
	3	80.85	0.19	80.49	81.22	614

Source: Own results of the authors.

The statistical analysis results presented in Table 2 indicate that for two speed levels v_1 and v_2, their average values amounted to 0.68 and 0.96 m·s^{-1}, respectively. The variance analysis, conducted for two speeds ($F_{v1=2, v2=2252}$ = = 15, at p <0.0001 for v_1 = 0.68 m·s^{-1}, $F_{v1=2, v2=1827}$ = 11, at p <0.0001 for v_1= = 0.96 m·s^{-1}) indicated a certain impact on repetitions on maintenance of forward speed. However, the variance coefficient calculated for all of the cases examined, which did not exceed 0.03% (Table 2) allows for assumption that forward speed levels of the stone rake rotor for the same levels during trial repetitions were maintained.

Correlations between peripheral speed of the working components and working speed have been described by the kinematic coefficient. For speeds v_1 and v_2, the calculated values of the kinematic coefficient amounted to λ_1 = 2.5 and λ_2 = = 1.7, respectively.

In order to verify the significance of differentiation of the torque value for individual measurement units, a two-factor analysis of variance was conducted. Differentiation of the torque was statistically significant for changes in the kinematic coefficient and the angle of positioning of the rotor in relation to the line perpendicular to the direction of mo-

TABLE 2. Average values ($v_{śr}$), standard deviation (SD) and 95% ranges for the speed of movement of the stone rake unit (v) at the rotational speed of the stone rake rotor shaft of 80 rpm for two speed levels of the unit v_1, v_2

Movement speed level (v)	Trial number	$v_{śr}$ [m·s^{-1}]	SD v [m·s^{-1}]	–95% v [m·s^{-1}]	+95% v [m·s^{-1}]	Number (N)
v_1	1	0.6764	0.0002	0.6761	0.6767	820
	2	0.6777	0.0002	0.6773	0.6780	703
	3	0.6772	0.0002	0.6769	0.6776	732
v_2	1	0.9633	0.0001	0.9631	0.9636	607
	2	0.9637	0.0001	0.9634	0.9639	609
	3	0.9642	0.0001	0.9639	0.9644	614

Source: own results of the authors.

tion and for interactions between them, which is indicated by statistical values of F_{obl} and the critical significance level (Table 3).

coefficient 2.5 and the rotor angle of positioning of 15° have been presented in Figure 3. Such great variability of torque should be explained by the nature

TABLE 3. Analysis of variance of factors that exert impact on torque

Source of variability	Sum of squares	Number of degrees of freedom	Average square	Statistical value of F_{obl}	Critical significance level
Kinematic coefficient: λ	82 603	1	82 603	176.14	<0.0001
Angle: α	55 381	4	13 845	29.52	<0.0001
Interaction: λ to α	23 088	4	5 772	12.31	<0.0001

Source: Own results of the authors.

As a result of the statistical analysis conducted, average values, standard deviations and 95% ranges for torque at various forward speeds of the stone rake unit were obtained for various angles of positioning of the rotor (Table 4). On the basis of data from Table 4, the variability coefficient calculated reached the value of 1.6 to 2%. In the entire range of analysis, the torque values ranged from 10 to 160 Nm. The exemplary course of changes in the torque for kinematic of work of the stone rake rotor. When moved along the rotor, the stones were hit by the tines several times and randomly moved forward.

On the basis of the average torque values (Table 4), a logical correlation can be found between increase in energy loads along with increasing of the forward speed of the stone rake unit, resulting from the greater number of rocks being moved by the rotor over time. The highest average torque value (71 Nm)

TABLE 4. Average values, standard deviation (SD) and 95% ranges for torque (M_{zg}) for two levels of the kinematic coefficient λ_1, λ_2

Kinematic coefficient level	Rotor positioning angle α [°]	M_{zg} (average), [Nm]	SD M_{zg} [Nm]	–95% M_{zg} [Nm]	+95% M_{zg} [Nm]	Number (N)
λ_1	15	63.67	1.00	61.71	65.62	470
	20	52.95	1.12	50.75	55.15	373
	25	55.00	0.93	53.17	56.82	541
	30	54.15	1.02	52.15	56.16	449
	35	51.25	1.05	49.19	53.32	422
λ_2	15	67.10	0.98	65.18	69.03	486
	20	71.30	1.26	68.83	73.77	295
	25	63.16	1.11	60.98	65.34	380
	30	63.52	1.18	61.20	65.85	333
	35	57.65	1.18	55.34	59.97	336

Source: Own results of the authors.

