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Analysis on the measurement results of the precision grades and their influence on the performance of the hydraulic pumps with spur gears

I G Ghionea[1], A L Ghionea[2], S Ćuković[3] and M I Pleşa[4]

[1] Manufacturing Engineering Department, University Politehnica of Bucharest, Faculty of Industrial Engineering and Robotics, 313 Splaiul Independentei, Bucharest 060042, Romania

[2] Machine and Production Systems Department, University Politehnica of Bucharest, Faculty of Industrial Engineering and Robotics, 313 Splaiul Independentei, Bucharest 060042, Romania

[3] Institute for Information Technologies, University of Kragujevac, Jovana Cvijica street, Kragujevac 34000, Serbia

[4] Doctoral School of Faculty of Industrial Engineering and Robotics, University Politehnica of Bucharest, 313 Splaiul Independentei, Bucharest 060042, Romania

ionut.ghionea@imst.pub.ro

Abstract. The paper presents results of experimental measurements on the main precision grades for a cylindrical spur gearing, from the assembly of a hydraulic pump integrated in a hydrostatic driving system. For the components of this assembly, the main stages of the design, material characteristics, constructive, precision and functional parameters that define the flow, the pressure of the hydraulic environment, the driving speed are briefly presented. The analysis was performed based on the data obtained by measuring the gears' deviations and tolerances. Among these, the most important data for complying the optimal requirements of the pump operating criteria were retained and analysed. The analysis of the measured results highlighted the main causes and effects of the deviations found. The presented results show the problem's complexity and are important in evaluating the performances that characterize these types of pumps, with wide use in many fields of hydraulic actuators.

1. Introduction

Hydrostatic systems (of actuation, lubrication, cooling, etc.) have a wide use, corresponding to the technical and performance data required for different industrial fields, such as: construction of machine-tools, automotive, transport equipment and installations, etc. The basic component of these systems is the hydraulic pump (generator), that is manufactured in many constructive and functional variants for a wide range of flow rates and pressures [1].

Hydraulic pumps are used in most of the hydraulic power transmission systems. These pumps convert mechanical energy into hydraulic energy, that is a combination of pressure and flow using a specific constructive solution. Different types of hydraulic pumps can be integrated into a wide variety of industrial hydraulic machines, being used on machine-tools, presses (for shearing, stamping, bending), excavators, cranes, loaders, tractors, forestry equipment, mining machinery, injection moulding machines etc.

There are two main groups of pumps: with a constant flow and with an adjustable flow. The main characteristics of the pumps refer to: constructive typology, overall dimensions, flow rate, working pressure, driving speed range and torque, efficiency, acquisition costs of some components, operation and maintenance, reliability, etc. The technical performances are determined by the following requirements, namely: the maximum driving speed and pressure level, the geometric volume (capacity), reduced pulses of the flow, increased reliability, low costs per unit of power [2]. Flows, discharge pressures and driving speeds are specific parameters, with values within wide limits, corresponding to the requirements of the installation in which the pump operates. Thus, it is appropriate to manufacture the pumps in standardized series [3], on a catalogue basis.

The spur gear pump assembly, analysed in this paper, was chosen from a constructive-functional wide range of pumps with different uses. This studied pump is new/unused (only ran for testing on the stand) and it is part of a manufacturing lot from a Romanian producer. The analysis presented aims to identify the fulfilment of the manufacturing and precision requirements imposed by the use.

The pump respects a series of requirements regarding the operation of the spur gearing in conditions of variable loads, of low vibration and noise, durability, safety in operation and optimum efficiency [4, 5]. These determine specific conditions of design, manufacture and control [6] of the two gears, their shafts and bearings blocks, of the functional and assembly surfaces of the pump components.

2. Stages of design and manufacture of the spur gear pump

Generally, the development of new products and/or the improvement of existing ones are important research and design activities in any company in the field of industrial manufacturing. Thus, in these activities there are certain risks and unexpected events that may reduce the technical characteristics. The causes can be numerous: deviations from the technical documentation, errors of processing or of measurement, defects of material, wear of the cutting tools, different elastic deformations, vibrations in the processes of manufacturing, quality and viscosity of the hydraulic fluid, impurities, variations of temperature, etc. All these affect the operating behaviour and decrease the pump performance [7].

The improvement of the constructive variant and the choice of materials require the simulation of the pump design (figure 1) in the stages of: computer-aided parts and technological design, creation of virtual 3D models, identification of several variants for the same product (using parameterized design) [8, 9], correct choice of manufacturing technologies, control methodologies and assembly of components, analysis of functioning behaviour. Appropriately, the manufacturing techniques are optimized and the costs and delivery times are established.

