

Performance of a Cam-Type Pulse Continuously Variable Transmission

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ABSTRACT

The aim of this paper is to assess the performance parameters of the cam-type pulse continuously variable transmission, which is a first attempt to investigate this issue in the presented manner. The transmission under investigation comprises two cams in cooperation with two units, each consisting of rollers connected to swing rods mounted on overrunning clutches. To accomplish this, a test stand was built and a detailed testing program was developed taking into consideration the influence of gear ratio, rotational speed, and load on the uniformity of motion and efficiency. Analysis of the obtained results leads to the conclusion that the tested transmission is particularly sensitive to changes in operating conditions, and its overall performance deteriorates as the reduction ratio increases. The best results were achieved with a load below the nominal one, at low rotational speeds and a low speed reduction. The transmission's structure design and operational principle result in significant power losses, as evidenced by an efficiency rating of less than 60%.

Keywords: pulse continuously variable transmission, cam-type continuously variable transmission, pulse infinitely variable transmission, efficiency, drive transmission.

INTRODUCTION

Continuously variable transmissions (CVT) are a well-known type of transmission, primarily used in vehicle drives and heavy equipment. This category of transmissions includes many different types, which due to considerable applicability, have garnered attention from numerous researchers. Authors investigate, e.g., dynamic [1], stress distribution [2], power losses and efficiency [8, 9] as well as heat transfer [10] in belt-type CVTs. Other studies concern geometric [3], kinematic [4] and performance [4, 11] analyses of ball-type CVTs. Similar research can be found in the design [5-7] and operation [12, 13] of toroidal CVTs. Among the various CVT variants, some, while being used in engineering, have received relatively limited scientific attention. An example of this is the pulse-CVT.

As per the definition [14], this transmission type involves the interaction of a linkage or cam mechanism with a unidirectional overrunning clutch. Due to its design, it can be categorized as

an infinitely variable transmission (IVT), which distinguishes it from a traditional CVT by its ability to generate a zero-transmission ratio. Common kinematic structures include those depicted in Fig. 1 and 2. The variant in Figure 1 consists of an input shaft 1 with a cam 2, cooperating with a roller 3 mounted on a swing rod 4. The swing rod is supported by an overrunning clutch 5 on the output shaft 7. Applying drive to the input shaft initiates the roller's motion on the cam's outer surface, inducing cyclic rotational movement of the swing rod and, consequently, driving the output shaft with pulsations. The gear ratio depends on the swing rod's angular deflection, determined by the position of bumper 8. To achieve a low gear ratio, the roller must roll around the entire active cam surface, while an increased speed reduction occurs when the swing rod's range of motion, limited by the bumper, allows the roller to engage only a portion of the cam surface. The largest gear ratio is obtained when the entire swing rod deflection cycle causes the overrunning clutch to rotate just one step resulting from its construction.

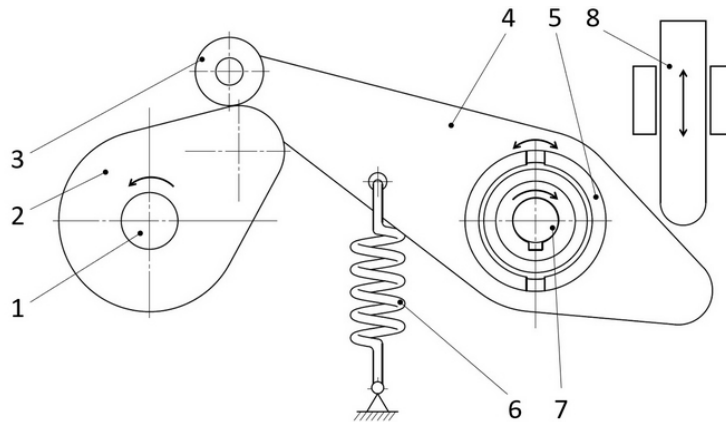


Fig. 1. First structure of a cam-type pulse-CVT: 1 – input shaft, 2 – cam, 3 – roller, 4 – swing rod, 5 – overrunning clutch, 6 – spring, 7 – output shaft, 8 – dumper

In the cam-type pulse-CVT shown in Figure 2, the cam 2 mounted on the input shaft 1 actuates the swing rod 3, with one end featuring a roller 4 rolling on the track 5, and the other end is connected to a rocker with overrunning clutch 6 mounted on the output shaft 7. The gear ratio depends on the tilt angle of the rocker, which results from swing rod’s range of motion. The gear ratio is changed by adjusting the inclination angle α of a raceway.

Over the years, various transmission designs have been developed, falling under the pulse-CVT classification, even if their structures differ from the aforementioned definition [14]. Transmissions equipped with planetary pawls, planetary gears or switch plates are discussed in [15, 16]. Additionally, there are designs that mechanism with a spatial cam with a planetary gear [17, 18] or combine a cam-follower with a grooved wheel-follower [19, 20]. The pulse CVT category

also includes the transmission described in [21], which transmits drive by controlling the engagement time of compact non-slip clutches (e.g., dog clutches).

