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This study aims to provide the recent advances in machining for modern manufacturing engineering, especially CNC machining, modern tools and machining of difficult-to-cut materials, optimization of machining processes, application of measurement techniques in manufacturing, modeling and computer simulation of cutting processes and physical phenomena.

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Table of Contents

PREFACE

Machining is one of the most popular technique to change shape and dimensions of the objects. Machining operations can be applied to work metallic and non-metallic materials such as ceramics, composites, polymers, wood.

Cutting tools have been used since ancient times to remove excess material from forgings and castings. Nowadays, metal cutting became one of the primary manufacturing processes for finishing operations. In the last few years we have observed a rapid development in automation of manufacturing processes, especially in automatic control systems. Progress in cutting stimulates a significant increase in the metal removal rate and achieving high accuracy in terms of dimensions and shape of machine parts. New materials, which play the key role here, are used to produce cutting tools.

To meet today's high demands concerning accuracy and efficiency of the manufacturing process of machine parts, it is necessary to use computer methods for designing of technological processes.

This study aims to provide the recent advances in machining for modern manufacturing engineering, especially CNC machining, modern tools and machining of difficult-to-cut materials, optimization of machining processes, application of measurement techniques in manufacturing, modeling and computer simulation of cutting processes and physical phenomena.

Wojciech Zębala

Chapter 5.3

INFLUENCE OF SOME KINEMATIC AND GEOMETRICAL PARAMETERS OF SPUR GEARS ON THE CHARACTERISTICS OF THE HERTZIAN CONTACT

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Abstract: *Several studies have been made on the variation of the Hertzian pressure, on the lubricant film thickness, on the variation of the bulk and surface temperature (for pinion and gear), on the principles presented by Wang and Cheng (1980). The following parameters have been considered variable: face width, rotational speed, gearing ratio.*

By processing the data obtained by numerical modelling, diagrams illustrating the variation of the tribological characteristics of the Hertzian contact in six points along the line of action have been obtained.

Keywords: *Tribological characteristics, Modelling, Hertzian pressure.*

Introduction

In industrial applications, the principal requirements of gear design have been focused on kinematic characteristics, load carrying capacity, dynamic performance and durability.

In order to develop a research program of the processes and phenomena which occur in the superficial layer of the teeth of metallic spur gears it became necessary to establish certain modelling criteria [1], [2]. These tribomodelling criteria presume the equality between the characteristic values of the gear (the tribosystem), in a specific zone on the tooth flank and, respectively, of the pair of disks (the tribomodel), considering the maximum hertzian contact pressure, p_0, the effective radius of curvature, R and the sliding[rolling ratio, $\xi=2.|v_1-v_2|[(v_1+v_2)$ as bases. By putting the principle into practice, a calculation program made possible the identification of pairs of rollers which model some points on the line of action of the gears under study. The tests have been performed on a Amsler type rig [3].

Geometrical and kinematic parameters

The main geometrical and kinematic parameters of the gear were calculated by Excel computational modules.

Input data

z_1	25
z_2	38
m (mm)	4,0
h_{a0}	1,0
c_0	0,25
α_0	20^o

Main parameters of the gear

Parameter	pinion (1)	gear (2)
d, (mm)	100	152
d_b, (mm)	93,96926	142,8333
d_a, (mm)	108	160
ε_α	1,657525	
p_b, (mm)	11,80853	
n, (rpm)	1000	657,8947
u	1,52	

Points on the line of action

Distance	Size, mm	Obs
AE	19,57293	ε_α,p_b
AC	10,05732	
AB	7,764404	
AD	11,80853	
N_1N_2	43,09454	
N_1C	17,10101	
N_1A	7,04369	$=N_1C-AC$
N_1A	7,04369	$=N_1N_2-AN_2$

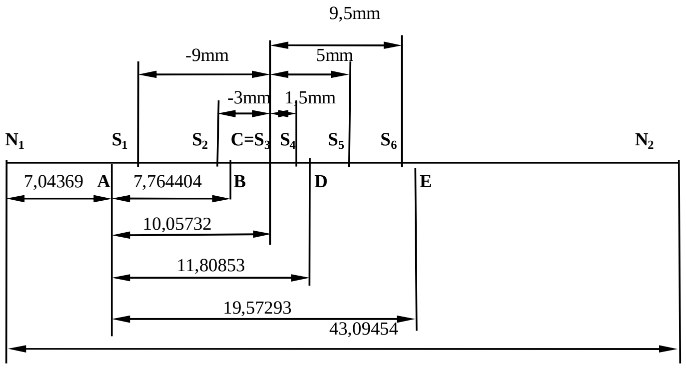

Fig. 1 Points of the line of action (N_1N_2)

Figure 1 represents the line of action (N_1N_2). It shows the positions of the starting and ending points of the engagement of double pairs of teeth of gear, A and E, the positions of points B and D (the limits of double pair of teeth engagement zone), the position of the pitch point, C, as well as the positions of points S_1.. S_6, which have been considered representative for the numerical modelling. Thus, the points S_1 and S_2, on one hand, as well as points S_5 and S_6, on the other hand, belong to the double pair of teeth engagement zone; S_3 is the pitch point; S_4 belongs to the single pair of teeth engagement zone.

