IMPROVING THE QUALITY OF THE METALLIC GEARS BY APPLYING RUNNING IN PROGRAMS

УЛУЩЕНИЕ КАЧЕСТВА ЗУБЧАТЫЙ МЕТАЛЛИЧЕСКИЙ ДЛЯ ПРИЛОЖЕНИЕ СЛЕД ОБКАТКА ПРОГРАМА ПЕРЕДАЧА

Prof. Dr. Eng. GHEORGHE D.[1],
Galați University, România[1]
E-mail: gheorghedum@yahoo.co.uk

Prof. Dr. Eng. JÂŞCANU M.[1],
Galați University, România[1]

Ass. Eng. BAROIU N.[1],
Galați University, România[1]
E-mail: nicusorbar2000@yahoo.com

Ass. Eng. ŞOLEA L.[1]
Galați University, România[1]
E-mail: lcsol@yahoo.com

Abstract: *The paper presents the technological aspects related to the running in of the metallic cylindrical gears by comparing two running in programs to study their effects on the flank micro-geometry as regards the profile deviations and teeth direction.*

KEY WORDS: FLANK SURFACE, GEAR, CYCLOGRAMES, GEOMETRICAL PARAMETERS.

1. Introduction

Gears are usually employed to transmit motion and power for a wide range of technical applications. Due to their spread use, they have known quire fast developments lately in terms of design and construction reflected by the continuous improving of their quality along with better performances [1, 2, 3, 4, 5].
This paper attempts to investigate how quality increases by improving the micro-geometry and some teeth accuracy indexes. This is achieved by applying two different running-in programs through loading cyclograms.
Since the running-in of the metallic gears is a fairly complex process, it is important to increase the life service of the friction couplings in general and the gear in particular.

2. The effect of running-in on flank micro-geometry

According to some researches [5, 7] it is considered that transmission of the power flow from one coupling element to another is achieved by micro-geometry of the two contacting surfaces.
When the coupling starts running, the roughness reached as a result of cutting, the technological roughness due to size and features has a negative effect on the lubricating film resistance (fig. 1), its max pressure (fig. 2) Also, the friction coefficient (fig. 3) and the wear (fig. 4) are strongly affected by the flank technological roughness [8].

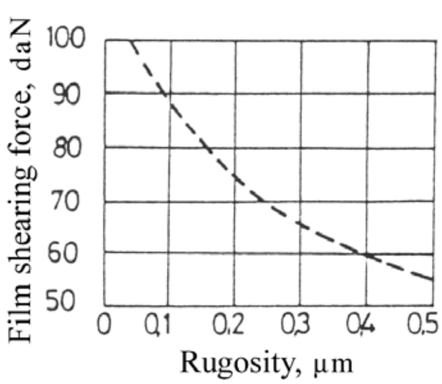

Fig. 1

When the coupling begins to rotate for the first time after having been installed, the technological roughness transformation into functional roughness is initiated (beginning of the running-in) shake down takes place along with a significant wear. This effect of the running in is also called initial wear.
Thus, when operating at the rated power, the gear improperly run in there is the possibility of cold or hot sticking [1, 2, 3, 5].
Sticking develops quickly and it is an undesired damage and is due to the local interruption of the lub film at high turbo-mechanical strains.