As previously mentioned, cam-type pulse CVTs have received limited attention in scientific studies. In the doctoral dissertation [22], the impact of pulse-CVT transmission operation on the uniformity of seed sowing by a grain seeder was examined. Computer Aided Design (CAD) models were developed to enable kinematic simulations of both types shown in Figures 1 and 2, leading to the determination of the gear ratio based on bumper position or raceway inclination angle. The dissertation author conducted experimental research to validate kinematic analysis results and to compare the uniformity of drive transmission in the gearbox according to Figure 1 with that of the Norton gearbox. Articles [13, 14] address analytical models of kinematic gears with a structure

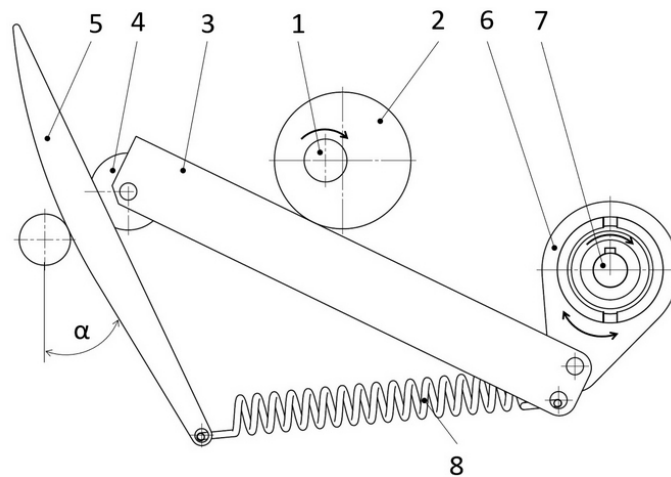


Fig. 2. Second structure of a cam-type pulse-CVT: 1 – input shaft, 2 – cam, 3 – swing rod, 4 – roller, 5 – raceway, 6- rocker with overrunning clutch, 7 – output shaft, 8 – spring, α – inclination angle of a raceway

as in Figure 2. The authors of [25, 26] describe an analytical linkage-type pulse-CVT energy model (in accordance with the definition of [14]) without simulation or experimental verification. This model includes energy losses in kinematic pairs, those resulting from lubrication resistance and energy losses due to damping in the transmission components. The analyses reveal that the highest power losses occur in the kinematic pair, with the lowest in material damping. While there are numerous studies on cam mechanisms in the literature, publications on overrunning clutches are relatively scarce. Among them, the article [27] stands out for describing an analytical model of sliding friction in an overrunning clutch, accounting for the EHL theory. The results presented determine friction in the bearing as a function of rotational speed and the course of power losses as a function of time based on rotational speed. However, it should be emphasized that this model lacks experimental validation.

The aim of this article is to test the operational parameters of the cam-type pulse CVT as depicted in Figure 1. As indicated earlier, existing research primarily focuses on kinematic analyses of this transmission type, with only a few experimental studies [e.g., 22] serving to validate them. Comprehensive performance metrics, such as drive transmission stability and efficiency, have not been reported. Conducting such research will help in identifying potential problems associated with the use of the cam-type pulse CVT and may uncover new areas of application.

The structure of the article is organized as follows. Section 2 discusses the research subject, the test stand, and presents the test program. Section 3 offers a graphical presentation of the obtained results and their analysis, and Section 4 provides a summary.

SUBJECT AND SCOPE OF RESEARCH

The tests were conducted on a cam-type pulse CVT, as depicted in Figure 1, which forms part of the grain seeder. Transmission structure is shown in Figure 3. It consists of an input shaft with two monolithic cams 2 featuring three local maxima and minima on the profile. These cams interact with two swing rods 4, which, via overrunning clutches 5, are connected to the output shaft 6. To ensure the transmission's uniform operation, the cams 2 are rotated 180° relative to each other. This configuration ensures that when one roller 3 loses contact with one cam's active surface, the other roller begins to roll on the active surface of the second cam. The gear ratio is set by changing the angular position of bumpers 7, which limit the swing rods' range of motion. The cams and bumpers are composed of polyamide, while the rollers are made of bronze. The swing rods are constructed from aluminum, and the shafts and axles are made of steel, while the body is cast iron.

The operation of the transmission is illustrated in Figure 4. Power transmission through the first cam-swing rod pair commences when

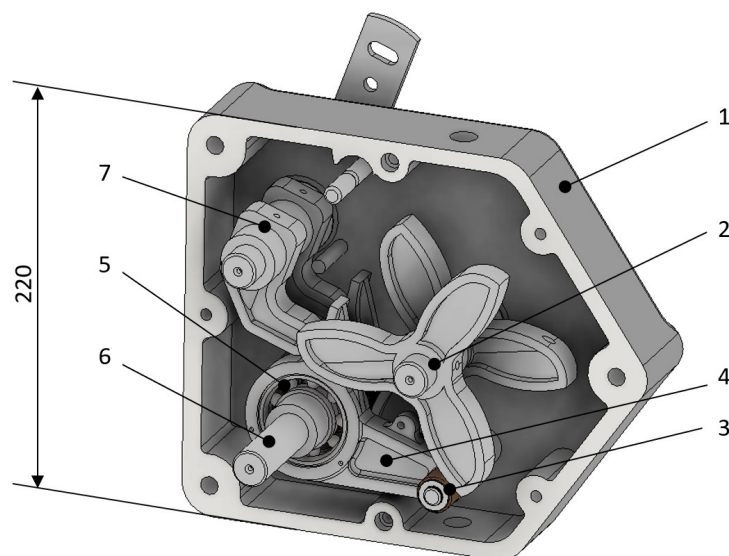


Fig. 3. CAD model of a cam-type pulse CVT: 1 – case, 2 – input shaft with cams, 3 – roller, 4 – swing rod, 5 – overrunning clutch, 6 – output shaft, 7 – dumper

