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Vibroacoustic testing of prototype hermetic harmonic drive

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Abstract

This study was designed to determine the vibroactivity parameters of a hermetic harmonic drive. A specially-prepared test bench was used to measure the normal velocity of vibrations and acoustic pressure generated by the unit. Piezoelectric sensors were applied to measure the characteristic values on the body of the prototype. In selected sections of drive, we determined the effective values of acceleration a_e and frequency f. For the same points, the effective values of vibration velocity V_c were determined for the corresponding frequencies. The findings presented in this paper enable the assessment of a toothed gear in terms of the quality of its workmanship and wear-related deterioration, which are very important due to the characteristic work of this special drive.

Introduction

Harmonic drives (Figure 1) are a specific type of gearing, not only because of their characteristic construction, but also because of their unique system for transferring loads. The main components include a flexspline, circular spline, and wave generator, and the basic layout of such a gearing is shown in Figure 2. The operation of a harmonic drive is based on the principle that the relative movement of the cooperating rings is caused by elastic strain in one of them. The ring deformed by the wave generator is called the flexspline (2), and the cooperating ring with teeth on the inside is called the circular spline (3). The rotation of the generator (1) produces a wave of deformations along the perimeter of the flexspline, whereby movement is transferred to the circular spline. The toothed rims of the flexspline and the circular spline in the harmonic drive come in contact only at two places, and away from these the teeth, the two rings do not mesh (Lasocki, 1986; Mijał, 1999; Ostapski, 2011). Each of the three basic elements of the assembly may function as a driver or as a follower, and the system may operate as a reducer, multiplier, or differential.

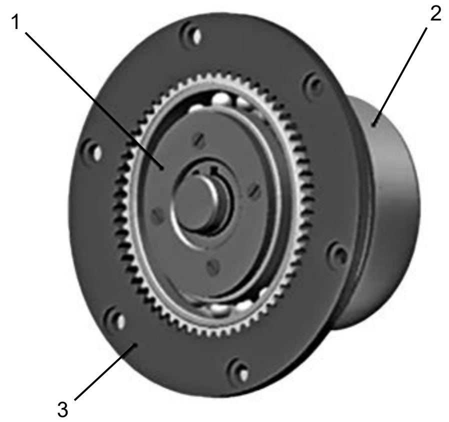

Figure 1. Solid model of harmonic drive; 1 – wave generator, 2 – flexspline, 3 – circular spline

Despite significant technological and operating difficulties, harmonic drives are used in the aviation industry and cosmonautics (Ueura & Slatter, 1999), and they are also increasingly used in automatic control engineering, robotics, as well as in optical, machine-tools, and precision industries (Bompos et al., 2007; Zhang & Yan, 2010; Gravagno, Mucino & Pennestrì, 2016). The fact that they are used in such highly specialised areas is associated with the unique characteristics of this type of transmission. Their unquestionable advantages include a potentially high gear ratio per one degree, high-precision kinematics, the low sliding velocity of the meshing teeth, low bending forces applied to shafts, as well as noiseless operation. Furthermore, only harmonic drives can transmit power between two separate working areas while maintaining tightness, which is achieved by hermetic transmission (Pacana, Witkowski & Mucha, 2017), which simultaneously retains all of the above advantages of harmonic drives.

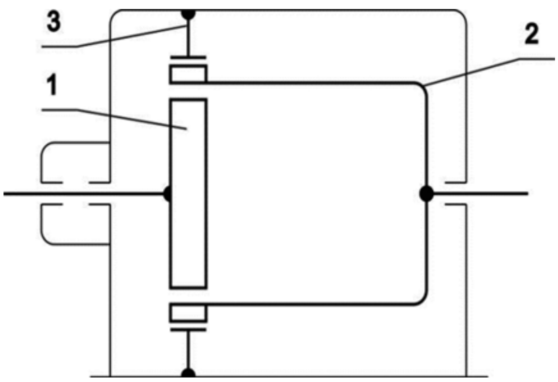

Figure 2. Layout of a strain wave reducer (PN-90/N-01358); 1 – wave generator, 2 – flexspline, 3 – circular spline

In harmonic drives, the load is simultaneously transmitted by 20–50% of the total number of flexspline teeth (Pacana & Budzik, 2004; Kalina, Mazurkow & Warchoł, 2017); therefore, a significant working pressure is possible, despite the relatively small dimensions of the assembly. On the other hand, as a result of such a design, the working conditions near the enmeshment are more demanding than in any other types of transmission. During operation, the teeth move not only along the perimeter of the ring but also in the radial direction, and they are turned relative to the axis due to the cyclic deflection of the flexspline.

