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OPERATIONAL TESTS OF WORM GEARBOX WITH ZK2 CONCAVE PROFILE

BADANIA EKSPLOATACYJNE PRZEKŁADNI ŚLIMAKOWEJ Z WKLĘSŁYM ZARYSEM ZK2*

The article presents an operational tests of worm gearboxes. Test bench trails were conducted for three gearbox types. Two of these gearboxes were manufactured using modern methods with conical endmills. The only difference between the two is the tooth profile. A ZK2 worm with a concave tooth profile and Archimedes' screw was used in the gearboxes. The third analyzed gearbox was a commercial gearbox with a ZK1 worm. When comparing the results of the analysis, the efficiency and load carrying capacity of the ZK2 worm gearbox is the highest and greatest respectively. The higher load carrying capacity of the ZK2 worm with concave teeth in comparison to the Archimedes' screw is confirmed by Hertz's theory. The results show, that the meshing area for ZK2 worm gearboxes is greater than Archimedes' screw. The confirmed increase of usage indicators of concave profile worm gearboxes can lead to their widespread production and application. The higher efficiency of the gearbox results in lower usage costs.

Keywords: *worm gearbox, worm, wormwheel, ZK2 concave profile.*

W artykule przedstawiono badania eksploatacyjne przekładni ślimakowych. Badaniom stanowiskowym poddano trzy przekładnie. Dwie z nich zostały wykonane nową technologią z wykorzystaniem stożkowych narzędzi trzpieniowych. Różnica pomiędzy nimi dotyczyła wyłącznie zarysu kół. Zastosowano przekładnie ze ślimakiem ZK2 o wklęsłym zarysie oraz ślimakiem Archimedesa. Trzecią badaną przekładnią była przekładnia handlowa ze ślimakiem ZK1. Z porównania otrzymanych charakterystyk wynika, że sprawność i obciążalność przekładni ze ślimakiem ZK2 jest najwyższa. Wyższa nośność przekładni z wklęsłym zarysem ZK2 w stosunku do zarysu Archimedesa znajduje potwierdzenie w teorii Hertza. Uzyskane charakterystyki pokazują, że obszar zazębienia dla przekładni ze ślimakiem ZK2 jest większy w porównaniu z przekładnią Archimedesa. Potwierdzony wzrost wskaźników eksploatacyjnych przekładni ze ślimakiem o zarysie wklęsłym może przyczynić się do powszechnej ich produkcji i stosowania. Wyższa sprawność przekładni to zarazem niższe koszty jej eksploatacji.

Słowa kluczowe: *przekładnia ślimakowa, ślimak, ślimacznica, zarys wklęsły ZK2.*

1. Introduction

Worm gearboxes belong to a group a screw transmissions with non-intersecting axes. In contrast to different types of gear transmission, they are characterized by their ability to transfer large ratios under beneficial conditions with high loading in a compact form. The kinematics in contrast to other types of transmissions differs because of the high level of meshing slip caused by concurrent meshing of a greater number of teeth. Due to the high loads carried and the type of meshing in the gear set, special attention must be paid to the phenomenon that occur during their operation, especially those that have a significant effect on wear [3, 5, 15, 19]. The main types of wear in the case include: teeth breaking, fatigue cracking and abrasion. Bending of the worm and the heat generated during operation are also considered in a wider scope due to their effect on transmission efficiency. Among the factors that affect wear in a correctly designed and used worm gearbox, the most significant ones are worm wheel tooth abrasion and surface fatigue wear. The factors that affect abrasive wear besides load include, slip direction and velocity between teeth and the surface finish of the gearwheel teeth.

The widespread use of worm gearbox has resulted in an increased amount of research being done on the topic. Several publications concern the material aspects and their effect on the gear set meshing conditions. Due to the significant role of slip in worm gear teeth, it is necessary to use materials that ensure a low coefficient of friction for worm wheels and worm screws. Fontanari and coauthors [14] present a tribological wear mechanism of a gear set made of steel-bronze. The resulting research shows that the identified wear occurrences are dependent on the applied load to the transmission. Additionally, Fontanari and coauthors [13] describe the possibility of using gearwheels made of spheroidal iron and hardened steel. The authors observed changes in the destruction of samples that resulted from pitting. Both the method of lubrication and the microstructure of the material exhibited a strong influence on the initiation and propagation of fractures. On the other hand, Simon [25-27] analyzes the load distribution in a worm drive in a steel-bronze configuration. They suggest to discretely divide the adhesion line into small segments, which allows for calculating the stress distribution and improving the usage parameters.