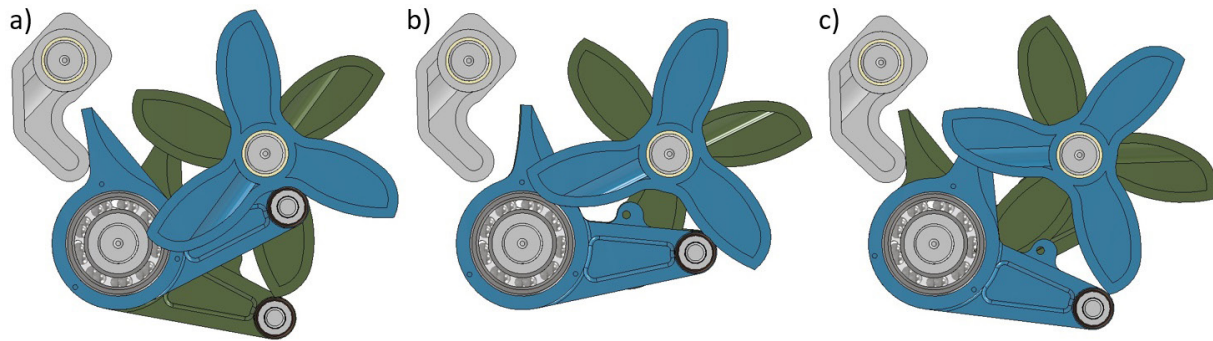


Fig. 4. Operation of a cam-type pulse CVT: a) the first pair starts cooperation, the second pair ends; b) drive transmission by the first pair; c) the first pair ends cooperation, the second starts

the roller begins rolling on the active surface of the first cam (Fig. 4a). Simultaneously, the roller of the second cam-swing rod pair rolls over the cam’s peak. Progressive cam rotation causes the first swing rod to swing outward, setting the over-running clutch, and hence the output shaft, in rotation (Fig. 4b). During this phase, the roller from the second swing rod rolls toward the minimum of the cam profile until the swing rod makes contact with the bumper. When the first roller reaches the cam’s peak (Fig. 4c), the roller from the second pair starts rolling on the second cam’s active surface. This cyclically maintains the drive transmission.

In order to determine the performance of the transmission, a research program was developed, as presented in Table 1. This program involves the measurement of the drive transmission’s continuity and efficiency for four different gear ratios (corresponding to four different bumper settings),

Table 1. Test program

Parameter	Symbol	Values
Gear ratio	i	$i_1; i_2; i_3; i_4$
Output shaft torque [Nm]	M_2	3.5; 7; 10.5; 14
Rotational speed of input shaft [rpm]	n	50; 100; 150; 200

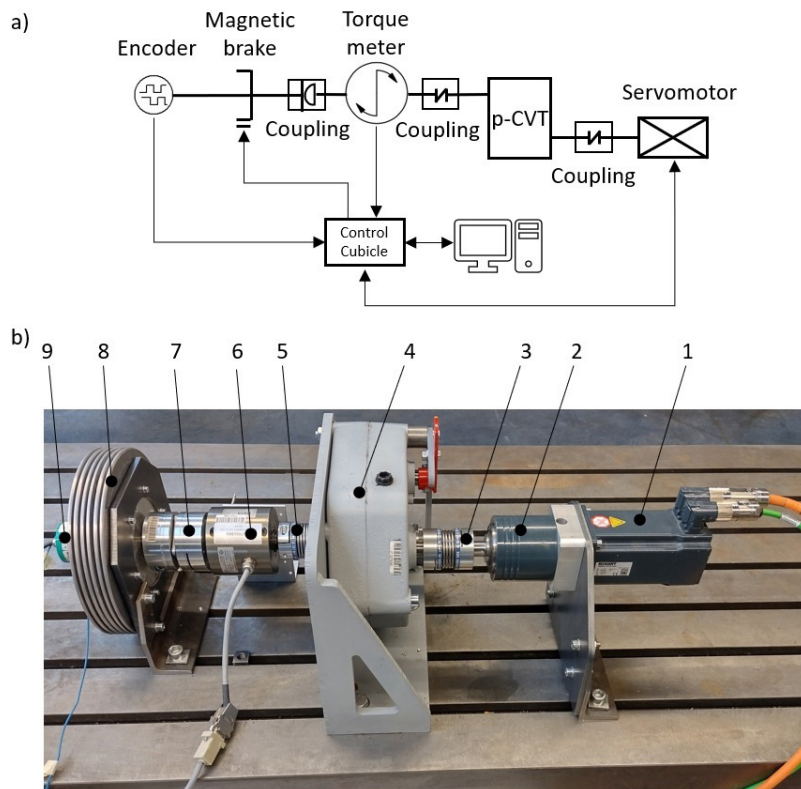


Fig. 5. Test stand: 1 – servomotor, 2 – planetary transmission ($i = 10, \eta_p = 97\%$), 3 – bellow coupling, 4 – cam-type pulse-CVT, 5 – bellow coupling, 6 – torque meter, 7 – laminae coupling, 8 – magnetic brake, 9 – encoder

four distinct loads, and four rotational speeds of the input shaft. The load values were determined experimentally by measuring the torque on the seeder sowing unit’s shaft, assuming that the maximum load would be twice the nominal torque. Due to the pulsating operation, the maximum speed of the input shaft n is limited to 200 rpm.