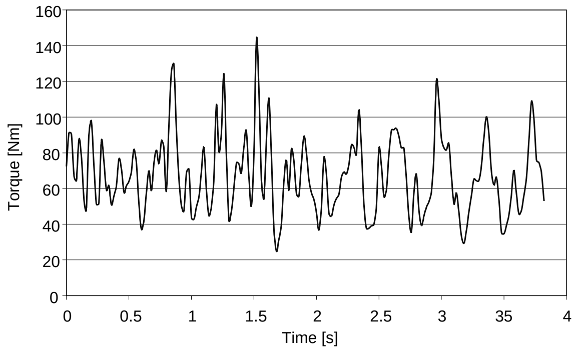

FIGURE 3. An exemplary course of changes in the torque of the stone rake rotor at the angle of 15° with kinematic coefficient λ_1 = 2.5

was achieved for kinematic coefficient λ_2 (greater forward speed of the unit) and the rotor placed at the angle of 20°, the smallest (51 Nm) for the kinematic coefficient λ_1 and the greatest angle of positioning of the stone rake rotor 35°. Smaller angles of positioning of the rotor result in slower movement of stones along the rotor and a bigger number of stones being hit by the tines, which increases the energy load. However, at these rotor settings, the stone rake working width can be increased. The test results presented indicate that the angle of positioning of the rotor, as well as its rotational and working speed exert significant impact on energy loads and working performance of the stone rake. Analyzing the impact of the kinematic coefficient on the torque on the stone rake rotor (Fig. 4), it can be concluded that its greater value (λ_1 = 2.5) resulting

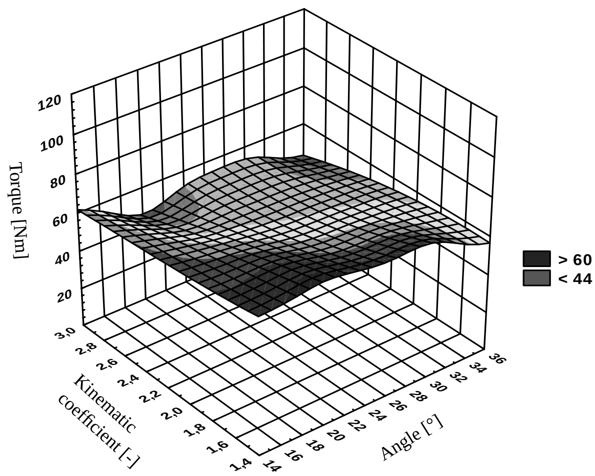

FIGURE 4. The impact of the kinematic coefficient and the angle of positioning of the rotor on the stone rake rotor torque

from lower speed of the stone rake unit ensured a torque below 44 Nm, at the rotor angles of 20 and 35°.

Greater loading of the stone rake rotor at smaller angles of positioning, in particular, at the angle of 15°, may be a result of a longer distance covered by the stones while moved and thus a greater number of stones moved over a specific time interval.

The lower value of the kinematic coefficient, obtained during the tests, is mainly due to higher speed of movement of the stone rake unit. The test results show that almost within the entire range of change of the angle of positioning of the stone rake rotor for the kinematic coefficient of 1.7, torque exceeded the value of 60 Nm, while a certain reduction of the load can be noticed after exceeding the angle of 30°. For higher values of the kinematic coefficient (above 2.5), a significant reduction in the torque of the stone rake rotor was observed. Tests of other machines with rotating components, such as rototillers [Beeny and Khoo 1970, Hendrick and Gill 1971c, Hendrick 1980, Waszkiewicz and Miszczak 1986, Tupkarii et al. 2014] indicate that the values of the kinematic coefficient to a different degree impact the demand for power to drive the working components of these machines and their energy consumption. According to these authors, reduction of the kinematic coefficient by increasing the forward speed results in increased power consumption while reducing power output. On the other hand, a lower value of the kinematic coefficient obtained through reduction of angular velocity of the rotor resulted in reduction of total and unit power. It should be noted that in the case of rototillers, changes in unit power are associated with the fact that the machine knives cut soil into pieces of varying volume [Gach et al. 1991]. Moreover, analysis of the impact of kinematic coefficient on energy load usually pertained to machines that cooperated with working components rotating in the direction consistent with the direction of movement of the machine, and not in the opposite direction, like in the case of stone rake.

