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DETERMINATION OF THE KINEMATIC DEPENDENCIES BETWEEN THE MOVEMENTS FOR A RETROFITTED HOB SHARPENINING MACHINE

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Abstract: *This paperwork presents some researches performed by the authors in the framework a national research grant regarding the retrofitting of a hob sharpening machine. The first step in this project was the identification of the movements involved in the process of hob sharpening, together with their kinematic dependencies. Because the references regarding these aspects are very poor and, furthermore, there is a very small number of hob sharpening machines in use these days, the process mentioned above was performed using simulation techniques. The known kinematic laws of motion and the shape of the imposed trajectory were combined into a simulation model in order to determine the unknown movements and the dependencies between them. After the simulation process, the structure of the retrofitted kinematic chains and the dependencies between the technological movements of the machine was determined.*

Key words: *hob sharpening, kinematic dependencies, laws of motion, simulation.*

1. INTRODUCTION

The hob gear is one of the most complex tools used in machine building industry. It is used mostly for manufacturing gears with various profiles, by the profiling method, on specialized machine tools [5, 7].

In the town were authors live there is a large manufacturing plant, which along other products, manufactures gear sets for the automotive industry. The competition between manufacturers on this market is significant, and consequently the quality demands for the manufactured products (gear sets) are also very high.

The parts are manufactured on numerically controlled machine tools using gear hobs as tools. The imposed productivity is also very high, so the cutting regimes are intense, leading to a very short life of the hob, until sharpening became compulsory.

The number of hob sharpening machines in the Romanian industry is quite low, and most of them are obsolete, not only as structure and performances but also as designing principles.

The above mentioned factory has to sent the hobs to be sharpened to another plant, at 100 km distance, this fact influencing directly upon the overall manufacturing costs.

Furthermore, the sharpening machine used is a very old one, and even if its performances are satisfactory, the time when the machine is down for repairs is a very long one, which is not acceptable for a factory that is requested to manufacture the gear sets on a “just-in-time” production philosophy [6, 7, 10].

Because of the implications of this fact (which may lead even to loosing the traditional clients) a decision had to be taken regarding the sharpening process.

Two choices were taken into consideration by the management of the manufacturing company: to buy a new sharpening machine or to reconfigure other machine tools for performing this task. Buying of a new sharpening machine was found to be economically challenging, so the second option was chosen.

In order to reconfigure a machine tool for hob sharpening operations, assistance was demanded and research project with national financial support (in the frame of a RELANSIN research program) was developed between the factory and a research team from “Lucian Blaga” University of Sibiu, Engineering Faculty [1].

2. THE KINEMATIC OF THE HOB SHARPENING PROCESS

The movements involved in the hob sharpening process are presented in figure 1, where:

- I represents the rotation of the sharpening tool (main cutting movement);
- II represents the movement for positioning the tool slide (auxiliary movement);
- III represents a continuous circular movement (circular feed);
- III’ represents a non-continuous circular movement used for moving across the helical channels of the hob;
- III’’ represents a non-continuous circular movement used for ensuring the removal of the sharpening stock;
- IV represents an axial movement (axial feed);
- V represents a positioning movement for ensuring the orientation of the sharpening tool towards the helical channels of the hob.

Movement I, II and III’, III’’ are independent movements, while movements III and IV (circular feed and axial feed) are depending on each other.

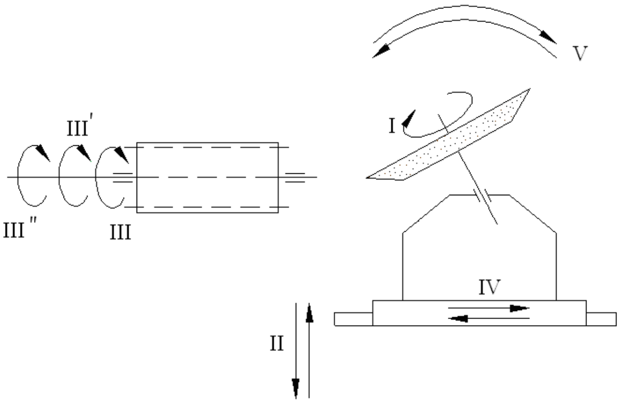

Fig. 1. The movements involved in the kinematics of the hob sharpening process.