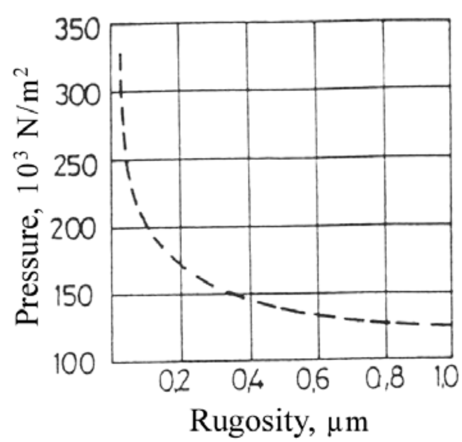

Fig. 2

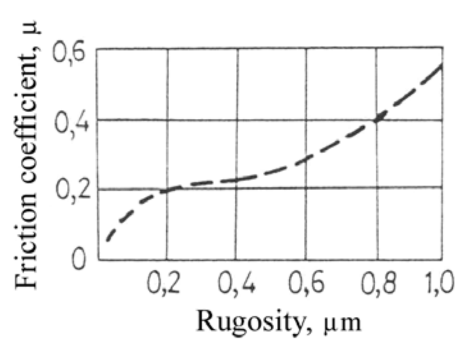

Fig. 3

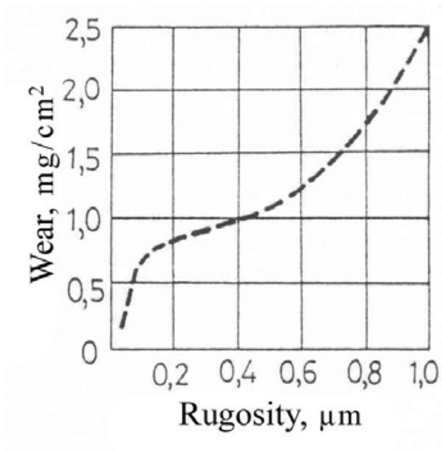

Fig. 4

3. Running in effect on the profile shape error and teeth direction

Out of the accuracy indexes for the three criteria, the error of the profile shape (f_{fr}), the tooth direction deviation ($F_{\beta r}$) and the step error on the basic circle (f_{pbr}), as defined in figures 5, 6 and 7 [1, 4, 10], determine the size and position of the contact face; this is a complex accuracy index which characterizes the heavy duty gears.
The running-in effects on the error of the profile shape (Y_F), the tooth direction deviation (Y_β) and the step error on the basic circle (Y_p) are taken into account by the design calculations as detailed in the literature [1, 9].
From the foregoing it is obvious the importance and opportunity of studying the gear running-in process which can improve those accuracy indexes on which the gear final capacity depends on.

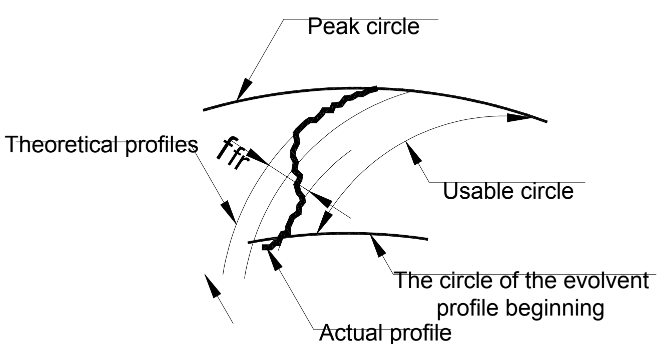

Fig. 5

4. Experimental researches

Conceiving running in programs able to eliminate possible damages and improve the gear quality was possible after a deeper insight into the process along with a large number of experimental tests [1].

The paper is focused on two running in programs. With the firs one, the loading cycle diagram is made up of five modules of total duration 175 min. (fig. 8), with the second one, there nine modules, exceeding the rated / nominal loading moment for the modules 6, 7 and 8 and reaching a total duration of 160 min. (fig. 9).

