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DYNAMICS OF TRACTOR AGGREGATED WITH CONTINUOUSLY OPERATING PLANTING HOLE DIGGER

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Digging holes for planting, which is a minimal tillage method, is beneficial in terms of reducing energy consumption and preventing erosion. Current popular hole diggers operate intermittently to give low productivity, and this problem can be solved by continuous hole digging machine with active tillage unit. On the basis of established principles, the kinematics of machine and its optimal parameters have been studied and determined. This paper presents the results of the study on dynamics of a tractor combined with a continuous excavator, as a basis for determining the combination's working parameters, such as the speed of the unit and the number of rows. The experimental data on digger's drag torque and the characteristics of Kubota B2530 tractor are the basis for simulating the working process at different speeds. The inequality of rotation speed and the moment of engine crankshaft is determined. Aggregates with single-row digger work stably in gears P1 to P6, while with double-row digger, stable operation is in gears P1 to P5. The productivity of a continuous operating digger is superior to that of an intermittent operating one. A continuous operating digger can reach 3500 pits-hour⁻¹ with a double-row or 2400 pits-hour⁻¹ with a single-row tractor aggregate.

Keywords: active tillage unit; tractor aggregate; dynamics; soil; continuous operation

Digging holes for planting is a minimal tillage method that can assist in reducing crop establishment costs through increasing timeliness of operations, potentially lowering fuel usage and increasing work rates. The use of minimal tillage can also help to control soil erosion by maintaining a higher proportion of crop residue on the soil surface than by conventional inversion tillage (Morris et al., 2010; Yuan et al., 2019).

To date, most of the widespread pit diggers are aggregated with intermittent excavators (Quynh, 2006; Thu, 2021). One tractor that aggregates with a digger has to stop moving before digging; therefore, its working productivity is negatively affected. In addition, existing continuous excavators create very long holes, for example PDI-1 (Kataoka et al., 1995; Phát et al., 1998), a Site Forming machine, can dig a pit with a length of up to 1.2 m.

With the advantages of increasing working productivity and eliminating the disadvantages of intermittent excavators, the principal model of continuously operating hole digging machine with an active soil tillage unit is introduced in Fig. 1 (Long et al., 2021). From the PTO through the cardan coupling, the movement of cutting blade is transmitted through conical gearbox [1] and two chain drives [2] and [3]. The rotation of the rotary body [7] is transmitted from the 5th wheel with grippers [6] through the chain drive [5].

Based on established principles, the kinematics of the machine has been studied (Beloev et al., 2022b), thereby, the optimal parameters of the machine can be determined, such as: the radius of rotary body R_q and the kinematic coefficient

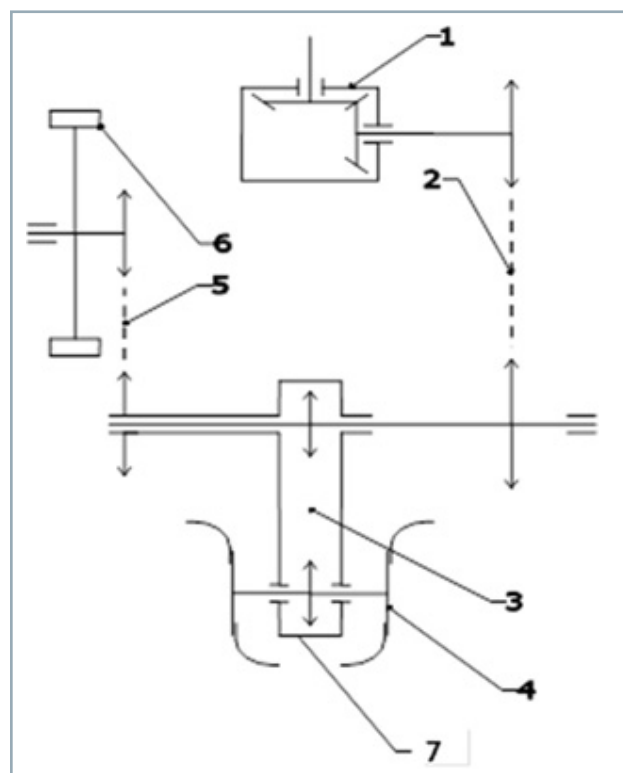


Fig. 1 Principal model of digger construction
1 – conical gearbox; 2 – chain gear 1; 3 – chain gear 2; 4 – cutter; 5 – chain gear for rotary body; 6 – fifth wheel; 7 – rotary body