The sliding[rolling ratio can be calculated with: $S = 2\cdot|\,v_1 - v_2\,|/(v_1 + v_2)$ or with $\xi = 2\cdot|\,r_1\cdot n_1 - r_2\cdot n_2\,|/(r_1\cdot n_1 + r_2\cdot n_2)$ ($v_{1,2}$ - velocity along the tangent line of the contact for pinion and gear; $r_{1,2}$ - local radius of tooth profile; $n_{1,2}$ - rotational speed for pinion and gear).

Effective radius of curvature $R= r_1 \cdot r_2[(r_1+r_2)$ and sliding[rolling ratio (for characteristic points on the line of action, situated at *x* distance from the pitch point, C) are presented in the Table 1.

Table 1. Values for characteristic points on the line of action

Point	ρ_1	ρ_2	X	ξ
A	7,04369	36,05085	-10,0573	-1,08409
B	14,80809	28,28644	-2,29291	-0,2280
C	17,10101	25,99353	0	0,0000
D	18,85222	24,24232	1,75121	0,1670
E	26,61662	16,47792	9,51561	0,84233

Several studies have been made on the variation of the Hertzian pressure, on the lubricant film thickness, on the variation of the bulk and surface temperature (for pinion and gear) with the help of the ADGEAR program (achieved by ACTIS, on the principles presented by Wang and Cheng, 1980 [4[, [5[).

ADGEAR provides advanced calculations for determining the surface pitting and scuffing tribological performance of gear sets [6[. This analytical tool can be used to predict distributions of dynamic load, the transient EHD film thickness, the flash temperature, the bulk temperature and the surface temperature of contacting gears teeth along the line of action and to evaluate the load capacity against pitting and scuffing failure in gear sets, based on tooth contact temperature.

Study on the variation of the tribological characteristics of the hertzian contact

By processing the data obtained by numerical modelling, diagrams illustrating the variation of the tribological characteristics of the Hertzian contact in six points along the contact line have been obtained. These six points have been selected from the immediate vicinity of the distinctive points of the line of action, A, B, C, D, E.

The following parameters have been considered variable: 1. face width, 2. rotational speed, 3. gearing ratio.

It was studied the influence of the variation of these parameters on the Hertzian pressure, lubricant film thickness, bulk temperature and surface temperature (for pinion and gear).

Figure 2 presents an example of graphical representation of the Hertzian pressure variation, along the line of action. Figure 3 illustrates the variation of the lubricant film thickness and figures 4 and 5 present the variation of the bulk and surface temperature, both for pinion and gear.

Influence of the face width

Figure 6 presents the variation with the following characteristics of the tribosystem versus the face width : the Hertzian pressure (fig.2a), the lubricant film thickness (fig.6b), the bulk temperature (for pinion – fig.6c and for gear – fig.6d), the surface temperature (for pinion – fig.6e and for gear – fig.6f).

As expected, the Hertzian pressure decreases together with the increase of the face width, as a consequence of the decrease of the load along the tooth length.

For all six points S1…S6, the film thickness increases with the width. For the point S1 the increase is slower than for the other points on the line of action. For an increase with 45% of the face width, the film thickness increases with 6% for S1 and with 10-12,6% for the other points.

The bulk and surface temperatures decrease, both for pinion and gear. The bulk temperature is more sensitive to the face width variation, due to the increase of the gear volume (because of the increase of the width). For the pinion, the bulk temperature has lower values than for the gear and the lowest decrease has the temperature of the point S1 (the starting engagement zone, double pair of teeth in meshing). For the gear, the similar phenomenon occurs on the conjugate zone, i.e. for the point S6.

Both, for the pinion and the gear, the highest surface temperatures are found at the ending points of the engagement zone, where the influence of the slidings is stronger than the decrease of the load (double pair of teeth engagement zone). The decrease of the temperatures is relatively small (about 12%) with a linear tendency.

Influence of the rotational speed

The second parameter under study was the rotational speed (fig.7). The following rotational speed range has been taken into consideration: 750; 900; 1000; 1200; and 1500 rpm.