For the research purposes, a test stand, as shown in Figure 5, was constructed. It consists of a servo motor 1 connected to a planetary gear 2, which drive the input shaft of the tested transmission 4, while the torque loading the output shaft is provided by a magnetic brake 8. Measurements of torque and rotational speed on the output shaft is performed using a torque meter 6 and an encoder 9, while aforementioned input shaft parameters are recorded by the servo drive controller.

The configuration of the test stand described above allows for the determination of the transmission operating parameters indicated in Table 1. The actual pulse CVT gear ratio was calculated according to the formula:

$$i_j = \frac{\overline{n_m}}{n_{2m}} \quad (1)$$

where: $\overline{n_m}$ – average rotational speed of the input shaft for m-th case (time: 1 min);
 n_{2m} – average rotational speed of the output shaft for m-th case (time: 1 min).

The transmission efficiency was determined based on the equation:

$$\eta_n = \frac{\overline{M_{2k}}}{i_j \cdot \overline{M_{1k}}} \cdot 100\% \quad (2)$$

where: $\overline{M_{1k}}$ – average torque on the input shaft for k-th case (time: 1 min);
 $\overline{M_{2k}}$ – average torque on the output shaft for k-th case (time: 1 min).

The above parameters describe the gear performance for each tested case, but do not provide information about the gear performance during a single measurement. Following other publications [28, 29], it was decided to determine the uniformity of motion on the basis of the coefficient of variation (CoV) for output shaft rotational speed, which can be described by the equation:

$$CoV(n_{2m}) = \frac{s_{n_2}}{\overline{n_2}} \cdot 100\% \quad (3)$$

where: s_{n_2} – standard deviation of n_2 .

The tests were executed according to a strictly defined procedure. For each gear ratio, the initial load was applied, and then the input shaft was set in motion at the first rotational speed. The one-minute measurement of the transmission’s operational parameters (with a resolution of 500 samples per second) commenced one minute after activating the drive. To ensure consistent test conditions, including maintaining a constant oil temperature, a 30-minute break was introduced before the next measurement cycle. Subsequently,

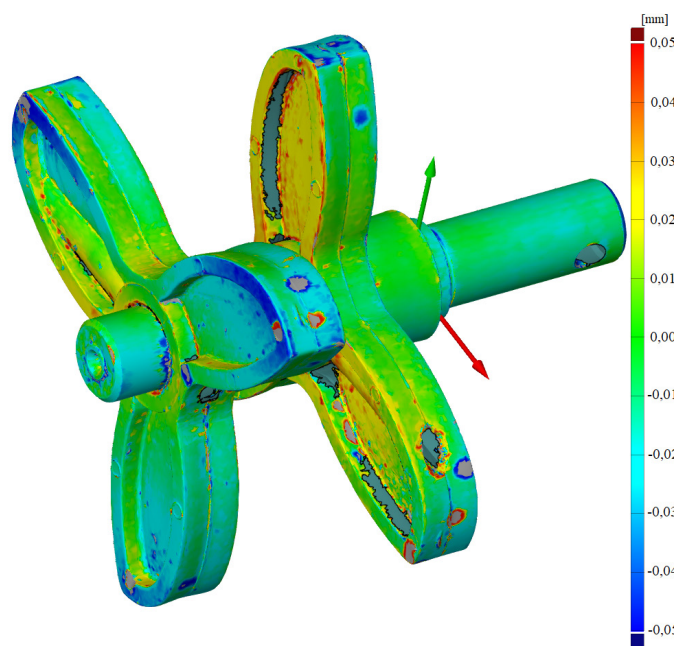


Fig. 6. Analysis of cam geometry after experimental tests in comparison with the reference model

