Dynamic Indicators of a Corn Ear Threshing Process Influenced by the Threshing-Separation Unit Load

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1. Introduction

Combine harvesters thresh grains by passing the crops between a rotating cylinder and a stationary concave of a threshing unit [1]. The effectiveness of combine harvesters depends on their design, reliability, and operating conditions [2, 3]. The primary functional objective of harvesting equipment is to collect crops in the shortest time with the lowest grain loss and grain damage possible and at the lowest energy and operating costs [1, 4, 5]. Fuel consumption and exhaust gas emissions are crucial factors for evaluating the environmental impact of combine harvesters [6].

A combine harvester operates at maximum efficiency only if there is a substantially constant crop material feed rate [2, 7]. The operator then optimally varies the harvester's ground speed to maintain a constant crop material feed rate [4]. The operator can also control the efficiency of the crop threshing unit by changing the clearance between the rotating cylinder and the concave and the rotational speed of the cylinder [8, 9]. Operating parameters, such as threshing cylinder speed and concave clearance, are the most important factors influencing the power consumption of the threshing apparatus and the grain loss [1, 10]. Usually, the operator attempts to adjust the ground speed to maintain the feed rate at a level that will maximize harvester capacity without overloading the harvester components [4, 11, 12]. The real-time working speed and output power of a combine harvester primarily depend on the feeding rate [13], which makes this parameter a key operating variable in such machines. During the operation, the feed rate of the crop material being inserted into the threshing unit of a combine harvester is highly variable [14], which is influenced by operator selections (such as ground speed) and feed variations due to changes in the crop density and the presence of weeds [7]. This variable feed rate presents a problem for the harvesting machine settings [15] as well as for the component life of concaves and other threshing devices [2].

The appropriate functioning of many working units of a combine harvester also depends on a constant rotation speed. Therefore, the engine is always operated at rated speed, even if the machine is not entirely loaded [16]. This results in operation points with unfavorable specific fuel consumption [16]. The cylinder speed may increase or decrease owing to variations in the engine speed resulting from load changes. A change in cylinder speed of more than ~5% from the ideal speed would normally be considered to be unacceptable [17].

A change in the crop material feed rate affects the operation of the threshing, separating, and cleaning units. A change in the quantity of material passing through a combine harvester can substantially change the efficiency of the harvesting operation, even if the weather and crop conditions remain unchanged. A decrease in the feed rate of material passing through the threshing cylinder can result in a substantial increase in the amount of damaged grain, while an increase in the feed rate through the harvester can overload the cleaning system and cause an increase in grain loss [4].

Furthermore, if the feed rate variation occurs before the operator can react, such as in the case of slug feeding, the threshing section undergoes extreme stress [11]. This stress leads to component failure in the mechanical systems [18, 19]; therefore, the mechanical components are designed to bear the highest possible loads, even though many machines will not handle such loads under normal operation and some of them will never be placed under such high loads.

The need to incorporate automatic systems into combine harvesters increases in order to lower the work load of the operator [20]. Since the feed rate significantly influences machine loading, the sensor may be an important tool for increasing the harvest efficiency [21]. Various studies on feed rates of combine harvesters have been performed with the main goal of achieving an appropriate speed control of the combine harvester. By measuring the torque on the feeding auger or the threshing cylinder, it is possible to estimate the feed rate [21]. Missotten [22] tested a few sensors. Two of them were mounted at the cross auger and one measured the mat thickness in the feeder house of a combine harvester.

Also, strain gauges have been used to record the torque of the threshing cylinder in a combine harvester. Two double-shear strain gauges, which were attached at an angle of 180°, were placed between the belt pulley and the bearing. The measuring points were calibrated under static conditions [16]. While filling the threshing apparatus with wheat crops, large increases of the cylinder torque were observed. This can be attributed to irregular feedings of the auger and the feeder as well as to the interaction between the threshing concave and the rasp bars [16].

In most modern combines, a hydraulically controlled variable-speed drive with load-sensing capabilities drives the threshing cylinder. Veal et al. [23] reported the development of a cylinder drive pressure sensor as an alternative to the feeder house-based sensor. This sensor measures the tension on the feed conveyor drive chain, which is related to the flow of crop through the feeder housing [23]. In some combine harvesters, both feeder house sensor data and threshing cylinder data are simultaneously used.

To evaluate emissions and fuel consumption, Tomita et al. [6] developed a torque-measuring device for its use in a combine's engine output shaft. During trials, an increased torque was observed at the initial harvesting stage; this increased torque resulted from the acceleration of the combine and the increasing mass of processed crop inside it [6].

Yu et al. [13] developed a device that can determine the feeding rate required to increase harvesting reliability and efficiency. However, using modern sensor technology to determine the feeding rate and other basic working parameters of combine harvesters in real time has been a challenge. Yu et al. [13] described a wireless feeding rate testing method for combine harvesters that was based on torque measurements of the auger sprocket wheel. Wireless torsional strain gauges were designed and fixed in the refitting sprocket, which was used to detect the flow rate of collected corn in the form of torque, and the wireless data transportation was achieved using appropriate wireless launching and receiving modules. The experimental results showed that the maximum test error did not exceed 10% [13]. Wheat harvesting was investigated fuzzy logic controller (FLC). The FLC automatically adjusts the cylinder speed, concave-to-cylinder clearance, fan speed of the cleaning shoe and forward speed of the combine based on the measured losses at the straw walker and sieve sections [11].

Nowadays, combine harvesters are equipped with mechanical or hydraulic drive systems to distribute power. Electrical drives offer an alternative to the above mentioned drives, and a threshing cylinder with integrated electric direct drive was developed in a feasibility study at TU Dresden in cooperation with the University of Applied Sciences Dresden. The number of transmission elements, which can be a measure of the complexity in harvesting machines, can be reduced by 60% by using electrical drive technology. Additional advantages are a high efficiency and a good controllability [24].