Novel applications appearing in recent years are associated with increasing requirements defined for harmonic drives and are related to rotational accuracy and fluidity, as well as load capacity. Indeed, the fact that they are used in controls, aviation and robotics proves that assemblies of this type are characterised by good kinematic accuracy. Harmonic drives must also comply with strict requirements related to quality and wear resistance because of the specific range of applications in aircraft and spacecraft controls (Ueura & Slatter, 1999; Krishnan & Voorhees, 2001).

The process of designing harmonic drives is highly complicated due to the exceptionally difficult operating conditions. Analytical calculations and computer simulations based on simplified models do not provide sufficient evidence to determine accuracy of the structures; therefore, it is necessary to perform bench tests using functional prototypes that represent actual operating conditions as accurately as possible (Dudley, 1962; Jeon & Oh, 1999; Li, 2016; Pacana & Markowska, 2016). Experiments involving harmonic drives are most commonly applied to determine their strength, wear-resistance and kinematic accuracy (Folęga, 2015; Gravagno, Mucino & Pennestrì, 2016). In the case of toothed gears, field tests examining their vibroactivity are also extremely important. Forced effects produced by joint operation of toothed wheels are transmitted by shafts and bearings onto the drive housing, generating high levels of vibroacoustic power (Wieczorek, 2008). Dynamic phenomena occurring in the enmeshment area are the main causes of vibrations that affect the gear housing. These, in turn, lead to a decrease in the wear resistance, reliability and accuracy of the machines and equipment whose operation is affected by them (Krishnan & Voorhees, 2001). These effects are important in the case of machine tools, since they ultimately impair the quality of items produced, and, in the case of robots, conveyors and manipulators, since they interfere with the positioning accuracy.

The level of the vibroacoustic signal depends on the accuracy with which the gear was manufactured, as well as the degree of its deterioration; therefore, measurements of the characteristic components of the forced effects enable the assessment of a gear unit in terms of its operating accuracy and deterioration level (Cempel, 1991; Folęga, 2010). By performing *in-situ* measurements of vibrations and noise generated during the real-life operation of toothed gearings, it is possible to determine whether these hazardous factors are present, and what effects they have on people.

Test bench and range of measurements

Vibroacoustic tests were performed to measure the noise and vibrations generated by a prototype hermetic harmonic drive, designed and built earlier. The purpose of the study was to identify the strength of the vibroacoustic signal and to determine the source of forced effects.