of rotary body λ_q . With banana, planting hole should be $R_q = 350$ mm and $\lambda_q = 1.15$. Based on the afore-mentioned principle, the parameters such as the kinematic factor of tillager and the number of blades are determined to ensure the size and the distribution of soil chip cuts (Beloev et al., 2022a). With the number of knives $Z = 4$ and the average size of soil chip cuts smaller than 25 mm, the kinematic factor of the cutter is $\lambda_f = 9$.

To assemble the tractor aggregates, it is necessary to determine the resistance of the working machine and the interaction between the tractor and the working machine. The working process of the rotary tiller is a complicated process, including the phenomena of cutting, breaking, and throwing of the soil, and it depends on many factors such as the speed of the rotor, forward speed, as well as the design parameters of the machine.

The soil preparation process with active rotary tillers has been experimentally studied by Phat et al. (1998), and Usaborisut and Prasertkan (2019). In their scientific works, the influence of design and operational parameters such as the mounting angle of a cutting blade, diameter, the speed of rotation of a rotary cutter on the ability to disperse soil, and the specific energy consumption of hoeing is investigated.

Using the laws of classical mechanics, Ahmadi (2017) developed a method to obtain the power requirement of a rotary tiller. The CAD modelling method proposed by Tolvaly-Rosca and Pásztor (2019) was implemented in a study of working processes of tilling blades disposed on a single rotor and the operation of two working blades on a rotor of vertical axis rotary harrow. In this way, the relationship between the surface of soil cutting chip and fuel consumption is determined. By means of DEM (Discrete Element Method), Li et al. (2021) simulated the motion and impact of soil chips, and determined that the trajectory of soil chips is affected by the parameters such as rotor speed and shield angle, etc. Salokhe and Ramalingam (2001) investigated the effect of reverse rotation of a rotary tiller on seed bed quality, and the incorporation of crop residues and weeds. The new type of blades was designed for the reverse rotary tiller; they evaluated their performance at the forward speed of 1.0, 1.5 and 2.0 km·h⁻¹. The assembly of tractor aggregates to ensure quality requirements and energy consumption with a rotary cutter was studied by Bulgakov et al. (2019). For machine-tractor aggregates to work optimally, they need a balance between the power of the tractor and the working machine, as well as their interaction with each other. The problem was studied by Mollazade et al. (2009) and Tarverdyan et al. (2017).

However, the above results are only suitable for conventional rotary tillers where the depth of tillage is constant. Due to the non-stationary nature of the operation of continuously planting hole digger, it is necessary to study the dynamics of digger's machine aggregate connected with a tractor in order to find the optimal operation parameters of the aggregate.

The purpose of this paper is to develop a method to determine the working regime of the tractor aggregate with continuously operating planting hole digger, and the number of diggers which can be aggregated with one specific tractor.

To do this, it is necessary to build a dynamic model of machine-tractor aggregate and investigate its stability in order to improve its productivity. Innovativeness is in building the dynamic model of machine-tractor aggregate to deal with the problem of non-stationary nature of continuous planting hole digger operation to improve its productivity.