In 2015, Tang et al. [10] investigated rice threshing using a multi-cylinder device, and although the rice-stem density was uniform throughout the entire threshing device, the rotation speed of the cylinder decreased during the process from 600 rpm to 10 rpm, fluctuating in a range of ±5 rpm. The authors stated that the threshing process was itself responsible for the decrease in rotation speed and its fluctuation. The torque values also fluctuated as well owing to instabilities in the threshing process. According to the results, the average torque value of the first threshing cylinder was 90 N m (with a fluctuation of ±70 N m) and that of the second cylinder 280 N m (with a fluctuation range of ±60 N m). The authors determined the changes in rotational speed and torque as a function of the threshing time, but their interdependencies, which are important for automatic control of the combine parameters, were not presented.

By means of analytical investigations, researchers have found that the evenness of rotation of threshing cylinders during operation is high if the moment of inertia of the cylinder, the angular velocity, and the diameter and flow of the crops fed into the threshing unit are high [25]. They suggest that increasing the torque of the internal combustion engine increases the unevenness of the cylinder rotation when an uneven crop flow is fed into the threshing unit.

Feeding the threshing unit with a variable (uneven) flow of cereal crops or corn ears causes a significant loss in the quality of operation of combine harvesters. The use of filler plates in the cylinder during corn ear threshing was found to have a substantial effect on the threshing efficiency [26, 27].

Since dynamic parameters such as the torque of resistance to rotation of the threshing cylinder, T_p, are suitable to establish a constant threshing separation unit load [25], it is appropriate to examine the influence of the corn ear feed rate on such parameters.

The objective of this paper is to establish the influence of the corn ear feed rate on the dynamic parameters of a threshing process (i.e., on the speed variations and torque of resistance to rotation of the threshing cylinder as well as the forces acting on the back of the concave) and find the relationship between those parameters.

2. Methodology and means of the experiments

The experiments were performed at the laboratory for investigation into technological processes of agricultural machinery of the Institute of Agricultural Engineering and Safety using a stationary tangential single-cylinder threshing unit [26] that comprised the following: a belt feeder (10 m in length and 1.2 m in width) for feeding the corn ear into the threshing unit; a tangential threshing cylinder of 1.2 m in width and 0.6 m in diameter with eight rasp bars attached to it, which was wrapped in the grate-bar type concave.

Corn ear threshing experiments were performed using a threshing cylinder containing filler plates made of stainless steel (width: 1.5 mm; weight: 2.54 ± 0.21 kg). These were attached to the cylinder by means of steel rivets. The threshing unit used in this study contained a standard cylinder with a cross-sectional separation between rasp bars of 95.94 cm^2 (this area was only 87.05 cm^2 after the filler plate was fitted, see Fig. 1a); this means that the area was reduced by 9.27%.

Fitting the threshing cylinder with filler plates whose operating plane (contact surface) formed a 36° angle to the cylinder radius led to a 21.12% decrease in cross-sectional separation, i.e., from 95.94 to 75.69 cm^2 (Fig. 1, b).
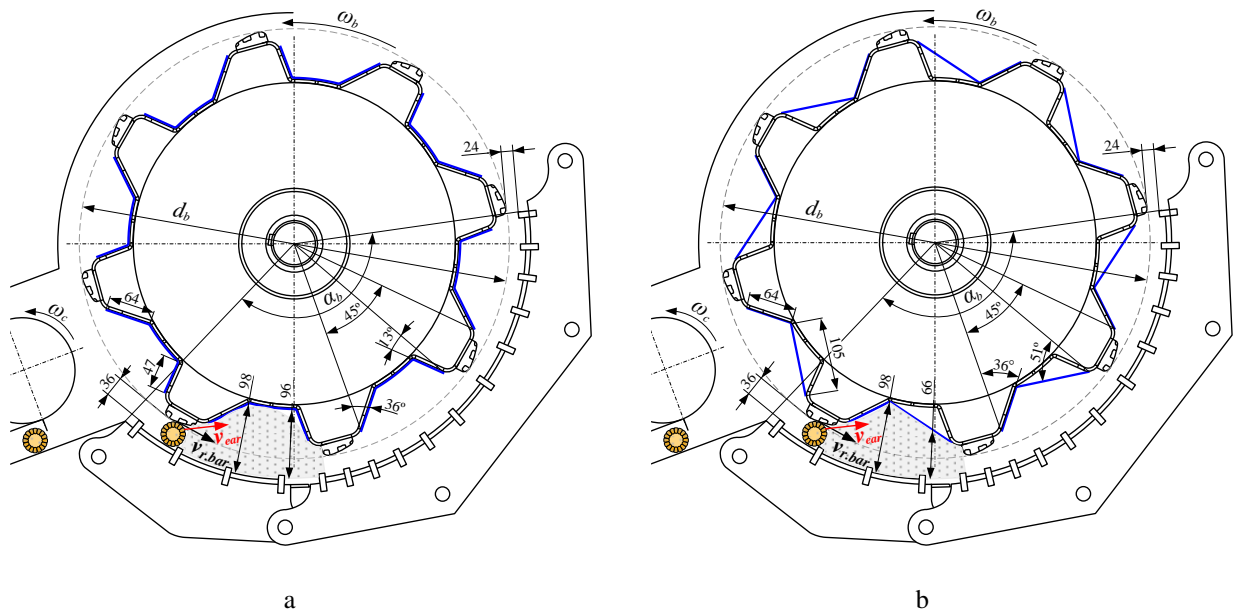
Fig. 1 Filler plates of a threshing cylinder: a) FP-I, b) FP-II [27] (ω_b: angular velocity of the cylinder; ω_c: angular velocity of the feeder conveyor; d_b: diameter of the cylinder; α_b: concave wrapping angle; $v_{r.bar}$: linear velocity of the rasp bars; and v_{ear}: speed of the corn ear)

A 15 kW electric motor was used to rotate the operating parts of the threshing unit. The cylinder speed was controlled by using a frequency transducer (*Delta VFD-C2000 SERIES*) and a belt drive.