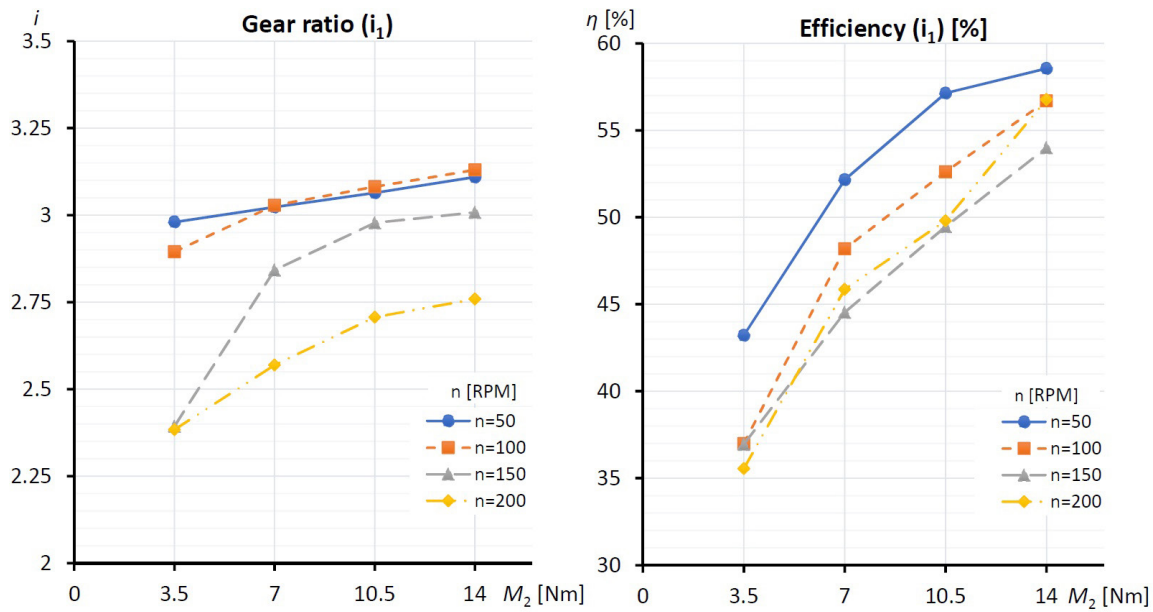


Fig. 7. Average gear ratio and efficiency measured for $i_1 = 2.87$ as a function of load M_2 and rotational speed n

the operational parameters were recorded for different rotational speeds, loads, and gear ratios, resulting in a total of 64 tests.

Given that the maximum load during the tests was twice the nominal torque, there was a risk of accelerated wear and permanent deformation of the cams. To ensure that these phenomena did not affect the measurement results, it was decided to analyze the cam geometry through scanning with a structured light scanner with a 0.01 mm resolution. Two measurements were taken: one before and one after the tests. The scans were analyzed in a dedicated software and the scan from before test was used as a reference model. The results shown in Figure 6 clearly indicate that there was no permanent deformation of the cams during the tests, as evidenced by the small deviation on the active surfaces, well within the scanner’s measurement resolution limits.

TEST RESULTS AND DISCUSSION

Figures 7-10 present the results of experimental tests of the cam-type pulse CVT for various gear ratios. The stability of the gear ratio and the efficiency of the transmission as a function of torque M_2 and the rotational speed of the input shaft n were examined. Upon analyzing the results for the first research case (Fig. 7), it can be seen that the average kinematic ratio was $i_1 = 2.87$. The lowest value ($i_1 \approx 2.4$) was recorded under the lowest load ($M_2 = 3.5$ Nm) and the highest

rotational speed ($n = 200$ rpm), while the highest value ($i_1 \approx 3.15$) was recorded under the highest load ($M_2 = 14$ Nm) and $n = 150$ rpm. Moreover, it is clear that the kinematic ratio increases with increasing load and decreases with increasing rotational speed. This trend primarily results from the flexibility of gear elements, mainly the cams, which experience greater elastic deformations under higher loads. At low rotational speeds ($n = 50$ – 100 rpm) the variation of the gear ratio as a function of the gear load is small ($i_1 = 2.9$ – 3.15), while at a speed of $n = 150$ rpm, the gear ratio change is more significant ($i_1 = 2.4$ – 3.0), indicating an increased variability of the kinematic ratio with higher rotational speeds. Analyzing the transmission efficiency for this case, it is observed that it increases with increasing load and decreases with increasing rotational speed. Importantly, similar characteristics were recorded for each of the tested speeds, demonstrating that rotational speed has a smaller impact than load on achieved efficiency. The cam-type pulse CVT operated with the highest efficiency ($\eta \approx 59\%$) at the lowest speed and the highest load, and with the lowest efficiency ($\eta \approx 36\%$) at the highest speed and the lowest tested load.

Similar observations can be made for the results shown in Figure 8. The average kinematic ratio for this case was $i_2 \approx 5.67$, slightly more than twice the gear ratio i_1 . The increase in gear ratio variability as a function of load and speed is very clearly visible. The smallest recorded kinematic ratio was $i_2 = 2.2$, and the largest $i_2 = 7.3$.

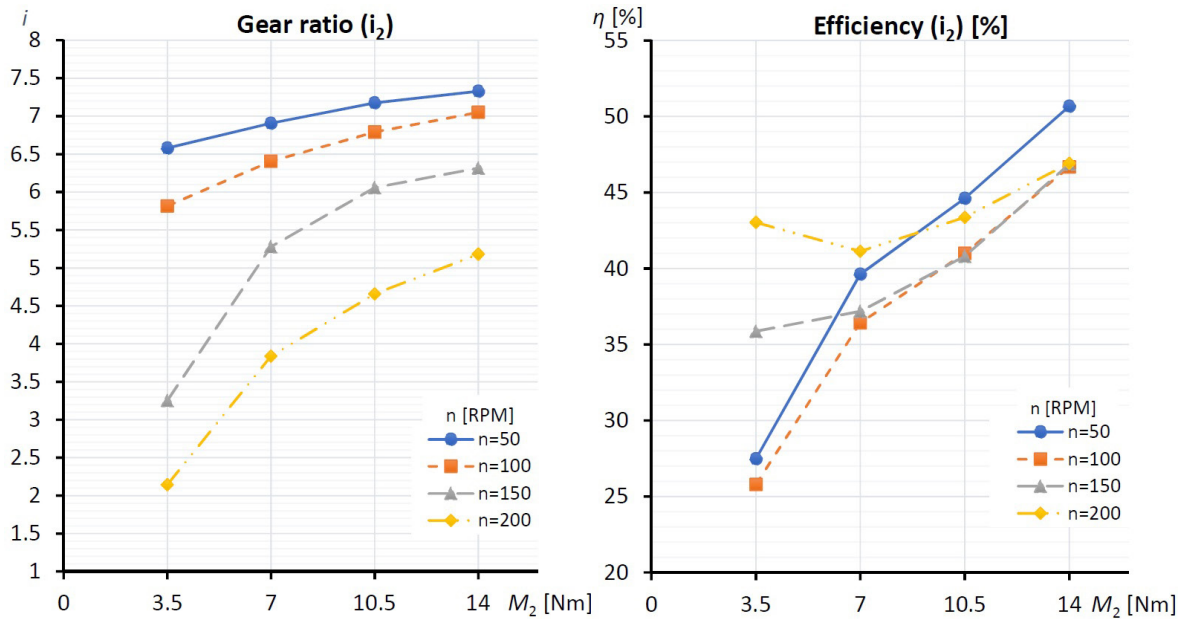


Fig. 8. Average gear ratio and efficiency for $i_2 = 5.67$ as a function of load M_2 and rotational speed n