Material and methods

Dynamic of hole digging machine

When operating a tractor with an active working tiller, the consumed power of the tractor is divided into two components: one component for the drive of the tractor's running system, and the other for the drive of the active tiller. Based on the force balance of the rotational movement of the engine crankshaft and the translational motion of the machine-tractor aggregate (MTA), the following equation is established:

$$J_d \dot{\omega}_e = M_e - \left[\frac{F_k \cdot R_k}{i_k \cdot \eta_k} + \frac{M_f}{i_p \eta_p} \right] \quad (1)$$

$$F_k = G_k \cdot f + F_T + m_k \dot{v}$$

where: J_d – mass moment of inertia of rotating parts, reduced to the crankshaft of engine (kg·m²); ω_e – angular velocity of crankshaft (rad·s⁻¹); M_e , M_f – effective moment of engine and the resistance moment of cutter (Nm); f – coefficient of self-movement; G_k , m_k – coupling weight (N) and the mass (kg) of tractor; F_k , F_T – driving and pulling force (N); v – speed of tractor (m·s⁻¹); R_k – dynamic radius of driving wheels (m); i_k , i_p , η_k , η_p – ratios and efficiency coefficient of transmission from engine to tiller rotor and tractor driving wheel

Driving force can be calculated by tractor wheel slip δ according to the empirical law by traction factor $T = \frac{F_k}{G_z}$ (Vin, 1998), where G_z is the vertical reaction of driving wheels (N).

$$\delta = A \cdot \ln \frac{B}{T_{\max} - T} \quad F_k = G_z \left(T_{\max} - B \cdot e^{-\frac{\delta}{A}} \right) \quad (2)$$

Tractor wheel slip is defined by the following formula:

$$\delta = 1 - \frac{v \cdot i_k}{\omega_e \cdot R_k} \quad (3)$$

$$F_k = G_z \left(T_{\max} - B \cdot e^{-\frac{\omega_e \cdot R_k - v \cdot i_k}{A \cdot \omega_e \cdot R_k}} \right) \quad (4)$$

The effective torque of engine can be calculated from the external characteristics of engine through ω_e . The external characteristic of engine depends on the speed of crankshaft, which consists of two branches, adjustable branch and overload branch.

With adjustable branch, the effective torque of engine can be considered linearly dependent on the speed of crankshaft, and it can be calculated by the formula:

$$M_e = \frac{M_n(\omega_{\max} - \omega_e)}{\omega_{\max} - \omega_n} \quad (5)$$

In the non-adjustable overload branch, torque can be considered as a quadratic equation:

$$M_e = \frac{M_n - M_{\max}}{(\omega_n - \omega_m)^2} (\omega_e - \omega_m)^2 + M_{\max} \quad (6)$$

where: M_n , $M_{\max}$ – rated torque and maximum torque of engine (Nm); ω_m – crankshaft speed at maximum torque ($\text{rad}\cdot\text{s}^{-1}$); ω_n , $\omega_{\max}$ – nominal and maximum engine shaft idle speed ($\text{rad}\cdot\text{s}^{-1}$)

From the above equation, we obtain the following system of differential equations:

$$\begin{cases} \dot{\omega}_e = \frac{M_e - \frac{R_k \cdot F_k}{i_k \cdot \eta_k} - \frac{M_f}{i_p \cdot \eta_p}}{J_d} \\ \dot{v} = \frac{(F_k - m_k \cdot g \cdot f - F_T)}{m_k} \\ F_k = G_z \left(T_{\max} - B \cdot e^{\frac{\omega_e \cdot R_k - v \cdot i_k}{A \cdot \omega_e \cdot R_k}} \right) \\ M_e = \frac{M_n(\omega_{\max} - \omega_e)}{\omega_{\max} - \omega_n} \quad \text{if } \omega_e \geq \omega_n \\ M_e = \frac{M_n - M_{\max}}{(\omega_n - \omega_m)^2} (\omega_e - \omega_m)^2 + M_{\max} \quad \text{if } \omega_e < \omega_n \end{cases} \quad (7)$$

This is a system of differential equations with 2 unknowns v and ω_e , with quantities F_k and M_e defined through v and ω_e . With the power of Runge-Kutta method and Matlab program package, we can solve differential

Eq. (7) knowing the moment M_f and obtain the value of the crankshaft speed ω_e , the tractor speed v , as a function of time. From these results, one can determine temporary speed, motor torque, motor torque reserve factor, and motion irregularity factor.