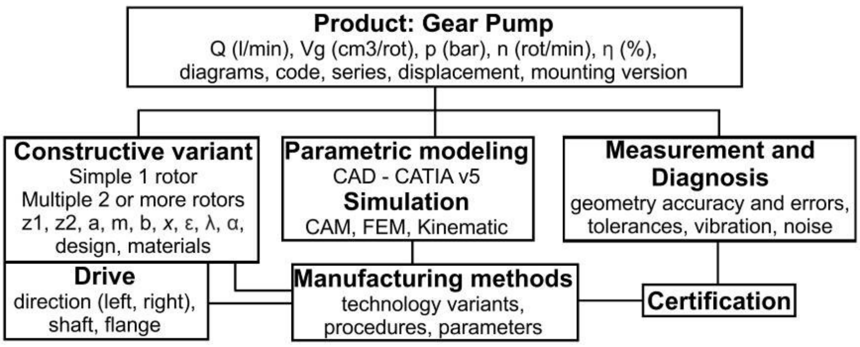

Figure 1. Design stages of the gear pump product

The main data that are involved in the parts and technological design stages are: geometric and operating parameters of the pump (Q_p - flow rate, V_g - geometric volume/displacement, p_p - working pressure, volume per one revolution, n_0 - speed, η - efficiency, other conditions and constraints, like: how is the end of the driving shaft, the shape and dimensions of the clamping flange, etc.). There are considered some constructive variants (with one, two or more pairs of gears, number of teeth, gear module, gear width, pressure angle). Generally, the pump is driven by a constant speed electric motor.

An important step in improving the performances of the pump is the optimization of the teeth flanks generation by: modification of the profile, flanking, rounding etc. [10] Thus, by parametric design, the 3D model of the gearing and of the pump assembly is obtained (figure 2). The applied methodology is based on numerous and complex parametric relations [1, 11] between certain dimensions that define shapes and positions of the pump components.

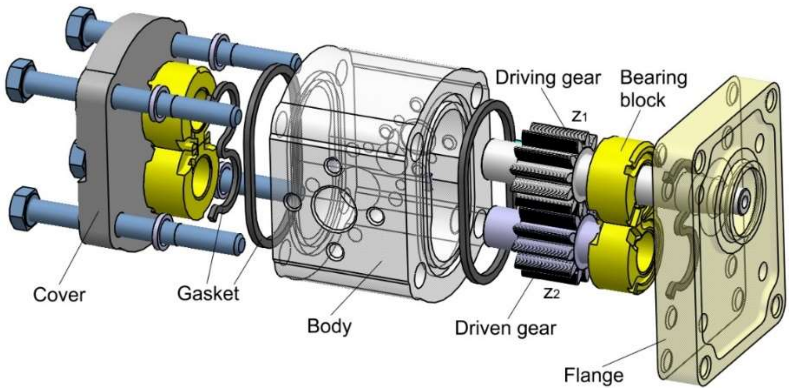

Figure 2. Parametric 3D model of the gear pump

The pump flow is provided by the size and number of gaps between the teeth of the cylindrical gear ($z_1 = z_2$) and its driving speed n_0 (rev/min). The hydraulic fluid is aspirated at the atmospheric pressure p_0 (bar) and discharged at a flow rate Q_p (l/min) and pressure p_p (bar) through the inlet, respectively, the outlet apertures of the pump body. The flow is determined by the geometric elements of the gears' teeth [5] supported by the bearings.

The 3D model of the pump in figure 2 was obtained using the following parameters: number of teeth $z_1 = z_2 = 12$, centre distance $A = 31.4$ mm, module $m = 2.54$ mm (normal specific displacements $x_n = 0.2$ mm pinion, $x_n = 0.199$ mm gear and frontal specific displacement $x_f = 0.2$ mm), gears width $b = 18.3$ mm, pitch circle diameter $d_d = 30.48$ mm, outside circle diameter $d_a = 37.413$ mm, rolling circle diameter $d_r = 31.4$ mm, base circle diameter $d_b = 28.642$ mm, pressure angle $\alpha = 20^0$, teeth height $h = 6.205$ mm, addendum $h_a = 1.261 \times m = 3.203$ mm, normal gear pitch p_n, normal arc of dividing the tooth/gap s_n/e_n, frontal contact ratio $\varepsilon = 1.494$, constant chord $s_c = 3.85$ mm. The parameter values are based on calculation methods [1, 7] and standards in the field [11]. These parameters should be found in the measurement process of the manufactured pump. Analysis of these data shows that the involute tooth gear is corrected.

High speeds, over $n_0 = 1500 \ldots 2000$ rev / min, lead to appearance of the cavity process, which reduces the pump flow, efficiency and also to gas occurrence in the hydraulic fluid.

The market demands and pump producers' concerns prove interest for the parametric design of pumps' series. Today's computer aided design techniques (CAD) offer many possibilities to edit the parameters in order to obtain other custom versions of the designed pumps. The validation of the pumps series is possible and is done by modern and complex simulations, such as: kinematics, FEM, CAM etc.

3. Measured data and results

Figure 3 presents the constructive solution of the pump type PRD2-223S, series 337 [3], having the geometric volume $V_g = 11$ cm^3 / rev, the suction pressure $p_a = 0.7\ldots 3$ bar, the nominal pressure $p_n = 250$ bar, the maximum allowable pressure $p_{max} = 280$ bar, minimum speed $n_{min} = 500 \ldots 1200$ rev / min, maximum speed $n_{max} = 3000 \ldots 3500$ rev / min.