Besides research on the topic material applications or wear simulation and stress distribution where the teeth mesh, multiple studies on the tooth shape and meshing analysis. Chen and Tsay [4] analyze the geometry and meshing of ZN profile in worm drives in contrast to ZA profiles using their own mathematical model. The developed mathematical model enables the testing of potential further analysis in the realms of sensitivity analysis, kinematic errors, and contact stress analysis. The resulting data is useful for designing. and generating and selecting operating parameters for the gearbox. A computerized approach for determining the contact surface and analyzing meshing in a Klingenberg parallel axes gear set was presented by Litvina and coauthors [17]. The present theory minimizes the error sensitivity re-

(*) Tekst artykułu w polskiej wersji językowej dostępny w elektronicznym wydaniu kwartalnika na stronie ***www.ein.org.pl*** sulting from not being coaxial. Dong et al. [10] describe a method of design a ZI worm screw while analyzing installation errors in a worm drive. The authors present a developed drive design method where they pay special attention to the parameters of the worm screw, which when selected in the correct range cause minimal translation of the contact line that result from installation errors. Tsay et al. [29] present a mathematical model ZE worm drive meshing that was prepared in a CAD environment.

Not only are issues related to geometry being covered in publications, methods of manufacturing are also being discussed. Unfortunately, the topic focuses solely around the use of hobbing tools for the manufacturing of worm drive gearwheels. The use of hobbing cutters in the manufacturing of a worm wheel is presented in the work by Fang and Tsay [12]. They suggested a mathematical model of a worm screw with a ZN profile based on the profile and machining parameters of a hobbing tool. Countless publications are dedicated to the machining of worm screws on universal CNC machines. Nieszporek and Boca [21] analyze the method of machining worm screws using spherical end mills. This approach permits for any profile regardless of the tool profile. On the other hand, Albu [1] and Albu and Bolos [2] developed an approach for creating tool paths for the machining of worm screws using cylindrical end mills on CNC lathes. Kacalak et al. [16] develops an methodology for analysis and modeling helical surface grinding processes using CAD/CAM systems and Matlab. The resulting methodology allows for conducting simulation tests in order to determine the precision of grinding while considering the positioning and geometric deviations of the set up and the run-out of the chuck and work.

Not including ZA, ZN, ZI, and ZK profiles, there are also concave profile worm screws used in worm drives [23]. Non-intersecting positioning of a tool with a linear profile enables the machining of this type of profile. Skoczylas and Pawlus [28] present a method of shaping worm screws with a concave profile by using special tooling. They demonstrated the superiority of a transmission with a concave profile worm screw over the traditional linear profile. In concave profiles, it is possible to transfer greater loads while maintaining lower k_H stress values on the surface.