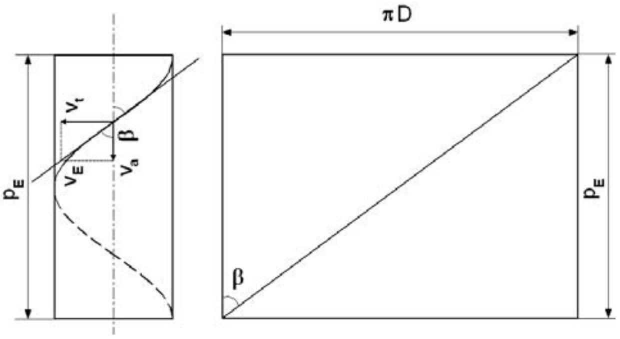

Fig. 2. The generation of the helical trajectory.

The dependence between the circular feed and the axial feed derives from the necessity of driving the sharpening tool towards a helical trajectory, mainly the helical channels of the hob along which the teeth of the gearing tools are placed.

Figure 2 shows the relationships between the axial speed (v_a) and the tangent speed (v_t) in order to align the resulting component speed (v_E) towards a helical trajectory [4, 5].

The notations in Fig. 2 represent:

- v_a – axial speed;
- v_t – tangent speed;
- v_E – resulting component speed;
- p_E – the step of the helical trajectory;
- β – the slope angle of the helical trajectory
- D – the diameter of the hob.

It is here to notice the fact that the values of angle β are between 2° – 4° degrees, this corroborated with the usual values for the diameter of the hob D leading to a step p_E usually in the vicinity of 3 meters.

The relation between v_a and v_t can be expressed as:

$$\tan\beta = \frac{v_t}{v_a} = \frac{\pi D}{p_E} \ . \tag{1}$$

3. SIMULATION

The dependence expressed by relation (1) ensures the fact that the sharpening tools will go toward the helical channel of the hob, by one have to notice the fact that only that will not be sufficient for completing the hob-gear sharpening process.

The sharpening tool has not only to follow the helical trajectory, but also has to remove the sharpening stock. Another movement (III'') will be necessary to be performed in order to ensure the stock removal.

In order to simulate the dependence between the tangent and the axial feed, the parametric equation of the motion were taken into consideration as:

$$x = r \cdot \cos\theta, \qquad z = \frac{r \cdot \theta}{tg\,\beta}, \tag{2}$$

where: x is tangent feed; $z-$ axial feed; r – radius of the hob; θ – current rotation angle of the axial plane which includes a mobile point characterised by coordinates $x-$ and z.

Based upon equations 2 a simulation diagram was built using the Matlab & Simulink software package [2]. The simulation diagram is presented in Fig. 3.

The simulation diagram from Fig. 3 was used for two purposes: the first one, for checking the fact that the correlation between the axial and tangent speed is correct (the sharpening tool is driven along the helical channel) and the second one, to found and check the necessary movement for stock removal.

During the simulation process, it was found that movement III'' should be an impulse movement, applied at every start of the combined axial and circular movement.

As a result of this impulse, the hob will move with a specific angle (which value depends of the sharpening stock value).

This angular movement will be added with the continuous circular feed and will drive the sharpening tool on a helical trajectory which is parallel with the initial one, but displaced with the value of the stock (divided by the overall number of sharpening tool passes).

Figure 4 shows both the generated helical trajectory and the helical trajectory necessary for sharpening.

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The kinematic chains of the hob-sharpening machine were retrofitted in order to attain the kinematic of the sharpening process (Fig. 1).

Consequently, the movements I to V are performed by the following drives:

- I induction motor drive controlled through frequency inverter with open loop speed control [8, 9];
- II induction motor drive with open loop position control (frequency inverter, PLC and limiters);
- III, III', III'' stepping motor drive controlled through parallel port of the PC;