The inequality of rotation speed and moment of engine can be determined by a coefficient:

$$k_n = 2 \frac{n_{e\max} - n_{e\min}}{(n_{e\max} + M_{e\min})} \quad (8)$$

$$k_M = 2 \frac{M_{e\max} - M_{e\min}}{M_{e\max} + M_{e\min}} \quad (9)$$

where: $n_{e\max}$, $n_{e\min}$ – maximum value and minimum value of n_e during working (min^{-1}); $M_{e\max}$, $M_{e\min}$ – maximum value and minimum value of M_e during working (Nm)

The torque reserve of engine is calculated using the coefficient μ according to the formula:

$$\mu = \frac{M_{e\max} - M_{\max}}{M_{e\max}} \quad (10)$$

A digger can be installed with a different number of working sections, i.e., single-row or multi-row digger. The productivity of digging aggregates is calculated by the formula:

$$z_{ho} = \frac{\varphi_q}{2 \cdot \pi \cdot t} \cdot z_f \quad (11)$$

where: z_f – number of rows of digger; t – time of operation of digger, it is also time of integration of Eq. (7)

Angle φ_q is determined from the speed of unit according to the differential equation:

$$\dot{\varphi}_q = \omega_q = \frac{v}{r_q} \cdot \lambda_q \quad (12)$$



Fig. 2 The prototype of the continuous-action digger

Experimental determination of the resistance of continuous digger

The resistance moment of digger in tractor aggregate is measured by the torque of PTO using the tensometric method. The prototype of continuous-action digger and measuring equipment are shown in Fig. 2.

The soil for the study is characterized as follows:

- type of soil: clay;
- soil moisture: 42%;
- pressure resistance: $83 \text{ N}\cdot\text{cm}^{-2}$.

The measurement result is presented in Fig. 3.

Using a regression method, the resistance torque of cutter is approximated by the formula:

$$M_f = (0.647 v^2 - 0.170 \cdot v + 0.845) \cdot (1368.8 \cdot \varphi_q - 1443.11 \cdot \varphi_q^2 + 563.08 \varphi_q^3 - 79.4 \cdot \varphi_q^4) \\ M_f = 0 \quad \text{if} \quad M_f < 0; \varphi_q < \pi/2 \quad (13)$$

where: φ_q – rotation angle of rotary body in rad; $\varphi_q = \frac{\pi \cdot \phi_q}{180}$

Study on the working process of tractor aggregated with digger

In this part, we study the assembly of a digger aggregated with a tractor, in order to find a suitable operation parameter and the number of diggers which can be aggregated with one particular tractor at the same time. For this purpose, a common brand of small tractors in Vietnam, that is Kubota B2530, is chosen. The tractor and engine parameters can be found in catalogue (NTNN, 2021). The tractor transmission has 9 gears, the speeds of which are shown in Table 1.

Work machine load is modelled for two cases:

- for the single-row digger, which has only one working body;
- for the double-row digger, which has 2 parallel working bodies.

For the single-row digger, the resistance moment of the cutter is periodically changed with the rotation angle period of the rotary body $\varphi_q = 2 \cdot p$. For the double-row digger, two working bodies do not work simultaneously but alternately, so load period is $\varphi_q = p$. In general, load period is $2 \cdot \pi/z_f$, where z_f is the number of rows.

With the above formulas, the moment of resistance of the cutter is calculated by Eq. (13) with an argument φ_{q1} , which is calculated by the formula:

$$M_f(t) = M_f(\varphi_{q1}) \quad \varphi_{q1} = \varphi_q - \text{int} \left(\frac{\varphi_q \cdot z_f}{2 \cdot \pi} \right) \cdot \frac{2 \cdot \pi}{z_f} \quad (14)$$

where: $\text{int} \left(\frac{\varphi_q \cdot z_f}{2 \cdot \pi} \right)$ is an integer part of $\left(\frac{\varphi_q \cdot z_f}{2 \cdot \pi} \right)$