The pump for which measurements were made (gears and shafts, body, bearing blocks) has an external cylindrical spur gearing with an involute profile [1, 10], the body is provided with two apertures: inlet for suction the fluid at the atmospheric pressure p_0 and outlet to discharge it in the hydraulic installation at the nominal pressure p_n. The bearing blocks have also the role of closing the gearing in the pump's body and all are assembled by the pump's cover and flange. The leakage of fluid outside the pump is prevented by rubber gaskets, according to figures 2 and 3.

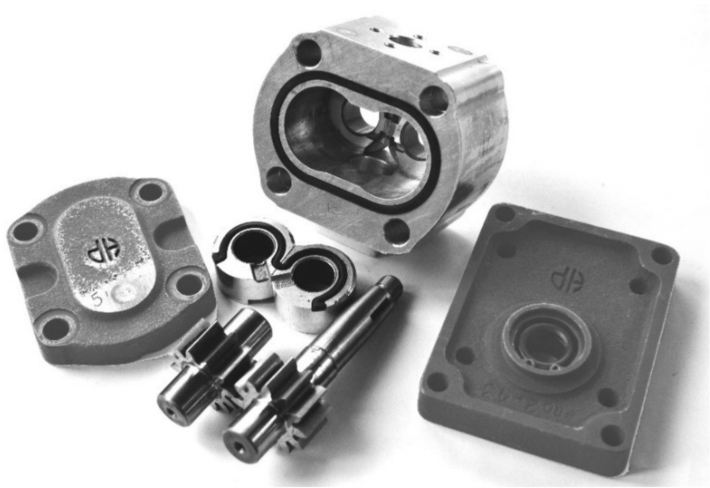

Figure 3. The analysed pump in disassembled representation

A metallographic analysis was performed on some components of the pump, which showed up the following materials: pump body (AlSi2MgTi), bearing blocks (AlSi6Cu4), flange and cover (EN-GJL-HB195), gears and their shafts (316 stainless steel).

In the research of the project [12], we have observed and studied, among others: the influence of the errors of manufacturing the main parts and of their assembly in the pump, the execution quality of teeth flank surfaces on the constructive and functional performances.

This paper is a continuation of the project and deals with the measurement of certain important parameters of the gears. The studies revealed that the errors and deviations of the gears elements and bearings occur during the different manufacturing stages of the body, gears, teeth flanks, shafts bearing blocks, when mounting the gearing in the pump assembly and under operating conditions. To determine the causes and sources of errors, specific steps, rules and methodologies presented in specialized standards [13, 14], workbooks and scientific papers [1, 7, 9] were applied.

In many analyses it was identified which of the sources have a higher relevance: the machine-tool, the cutting tools, instruments and control devices. These are basic components of the technological system. Also, it was necessary to analyse the process and manufacturing technology and, implicitly, the values of the process parameters.

For this pump, the research in the project was carried out, mainly, in three directions, regarding:

The processing precision of the spur gears, corresponding to their type-dimensions, that is defined by the precision grade, the roughness of the flanks and the clearance between the flanks.

Precision indices regarding the names and their notation, as well the sizes of the limit deviations and the specific tolerances, are established by standards [15]. These indicate individual errors and cumulative errors.

Precision criteria. In defining them, we consider the main areas of gears use and the grouping of the precision grades indices. According to the three corresponding precision criteria: kinematic, smooth running and contact between teeth [14], measurements were made using the Klingelnberg P150 centre. For each criterion there are established indices and groups of precision indices.

Thus, for this type of hydraulic pumps, the following deviation values are recommended [7], being important and normal in the assembly process and during the pump operation: the front clearance between the front surfaces of the gears and bearings: 10 ... 50 µm, the tolerance of the distance between the gears axes: max. 0.02 mm, the eccentricity between the gears outside diameters and the bores diameters of the body: max. 0.01 mm, radial clearance of the pump > 0.0015 × outer diameter of the gears, parallelism of the front surfaces should not exceed a tolerance of 0.01 mm.

The measurements were made for each gear in the transverse plane (figure 4), on the left and on the right flanks of the 12 teeth.

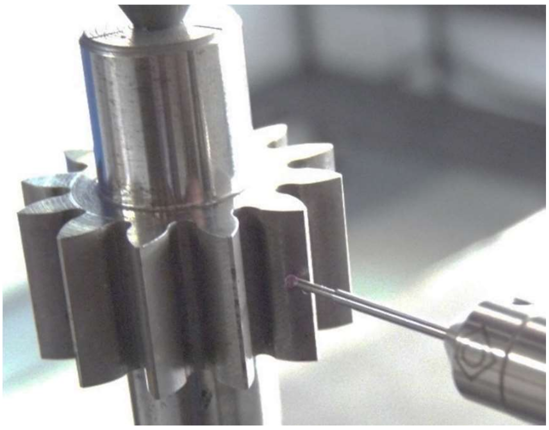

Figure 4. Control of the profile and of the teeth flank line

Diagrams in figure 5 (for the driving gear/pinion z_1) present results of these measurements on flanks and the value of the tolerance prescribed for the precision grade 5.