The design and development of threshing devices are highly dependent on the agronomic characteristics of the crops [10]. Corn ears (*Rodni* species) were threshed in the threshing unit at their physiological maturity stage with a grain moisture content of $30.49 \pm 0.6\%$, a moisture content of the ear cobs of $54.85 \pm 1.31\%$, and a moisture content of the corn ear covering leaves of $27.51 \pm 2.41\%$.

After the corn ears on a digital weighing scale (CAS DB-1H, maximum load 60 ± 0.02 kg, minimum load 400 ± 20 g), they were evenly distributed on a 7-m long feeder belt and were fed into the tangential threshing unit at a speed (feed rate) of 1.0 m·s^{-1}. Corn ear threshing trials were performed at varied feed rates ranging from 4 to 12 kg s^{-1}.

The torque of the threshing resistance was measured using a force sensor (*Scaime ZFA* 2000 N), which was placed between the unit frame and the lever connected to the stator of the electric motor [27]. The rotor shaft rotates in two bearing units attached to the frame of the threshing unit using bolts. In this manner, depending on both the loading of the electric motor and that of the threshing cylinder, the stator of the electric motor through a fixed lever compressed a force sensor. The total error of the sensor and the amplifier was 0.03% of the maximum measurement value. The 12-bit ADC with resolution of 1/2LSb was used (0.488 N m).

The output signal of the sensor was amplified using a *Scaime CPJ* amplifier and subsequently transmitted to the modified converter board (K8055N). The force applied to the rear part of the concave was measured using two *Scaime ZFA* 5000 N tension-compression force sensors 17 [27] fitted on rods holding the rear part of the concave. These sensors were capable of measuring loads of up to 5000 N. The total error of the sensor and the amplifier was 0.03% of the maximum measurement value. The

12-bit ADC with above mentioned resolution was used (4.79 N).

The output signal of each sensor was strengthened using a *Scaime CPJ* amplifier and transmitted to the modified converter (*K8055N*), which turns the voltage into a 12-bit data stream. Then, the signal was transmitted (via USB) from the converter to the computer (20), where it was further processed using the *ThreshLab* software [27], which calculates the mean value of the force at pre-set time intervals (minimum recommended: 20 ms). For this purpose, while measuring at a 1 kHz frequency, a sample with a pre-established size (5 values) was collected for each sensor.

Optical encoders were attached to the shafts of the motor and the cylinder to measure the rotational speed of the electric motor and the threshing cylinder. The encoder pulses obtained per time interval during the rotation of the shafts were calculated using the digital counter of a modified *K8055N* converter. To improve the accuracy of the measurements, the time intervals were calculated by counting the pulses generated by a fixed frequency source using a second counter (resolution: 0.366 ms).

2.1. Finding the normal force exerted by the portion of corn flow on the rear part of the concave

The force was measured using two sensors fitted on rods holding the rear part of the concave. Since both sensors recorded force variations (F_{g1} and F_{g2}) synchronically, adding the averages of their readings allowed finding the total force F_g applied to the rear part of the concave.

Since each trial was repeated three times, $F_{g1\text{-}2}$ was obtained from six sensor readings.

For different corn ear feed rates, the force variations at the rear part of the concave (F_{g1} and F_{g2}) can be divided into five time periods (Fig. 2):

1 – Force under rotation of a non-loaded (empty) threshing cylinder ($F_g = 0$ N).

2 – Increase in force when starting to feed the threshing unit with corn ear (t_1–t_2).

3 – Force variation during the settled operating mode (t_2–t_3) under a constant feed rate.

4 – Decrease in force when the corn near is no longer fed to the threshing (t_3–t_4).

5 – Force under rotation of a non-loaded (empty) threshing cylinder ($F_g = 0$ N).

In periods 1 and 5, F_g is equal to zero as the concave is not loaded when the threshing unit is not fed with corn ears. In periods 2 and 4, the increase (for t_1–t_2) or decrease (for $t_3 - t_4$) in the force is estimated in the order or percentage of the average, F_{g1-2}, or peak (maximum) force, $F_{g1-2\ max}$. The above-mentioned time periods last 1 s. Threshing takes place in period 3 (t_2–t_3), and its duration is ~5 s. The value of F_{g1-2} during the threshing process, even under the well-established operating mode (t_2–t_3), is variable owing to the different physical–mechanical characteristics of the corn ear flow as well as other random factors.
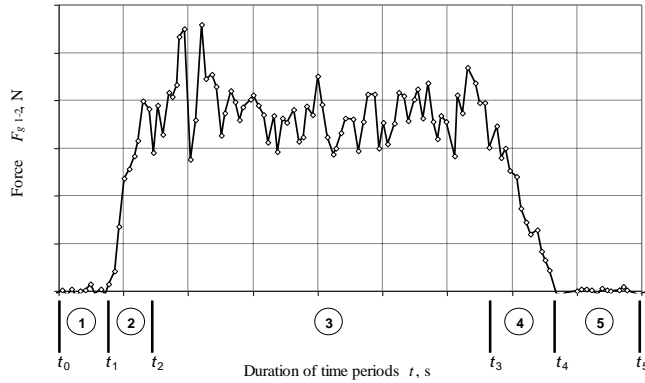

Fig. 2 Time periods of the variations in force applied to the rear part of the concave

Force variations can be considered using the coefficient of fluctuation and amplitude described in the Methodology section [27]. Force F_{g1-2} during process of operation is randomly achieved maximum and minimum numerical values that are different. To estimate the variation of force F_{g1-2} more adequately, it is worth to calculate the coefficient of fluctuation as follows:

$$\delta_{Fg1-2} = \frac{F_{g1-2\max 10} - F_{g1-2\min 10}}{F_{g1-2}}, \qquad (1)$$

where: $F_{g1-2\max10}$ – arithmetic average of maximum numerical values of force F_{g1-2} (10 maximum values selected) (in N); $F_{g1-2\min10}$ – arithmetic average of minimum numerical values of force F_{g1-2} (10 minimum values selected) (in N); F_{g1-2} – arithmetic average of all numerical values of force F_{g1-2} (in N).