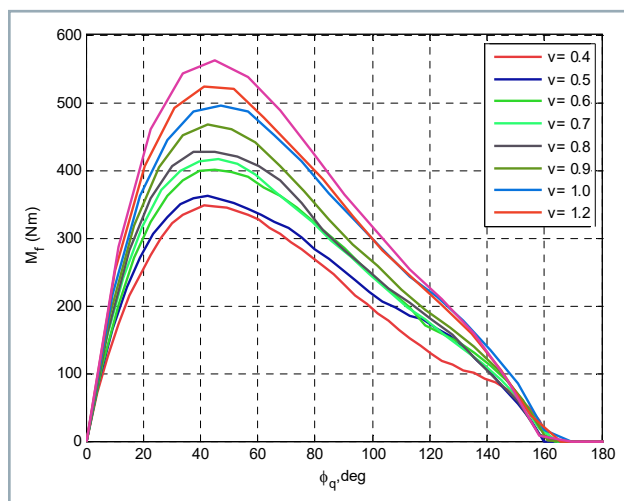


Fig. 3 The change of cutter shaft's torque by the angle of rotation of rotary body ϕ_q (deg) at aggregate's different speed

Results and discussion

The modelling of the digging process is carried out for two cases of single-row aggregated and double-row aggregated digger, for 15 seconds and at different gear. Figure 4 shows the changes in the revolution of tractor's engine crankshaft at different gears, when it is working with a single-row digger. From the nature of the curves, it can be seen that when nominal feed rate is increased, the stabilization time of the process is increased. For gears P1 to P6, stabilization time is less than 1 second, and for gear P6, stabilization time is about 2 seconds. For gears P7 onward, the operation process is unstable.

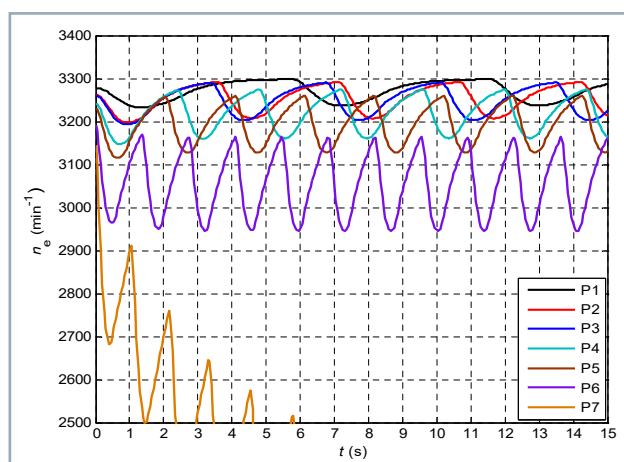


Fig. 4 Variation of the speed of tractor's crankshaft with the single-row digger over time in different gears

Table 1 Nominal speed in different gears of the tractor Kubota B2530

Gear	P1	P2	P3	P4	P5	P6	P7	P8	P9
$V (\text{km}\cdot\text{h}^{-1})$	1.1	1.8	1.9	2.7	3.2	5.0	7.1	11.7	18.0
i_k	771.1	471.2	446.4	314.2	265.1	169.6	119.5	72.5	47.1

At low gears, the change of the engine from the loaded mode and idling mode to the engine idle mode is clearly visible, until at high gears (e.g., P6). Although the MTA idles for over half of rotary body's rotation period, crankshaft revolution at gear P6 is only changed from 2990 rpm to 3100 rpm. That is close to rated engine speed, and it is due to high inertia of the engine and transmission.

The same pattern is seen for the aggregate with a double-row digger (Fig. 5), as for a single-row digger. However, with the aggregate with the double-row digger, the unevenness of crankshaft rotation is clearer due to a more uniform load on the working machine. With the double-row digger, the time of idling is insignificant.

The aggregate with the double-row digger works steadily at gears P1 to P5. At higher gears, the operation of the unit is unstable, and the speed of the crankshaft is lower than that at the maximum moment of the engine.

Coefficient k_n calculated by Eq. (8) is used to evaluate the uniformity of engine's speed. In Fig. 6, the coefficient of unevenness of crankshaft's speed of engine for the double-row unit is smaller than that for the single-row unit. The reason is that with the double-row digger, the resistance moment of the working machine alternates in one revolution of rotary body's rotation.