According to figure, the notations are: F_p - total cumulative pitch deviation (difference between the most positive and the most negative index value for all teeth), f_p –single pitch deviation (difference between two adjacent teeth index values +, -), $f_{p\ max}$ – maximum single pitch deviation (maximum difference between two adjacent teeth index values +, -), $f_{u\ max}$ – maximum difference between adjacent pitches (maximum difference between actual dimensions of two successive right or left flank transverse pitches), R_p – range of pitch error, pitch variation (difference between the largest and the smallest actual size of the transverse pitches of a given flank), $F_{p\ z/8}$ – total cumulative pitch deviation for eight teeth (maximum difference in cumulative deviations of actual values and those nominal for eight teeth), F_r – runout error (radial position difference of a probe contacting all teeth at measuring diameter – pitch diameter, it combines tooth eccentricity in relation to the datum axis and the tooth spacing error), R_s – variation of tooth thickness (difference between the largest and the smallest tooth thickness of a gear) [14, 15].

The results of the measurements presented in the diagrams (figure 5) for the deviations f_p and F_p do not correspond (f_p = 21.7 µm and f_p = 27.5 µm, the left flank, respectively, the right flank) because the left flank has an addendum and a dedendum relief/flanking. The maximum value prescribed in the standard [15] for f_p is 5 µm in the precision grade 5, but the values resulting from the measurements correspond to the precision grade 10. Similarly, the values measured for the deviation F_p (F_p = 40.9 µm and F_p = 52.1 µm, the left flank, respectively, the right flank) do not correspond to the prescribed value of 16 µm according to the precision grade 5, being (as values) in the precision grade 9.

There were determined total cumulative pitch deviations for eight teeth $F_{p\ z/8}$ = 40.2 µm and $F_{p\ z/8}$ = 44.1 µm for the left and right flank, respectively, and correspond to precision grade 10. DIN 3962 [15] provides a maximum value of 10 µm for these indices, according to the precision grade 5.

The runout error F_r defines the runout of the pitch circle, it is the error in radial position of the teeth. Most often it is measured by indicating the position of a pin or a ball inserted in each tooth space around the gear and taking the largest difference. This error causes a number of problems: noise and wear of gears and bearing blocks. Its source is most often insufficient accuracy and ruggedness of the

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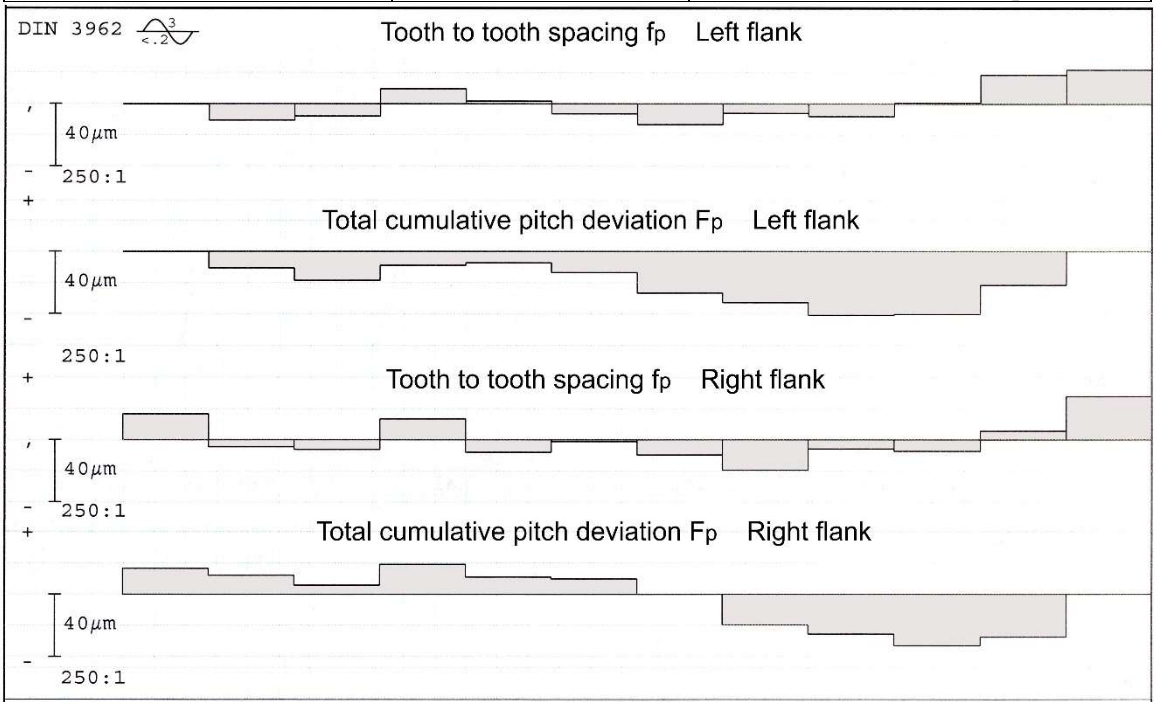
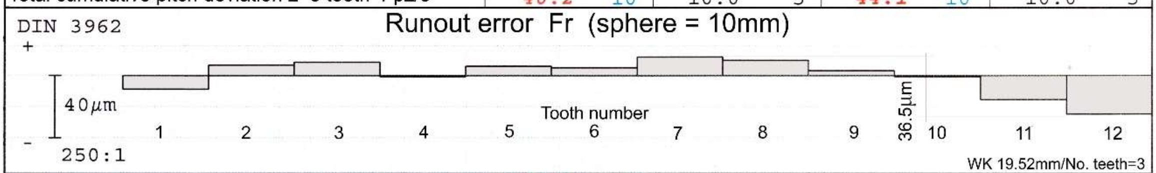
driving shaft and of the tooling system. In the size of the runout error, the eccentricity of the gear influences the general shape of the gear kinematic error diagram. This is used to determine several errors and individual deviations.