By the decrease of the load along the length of the tooth and the influence on the teeth stiffness, the increase of the rotational speed determines a stronger decrease of the Hertzian pressure then those due to the face width variation. The more sever relative decrease (cca. 47%) can be noticed in the double pair of teeth engagement zone, at the starting of the engagement (point S1) and at the pitch point, S3 (cca.33.6%). The decreasing values are obvious in the point S2 with 30%, respectively: S6 – 29%, S5 – 27,4%, S4 – 25,7%.

For a two times increase of the rotational speed, the lubricant film thickness increase by 51,3% for S1 and by 71,7% for S2, respectively: 69,6% for S3, 67,8% for S4, 73,2% for S5, 71,4% for S6. The increase of the lubricant film thickness can improve the lubrication in contact area.

The surface temperatures present small decreases, i.e. 2 - 3°C. At the ending zone of the line of action highest temperatures were found, higher for the pinion than for the gear, as expected.

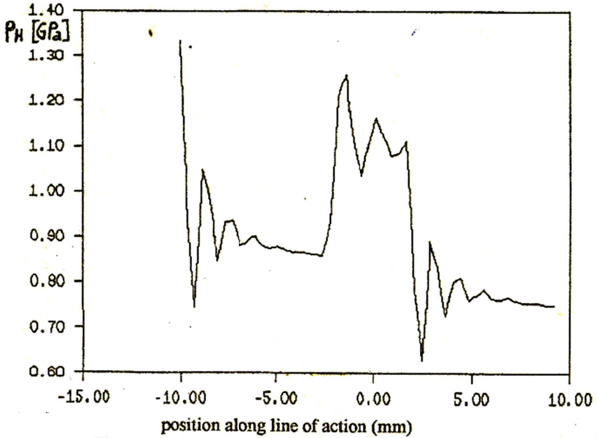

Fig. 2 Hertzian presure variation along line of action

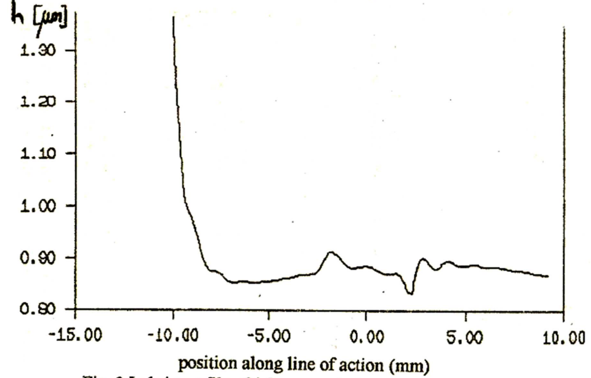

Fig. 3 Lubricant film thickness variation along line of action

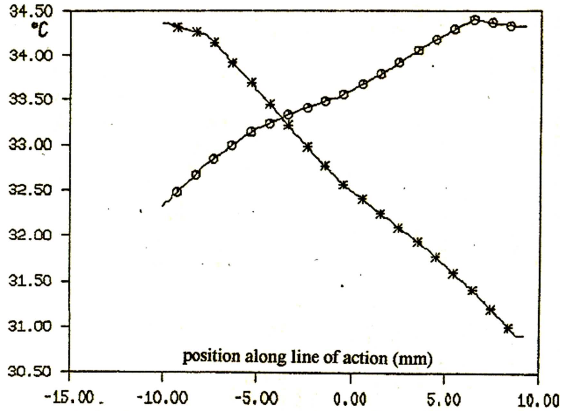

Fig. 4 Bulk temperature variation along line of action (for pinion (o) and gear (*))

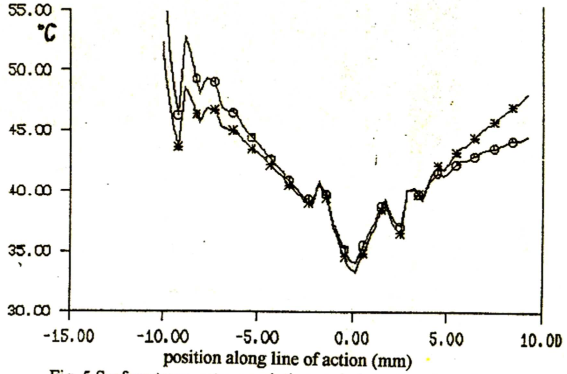

Fig. 5 Surface temperature variation variation along line of action (for pinion (o) and gear (*))