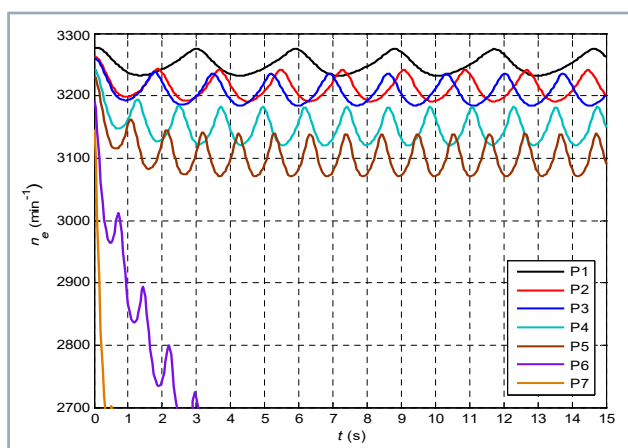


Fig. 5 Variation of the speed of crankshaft with the double-row digger over time in different gears

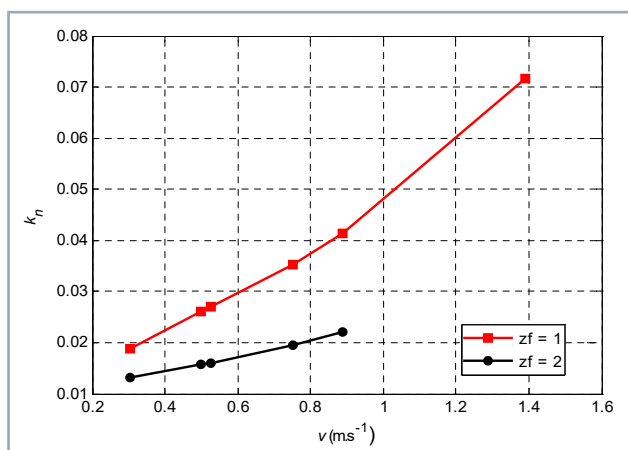


Fig. 6 Coefficient of inequality of crankshaft speed at different gears (speed)

It can be also seen that at higher gears, the coefficient of unevenness of crankshaft's speed of engine increases. The reason for this is at a higher gear, the resistance torque of the cutter increases and the engine speed decreases. In general, the unevenness coefficient of engine crankshaft speed is very small and not over 8%.

Figures 7 and 8 show the changes in the effective moment of engine's crankshaft of one aggregate with both the single-row digger and double-row digger at one time and at different nominal velocity.

The figures also show that at velocity P1 to P6, the work is stable with the single-row aggregate, and at velocity P1 to P5, the work is stable with the double-row aggregate. At higher speed, engine torque exceeds even the maximum engine torque and leads to engine overload.

From Fig. 9, it can be seen that the effective torque inequality of the engine crankshaft in the single-row ($z_f = 1$) aggregate is higher than that in the double-row ($z_f = 2$) aggregate. The reason is the same with the case of engine's angular speed, that loading on working machine is inequal.

In addition, at a high forward speed of aggregate, the unevenness coefficient of crankshaft's effective moment is reduced. The reason is that, at higher speed, the frequency

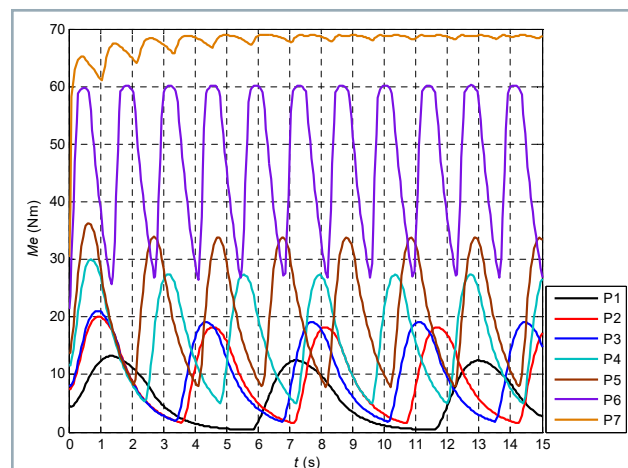


Fig. 7 Variation of crankshaft torque of the single-row digger tractor over time in different gears