Spur gear division		
Name: Driving gear/Pinion	No. of teeth: 12	Pressure angle: 20⁰00'00''
Drawing no. Pinion z_1=12	Module m=2.54 mm	Flank line inclination angle: 0⁰0'0''
Order no. 01	Condition: final	Place of measurement: Neptun



Pitch circle measure: 34.622 z=9mm	Left flank				Right flank			
	Measured value	Precision	Measured value	Precision	Measured value	Precision	Measured value	Precision
Max. single pitch deviation fp max	21.7	10	5.0	5	27.5	10	5.0	5
Max. difference between adjacent pitches fu max	21.9	9	6.0	5	22.0	9	6.0	5
Range of pitch error Rp	34.9				47.2			
Total cumulative pitch deviation Fp	40.9	9	16.0	5	52.1	9	16.0	5
Total cumulative pitch deviation z=8 teeth Fpz/8	40.2	10	10.0	5	44.1	10	10.0	5



Runout error Fr	36.5	9	11.0	5	Limit value	19.47	19.43
Variation of tooth thickness Rs					Measured value	19.479	19.441

Figure 5. Individual and cumulative pitch deviations and runout

The measured values for F_r (figure 5), greater than those prescribed, generate noise during the pump's operation. The measured value of F_r = 36.5 µm corresponds [15] to the 9th grade of precision. The requirements for precision of this spur gear pump gearing is that it should correspond to the precision grade 5, at the imposed value of F_r = 11 µm.

According to the laboratory measurements using the Klingelnberg P150 centre for the driven gear z_2, there were found errors and similar values to those of the pinion z_1.

We consider that this gearing, from the point of view of its functional role, shall meet the criterion of smooth operation. The limit deviation tolerance for F_r is determined [14, 15] depending on the precision grade of the gearing.

The evaluation of the pump's gearing precision also required the evaluation of the specific deviations [16] regarding the profile and the flank line for each of the two gears (figures 6 and 7).

Name: Pinion	No. of teeth: 12	Pressure angle: $20^0 00' 00''$
Drawing no. Pinion z_1=12	Module m=2.54 mm	Flank line inclination angle: $0^0 0' 0''$
Order no. 01	Base circle diameter d_b=28.6418 mm	Face width b=18.3 mm Profile evaluation range L_a=9.28 mm
Condition: final		Flank line length L_b=14.64 mm
	Temperature: 21.4^0	Probe rod ball diameter: 2 mm