There are practically no publications dedicated to testing the operating properties in worm drives. Czerniec et al. [8] present a method of calculating the effect of tooth correction of a worm drive with an Archimedes' screw on the contact strength, wear, and durability of the worm gearwheel teeth. The results of the study establish the correctness of the correction effect on the contact parameters and tribological contact. Czerwiec and Kiełbiński developed a method of analyzing the wear kinematics of a worm drive with an Archimedes' screw [7]. Based on the aforementioned method of testing the wear kinematics of materials due to slip friction, they present a method of estimating the lifespan of a worm drive with an Archimedes' screw. Based on the numerical solution, a relationship between the drive resource and wear was determined. Also, Czerwiec and Kiełbiński present a method of calculating the lifespan of a worm drive with a involute worm screw [8]. The result of the numerical solution was used to determine the characteristics of the dependency between the lifespan of the transmission and the linear wear of the worm gearwheel teeth. They determined that changes in the wear of worm gearwheel teeth along the profile, where the maximum contact pressure and slip velocity was. The effect of the module and diameter indicator on select parameters. In turn, Waqar and Demetgul [30] use Fourier transform and neural network to diagnose the damages of toothed elements in worm drives. Vibrations and sound waves that arise during the operation of the transmission are detected using sensors. The data is then used to teach the network. The use of neural networks for the prognosis of damages in the drive system is used by Shao et al. [24]. In order to reduce the variability of vibrations and the accuracy of counteracting the residual durability of the drive system, a method of predicting was proposed, which combines a neural network of radial base functions and recurrent initial processing. The results of the study show that the presented method can be used to optimize traditional prediction methods. The use of the research techniques is a effective way of extracting valuable operation properties. Early acquired information of ongoing degradation processes allow for planing service intervals and repairs correctly. Thus, improving the reliability of all of the elements in the kinematic chain. Developing methods that can be used for early identification of damage in the form of pitting of work surfaces, face chipping, tooth root cracking and partial fracturing was the topic of the paper by Łazarz et al. [18]. They conducted a study on the effectiveness of selected methods on the processing of vibroacoustic signals in the process of detecting faults in gearwheel with concurrent bearing damage of drive systems working in various conditions. Initially the converted vibration signals were analyzed in the framework of time and frequency to be the basis for developing a diagnostic metric that is sensitive to earlier tooth damage. Elforjani et al. [11] indicate that using acoustic emission techniques offers more diagnostic capability of worm drives during operation than vibration analysis. Monitoring the research has shown that the acoustic emission parameters and energy are more reliable, durable, and capable of detecting defects than the corresponding vibration parameters.

The literature covering worm drives consists of many titles that approach the issue in a purely theoretical approach. The papers in this field describe mathematical issues related with the analysis of meshing geometry of worm drives with various worm screw profiles. Very little attention is placed on the possibility of shaping concave profiles that demonstrate a significant level of usefulness when improving the operating parameters of drive system. The descriptions of the machining methods lack information regarding manufacturing issues related with the effects of surface and the precision of the machined work surface. There is also a very small number of publications that approach the issues of analyzing the operating properties of worm dirve with various worm screw profiles that has a significant effect on the durability and reliability of the codependent toothed parts.

2. The effects helical geometry of the worm screw on gearbox usage problems

When looking at fatigue, the factors that affect abrasive wear (load, gearwheel material) should include the curvature of the teeth and the length of the contact line. It can be noticed that some of the aforementioned factors are dependent on the geometry of the gear screw and gearwheel teeth. Improper manufacturing of the worm screw can lead to accelerated tooth wear of the worm gearwheel. The result can be observed in fig. 1.

Fig. 1. Wormwheel tooth damage resulting from excessive loads.

The shape of the teeth determines the length and position relative slip velocity of the contact line. This effectively changes the lubrication conditions in contact area, the efficiency and gearbox wear

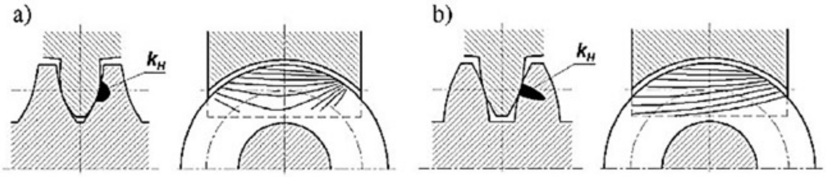

Fig. 2. The meshing of worm gearbox teeth for: a) concave profiles b) convex profiles.

rate. The shape of the teeth also affect the substitute radius curvature, which is an important parameter because of the large contact stress. Thus, the correct shape of the teeth can significantly affect the usage parameters of a gearbox.

The ability to modulate the shape of the helical surface of the worm screw is greatly increased by the non-intersecting positioning of the end mill in relation to the axis of the worm screw. The shape of the profile is the result of proper tool positioning. The advantages of a concave profile in comparison to a convex one are presented in fig. 2

The greater load carrying capacity of a concave profile worm screw results primarily from the lesser contact stress values on the tooth surfaces of the gearwheels. In addition, the large angle between the contact line and the circumferential speed of the worm screw is beneficial for lubricating the contact area.