CONCLUSIONS

1. At constant parameters of the soil and the rock content, as well as working depth of the stone rake rotor, a change in the kinematic coefficient within the range of 1.7–2.5 and the angle of positioning of the rotor perpendicular to movement of the stone rake unit within the interval of 15–30° exerted a statistically significant impact on the value of torque on the rotor shaft. Interaction between these factors also turned out to be statistically significant.
2. Thanks to a greater value of the kinematic coefficient, resulting mainly from reduction of speed of the stone rake unit, set at the angle of 20°, upon increase of kinematic coefficient from 1.7–2.5, a 25% reduction of the torque was achieved.

3. The angle of positioning of the stone rake rotor shaft, amounting to 35° in relation to direction perpendicular to motion of the tool is beneficial in the context of energy loads from the rotating rotor. However, taking into account the working width of the stone rake, for practical reasons, the recommended stone rake rotor angle can be within the range of 25–30°.
4. The research conducted indicates that for higher speeds of the stone rake unit, higher rotational speeds of the rotor should be applied. Nevertheless, it should be noted that increased rotor speed may result in greater demand for power, therefore further research should be conducted, using other ratios of rotor peripheral speed to speed of movement of the stone rake unit.

Acknowledgments

This work was supported financially by NCBiR and Kongskilde, for which we are grateful.

REFERENCES

BEENY J.N., KHOO D.C.P. 1970: Preliminary investigations into the performance of different shaped blades for rotary tillage of wet rice soil. Journal of Agricultural Engineering Research 15 (1): 27–33.

BULIŃSKI J., KLONOWSKI J., SERGIEL L. 2010: Wykorzystanie kanału glebowego do badań zespołów roboczych narzędzi i mechanizmów jezdnych. Inżynieria Rolnicza 1 (119): 93–98.

BULIŃSKI J., SERGIEL L. 2014: Effect of moisture content on soil density – compaction relation during soil compacting in the soil bin. Annals of Warsaw University of Life Sciences – SGGW, Agriculture (Agricultural and Forest Engineering) 64: 5–13.

CZERKO Z., GOLISZEWSKI W., JANKOWSKA J., LUTOMIRSKA B., NOWACKI W., TRAWCZYŃSKI C., ZARZYŃSKA K. 2014: Metodyka integrowanej produkcji ziemniaków. Państwowa Inspekcja Ochrony Roślin i Nasiennictwa. Główny Inspektorat, Warszawa. Retrieved from https:// piorin. gov.pl/publikacje/metodyki-ip (access 08.06.2015).

GACH S., KUCZEWSKI J., WASZKIEWICZ Cz. 1991: Maszyny rolnicze. Elementy teorii i obliczeń. Wydawnictwo SGGW, Warszawa.

GACH S., MISZCZAK M., WASZKIEWICZ Cz. 1989: Projektowanie maszyn rolniczych. Wydawnictwo SGGW-AR, Warszawa.

GRUCZEK T. 2000. Technologia produkcji ziemniaka jadalnego i dla przetwórstwa spożywczego w warunkach gleb zakamienionych. W: Ziemniak spożywczy i przemysłowy oraz jego przetwarzanie. Mat. Konf. Nauk. Polanica Zdrój, 8–11.05.2000. AR Wrocław: 89–90.

GRUCZEK T. 2002: Efektywność produkcji ziemniaka na glebach zakamienionych. Zesz. Probl. Post. Nauk Rol. 489: 137–146.

HENDRICK J.G. 1980: A rotary chisel. Transactions of the ASAE 24 (6): 1349–1352.

HENDRICK J.G., GILL W.R. 1971a: Rotary tiller design parameters: Part I-Direction of rotation. Transaction of the ASAE 14 (4): 669–674.

HENDRICK J.G., GILL W.R. 1971b: Rotary tiller design parameters: Part II – Depth of tillage. Transaction of the ASAE 14 (4): 675–678.

HENDRICK J.G., GILL W.R. 1971c: Rotary tiller design parameters: Part III – Ratio of peripheral and forward velocities. Transaction of the ASAE 14 (4): 679–683.

JAVADI A., SEYEDI E., MOHAMADIGOL R., SHAHIDZADEH M. 2012: Effect of a modified and common disc openers on soil failure and forces using for direct planting. Global Journal of Medicinal Plant Research 1 (1): 26–32.