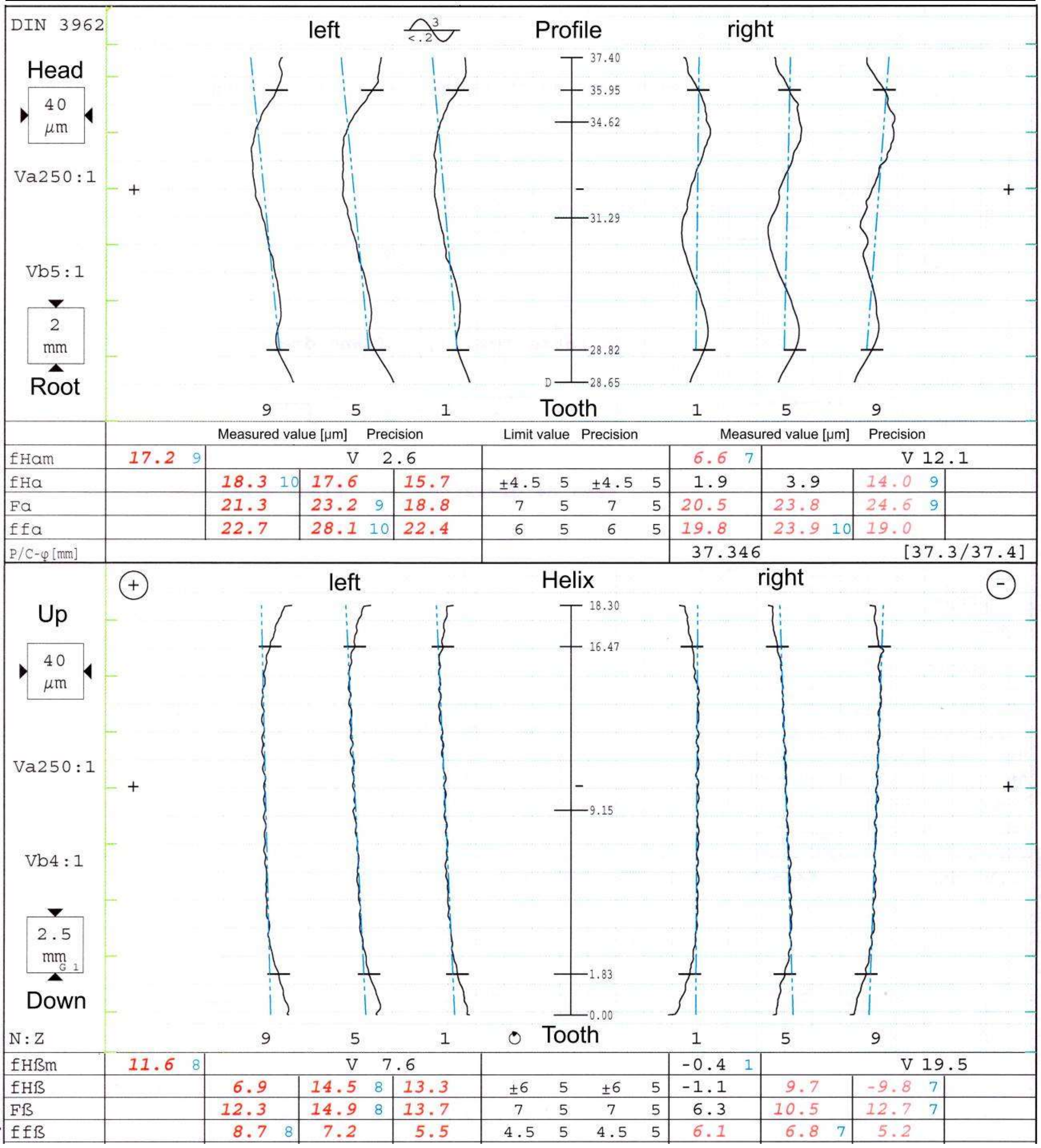


Figure 6. Diagrams and deviations of the teeth profile and of the flank line – pinion

Name: Gear	No. of teeth: 12	Pressure angle: 20⁰00'00"
Drawing no. Driven gear z_2=12	Module m=2.54 mm	Flank line inclination angle: 0^0
Order no. 01	Base circle diameter d_b=28.6418 mm	Face width b=18.3 mm Profile evaluation range L_a=9.28 mm
Condition: final		Flank line length L_b=14.64 mm
	Temperature: 21.3^0	Probe rod ball diameter: 2 mm