Analyzing the efficiency diagram, it can be seen that at the lowest load $M_2 = 3.5$ Nm the highest efficiency $\eta = 43\%$ was achieved at the highest rotational speed $n = 200$ rpm, while for the highest load, the gear efficiency was highest at the lowest speed ($\eta = 51\%$). In this case, depending on the rotational speed and load, the transmission operated with an efficiency of $\eta = 26\text{--}51\%$.

In the third case considered (Fig. 9), the average kinematic gear ratio was $i_3 = 18.47$, with the lowest value recorded for $M_2 = 3.5$ Nm and $n = 200$ rpm ($i_3 \approx 5.3$), and the highest for $M_2 = 14$ Nm i $n = 100$ rpm ($i_3 \approx 33.5$). Unlike the previous

cases, a more rapid increase in gear ratio was observed with increasing load. At low rotational speeds, an increase in efficiency with increasing load was observed (up to maximum values of $\eta \approx 27\text{--}30\%$ at $M_2 = 10.5$ Nm), while at high speeds, the efficiency as a function of load tends to decrease.

In the fourth and final research case (Fig. 10), the highest possible gear ratio was set, resulting in an average kinematic ratio of $i_4 = 183.86$. It should be emphasized that the gear ratio varied in the range of $i_4 = 50\text{--}440$ depending on the load and rotational speed. Additionally, it is worth

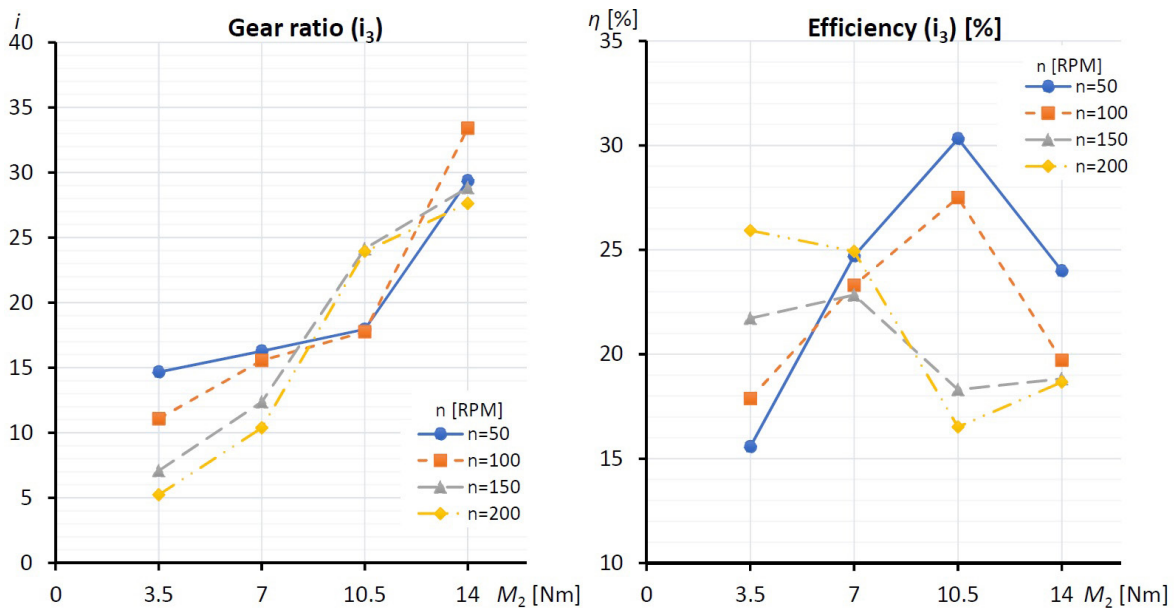


Fig. 9. Average gear ratio and efficiency for $i_3 = 18.47$ as a function of load M_2 and rotational speed n