Generalized maximum and minimum numerical values were used in calculations, i.e. arithmetic averages of 10 minimum and 10 maximum values. Having calculated averages of minimum and maximum numerical values of force F_{g1-2} allows for finding the average amplitude of force variation f

rom numerical values of arithmetic average of force:

$$A_{Fg1-2} = \frac{F_{g1-2\max 10} - F_{g1-2\min 10}}{2}. \qquad (2)$$

Variations in the resultant force of corn flow resistance (F_{g1-2}) can also be characterized by the period of variation, i.e., the time between two adjacent maximum numerical values of F_{g1-2}. Similar to frequency variations, this parameter is also difficult to determine because its value changes during the threshing process.

Once sufficient measurements have been made (in this particular investigation, 13.4 times per second or each 0.075 s), the force variations can be precisely determined by considering the dispersion of the measurement values through the standard deviation, σ_{Fg1-2}, or the confidence interval of the average (for example, by the selected level of probability 95%). The coefficient of variation, v_{Fg1-2}, can also be calculated.

2.2. Finding the total torque of the resistances during corn ear threshing

The motor of combine harvester power must be related to the crop flow and to the moment of inertia and diameter of the threshing cylinder.

The total torque of the resistances of a threshing cylinder is given as follows:

$$T_p = T_k + T_{pn} = F_p \cdot r_b + T_{pn}, \qquad (3)$$

where: T_k is the torque required to thresh the crops (in N m), F_p is the resultant crop flow resistance force (in N), r_b is the threshing cylinder radius (in m), and T_{pn} is the total torque of negative (detrimental) resistances due to friction in bearings and air flow within the threshing unit (in N m):

$$T_{pn} = T_{tr} + T_o, \qquad (4)$$

where: T_{tr} is the torque of friction forces (in N m) and T_o is the torque needed to overcome air resistance (N m).

The power required for rotation of an empty threshing cylinder is given as follows [17]:

$$N_{pn} = A\omega_b + B\omega_b^3, \qquad (5)$$

where: A is the coefficient accounting for the resistance of friction forces in the bearings ($A = 0.2$ N m per 100 kg of threshing cylinder weight, i.e., $A = 0.41$ N m, when $m = 204.2$ kg); B is the coefficient accounting for the air flow ($B = 0.00097$ N m s^2 per meter of threshing cylinder length l_b, i.e., $B = 0.001164$ N m s^2, when $l_b = 1.2$ m); and ω_b is the angular velocity of the threshing cylinder (in s^{-1}).

Consequently,

$$T_{pn} = A + B\omega_b^2. \qquad (6)$$

From the above information, a differential equation for the threshing cylinder is represented as follows:

$$I_\Sigma \frac{d\omega_b}{dt} = T_v \cdot i \cdot \eta - T_k - T_{pn}, \qquad (7)$$

where: I_Σ is the moment of inertia of the system containing the threshing cylinder (in kg m^2) and ω_b is the angular velocity of the threshing cylinder (in s^{-1}).

Conditions ensuring an even rotation of the threshing cylinder are practically impossible because the main portion of the total torque of resistances T_p is comprised of the torque of crop flow resistance ($F_p \cdot r_b$), which represents a value that varies in time and is determined by the uneven the crop flow rate.

The torque required for threshing the crops was measured indirectly. A force sensor (*Scaime ZFA* 2000 N), attached to the electric motor, was used to measure the reaction force of stator. The force, multiplied by lever length (the length of the lever attached to the stator of the electric motor does equal to 0.5 m), gives the torque of the resistance to rotation of the rotor of the electric motor, $T_{v.st}$. Keeping the above-mentioned values constant, the total torque of resistances applied to threshing cylinder, T_p, can be calculated as follows:

$$T_p = T_{v.st.} \frac{n_v}{n_b} \eta_d , \qquad (8)$$

where: n_v is the rotation frequency of the electric motor shaft (in min^{-1}), n_b is the rotation frequency of the threshing cylinder shaft (in min^{-1}), and η_d is the efficiency coefficient of the belt drive ($\eta_d = 0.98$).

Since the rotation frequencies of the electric motor shaft and the threshing cylinder shaft were measured simultaneously, expressing the ratio between these values in percentage gives the coefficient of belt slipping of the belt drive. The torque required for threshing the crops, T_k, can be calculated by subtracting a constant numerical value ($T_{pn} = 51.0 \pm 0.9$ N m) from the total torque of resistances, T_p. This T_{pn} value was determined from experimental trials and corresponds to the total torque of negative (detrimental) resistances, which are due to friction in the bearings and air flow within the threshing unit.

T_p was calculated in a similar manner to the forces applied to the rear part of the concave; the only difference in this case is that the arithmetic average was obtained from three readings of the sensors as each trial was repeated three times.

3. Results and discussion

3.1. Force acting on the rear part of the concave

Our results were obtained from two *Scaime ZFA* 5000 N force sensors fitted on rods at the rear part of the concave. The experiments were repeated three times and the results of all three measurements were included in the plots of the forces acting on the rear part of the concave. It is obvious that at a corn ear feed rate of 6 kg s^{-1} (with the spaces between the rasp bars covered with filler plates), there is no significant difference between the individual repetitions. This is true for corn ear flow threshing at a feed rate of 12 kg s^{-1} (Fig. 3).
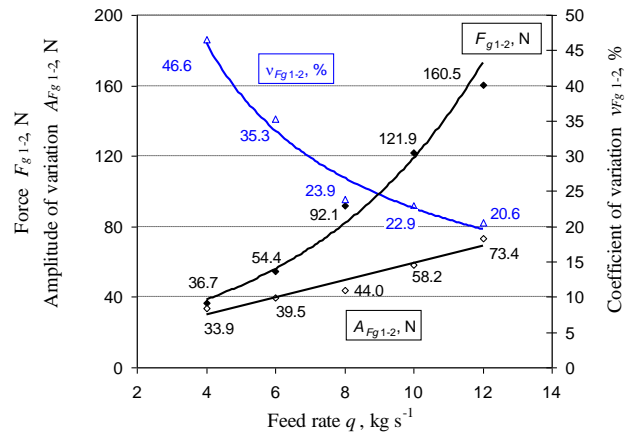