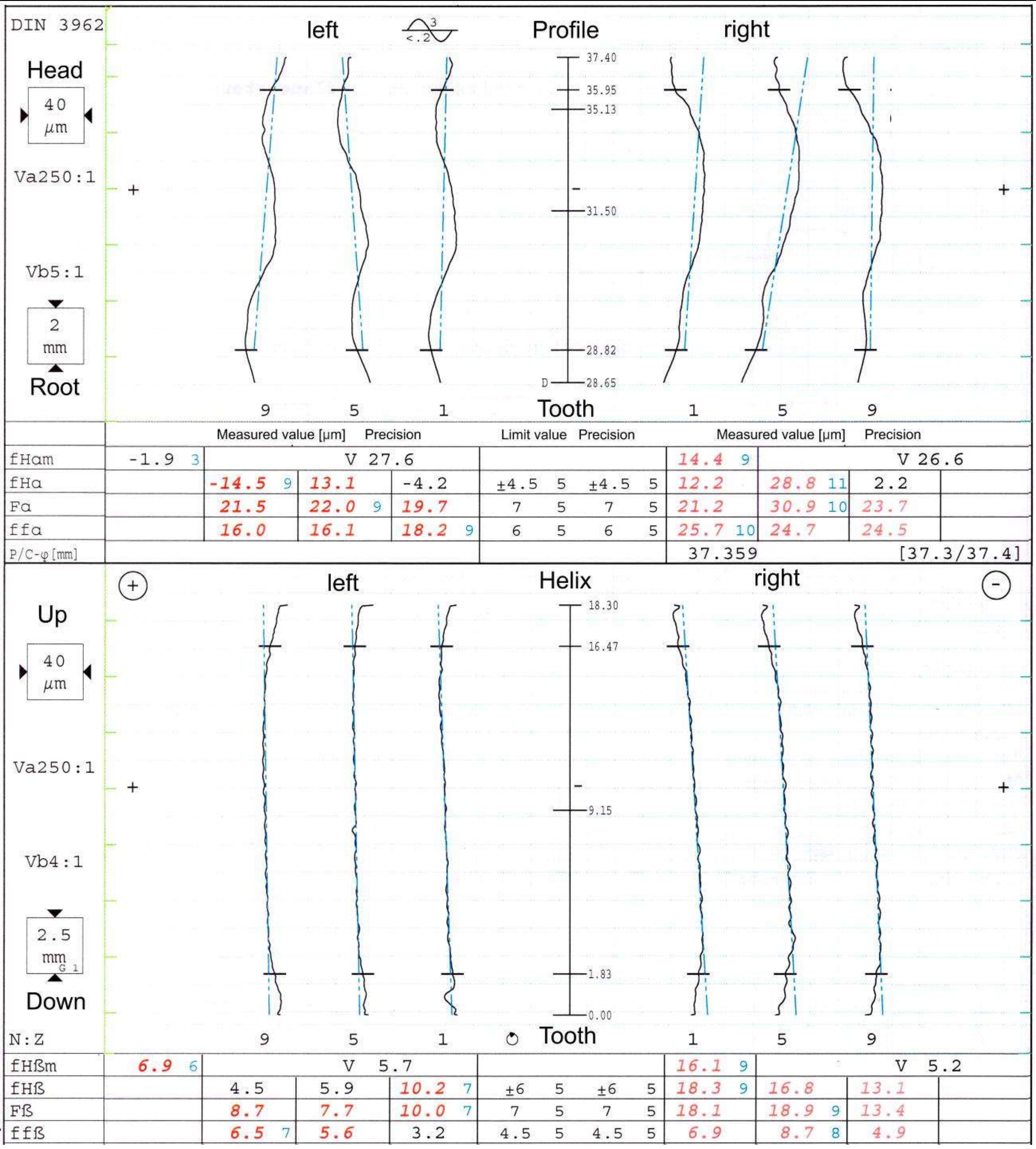


Figure 7. Diagrams and deviations of the teeth profile and of the flank line – driven gear

The measured values refer to: $f_{H\alpha m}$ – mean profile slope deviation (arithmetic mean of the profile slope deviation of three or more equally spaced tooth flanks around the gear's circumference), $f_{H\alpha}$ – profile slope deviation/profile angular error (distance between two nominal profiles that intersect the average profile at start and end points of the profile range), F_α – total profile deviation/profile total error (distance between two nominal profiles enclosed within the profile test range, $f_{f\alpha}$ – profile form deviation/profile form error (distance between two involutes of the actual base circle, that enclose the actual involute profile within the profile inspection range), $f_{H\beta m}$ – mean helix slope deviation (arithmetic mean of the helix slope deviations of three or more equally spaced tooth flanks around the gear's circumference), $f_{H\beta}$ – helix slope deviation/tooth alignment tolerance (distance in transverse plane between two nominal leads that intersect the average lead – helix at start and end points of the lead inspection range), F_β – total helix deviation/lead total error (distance between the two nominal leads enclosed within the lead inspection range), $F_{f\beta}$ – helix form deviation/lead form error (distance between two helical lines that enclose the actual lead within the lead inspection range).

In figure 6, the total helix deviation of the tooth direction (flank line) F_β (for teeth 1, 5 and 9, left-hand measurement of the pinion z_1) has the values 13.7 µm, 14.9 µm and 12.3 µm, according to the criterion of contact between the teeth. These measured values correspond to precision grades 7 or 8 [14, 15], and the prescribed value (precision grade 5) for F_β deviation is 11 µm.

The measuring on each tooth profile is properly performed according to the tooth height, between the base circle and the head circle. The highest measured value has the deviation $f_{f\alpha}$ = 28.1 µm for the 5th tooth on the left flank (figure 6), which corresponds to the precision grade 10, a value that cannot be accepted because it generates an inappropriate operation of the gearing. Alongside are presented, for comparison, the limit values of deviations and tolerance F_α for the 5th grade of precision. Also, the measured values are indicated for the right flank. Figure 7 presents measurements for the teeth 1, 5 and 9 of the driven gear z_2. The figure also shows the highest value of the tolerance F_α = 30.9 µm, which corresponds to the precision grade 10. The analysis of the profile shapes for the two gears presents a certain similarity, corresponding to the left and to the right flank. We believe that this may be due to the generating profile of the tool's cutting edges, but also to the generating process of the teeth profile.

On diagrams from figures 6 and 7 are identified some specific points of the profiles measured on the left and right flank for those three teeth considered. Thus, it is possible to identify, without being noted, the following characteristic points [11]: the start of the measurement (diameter 28.65 mm), the start of the generated profile (diameter 28.82 mm), the start of flanking (diameter 34.6 mm - pinion and 35.13 mm – driven gear), of singular gearing on the tooth head (diameter 35.95 mm) and of profile limit on the tooth head (diameter 37.4 mm). From the profiles diagrams we observed that it was applied a modification by flanking on the left flanks to the pinion and on the right flanks to the driven gear. On the other corresponding flanks, right on the pinion and left on the gear it is also applied a modification called roundness on the tooth width.