The maximum stress resulting from cylinder contact can be calculated using the following formula (1) [20]:

$$k_H = \frac{F_N}{2L\mathrm{r}_{zr}} \tag{1}$$

where: F_N – normal force at the contact point, L – contact line length, ρ_{zr} – reduced curvature radius contacting the surface.

Based on this and when the normal forced is accepted, the magnitude of pressure depends on the product of the contact line length and the reduced curvature radius of the cylinders. The area of contact of gearwheels in the transmission can be considered to be very variable. The contact line of the teeth, along with the change of the curvature radius, will change its shape and length. However, the value of reduced curvature radius change along the analyzed contact line. The load carrying capacity will be determined by the minimal value of the aforementioned parameters (calculations are required for the whole meshing range).

3. Meshing characteristics of the analyzed worm screw profiles

Test bench studies were preformed for three gearboxes. Two of them were manufactured using new techniques that use conical end mills. The difference between these gearboxes is limited to the profiles of the gearwheels (gearboxes with concave ZK2 worm screw profiles and an Archimedes' screw). The third gearbox was a commercial transmission with a ZK1 worm screw. The basic parameters of the analyzed gearwheels, in accordance with their respective standards[9, 22], are listed in Table 1.

The decision to analyze ZK2 profiles results primarily from the large possibility of a non-intersecting tool position affecting the thread profile of the worm screw, including shape of concave profiles. Assuming a constant worm screw axial angle profile, calculations were done for the concave ZK2 worm screw profile with the parameters listed in Table 1. The tool parameters and its position angle in specific shaping cases along with the helix angle and thread curvature angle on the pitch diameter of the resulting worm screw are presented in Table 2.

Table 1.
Analyzed worm drive gearwheel parameters.

#	Parameter	Value
	worm gear	
1	Axial module	4 mm
2	Diametrical indicator	10
3	Thread start count	1
4	Tip height factor	1
5	Root height factor	1.2
6	Axial angle profile	20°
7	Helix angle	5.7106°
	Wormwheel	
8	Number of teeth	30
9	Tooth correction factor	0
10	Tip height factor	1
11	Root height factor	1.2
12	Tooth heigh	2mm
13	Width	30mm

Table 2. Tool parameters and positioning angles.

Profile type	Tool				Worm screw
	α_N	d_N	ϕ_N	χ_N	α_o
	[°]	[mm]	[°]	[°]	[°]
Concave	8	5	-36	9,92	20
Linear	8	5	12	90	20