Fig. 3 Dependencies of the force acting on the rear part of the concave, $F_{g1\text{-}2}$, the amplitude of force, $A_{Fg1\text{-}2}$, and the coefficient of variation in force, $v_{Fg1\text{-}2}$, on the feed rate, q, when using a threshing cylinder with FP-I:

$$F_{g1-2} = 18.11e^{0.188q} ; \qquad R^2 = 0.98 ;$$

$$v_{Fg1-2} = 135.23q^{-0.777} ; \qquad R^2 = 0.96 ;$$

$$A_{Fg1-2} = 4.89q + 10.69 ; \qquad R^2 = 0.94$$

Table 1

Effect of the feed rate (q) on the dynamic indicators of force ($F_{g1\text{-}2}$) using cylinders fitted with FP-I and FP-II

Indices	Feed rate q, kg s^{-1}									
	4		6		8		10		12	
	FP-I	FP-II	FP-I	FP-II	FP-I	FP-II	FP-I	FP-II	FP-I	FP-II
Average of force with confidence level at 95% $F_{g1\text{-}2}$, N	36.66 ±2.31	52.32 ±2.47	54.38 ±2.59	76.93 ±2.32	92.13± 2.97	105.99 ±3.46	121.89 ±3.77	151.23 ±3.72	160.45 ±4.46	184.72 ±4.58
Standard deviation $\sigma_{Fg1\text{-}2}$, N	17.09	18.29	19.18	17.18	21.98	25.64	27.94	27.51	32.99	33.94
Coefficient of variation $v_{Fg1\text{-}2}$, %	46.62	34.95	35.27	22.33	23.86	24.19	22.92	18.19	20.56	18.37
Amplitude of variation $A_{Fg1\text{-}2}$, N	33.86	37.46	39.49	35.51	43.97	53.81	58.16	56.22	73.38	73.87
Coefficient of fluctuation $\delta_{Fg1\text{-}2}$	1.847	1.432	1.452	0.923	0.955	1.015	0.954	0.743	0.915	0.800

Fig. 3 and Table 1 show that at a feed rate of 12 kg s^{-1}, the force variation (73.38 ± 12.69 N) is higher than that at 6 kg s^{-1} (39.49 ± 5.88 N). The force acting on the rear part of the concave also increases (from 54.38 ± 2.59 to 160.45 ± 4.46 N, i.e. approximately three times, see Table 1) upon raising the feed rate from 6 to 12 kg s^{-1}.

An analysis of the operation of a threshing cylinder fitted with FP-II revealed that an increase in feed rate from 6 to 12 kg s^{-1} causes a force increase from 76.93 ± 2.32 to 184.72 ± 4.58 N (i.e. ~2.5 times). It is worth noting that the increase in force when the spaces between the rasp bars are covered is threefold while that observed when using FP-II is about 2.5-fold. However, the total increase in force acting on the rear part of the concave obtained using both types of filler plates was somewhat above 100 N.

The data provided herein were obtained by analysis of trials performed in period 3 (time interval t_2–t_3)

(Fig. 2). Integrating the data and further statistically processing them provides the average forces acting on the rear part of the concave for all the repetitions. The amplitudes of the force variations along with the variation coefficients and the degrees of fluctuation were calculated depending on the feed rate q, which ranged from 4 kg s^{-1} to 12 kg s^{-1} (Table 1).

Corn ear flow threshing experiments performed using a cylinder fitted with filler plates (FP-I) showed that the force acting on the rear part of the concave increased from 36.7±2.31 to 160.5±4.46 N (Fig. 3). Is was also observed that increasing the feed rate from 4 to 12 kg s^{-1} caused the amplitude of the force acting on the rear part of the concave to increase from 33.9 ± 3.6 to 73.4 ± 12.7 N. However, it is worth noting that the corresponding variation coefficient, v_{Fg1-2}, decreased from 46.6% to 20.6% under the same corn ear flow into the threshing unit.

Figure 4 shows that while operating a threshing cylinder fitted with FP-II, increasing the feed rate from 4 to 12 kg s^{-1} causes an increase in the force acting on the rear part of the concave (from 52.32±2.47 to 184.72±4.58 N). Also, increasing q causes an increase in the amplitude of force acting on the rear part of the concave.
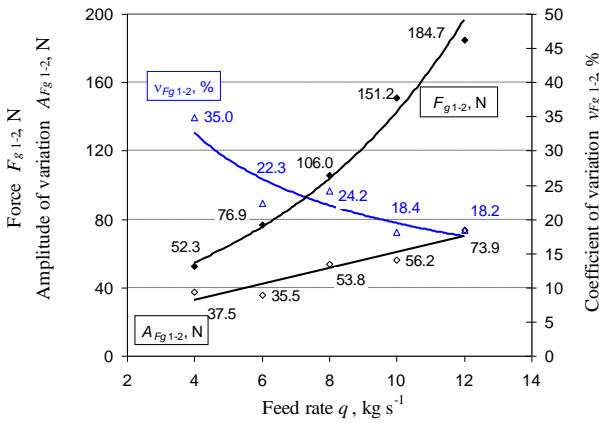

Fig. 4 Dependencies of the force acting on the rear part of the concave, F_{g1-2}, the amplitude of force, A_{Fg1-2}, and the coefficient of variation in force, v_{Fg1-2}, on the feed rate, q, when using a threshing cylinder fitted with FP-II:

$$F_{g1-2} = 28.81 e^{0.1599q} ; \qquad R^2 = 0.99 ;$$

$$v_{Fg1-2} = 71.50 q^{-567} ; \qquad R^2 = 0.85 ;$$

$$A_{Fg1-2} = 4.67 q + 13.96 ; \qquad R^2 = 0.89$$

However, the coefficient of variation decreases from 34.95% to 18.37% (Fig. 4), similar to what happens when using a cylinder with covered spaces between the rasp bars. Table 1 shows that increasing q within the above-mentioned range leads to a decrease in the degree of fluctuation, δ_{Fg1-2}, from 1.847 to 0.915 (using FP-I) and from 1.432 to 0.800 (using FP-II). This proves that the threshing unit operates consistently (evenly) at high feed rates (q). It should also be noted that the amplitude of variation (A_{Fg1-2}) is two times greater than the standard deviation, σ_{Fg1-2}, which indicates that the suggested methodology can be considered to be correct.