According to [13, 15] for evaluation of the flanks direction deviations (figures 6 and 7) there were measured every three flanks left - right on pinion and on driven gear. For these deviations there are indicated actual values and limit values. Two areas are identified on every gear's face width (b = 18.3 mm): the first at the flanks ends (each on a 1.8 mm width) and the second on a length ($L_b \approx$ 15 mm). The active area of the flanks presents small deviations corresponding to the precision grades 7 and 8, which we consider that are acceptable. The flank on its width may be considered to have a slight longitudinal swell modification, which leads to a functional improvement. Also, this modification may diminish the deviations influence on axes not being parallel [17].

4. Conclusions

The measurement of the geometrical elements of the two gears and their deviations was performed with a specialized precision equipment, according to the methodology indicated in standards [14, 15]. The graphical and numerical results indicate that the two gears' teeth are corrected and correspond to the measured data and to the geometrical calculations made using the known relations [7, 11].

The measured deviations have some values that are different from those prescribed in the technical documentation [3]. The values of the precision grades of the limit tolerances and deviations have some common causes: constructive, of manufacturing and of assembly. To these are added the wear of the cutting tools, the inaccuracy of the gear cutting machine-tool [18], the positioning of the tool on the work surface, the speeds of the generation movements, the inaccuracy or lack of rigidity of the mechanical orientation, fixture and driving devices. The choice of the precision grades and of the teeth flanks roughness [19] should correspond to the pump's operating conditions. The deviations found, with values higher than those allowed for this type of pumps, determine the occurrence of inconsistencies in operation (vibration, noise, wear of the teeth flanks and of the bearing blocks). The beneficiary of a lot of such pumps noticed these problems and wanted to know their cause.

A next stage of research may consist in the elaboration of a measurement methodology of many other parameters with the pump in operation (the required driving power, noise and vibration levels, correlations between the viscosity of the hydraulic fluid and the driving torque, etc.). Consideration will also be given to the determination of the optimal variant of the technological processing parameters.

5. References

[1] Zhao X and Vacca A 2017 Formulation and optimization of involute spur gear in external gear pump *Mech. Mach. Theory* **117** 114-132
[2] Egbe E A 2013 Design analysis and testing of a gear pump *Int. J. of Eng. & Sci.* **3** 01–07
[3] UM Hidraulica Plopeni; Hesper 2019 Gear pumps & motors, *Company catalogs*
[4] Mucchi E, D'Elia G and Dalpiaz G 2012 Simulation of the running in process in external gear pumps and experimental verification *Meccanica* **47** 621–637
[5] Nagamura K, Ikejo K, Toyoshima T and Ueda M 2001 Design and performance of gear pumps with non-involute tooth profile *JSME Intern. Conf. on Motion & Power Transm.* 689-694
[6] Mucchi E, Dalpiaz G and Rivola A 2010 Elastodynamic analysis of a gear pump. Part II: Meshing phenomena and simulation results *Mech. Syst. & Signal Process*. **24(7)** 2180-2197
[7] Oprean A, Marin V and Dorin Al 1976 *Hydraulic power systems* (in Romanian: *Acţionări hidraulice*), (Editura Tehnică Bucureşti)
[8] Ghionea I 2010 Research on optimization by simulation of the industrial products design *PhD. Thesis*, University Politehnica of Bucharest
[9] Casoli P, Vacca A and Berta G L 2008 Optimization of relevant design parameters of external gears pumps *Proc. 7th JFPS Intern. Symp. on Fluid Power* Toyama 277-282
[10] Colborne J R 1987 *The geometry of involute gears* (Springer Verlag)
[11] SDP/SI 2014 *Elements of Metric Gear Technology*
[12] Project POSDRU/159/1.5/S/138963 2014 Parameterized design of elements from integrated product systems and technological manufacturing structures, Univ. Politehnica of Bucharest
[13] ISO 18653 2011 *Gears – evaluation of instruments for the measurement of individual gears*
[14] STAS 6273-81 1981 *Cylindrical gears. Tolerances*. Romanian standard
[15] DIN 3962 1978 *Tolerances for cylindrical gear teeth*. Part 1: *Tolerances for deviations of individual parameters*, Part 2: *Tolerances for tooth trace deviations*
[16] Lianfu H, Changfeng F, Wang J and Tang W 2013 Outlier detection and correction for the deviations of tooth profiles of gears, *Meas. Sci. Rev.* **13**, *J. of Inst. of Meas. Sci.*, Slovak Academy of Sciences, Walter de Gruyter GmbH, Berlin
[17] Velex P and Maatar M 1996 A mathematical model for analyzing the influence of shape deviations and mounting errors on gear dynamic behaviour *J. of Sound & Vibration* **191** 629–660
[18] Rădulescu Gh, Miloiu Gh, Gheorghiu N, Munteanu C, Vişa Fl, Ionescu N, Popovici Vl, Dobre G and Raşeev M 1986 *Practical guide for machine-tools design* (in Romanian: *Îndrumar de proiectare în construcţia de maşini*) **III**, (Editura Tehnică, Bucureşti)
[19] Akbarzadeh S and Khonsari M M 2008 Performance of spur gears considering surface roughness and shear thinning lubricant *ASME, J. of Trib.* **130(2)** 021503

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