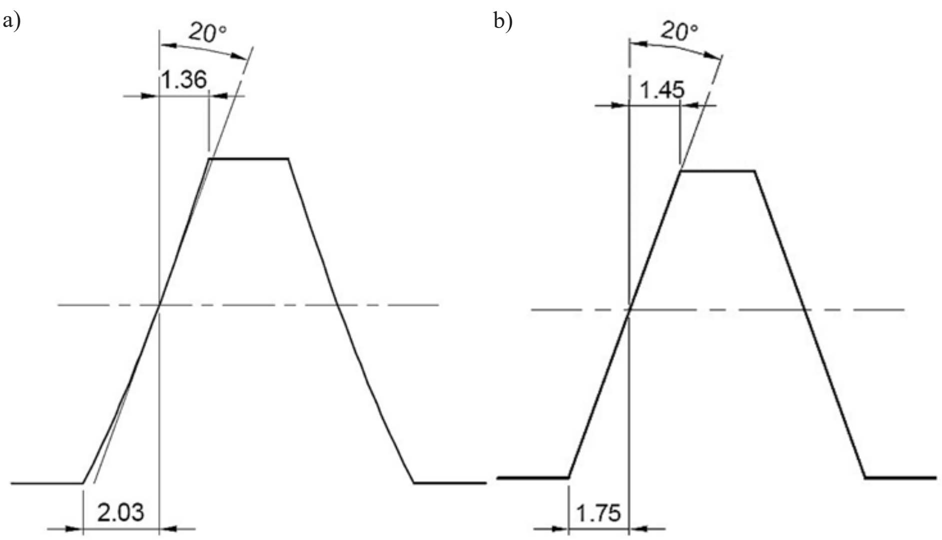

Fig. 3. Axial tooth profile: a) concave b) linear

Positioning the tool in regards to the axis of the worm screw was done to achieve profile that is as concave as possible and to avoid undercutting during meshing with the gearwheel. In order to more precisely visualize the change in shape of the ZK2 profile, fig. 3 presents the concave profile with a Achimedes' profile (ZA). Figure 3a show the visible difference in shape between the concave profile in comparison to the linear profile displayed in fig. 3b.

The most characterizing dimensions are the tip diameters and root diameters. In neither of the aforementioned cases does the thread profile angle match the angle of the tool of 20°.

4. Materials and gearbox loading analysis

4.1. Test bench

The complexity of the phenomena describing the meshing of worm drive results in loading tests being the most objective evaluation of the effect of worm screw profile on the operating parameters. In order to accomplish this, a test bench that assessed actual worm drive was prepared. The block diagram representing this setup and test bench are presented in fig. 4.

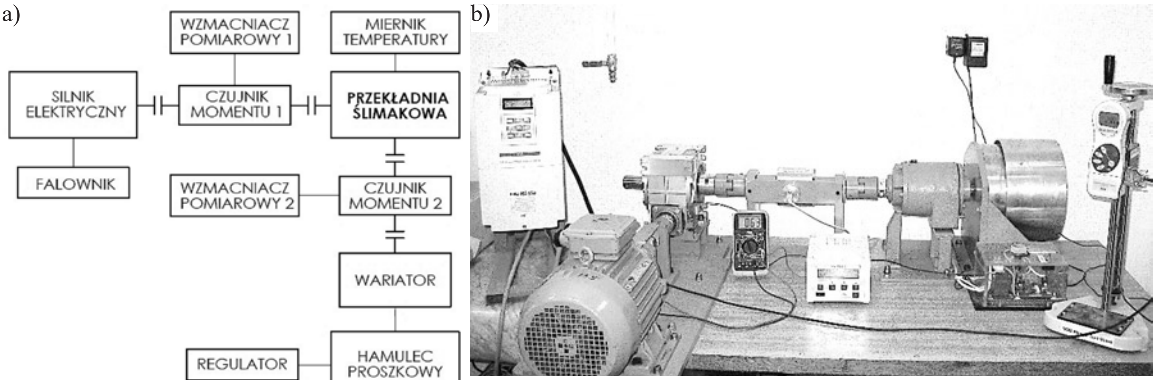

Fig. 4. Gearbox loading test bech: a) block diagram b) actual view

The configuration of the test bench assumes that only the gear set geometries will be variable during the tests. The propulsion and measurement systems were selected in order to appropriate to the size of the gearboxes. The commercial housing was modified to enable visual observation of the gearwheel condition. The housing of the gearbox (fig. 5) is made from aluminum with an axis base of 80mm. The housing material is a standard choice for mass produced gearboxes of this size.

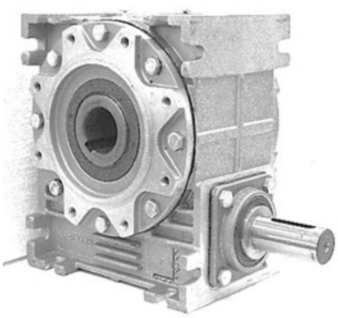

Fig. 5. Analyzed worm drive

A 4kW 3-phase induction motor with nominal speed of 2815 RPM was selected as the propulsion unit of the test bench. The motor control was done using an inverter. The inverter allows for linear scalar control (U/f) as well as sensorless and sensor based controlling of torque and speed. The load was provided by a powder brake with dispersion power of 2kW. The torque of the brake is proportional to the voltage of the electromagnetic coil that is regulated by an electronic system. The brake properties enables constant work with slip that allows for long term tests. Due to the fact that resulting speed out of the worm drive was lower than the recommended operating speed of the brake (50 RPM), the test bench was equipped with an additional gearbox that functioned as a variator. All of the shafts in the drive and loading systems were connected using elastic Rotex clutches (KTR).