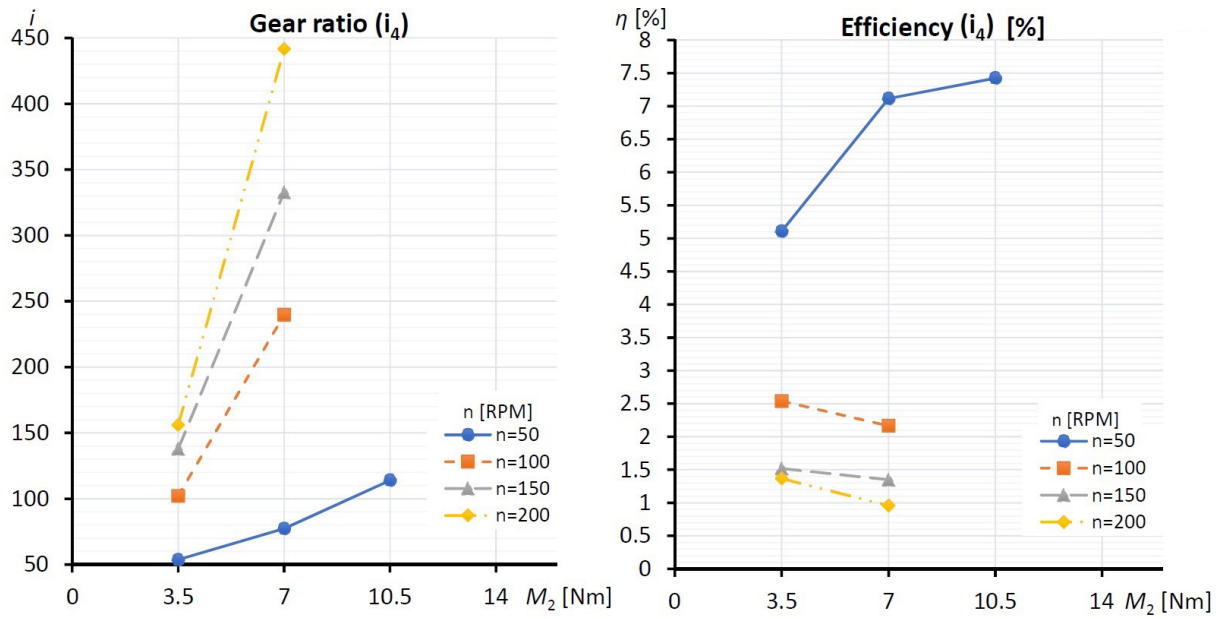


Fig. 10. Average gear ratio and efficiency for $i_4 = 183.86$ as a function of load M_2 and rotational speed n

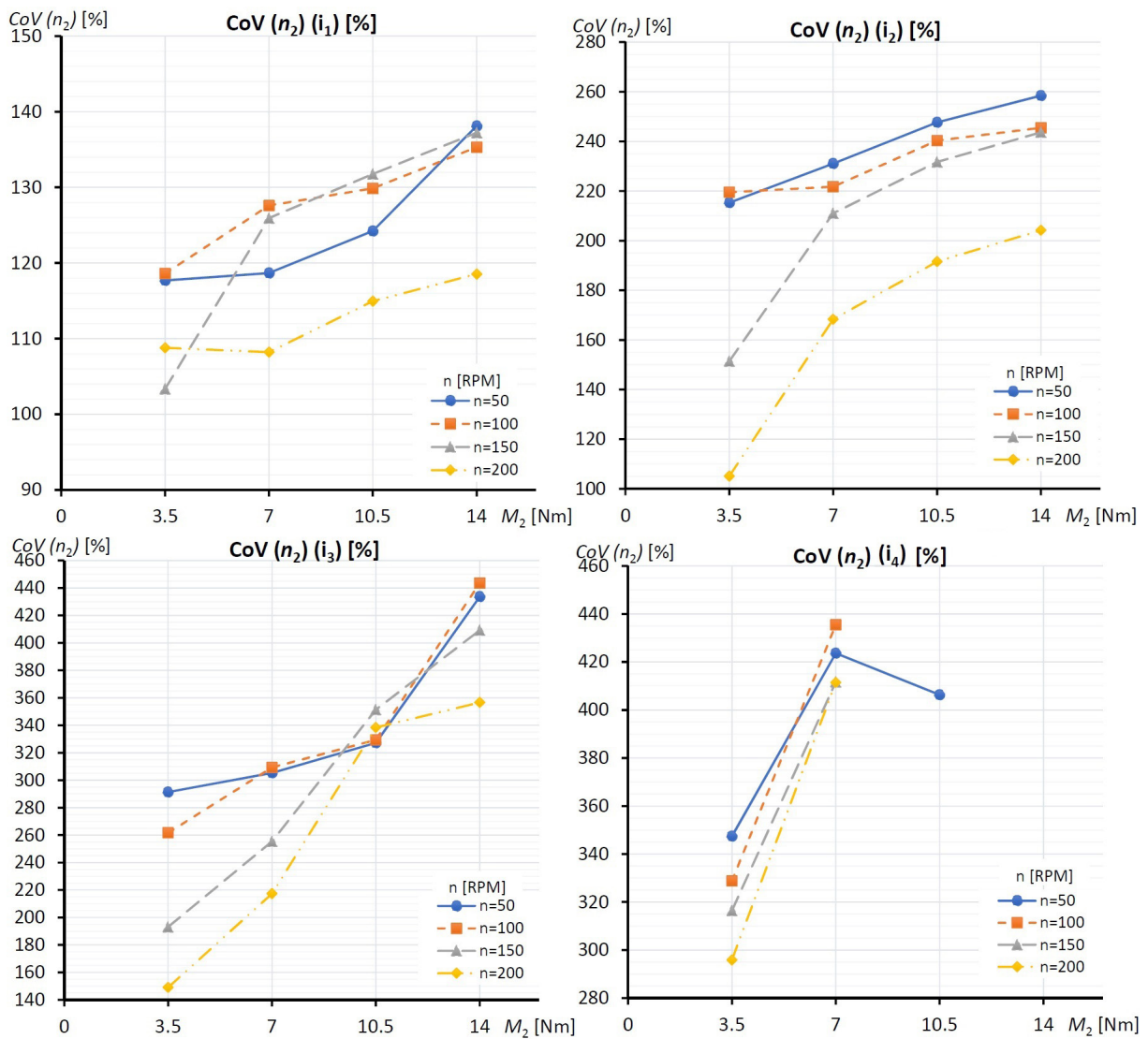


Fig. 11. $CoV(n_2)$ for tested gear ratios as a function of load M_2 and rotational speed n