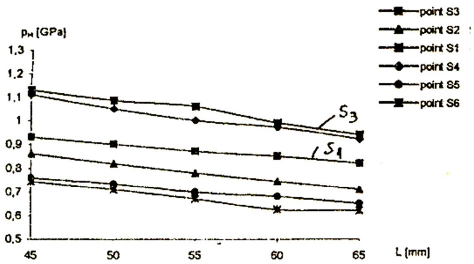

Fig. 6a Effect of gear face widh on Hertzian presure

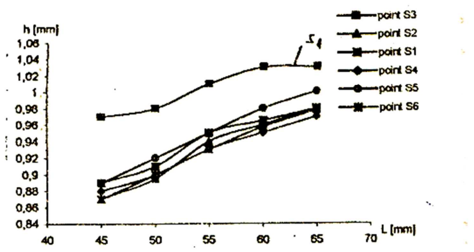

Fig. 6b Effect of gear face widh on film thickness

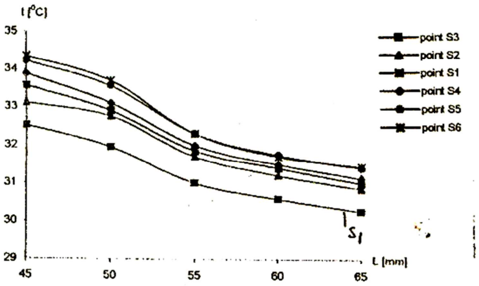

Fig. 6c Effect of gear face widh on bulk temperature (for pinion)

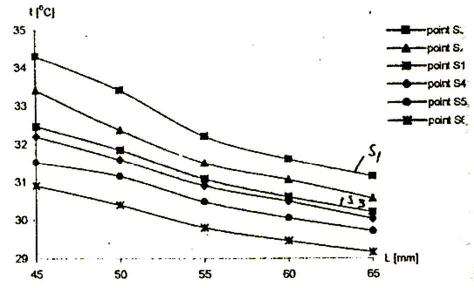

Fig. 6d Effect of gear face widh on bulk temperature (for gear)

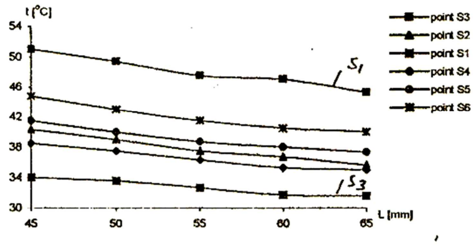

Fig. 6f Effect of gear face widh on surface temperature (for pinion)

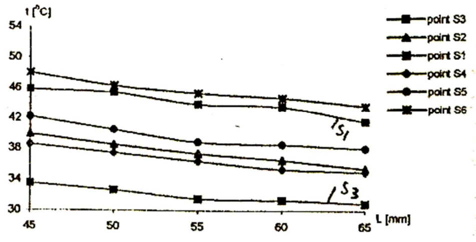

Fig. 6g Effect of gear face widh on surface temperature (for gear)

The influence of the gearing ratio

The variation of the tribological characteristics with the gearing ratio is presented in figure 8. The increase of the gearing ratio, u, was obtained by increasing the number of the gear teeth, for the same face width.

The Hertzian pressure and the lubricant film thickness are not significantly influenced by the increase of the gearing ratio. The surface temperature decreases by a few degrees due to the fact that the gearing ratio increase determines the increase of the gear volume.

Conclusions

By processing the data obtained by numerical modelling, several studies have been made on the variation of the Hertzian pressure, of the lubricant film thickness, on the variation of the bulk and surface temperature (for pinion and gear), along the line of action. The following parameters have been considered variable: gear face width, rotational speed, gearing ratio.