The torque before and after the worm gearbox was measured on the test bench as well as the operating temperature. Measuring the torque enables the monitoring of the load levels on the gearbox and limiting power loss. A measuring shaft with a torque sensor was using to measure the rotational moment of the gearbox. Connecting both measuring shafts with a computer allows for electronic registering of torque values before and after the gearbox. The operating temperature measurements were done using a resistance sensor located on the housing of the gearbox.

The described test bench is prepared primarily for tests that determine the size of a worm drive, where the modular design permits quick modifications for testing different gearboxes.

4.2. Research object

Manufacturing the gearwheels makes it possible to conduct experimental gearbox tests. Two worm screw profiles were manufactured and both were shaped using cylindrical end mills. The worm screws were made of 42CrMo4 steel intended for heat treating. The chemical composition of the selected steel alloy is presented in Table 3. This steel is intended for the manufacturing of machine parts with high strength, ductility, and variable loading such as: axles, cranks, gearwheels, discs, rotors, levers and other similar items.

Table 3. Alloy composition [%] of 42CrMo4 steel.

C	Mn	Si	P	S	Cr	Ni	Mo	W	V	Co	Cu
0,38 – 0,45	0,4 – 0,7	0,17 -0,37	max 0,035	max 0,035	0,8 – 1,2	max 0,3	0,15 – 0,25	max 0,2	max 0,05	-	max 0,25

The manufacturing process included centering, roughing, shaping, heat treatment, re-centering, and finishing of the teeth and cylindrical surfaces for bearing seating and clutch fitment. The heat treatment included hardening that was done by heating the metal to 840°C and quenching in oil and annealing at 550°C followed by oil quenching. The resulting hardness was 55HRC. The finishing was done on a five-axis Haas VF-2 mill.

The meshing of concave and linear profile worm screws requires the manufacturing of two worm gearwheels. The blank for the worm gearwheels was heterogeneous and made of a tin-phosphorus bronze ring cast on a iron hub. The chemical composition of the material is presented in Table 4.

Table 4. Alloy composition[%] of CuSn10P bronze.

Sn	Pb	Mn	Al	P	Zn
9,0 – 11,0	–	–	–	0,8 – 1,2	–

The manufacturing process was mostly traditional besides the tooth cutting operation. The tooth cutting was done on a five-axis Haas VF-2 mill with universal cylindrical and spherical end mills.

4.3. Preparing the test bench

The assembled gearboxes, after filling it with Synlube CLP 220 oil, underwent further test on the loading test bench. The effect of tooth profile on the operating parameters of the gearbox were assessed, where the primary indactor was selected to be gearbox efficiency. The test bench trials were conducted for one gearbox size with an axis base of 80mm. The selected gear set parameters enabled testing in a commercial gearbox housing.

The efficiency tests were also done for the commercial gearbox. The aim was to compare the results of the manufactured gear sets with the mass produced gearbox with comparable parameters. The gear ratio of the commercial gearbox was 31 using a ZK1 worm screw profile. Efficiency testing was done following a break in period of 400 hours. This process was conducted at 1000 RPM and load that results in operating temperatures around 65°C. The efficiency measurements were done at constant worm screw RPM of 1400 RPM and ambient temperature of 23°C. The load level was gradually increased. After every change and when the temperature stabilized, an efficiency measurement was done. The trails were ended when the gearbox oil temperature reached 110°C.

5. Results

The efficiency test results (solid line) and temperature measurements (dashed line) of the gearbox is presented in fig. 6.