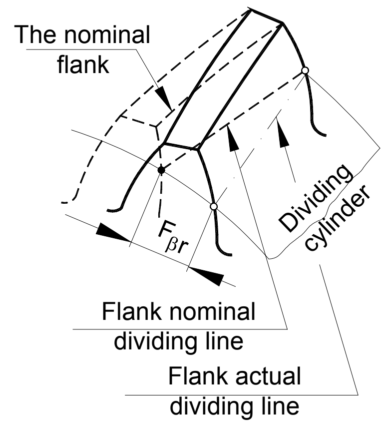

Fig. 6

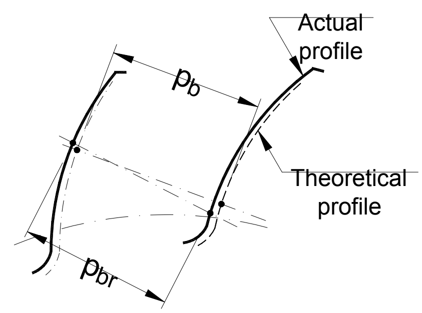

Fig. 7

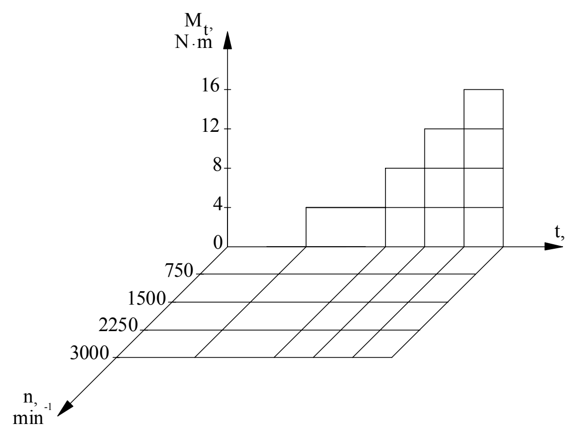

Fig. 8

The equivalent number of cycles is calculated according to [9], as follows:

$$N_{e_1} = \sum \left(\frac{M_{tj}}{M_e} \right)^{\frac{13}{2}} \cdot N_{H_j} = 87406 \quad (1)$$

$$N_{e_2} = \sum \left(\frac{M_{tj}}{M_e} \right)^{\frac{13}{2}} \cdot N_{H_j} = 1716312 \qquad (2)$$

where:

M_e - the basic torque, $M_e = 16$ N · m; M_{tj} - torque of the j th step; N_{H_j} - number of cycles of the j th step.

The experiments were carried out with identical samples, the gear sample having two crowns: the (5-module) cyclogram was tested on both flanks on the first crown and the (9-module) cyclogram was tested on the second crown.

The material used was 41MoCr11 enriched, HRC = 54 ± 2, the geometric elements are those in the execution drawing [1].

The lubricator was the same for both programs, T90EP2, the bath temperature was kept within 60-70 degrees.

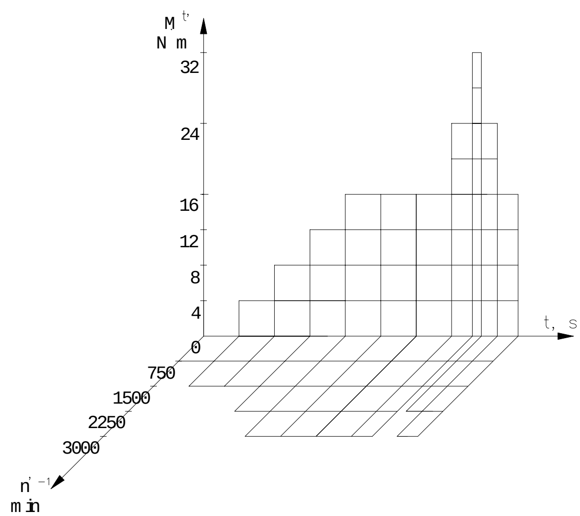

Fig. 9

The flank micro-geometry was determined from the profile diagrams as recorded with proper devices (universal tool MAAG SP 60) fitted with sapphire sensor of tip radius r = 5 μm.

The physical parameters of the flank roughness R_a and R_z before and after applying the two running in programs took the values given in table 1.