LIBIN Z., JIANDONG J., YANBIAO L. 2010: Agricultural rotavator power requirement optimization using multi-objective probability parameter optimization. International Agricultural Engineering Journal 19 (3): 15–22.
LISOWSKI A. (Ed.) 2009: Efekty działania elementów wspomagających rozdrabnianie roślin kukurydzy a jakość kiszonki. Wydawnictwo SGGW, Warszawa.
MAJEWSKI Z., ROSZKOWSKI H., WASZKIEWICZ Cz. 1982: Wpływ parametrów pracy glebogryzarki na wielkość zapotrzebowania energetycznego. Maszyny i Ciągniki Rolnicze 3: 27–29.
MANDAL S., BHATTACHARYYA B., MUKHERJEE S., KARMAKAR S. 2014: Soil-blade interaction of a rotary tiller: soil bin evaluation. International Journal of Sustainable Agricultural Research, 1 (3): 58–69.
MARENYA M.O. 2009: Performance characteristics of a deep tilling rotavator. Doctor thesis. University of Pretoria.
MARENYA M.O., Du PLESSIS H.L.M., MUSONDA N.G. 2003: Theoretical force and power prediction models for rotary tillers – a review. Journal of Engineering in Agriculture and the Environment 3 (1): 1–10.
MITRUS J. 1979: Usuwanie kamieni z pól. Pr. PIMR 7, 43.
MITRUS J. 1980: Procesy przemieszczania kamieni w glebie przez elementy robocze maszyn zbierających. Doctor thesis. IBMER, Warszawa.
NOWAKOWSKI T., LISOWSKI A., STRUŻYK A., KLONOWSKI J., SYPUŁA M., CHLEBOWSKI J., KOSTYRA K., KAMIŃSKI J., ŚWIĘTOCHOWSKI A., GREEN O., LAURYN D., MARGIELSKI J. 2015: Impact of technical parameters on the horizontal resistance component when slicing soil with a duckfoot share. Annals of Warsaw University of Life Sciences – SGGW, Agriculture (Agricultural and Forest Engineering) 65: 5–13.
PIOTROWSKA E., KLONOWSKI J. 1996: Badania modelowe wąskich narzędzi do uprawy głębokiej. Przegląd Techniki Rolniczej i Leśnej 12: 9–12.
POWAŁKA M., BULIŃSKI J. 2014a: Changes in soil density under influence of tractor wheel pressures. Annals of Warsaw University of Life Sciences – SGGW, Agriculture (Agricultural and Forest Engineering) 63: 15–22.
POWAŁKA M., BULIŃSKI J. 2014b: Effect of compacting soil on changes in its strength. Annals of Warsaw University of Life Sciences – SGGW, Agriculture (Agricultural and Forest Engineering) 63: 5–14.
SALOKHE V.M., RAMALINGAM N. 2003: Effect of rotation direction of a rotary tiller on draft and power requirements in a Bangkok clay soil. Journal of Terramechanics 39: 195–205.
TUPKARI P.D., SHARMA P.K., SINGH A. 2014: Computer aided design and analysis of rotavator blade. International Journal of Advanced Technology in Engineering and Science 2 (5): 1–6.
WASZKIEWICZ Cz.. MISZCZAK M. 1986: Analiza wpływu parametrów roboczych na zapotrzebowanie mocy i jakość pracy glebogryzarki. Maszyny i Ciągniki Rolnicze 5–6: 30–32.

Streszczenie: *Wpływ parametrów pracy na obciążenia wirnika zgarniającego kamienie*. Badania przeprowadzono w kanale glebowym ze specjalnie wykonanym zespołem roboczym wyposażonym w wirnik zgarniający kamienie. Zespół roboczy zamocowano do ramy wózka narzędziowego kanału glebowego. Badania wykonano dla dwóch wartości współczynnika kinematycznego przy stałej prędkości obrotowej wirnika zgarniającego 80 obr.$\cdot$min^{-1} oraz dla pięciu kątów ustawienia wirnika do kierunku prostopadłego względem ruchu narzędzia w zakresie od 15 do 35°. Z całości przeprowadzonych badań wynika, że kąt ustawienia wału wirnika zgarniającego kamienie wynoszący 35° jest korzystny ze względu na obciążenia energetyczne pochodzące od obracającego się wirnika. Jednak biorąc pod uwagę szerokość roboczą zgarniacza, dla praktyki można zaproponować kąt ustawienia wirnika zgarniacza w zakresie od 25 do 30°.

MS received July 2015

Authors' address:
Jarosław Chlebowski, Aleksander Lisowski, Adam Strużyk, Jacek Klonowski, Michał Sypuła, Tomasz Nowakowski, Krzysztof Kostyra, Jan Kamiński, Adam Świętochowski, Daniel Lauryn, Jarosław Margielski
Wydział Inżynierii Produkcji SGGW
Katedra Maszyn Rolniczych i Leśnych
02-787 Warszawa, ul. Nowoursynowska 164
Poland
e-mail: jaroslaw_chlebowski@sggw.pl

Ole Green
Kongskilde Industries A/S
4180 Sorø, Skælskørvej 64
Denmark