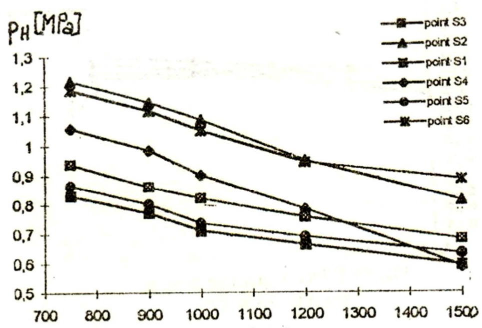

Fig. 7a Effect of rotational speed, n, on Hertzian presure

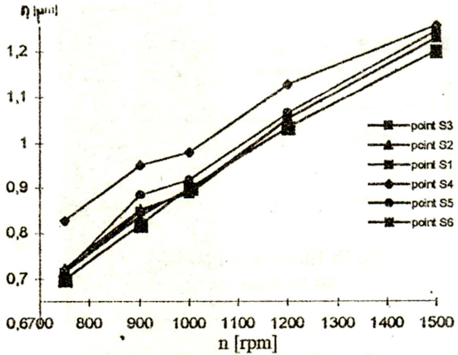

Fig. 7b Effect of rotational speed, n, on film thickness

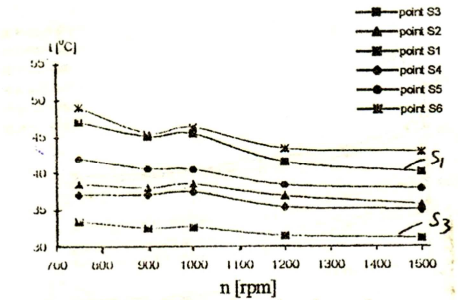

Fig. 7c Effect of rotational speed, n, on surface temperature (for pinion)

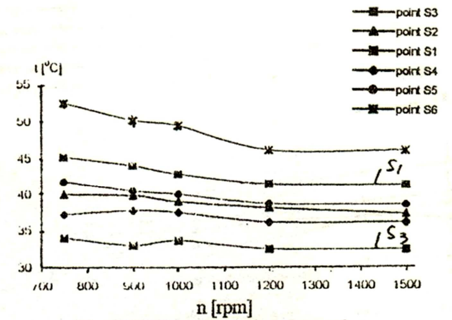

Fig. 7d Effect of rotational speed, n, on surface temperature (for gear)

As expected, the Hertzian pressure decreases together with the increase of the face width. The increase of the rotational speed determines a stronger decrease of the Hertzian pressure then those due to the face width variation. The more sever relative decrease can be noticed in the double pair of teeth engagement zone, at the starting of the engagement (point S1) and at the pitch point, S3.

The film thickness increases with the gear face width and more with the rotational speed.

The surface temperatures decrease and the highest surface temperatures are found at the ending points of the engagement zone, where the influence of the slidings is stronger than the decrease of the load (double pair of teeth engagement zone), both for pinion and gear.

The gearing ratio increase determines no significantly variations of Hertzian pressure and film thickness; only the superficial temperature decreases by a few degrees.

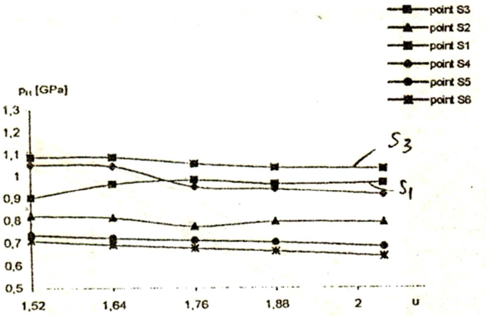

Fig. 8a Effect of gearing ratio, u, on Hertzian presure

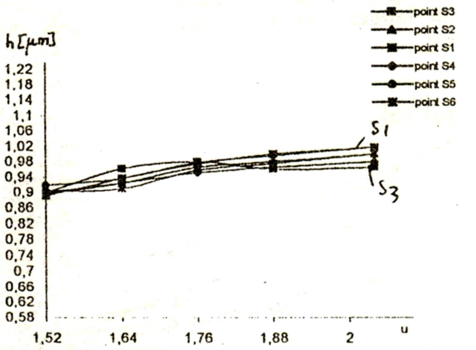

Fig. 8b Effect of gearing ratio, u, on film thickness

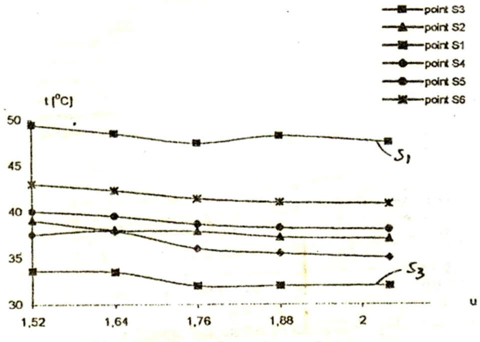

Fig. 8c Effect of gearing ratio, u, on surface temperature (for pinion)

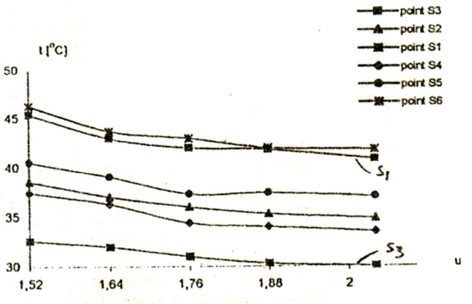

Fig. 8d Effect of gearing ratio, u, on surface temperature (for gear)

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