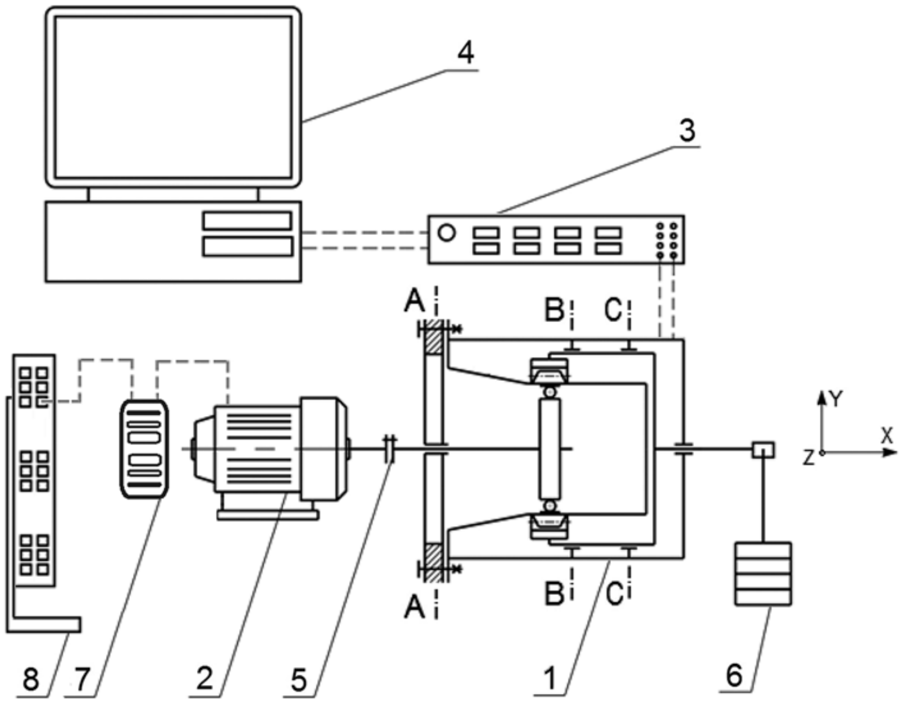

Figure 3. Layout of the station designed for testing a hermetic harmonic drive: 1 – hermetic harmonic drive, 2 – direct current motor, 3 – measuring tool, 4 – PC-type computer, 5 – coupling, 6 – external load, 7 – autotransformer, 8 – control panel

The layout of the open test bench, designed and built for the needs of the study assessing the vibroacoustic properties and load capacity of the hermetic harmonic drive is presented in Figure 3. The harmonic drive (1) is powered via a coupling (5) by a direct current motor (2) with regulated rotation. A straining