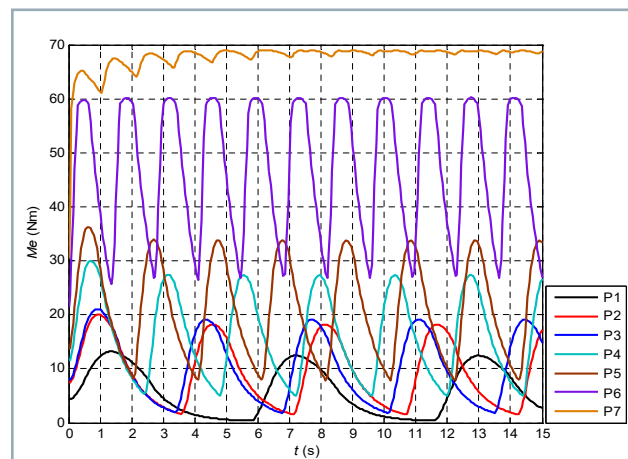


Fig. 8 Variation of crankshaft torque of the double-row digger tractor over time in different gears

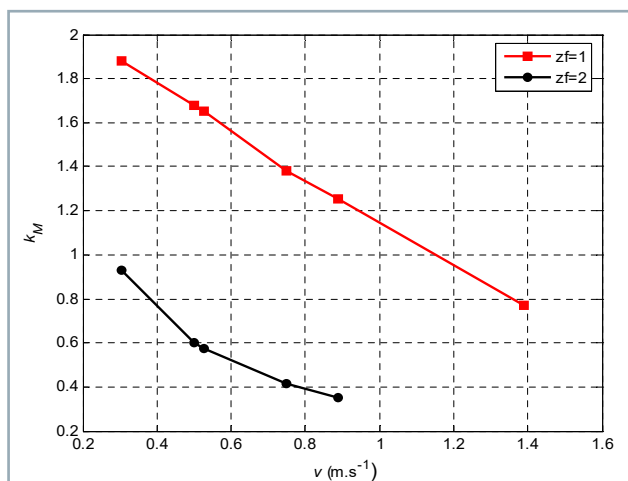


Fig. 9 Unevenness coefficient of the effective moment of engine's crankshaft with the single-row ($z_f = 1$) digger and double-row ($z_f = 2$) digger

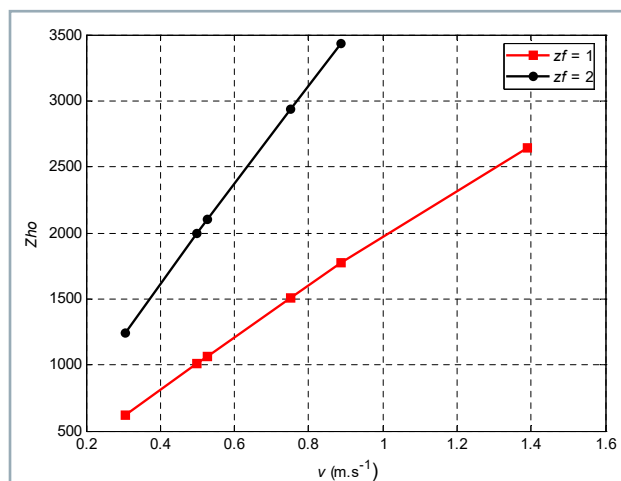


Fig. 11 Theoretical productivity of the single-row ($z_f = 1$) digger and double-row ($z_f = 2$) digger at different gears (speed)