Table 1

Cycles diagram	Time of evaluation	R_z [μm]	R_a [μm]	Difference R_a
1	Before	3,97	0,71	0,14
	After	3,19	0,57	
2	Before	4,31	0,77	0,21
	After	3,13	0,56	

The direction and shape deviations of the three teeth of the crown sample were determined by means of the MAAG SP 60.

The results obtained are presented in table 2 and fig. 10.

Table 2

Accur. index	Sample	Measured tooth no	Cyclogram 1 before	Cyclogram 1 after	Cyclogram 2 before	Cyclogram 2 after
f_{fr}	1	z_1	44	40	32	28
		z_8	48	48	40	32
		z_{15}	36	32	32	28
	2	z_1	92	88	32	24
		z_8	60	60	40	24
		z_{15}	60	56	48	32
$\bar{f}_{fr} = \frac{1}{n}\sum f_{fri}$			56,6	54	37,3	28
$F_{\beta r}$	1	z_1	30	12	36	12
		z_8	28	12	16	8
		z_{15}	12	8	24	12
	2	z_1	18	8	28	8
		z_8	30	12	8	6
		z_{15}	12	8	16	8
$\bar{F}_{\beta r} = \frac{1}{n}\sum F_{\beta ri}$			21,6	10	21,3	9

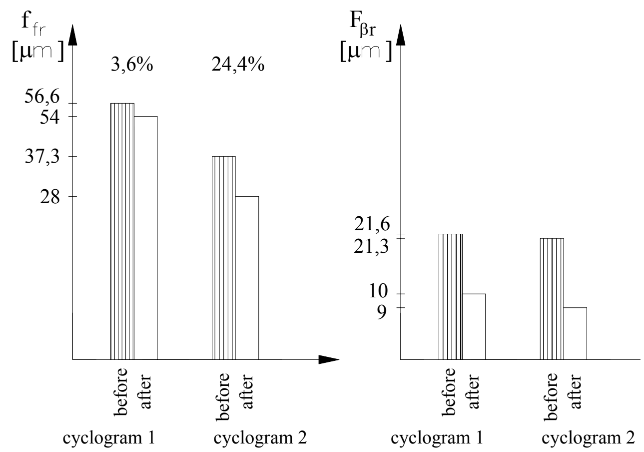

Fig. 10

5. Conclusions

The researces results are positive but slightly different. They prove the validity of the program 2 (fig. 9). Although of shorter duration, this number of cycles is larger than program 1 (fig. 8) which implies higher corrections to the analyzed accuracy indexes.

Increasing the micro- and macro accuracy of the gears through these programs as designed by the authors has a favorable influence on their performance.

References

1. Gheorghe, D. - "Some contribution to the running growth of the portant capacity of the devices with application to the metallic cylinder devices" Ph. D. Thesis, Galați, 1998;
2. Gheorghe, D. - "The improvement of the micro and macro-geometrical parameters of the teeth in the process of running", TCMM, Technical Ed., Bucharest, 1996;
3. Gheorghe, D., Jâșcanu, M. - "Comparative analysis of 3 original running cyclograms as regards their effect on the macro and micro-geometry of the teeth flanks", Part. I, II, The Annals of "Dunărea de Jos" University of Galați, Fascicle VIII, Tribology, ISSN 1220-0824, 1998;
4. Gheorghe, D., Georgescu, C., Baroiu, N. – "Tolerances and dimensional control", Scorpion Ed., Galați, ISBN 973-85803-0-7, 2002;

5. Michalis, K. - “Testing Procedure for Gear Lubricants with FZG – Test Rig, Industrial Lubrication and Tribology”, 1974;
6. Tudor, A. - “The real contact of the friction surfaces”, Romanian Academy Ed., Bucharest, 1990;
7. Lefevre, R. - “Graissage et Tribotechnique”, Ed. Technique, vol. I, II, IV, 1975;
8. *** STAS 12268 – 84 - Cylindrical toothed gears;
*** STAS 6273 – 83 - Cylindrical gears. Tolerances.