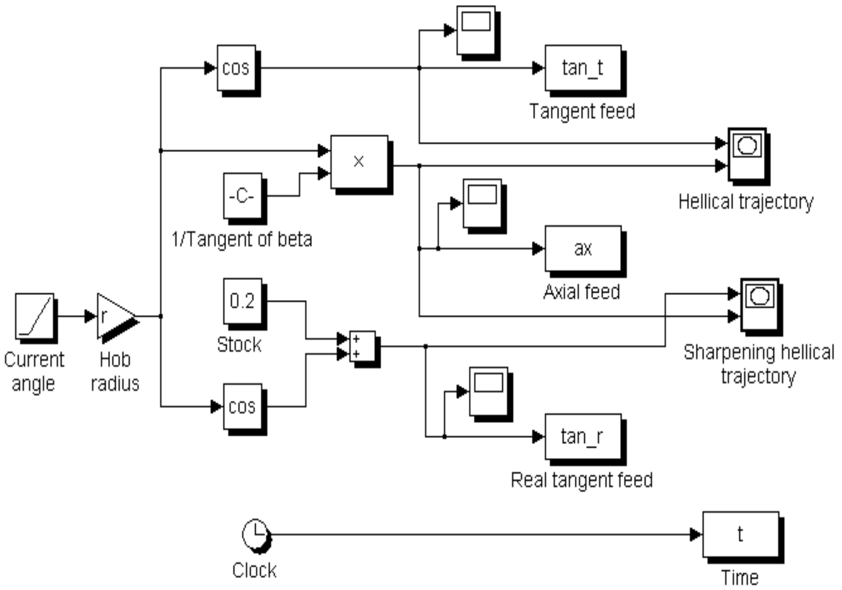

Fig. 3. The simulation diagram.

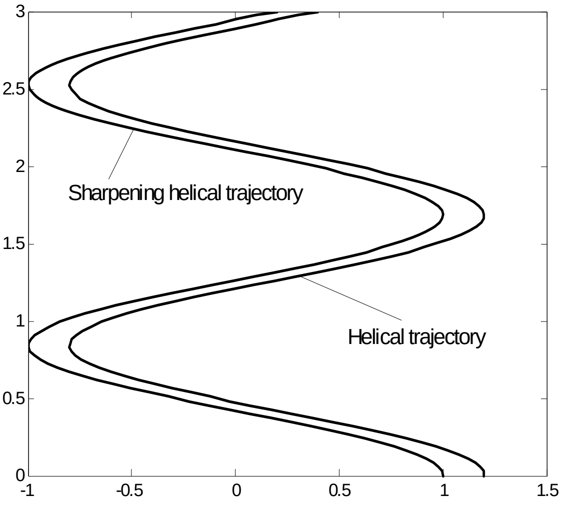

Fig. 4. Simulated helical trajectory and sharpening helical trajectory.

- IV stepping motor drive controlled through parallel port of the PC;
- V unmodified drive, manual actuation.

4. THE LONGITUDINAL FEED KINEMATIC CHAIN

The longitudinal feed kinematic chain uses as actuation drive a stepping motor controlled by means of pulses generated through the parallel port of the PC.

The change between rotational movement and translation movement is performed by a pinion-rack mechanism (Fig. 5). In order to calculate the pulse frequency f_L, the following input data will be considered:

θ_{ML} - angular step of the stepping motor),
$\theta_{ML} = 1.8^{\circ}$ (200 pulses/rev),
z_p - number of teeth of the pinion ($z_p = 30$ teeth),
p_p - step of the rack,
m – gear modulus,
$p_p = \pi \cdot m$ (m = 1.5 mm).

First the linear displacement of the table x_L at a step performed by the stepping motor will be calculated:

$$\theta_{ML} = \frac{x_L}{z_p \cdot p_p} \Rightarrow x_L = \theta_p \cdot z_p \cdot p_p, \tag{3}$$

$$x_L = \frac{1.8^\circ \cdot 2\pi}{360^\circ} \cdot 30 \cdot 1.5\pi = 0.45 \cdot \pi^2 \text{ mm},$$

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In order to calculate the pulses frequency f_L one will start from the expression for the angular speed of the motor ω_{ML}:

$$\omega_{ML} = \frac{v_L}{z_p \cdot p_p}, \text{ [rad/sec]} \tag{4}$$

where v_L – longitudinal speed [m/s].

From the above relation it is possible to express the longitudinal speed v_L as:

$$v_L = \omega_L \cdot z_p \cdot p_p. \tag{5}$$

The motor speed n_L may be expressed as:

$$n_L = f_L \cdot \frac{\theta_M}{360^\circ} \cdot 60, \text{ [rev/min].} \tag{6}$$

Considering:

$$N_{pL} = \frac{360^\circ}{\theta_M}, \text{ [steps/rev],} \tag{7}$$

$$N_{pL} = \frac{360^\circ}{1.8^\circ} = 200, \text{ [steps/rev].}$$

Replacing in the initial expression of the motor speed we obtain:

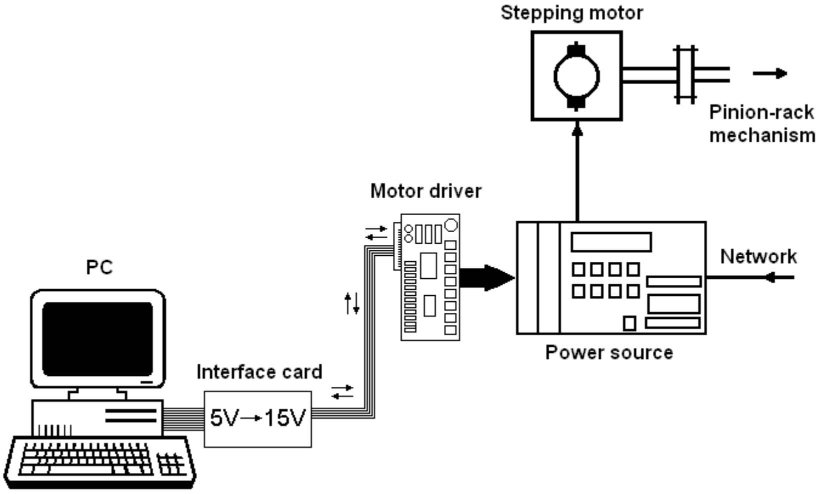

Fig. 5. The longitudinal feed kinematic chain.

$$n_L = \frac{f_L \cdot 60}{N_{pL}} \quad . \qquad (8)$$

The angular speed of the motor may also be expressed as:

$$\omega_{ML} = \frac{\pi \cdot n_L}{30} = \frac{\pi \cdot f_L}{30} \cdot \frac{1}{N_{pL}} \cdot 60 = \frac{2\pi \cdot f_L}{N_{pL}} \cdot \qquad (9)$$

Replacing the expression of ω_{ML} in the initial relation we obtain:

$$\frac{2\pi \cdot f_L}{N_{pL}} = \frac{v_L}{z_p \cdot p_p} \,. \qquad (10)$$

From the above expression we may express the longitudinal speed v_L as:

$$v_L = \frac{2\pi \cdot f_L \cdot z_p \cdot p_p}{N_{pL}} \text{, [m/s].} \qquad (11)$$

In order to determine the pulses frequency f_L, it will be expressed by means of the angular step of the motor, the linear longitudinal speed, and the teeth number of the pinion and the step of the pinion.

Thus the pulses frequency f_L may be expressed as:

$$f_L = \frac{N_{pL} \cdot v_L}{2\pi \cdot z_p \cdot p_p} \text{, [Hz].} \qquad (12)$$

We will transform the linear longitudinal speed from [m/min] to [m/s]:

$$f_L = \frac{200 \cdot \frac{v_L}{60}}{2\pi \cdot 30 \cdot 1.5 \cdot 10^{-3} \cdot \pi} = \frac{200 \cdot v_L \cdot 1000}{\pi^2 \cdot 90 \cdot 60} \,.$$

We impose pulses frequency f_L to be $f_L = 40$ Hz.

Replacing f_L in:

$$f_L = \frac{200 \cdot 1000 \cdot v_L}{90 \cdot 60 \cdot \pi^2} \,,$$

we have:

$$\frac{200 \cdot 1000 \cdot v_L}{90 \cdot 60 \cdot \pi^2} = 40 \,.$$

The longitudinal speed will be:

$$v_L = \frac{40 \cdot 90 \cdot 60 \cdot \pi^2}{200 \cdot 100} = \frac{216 \cdot \pi^2}{200} \cong 10 \text{ , [m/min]}$$

The period of the pulses may be expressed as:

$$T_L = \frac{1}{f_L} = \frac{1}{40} = 0.025\,[s] = 25\,[ms] \cdot \qquad 13)$$

5. THE TANGENTIAL FEED KINEMATIC CHAIN

The tangential feed kinematic chain uses as actuation drive a stepping motor controlled by means of pulses generated through the parallel port of the PC. In order to determine the pulse frequency f_T the following input data will be taken into consideration:

θ_{MT} - angular step of the stepping motor (0.36º)

$N_{pT} = 1000$ steps/rev

β – tilting angle of the hob-gear helix

A calculus for an angle of β = 4°17'9" = 4.28° will be presented

Considering:

$$\tan\beta = \frac{v_T}{v_L} \,, \qquad (14)$$

where:

v_T – tangential speed,
v_L – linear speed,
for the considered example tanβ = 0,074941741.