A comparison of the data obtained for variations in threshing cylinder speed, n_b, with those obtained for variations in the force acting on the rear part of the concave, F_{g1-2}, reveals a linear relationship between these two parameters, irrespective of the feed rate or type of filler plates (spaces between rasp bars) (see Figs. 5 and 6). The data provided in this study were obtained by analyzing periods 2–4 (time intervals t_1–t_2, t_2–t_3, and t_3–t_4) (Fig. 2).
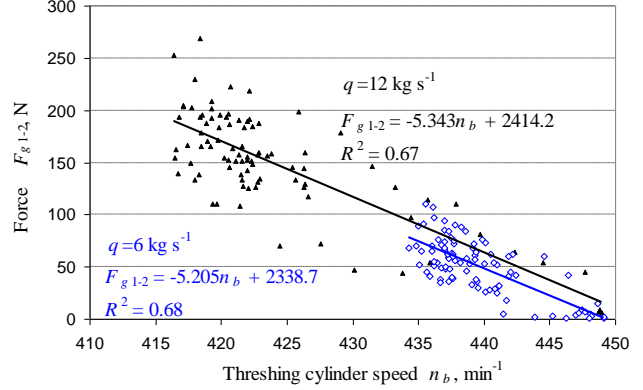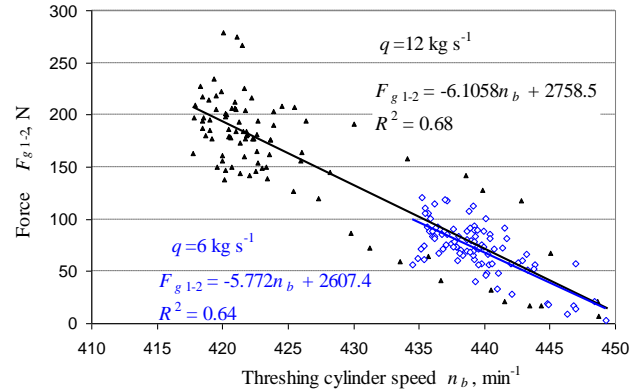

Fig. 5 Variations in the force acting on the rear part of the concave, F_{g1-2}, depending on the threshing cylinder speed, n_b, at corn ear feed rates of 6 and 12 kg s^{-1} using a threshing cylinder with FP-I.

Fig. 6 Variations in the force acting on the rear part of the concave, F_{g1-2}, depending on the threshing cylinder speed, n_b, at corn ear feed rates of 6 and 12 kg s^{-1} using a threshing cylinder fitted with FP-II
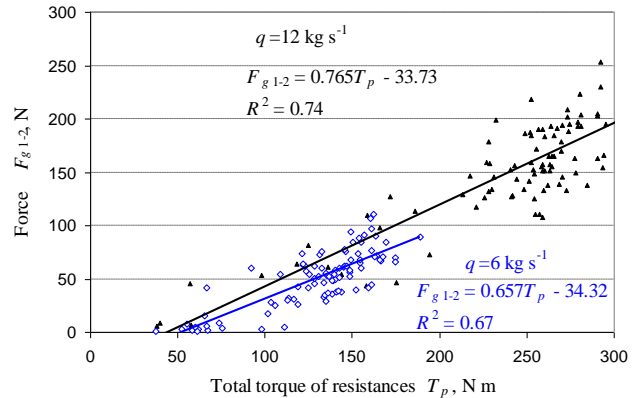

Fig. 7 Variations in the force acting on the rear part of the concave, F_{g1-2}, depending on the variations in resistance to threshing, T_p, at corn ear feed rates of 6 and 12 kg s^{-1} using a threshing cylinder with FP-I

However, a comparison of the variations in F_{g1-2} with the data obtained for variations in the total torque of resistances, T_p, shows that these parameters are also linearly dependent or are linearly correlated when varying the feed rate. This trend is observed when using cylinders with both types of filler plates/spaces between the rasp bars (see Figs. 7 and 8).
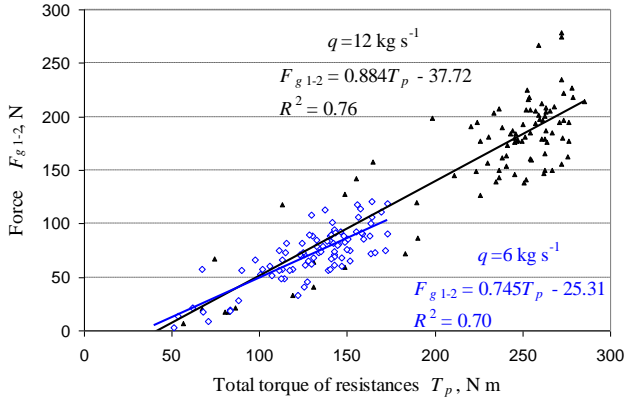

Fig. 8 Variations in the force acting on the rear part of the concave, F_{g1-2}, depending on the variations in resistance to threshing, T_p, at corn ear feed rates of 6 and 12 kg s^{-1} using a threshing cylinder fitted with FP-II

3.2. Torque of resistance to threshing cylinder rotation

The experiments to determine the torque of resistance to threshing cylinder rotation were repeated three times, and their results are summarized in Figure 9. Contrary to the force acting on the rear part of the concave, the torques of resistance depend only slightly on the type of filler plate used. At a corn ear feed rate of 6 or 12 kg s^{-1} the torques are almost equal; $T_p=143.59 \pm 2.14$ N m and $T_p=141.56 \pm 2.01$ N m; and $T_p = 261.21 \pm 3.41$ N m and $T_p = 255.37 \pm 2.35$ N m, respectively. By statistically processing the torque of resistance data, it was possible to obtain average values for each repetition as well as the total average for all the repetitions depending on the feed rate. The amplitudes of variation of the torques and the variation coefficients, and degrees of fluctuation were also calculated (Table 2).