Table 2. Overall test results

Parameter	Symbol	Value			
		i_1	i_2	i_3	i_4
Average gear ratio	\bar{i}_j	2.87	5.67	18.47	183.86
Standard deviation of gear ratio	$s(i)$	0.24	1.47	8.24	122.24
Average CoV(n_2) [%]	CoV_{n_2}	122.43	211.66	310.74	375.22
Maximum efficiency [%]	η_{max}	58.56	50.67	30.32	7.42

mentioning that the pulse-CVT transmitted drive successfully only for M_2 loads less than or equal to 7 Nm. For $M_2 = 10.5$ Nm, motion was recorded only at rotational speed $n = 50$ rpm, and in all subsequent cases, the output shaft did not rotate ($i_4 = 0$). The inability to transmit drive results from the elastic deformations of the cams, which, for large gear ratios, maintain contact with the rollers on the swing rod only near the cam profile peaks. It is also challenging to interpret the efficiency changes concerning load and speed, but it can be observed that significant power losses occurred, and the recorded efficiency values ranged from $\eta = 1-7\%$.

The observations described above lead to the conclusion that the kinematic gear ratio of the tested cam-type pulse CVT varies with rotational speed and load. To precisely determine the transmission's uniformity of operation, an analysis of the coefficient of variation of n_2 values for the tested gear ratios (Fig. 11) was conducted. Since the drive is not continuously transmitted, and the output shaft rotates cyclically, the $CoV(n_2)$ values for each case exceed 100%, even reaching 440%. The analysis of the charts reveals that drive transmission is most uniform (with the smallest CoV values) at the highest tested rotational speeds ($n_2 = 200$ rpm), while the highest $CoV(n_2)$ values occur at the lowest speed ($n_2 = 50$ rpm). This behavior can be attributed to two factors: the operation at higher rotational speeds involves greater dynamics within the system (higher accelerations of the active transmission's components), resulting in more even movement (additional dynamic forces compensate for uneven operation). Simultaneously, the resolution of the measurement system decreases at higher rotational speeds, affecting motion registration accuracy.

The results are easier to interpret concerning the tested gear ratios. The gear was characterized the highest uniformity of output shaft movement for small gear ratios ($i_1 = 2.87$ and $i_2 = 5.67$). It is noteworthy that doubling the gear ratio (from i_1 to i_2) nearly doubled the $CoV(n_2)$ value. However,

for large gear ratios ($i_3 = 18.47$ and $i_4 = 183.86$), the operation's unevenness was significant due to the transmission principle (shown in Fig. 1 and Fig. 4). Achieving a high degree of reduction involves deflecting the swing rod using a bumper, causing the roller to cooperate only a small part of the cam contour (near the peak). In such cases, the output shaft experiences very high angular acceleration, resulting in substantial variations in rotational speed n_2 .

The extensive results of the experimental research described above are summarized in tabular form (Table 2).

CONCLUSIONS

The cam-type pulse CVT is a transmission with a relatively simple and compact structure, allowing for a wide range of gear ratios. However, its operational principle poses several problems regarding its performance. Through the research, the influence of rotational speed and transmission load on movement uniformity and efficiency was assessed for four different gear ratios. Detailed analysis of the results yields the following conclusions:

- The kinematic gear ratio of the transmission (with the bumper's position unchanged) increases with the load of the out shaft.
- The rotational speed of the input shaft affects the kinematic gear ratio. For small reduction ratios, higher speeds lead to smaller gear ratios, while for large reduction ratios, an increase in speed results in a higher gear ratio.
- The variability of the kinematic gear ratio during transmission operation increases with the reduction ratio.
- Cam-type pulse CVT operate with the highest efficiency at low rotational speeds ($n = 50$ rpm).
- An increase in transmission efficiency with load is observable.
- As the gear ratio increases, its efficiency decreases dramatically.

- An increase in the input shaft's rotational speed positively affects the uniformity of the output shaft's movement.
- An increase in the torque M_2 negatively impacts drive transmission uniformity.
- Load has a more significant impact on transmission performance than rotational speed.

It is evident that the cam-type pulse CVT is particularly sensitive to changing operating conditions. To ensure its proper operation, primarily in terms of gear ratio stability, maintaining a constant rotational speed and load is essential. Moreover, it's worth noting that the best results were achieved with a load lower than the nominal load, at low rotational speeds, and with a low gear ratio. Increasing the reduction ratio significantly deteriorates transmission performance.

It remains challenging to determine whether the transmission described in this article meets the requirements for modern machines and devices. Its structure and operational principle contribute to substantial power losses, as evidenced by low efficiency ($\eta = 1-58\%$ recorded during the tests). For comparison, belt-type, ball-type, and toroidal CVT transmissions achieve efficiencies above 90% [3, 8, 13]. In the author's opinion, slight improvements in performance can be achieved by altering the cam geometry and the number of active pairs of elements in the drive transmission.

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