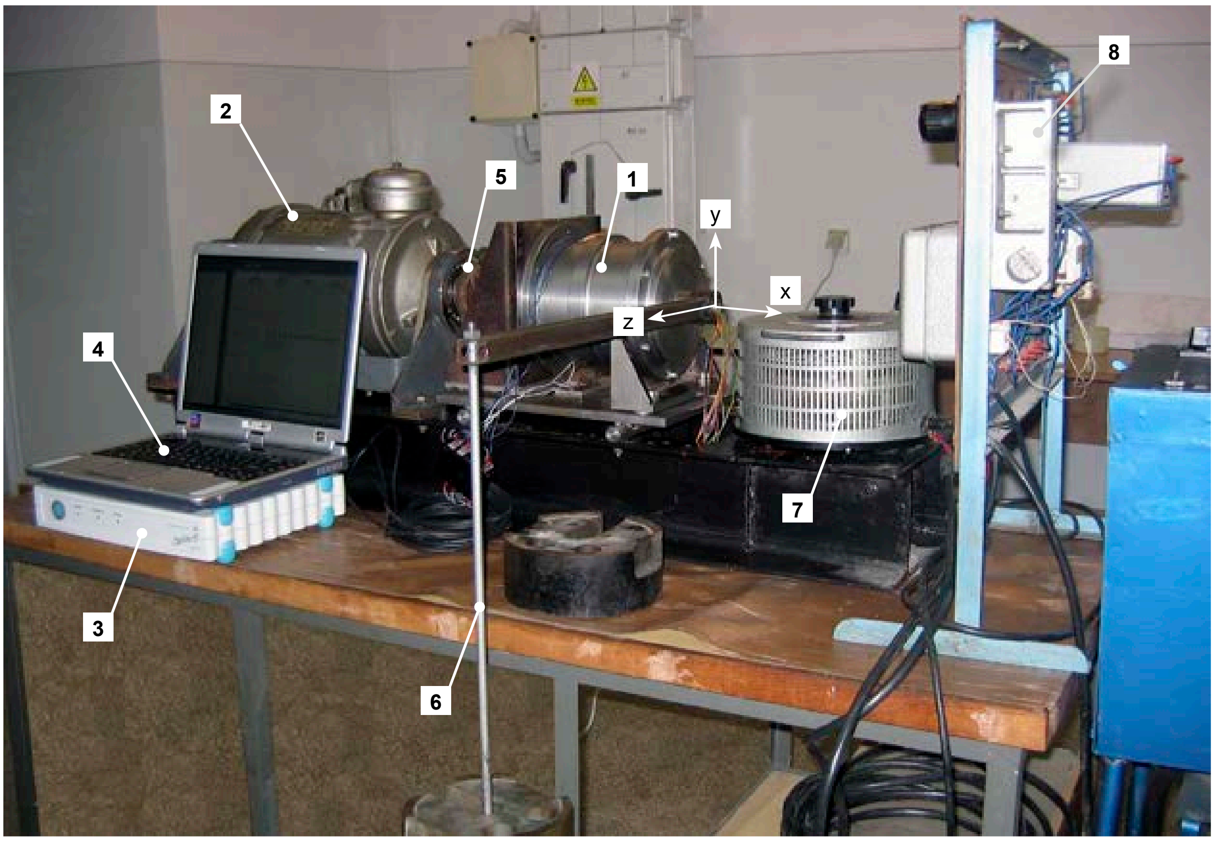

Figure 4. Station designed for testing a hermetic harmonic drive (elements numbered the same as in Figure 3)

torque loading is implemented using weights positioned on the lever (6) mounted onto the output shaft. The test bench also includes a measuring tool (3) and a PC-type computer (4) for the operation of the station and recording of the results.

A view of the test bench during measurements of the hermetic harmonic drive is shown in Figure 4. The tested drive transmits a torque of 150 N·m and a gear ratio of 58. It has a two-wave generator and a flexible toothed rim with a modulus of 0.7 mm and 114 teeth. Besides vibroacoustic testing, the station enables tensometric measurements of the flexspline body at different velocities of the generator rotation and different values of the harmonic drive loading. The measurement results can be followed on the computer screen as they are taken, and they can be recorded for future analysis.

Mechanical vibrations of gear units should be determined in compliance with the guidelines specified in PN-ISO 8579-2. According to the above standard, as well as PN-90/N-01358, gear housing vibrations should be measured on rigid parts of the housing at all accessible bearing locations of the gear unit. Measurements should be performed in three mutually perpendicular directions, two of which are located in a horizontal plane. The following requirements should be met during the measurements:

- gear unit testing should be performed during its operation at its designed speed;
- the direction of rotation should be the same as that designed for operation;
- gear unit should be tested with no load, or with a small load required for proper operation;
- lubrication should be applied during tests;
- measurements should be performed when the gear unit operates at the temperature range defined by the engineering design.

The measurement tool should have the capacity to measure shaft displacement frequencies in the range from 0 Hz to 500 Hz, as well as higher tooth mesh frequencies ranging from 10 Hz to 10,000 Hz. The read-out instrument should be checked against a reference signal before the test is initiated.

The above standard recommends measurements of the effective value of vibration velocity V_c from 10 Hz to 10,000 Hz, split into frequency components using a fast Fourier transform. However, if it is impossible to read the frequency spectrum data, the tests may apply unfiltered velocities of housing vibrations, which should be compared with the permissible values for gear units of a given class.

The vibrations measured on the gear housing may be assessed by comparing the measured values with normative values, which makes it possible to determine the condition of the unit and its class.

Measurement process

Vibroactivity was measured using piezoelectric sensors, while a probe amplifier and digital oscilloscope from Tektronix were used to process and read out the results. The oscilloscope enables the measurement of timelines, and after signal conversion, it determines the frequency spectrum. The measurements were performed in conformity with the requirements presented above, at a generator rotational speed of 600 rpm. Frequency components were measured from 0 Hz to 12,500 Hz, which is a wider range than defined by the standard.

The measurements were performed in three mutually perpendicular directions (X, Y, and Z), on the housing of a hermetic harmonic drive, at cross-sections A, B, and C as illustrated in Figure 3. The locations were selected for measurements because they were near the roller bearings supporting the flexspline and generator. Effective values of vibration acceleration a_e and frequency f were read at the defined points for an undivided signal. Subsequently, the program started in the oscilloscope converted the complex vibrations into frequency components. For the whole duration of the measurements, the gear unit operated at a constant speed under the same conditions and was not switched off. The measurements of the acceleration of complex vibrations were only used to verify the results obtained after the signal was divided into frequency components. The entire measurement cycle was performed three times, and all measurements were recorded.

The effective values of vibration velocity V_c were determined for the corresponding frequencies at each point of measurement. The mean analysis results of the first five frequency components of vibration at the predefined locations of the hermetic harmonic housing drive are shown in Tables 1–3.