Figs. 9 and 10 were prepared using the data presented in Table 2. Irrespective of the shape of the filler plates, increasing the feed rate from 4 to 12 kg s^{-1} led to an exponential increase in T_p. When using a cylinder with covered spaces between the rasp bars (FP-I), T_p was found to increase from 114.7 ± 1.79 to 261.21 ± 3.41 N m, and when the cylinder was fitted with FP-II, it increased from 111.75 ± 1.85 to 255.37 ± 2.35 N m. Upon increasing q within the same range, the amplitude of the torque of resistances was also found to increase from 25.9 ± 2.7 to 58.6 ± 9.6 N m (using FP-I). The amplitude of variation of the torque of resistance was found to change from 28.06 ± 3.6 N m for $q = 4$ kg s^{-1} to 35.15 ± 3.3 N m for a feed rate of 12 kg s^{-1}. Table 2 shows that the degree of fluctuation, δ_{Tp}, decreases from 0.502 to 0.275 in the case of covered spaces, while in the case of FP-II, even if the feed rate (q) is increased three times, this coefficient remains constant at ~0.45. However, when using a cylinder with covered spaces between the rasp bars, the variation coefficient decreases from 11.53% to 9.65%, while in the case of FP-II, it decreases from 12.25% to 6.82%.

Table 2

Effect of the feed rate (q) on the dynamic indicators of the torque of resistance (T_p) using a cylinder fitted with FP-I and FP-II

Indices	Feed rate q, kg s^{-1}									
	4		6		8		10		12	
	FP-I	FP-II	FP-I	FP-II	FP-I	FP-II	FP-I	FP-II	FP-I	FP-II
Average of torque with confidence level at 95% T_p, N m	114.70 ±1.79	111.75 ±1.85	143.59 ±2.14	141.56 ±2.01	181.11 ±2.43	176.90 ±2.72	221.08 ±2.94	218.80 ±2.77	261.21 ±3.41	255.37 ±2.35
Standard deviation σ_{Tp}, N m	13.22	13.69	15.81	14.88	18.01	20.17	21.79	20.52	25.21	17.41
Coefficient of variation ν_{Tp}, %	11.53	12.25	11.01	10.51	9.95	11.40	9.86	9.38	9.65	6.82
Amplitude of variation A_{Tp}, N m	25.89	28.06	33.01	30.70	40.00	41.51	45.57	41.26	58.58	35.15
Coefficient of fluctuation δ_{Tp}	0.451	0.502	0.460	0.434	0.442	0.469	0.412	0.377	0.449	0.275

These results suggest that the corn ear threshing process is stable when a cylinder fitted with FP-II is used. It is known that the speed of the threshing cylinder decreases with an increase in the threshing separation unit load [10, 17, 27]. Therefore, it is appropriate to establish a relationship between the torque of resistance to rotation of the threshing cylinder, T_p, and the rotation speed of the threshing cylinder, n_b.

In this study, a highly linear correlation was found between these two parameters (Figs. 11 and 12), regardless of the type of filler plate used. However, increasing the feed rate caused correlative interdependency of the above-mentioned parameters to increase. Approximation of the data resulted in linear dependencies with correlation coefficients close to one.

Sensing devices are commonly used to monitor the feed rate of crop materials in combine harvesters. In this manner, it is possible to control the forward speed of the combine harvester and adjust particular mechanisms in the machine [16, 28].

In this study, we found that the threshing cylinder torque is proportional to the amount of crop being threshed in a combine harvester; therefore, this is an acceptable parameter for monitoring and controlling the operation conditions of such machines. However, by the time the crop material reaches the threshing cylinder, it may have been inside the harvesting machine for ~5 s, which means that there is a lag of ~5 s and this must be accepted.
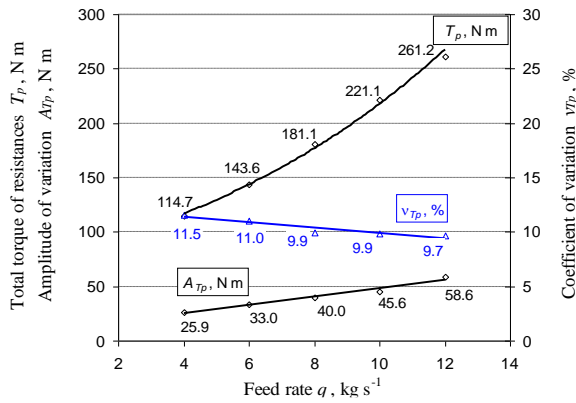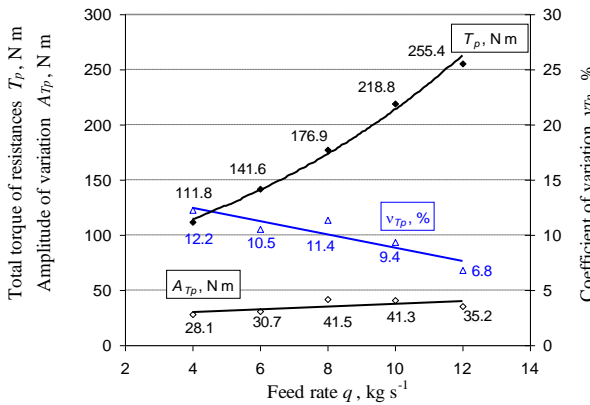
Fig. 9 Total torque of resistance, T_p, amplitude of variation of the torque, A_{T_p}, and dependency of the coefficient of variation of the torque, v_{T_p}, on the corn ear feed rate, q, using a threshing cylinder with FP-I:

$$T_p = 76.97\,e^{0.1039q}\;; \qquad R^2 = 0.99\;;$$

$$v_{T_p} = -0.246\,q + 12.37\;; \qquad R^2 = 0.89\;;$$

$$A_{T_p} = 3.90\,q + 9.43\;; \qquad R^2 = 0.97$$

Fig. 10 Total torque of resistance, T_p, amplitude of variation of the torque, A_{T_p}, and dependency of the coefficient of variation of the torque, v_{T_p}, on the corn ear feed rate, q, using a threshing cylinder fitted with FP-II:

$$T_p = 75.17\,e^{0.1044q}\;; \qquad R^2 = 0.99\;;$$

$$v_{T_p} = -0.599\,q + 14.87\;; \qquad R^2 = 0.81\;;$$
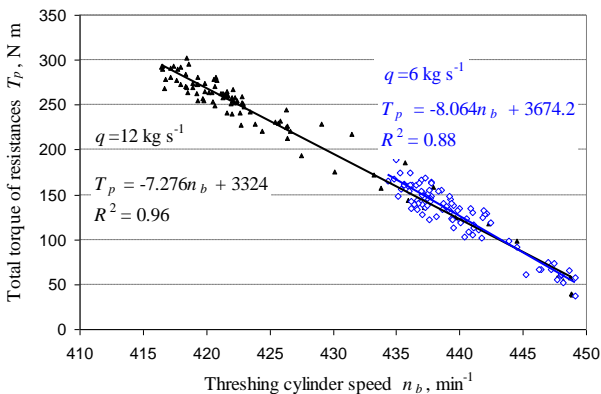
$$A_{T_p} = 1.237\,q + 25.44\;; \qquad R^2 = 0.67$$

Fig. 11 Variations in the total torque of resistance to rotation as a function of the threshing cylinder speed, n_b, at feed rates of 6 and 12 kg s^{-1} using a threshing cylinder with FP-I
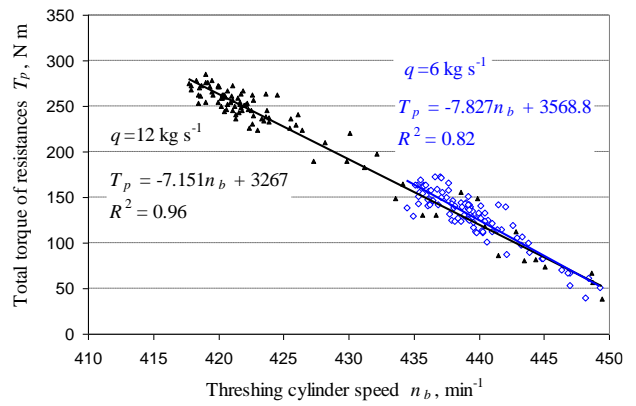

Fig. 12 Variations in the total torque of resistance to rotation as a function of the threshing cylinder speed, n_b, at feed rates of 6 and 12 kg s^{-1} using a threshing cylinder fitted with FP-II

4. Conclusions

The force acting on the rear part of the concave ($F_{g1\text{-}2}$) was found to increase about four times upon increasing the feed rate (q) from 4 to 12 kg s^{-1}. At a feed rate of 12 kg s^{-1}, the value of the amplitude ($A_{Fg1\text{-}2}$) of variation in force ($F_{g1\text{-}2}$) was higher than that at $q = 4$ kg s^{-1}. Increasing q within the above-mentioned range led to a decrease in the degree of fluctuation, $\delta_{Fg1\text{-}2}$, from 1.847 to 0.915 (in the case of FP-I) and from 1.432 to 0.800 (in the case of FP-II). The coefficient of variation showed a similar trend. This evidences that the threshing unit tends to operate consistently (evenly) at higher feed rates.

Experiments performed using FP-II showed that the torque of resistance to cylinder rotation varied from 28.06 N m at $q = 4$ kg s^{-1} to 35.15 N m at $q = 12$ kg s^{-1}. Moreover, when using a cylinder with covered spaces between the rasp bars (FP-I), the coefficient of variation was found to decrease from 11.53% to 9.65%, and when using a cylinder fitted with FP-II, it changed from 12.25% to 6.82%. These results indicate that the corn ear threshing process is stable (or even) when the threshing cylinder is fitted with FP-II.

A comparison of the results obtained for the variations in the forces acting on the rear part of the concave ($F_{g1\text{-}2}$) with those obtained for the total torque of resistance (T_p) and the cylinder speed (frequency of rotation) (n_b) revealed a linear correlation between these parameters at varied feed rates. A high correlation was found between T_p and n_b. This trend was observed for both types of filler plates, and an increase in the feed rate resulted in an increased correlation dependency of the above-mentioned parameters. These dependencies can be further used to improve automatic speed control systems for increasing the efficiency and cost-effectiveness of combine harvesters. Besides, crop flow sensing at the threshing cylinder might complement existing technologies to improve the quality of data monitoring.

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DYNAMIC INDICATORS OF A CORN EAR THRESHING PROCESS INFLUENCED BY THE THRESHING-SEPARATION UNIT LOAD

S u m m a r y

Corn ear feed rate variations in the threshing-separation unit of a combine harvester have a significant impact on the dynamic indicators of the threshing process. In this study, we conducted experiments using a stationary tangential threshing device and measured the forces acting on the rear part of the concave as well as the torque of the rotating cylinder during a threshing process. The *ThreshLab* software was developed for this purpose. We found that increasing the corn ear feed rate made the threshing process more even, due to decreasing the fluctuation of forces acting on the rear part of the concave and torque of the rotating cylinder. The corn ear threshing process may be more stable if the threshing cylinder is fitted with filler plates (FP-II). A comparison of the results obtained for variations in the forces acting on the rear part of the concave with those obtained for variations in the total torque of resistance and the speed of the threshing cylinder revealed a linear correlation between these parameters at varied feed rates.