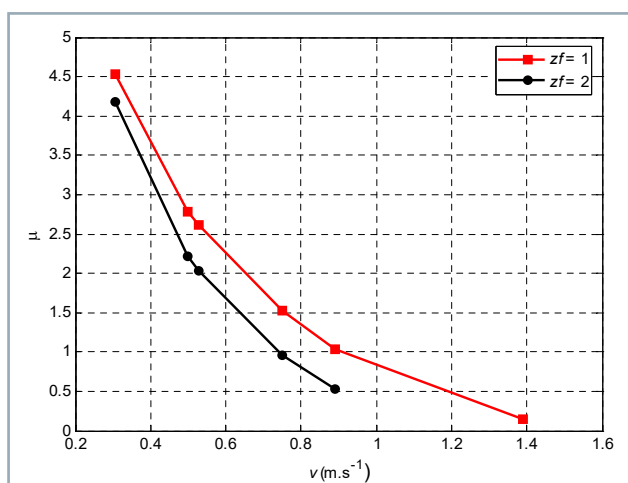


Fig. 10 Reserve coefficient of crankshaft torque at different speed

of loading change on the working machine increases, which leads to an increase in inertial forces applied to crankshaft.

Figure 10 presents the reserve coefficient of crankshaft torque at different speed. These data are important for choosing the gears with which the unit can work in specific conditions of soil.

Working on different resistances of soil, one unit requires different values of the reserve coefficient of torque. For example, on mountain condition, higher value of the reserve coefficient of torque should be chosen, and on delta condition, a lower coefficient value should be applied. For P6 speed, crankshaft torque reserve factor is 14%, while at higher speed, the value of coefficient is almost zero or less than zero.

The most important indicator of a unit is its productivity. Figure 11 shows the productivity of the single-row ($z_f = 1$) digger and double-row ($z_f = 2$) digger at different speed.

The figure shows the advantages of continuous operation diggers. Ordinary diggers with intermittently working drills have much lower productivity than that of continuous operation diggers. For example, a Yanmar F535

tractor mounted with a screw digger (39 kW) working on soil hardness of 20–25 kg·cm⁻² has a productivity of about 200 pits·hour⁻¹ (Thu, 2021). The productivity of the studied pit diggers is many times higher (up to 3500 pits·hour⁻¹).

It is likely that the double-row digger is preferred because of its higher productivity. The single-row digger can also be selected in some specific conditions when the working condition, such as small field areas or uneven terrains, etc., is inappropriate for the double-row digger.

The above are only results for the particular tractor (Kubota B2530). However, with the built dynamics model (Eq. 7), it is possible to investigate other tractors when the characteristics of the engine are known, thereby to evaluate and compare the effectiveness of the MTA with other available tractors, as the basis for the optimal selection. In addition, the above model can investigate the dynamics of active working machines with unstable loads.

Conclusion

1. Load on the digger is uneven. With Eq. (7), the dynamics of the MTA is studied, from which the value of crankshaft speed and tractor's speed are obtained, as a function of time. Furthermore, the momentary torque of the engine, the unevenness of speed and load, and engine reserve torque are determined.
2. Through experimental research, the resistance moment of digger's working body is obtained, which is influenced by the angle of rotary body's rotation and unit's forward speed. By using the Table Curve 3D program, the resistance moment of digger's working body can be approximated by Eq. 13.
3. By digital modelling performed using the Matlab program, the MTA work process based on widespread Kubota B2530 tractor is studied. The results are as following:
 - aggregates with the single-row digger work stably in gears P1 to P6. Aggregates with the double-row digger work stably in gears P1 to P5. At higher gears, the operation of the working process is unstable;

- the inequality of rotation speed and engine crankshaft moment in the single-row ($zf = 1$) aggregate is higher than that of the double-row ($zf = 2$) digger;
- as nominal forward speed (at higher gears) increases, engine angular velocity inequality coefficient increases, and effective moment unevenness coefficient decreases.

4. The productivity of a continuous-action machine is superior to that of an intermittent-action one. While the productivity of one intermittent digger is about 200 pits-hour⁻¹, the productivity of a double-row continuous digger can reach 3500 pits-hour⁻¹ or of a single-row MTA digger can reach 2400 pits-hour⁻¹.

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