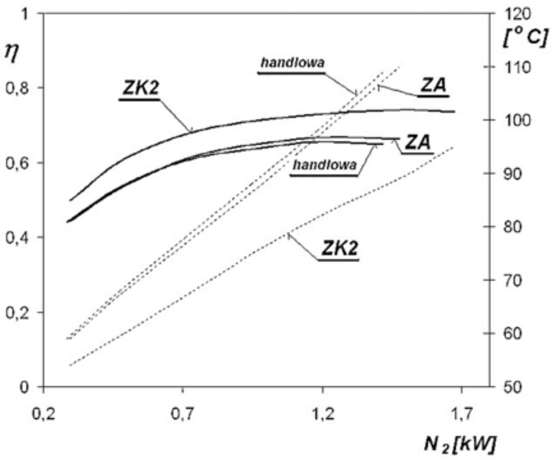

Fig. 6. Gearbox efficiency as a function of load

Comparing the characteristics of the graph (fig. 6), it shows that the efficiency and load carry carrying capacity of the ZK2 gearbox is the greatest. This phenomenon was noticeable during the break in period, when the gearbox with the concave profile had 20% greater load while maintaining the same temperature levels. The efficiency of the commercial gearbox in comparison to the gearbox with Archimedes' screw, besides the similarities of both worm screws, is significantly smaller. This probably results from lesser stiffness of the commercial worm drive, which due to the high gear ratio had a smaller diameter. The temperature characteristics show that the main limiting factor of the Archimedes' screw and commercial gearboxes was temperature. Only the concave profile gearbox did not reach the temperature barrier and could undergo greater loads. Further tests exceeded the possibilities of the brake. The test range was sufficient, which can be observed by the extremes in efficiency.

The greater load carrying capacity of the ZK2 concave profile in comparison to the Archimedes' screw can be confirmed by Hertz's theory. Using a specially developed program, calculations were done to find the product value of the contact line curve and reduced curvature radius of the gearbox teeth. The results of these calculations are presented in fig. 7.

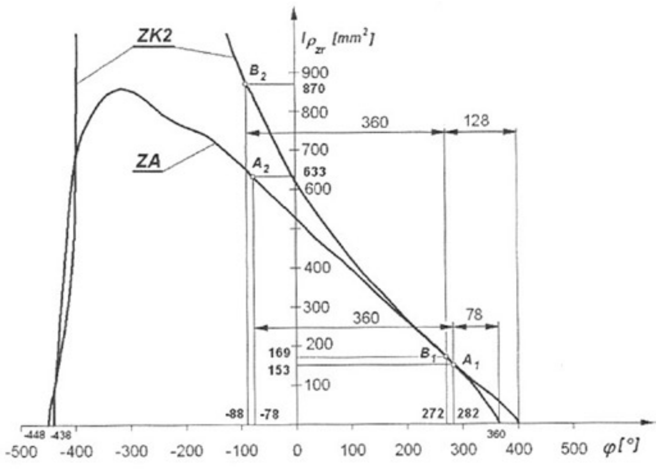

Fig. 7. The dependency of contact line length and curvature radius from worm screw rotation

To maintain the readability of the graph, part of the ZK2 parameter product values were excluded because they reached a few thousand. In the area of decoupling the separation between the lines in lesser, and even comparable. The resulting characteristics show that the meshing area for ZK2 worm screw gearbox is greater that Archimedes' screw drives. The complete meshing indicator for ZK2 is 2355 while ZA is 2216. In relation to the load carrying capacity, the greatest pressure occurs at points with two pair engagement with the lowest contact line length and reduced curvature radius product. For ZA gearbox these are A_1 and A_2, while B_1 and B_2 for ZK2. The total product value for the ZA worm screw is 786mm^2, while it was 1039mm^2 for the ZK2 worm screw. In conclusion, this illustrates the greater load carrying capacity of the ZK2 worm screw compared to the ZA worm screw.

6. Conclusion

Besides long standing use of worm drives, the majority is manufactured using a profile that is easy to shape. These are mostly involute worm screws or conically shaped ZK1 that have a helical thread shaped by a hobbing tool with a linear profile.

The trails show that convex profiles do not ensure a gearbox with maximum power transmission or efficiency. This evidenced by the greater efficiency and load carrying capacity of new concave ZK2 profile worm screws, which illustrate their usefulness. It should be noted that the test bench trails were conducted on a basic variat of the gearbox. Modifying the parameters of the gearbox can additionally affect an increase in usage indicators, which require further experimental trails.

The confirmed increase in usage parameters of the concave worm screw gearbox can lead to its widespread production and use. The higher efficiency of the gearbox results in lower operating costs.

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