It is evident that the fundamental frequency component, at all points of measurement, is in the range from 500 to 600 Hz, and the corresponding vibration velocities were significantly greater than the velocity at the remaining frequencies. Greater vibration velocities were also observed near a frequency of 3500 Hz. Vibrations in the higher-frequency band may be produced by the meshing of the splines of the examined drive unit, while fundamental frequency vibrations were determined by the rotating input shaft.

If the identified effective values of vibration velocity are compared with normative criteria, the

Table 1. Results of vibration measurements at cross-section A of the hermetic harmonic drive

Vibrations in cross-section A					
Towards axis X		Towards axis Y		Towards axis Z	
Frequency	Velocity	Frequency	Velocity	Frequency	Velocity
f [Hz]	V_c [mm/s]	f [Hz]	V_c [mm/s]	f [Hz]	V_c [mm/s]
580	0.445	610	0.360	600	0.375
2220	0.000	2705	0.006	2250	0.002
3580	0.030	3570	0.009	3600	0.039
3905	0.012	4200	0.008	4000	0.027
7320	0.000	6400	0.010	6750	0.000

Table 2. Results of vibration measurements at cross-section B of the hermetic harmonic drive

Vibrations in cross-section B					
Towards axis X		Towards axis Y		Towards axis Z	
Frequency	Velocity	Frequency	Velocity	Frequency	Velocity
f [Hz]	V_c [mm/s]	f [Hz]	V_c [mm/s]	f [Hz]	V_c [mm/s]
570	0.616	500	0.857	540	0.512
2450	0.009	2490	0.035	2500	0.014
2800	0.031	3450	0.080	3280	0.009
4750	0.023	4650	0.009	4600	0.021
7000	0.022	6200	0.000	6900	0.002

Table 3. Results of vibration measurements at cross-section C of the hermetic harmonic drive

Vibrations in cross-section C					
Towards axis X		Towards axis Y		Towards axis Z	
Frequency	Velocity	Frequency	Velocity	Frequency	Velocity
f [Hz]	V_c [mm/s]	f [Hz]	V_c [mm/s]	f [Hz]	V_c [mm/s]
540	0.775	600	0.440	550	0.710
1300	0.035	1200	0.034	1250	0.037
3500	0.067	3500	0.097	3500	0.061
4800	0.004	4700	0.003	4700	0.001
5700	0.000	5800	0.001	5500	0.001

hermetic harmonic drive may be classified in the VR3.15 category. This is the lowest class defined by the standard, and the symbol shows that the values of vibration velocity V_c of the gear unit from 45 Hz to 1590 Hz did not exceed 3.15 mm/s.

Additionally, based on other normative recommendations (in PN-90/N-01358) the condition of the hermetic harmonic drive in question may be assessed by taking into account the level of the generated vibrations, and it may be classified in "Group I", which includes machines with a power up to 15 kW. Based on the highest measured values of vibration velocity, $V_c = 0.857$ mm/s, its operating condition can be considered satisfactory. "Group I" gear units, whose operating condition is defined as 'good', must achieve vibration velocity values lower than $V_c = 0.71$ mm/s, but this requirement was not met in the case under consideration. However, the vast majority of the vibration velocity components, as well as the mean values in the three directions taken into account, were lower than the threshold value. Given this, the general operating condition of the relevant hermetic harmonic drive should be rated as good.

Conclusions

The article presents the findings of bench tests involving a prototype hermetic harmonic drive. The tests investigated its vibroactivity under real-life operating conditions, under a nominal load. The effective values of vibrations a_e and frequency f were measured at key locations on the gear unit housing. The findings definitively show that the highest vibration velocities occurred at the bandwidth corresponding to rotations of the input shaft. Increased vibration velocities were also observed for one other higher frequency, and these were caused by the meshing of the teeth in the splines of the relevant transmission unit.

Measurements were performed in real-life operating conditions (*in-situ*) for an unloaded and loaded hermetic harmonic drive. Loading of the gear unit slightly increased the value by approximately 5%. The identified values were significantly lower than the corresponding values in gear units of this type approved for use in laboratory conditions.

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