The tangential speed may be also expressed as:

$$v_T = \omega_{MT} \cdot \frac{d_f}{2}, \tag{15}$$

where:

d_f – hob-gear diameter (for the considered example $d_f = 111.167$ mm)

The motor angular speed ω_{MT} is:

$$\omega_{MT} = \frac{2\pi \cdot f_T}{N_{pT}}. \tag{16}$$

This value for the angular speed will be replaced in the tangential speed formula obtaining:

$$v_T = \frac{2\pi \cdot f_T}{N_{pT}} \cdot \frac{d_f}{2}. \tag{17}$$

Returning to the formula of tg β and replacing with the corresponding values of the two speeds we obtain:

$$\tan\beta = \frac{v_T}{v_L} = \frac{2\pi \cdot f_T \cdot d_f}{2 \cdot N_{pT}} \cdot \frac{N_{pL}}{2\pi \cdot f_L \cdot z_p \cdot p_p}. \tag{18}$$

Replacing the following values:

N_{pL} = 200 steps/rev

N_{pT} = 1000 steps/rev

we obtain:

$$\tan\beta = \frac{2\pi \cdot f_T \cdot d_f}{2 \cdot 1000} \cdot \frac{200}{2\pi \cdot f_L \cdot z_p \cdot p_p} =$$

$$= \frac{f_T \cdot d_f}{10 \cdot f_L \cdot z_p \cdot p_p}. \tag{19}$$

In order to determine the pulse frequency f$_T$, we will extract this value from the above formula:

$$f f_T = \frac{\tan\beta \cdot 10 \cdot f_L \cdot z_p \cdot p_p}{d_f} =$$

$$\frac{0.0749 \cdot 10 \cdot 40 \cdot 30 \cdot 1.5 \cdot \pi}{111.167} = 38.08 \text{ Hz.}$$

It is here noticeable the fact that a difference of 1,92 Hz exists between the two pulse frequencies f_L = 40 Hz and f_T = 38.08 Hz.

From hardware and software points of view regarding the generation of the control pulses through parallel port of the computer it is recommended that the two values f_L and f_T to be equal.

The tilting angle β error could be calculated if using for f_T the value of 40 Hz.

The recalculated tangential speed v_{Treal} will be:

$$v_{Treal} = \frac{2\pi \cdot f_T}{N_{pT}} \cdot \frac{d_f}{2} = \frac{2\pi \cdot 40}{1000} \cdot \frac{111.167}{2} \cdot \frac{60}{1000} =$$

$$= 0.83817943325 \text{ [m/min]}$$

Replacing this values in the formula:

$$\tan\beta = \frac{v_T}{v_L},$$

we obtain:

$$\tan\beta_{real} = \frac{0.83817943325}{\frac{216}{200} \cdot \pi^2}$$

and consequently the real tilting angle of the hob gear helix $\beta_{real} = 4.49°$.

Depending on the imposed accuracy conditions this error may be accepted, or different compensation measures may be taken in order to reduce it.

Complex interpolation algorithms for controlling simultaneous movements may also be used in order to work with different longitudinal and tangential frequencies, an approach which will be considered in future researches.

6. CONCLUSIONS

Taking part from a larger research project co-financed by the national research authorities in the frame of RELANSIN program, this paperwork presented the problems appeared in establishing the kinematic of the hob sharpening process.

The problems tackled in this research were stated by an industrial company which acts as an important supplier in the automotive industry. The above mentioned company manufactures gear sets for a large number of automotive companies. An important part of the gear sets manufacturing process is based upon specialised machine tools which use the hob gear as the main tool.

Dynamic simulation techniques were used during this research for solving the problems posed by the hob sharpening process and checking the solutions.

The proposed developed low costs drives both for the main drive (induction motor controlled through frequency inverters) and feed drives (stepping motors controlled through pulses generated from the personal computer as actuating devices) were presented and implemented in a retrofitted hob-sharpening machine tool kinematic chains.

The problems presented in this paperwork represent only a part of a larger program of developing low cost solutions for retrofitting machine tools.

The proposed developed low cost drives, both for the main drive (induction motor controlled through field controlled frequency inverters) and feed drives (stepping motors controlled through pulses generated from the personal computer as actuating devices) were implemented in retrofitted hob-sharpening machine tool kinematic chains.

Preliminary tests performed on the retrofitted machine had shown that the performances are quite satisfactory and the quality of sharpened hobs is in the range of the geometrical dimensions prescribed by the standards. Both Romanian and European standards were taken into consideration by this point of view.

Furthermore, the preliminary tests also shown that the reliability of the retrofitted machine (seen as the time interval between two break downs) seems to be higher than before the retrofitting process.

The performances demonstrated by the retrofitted drives of the hob sharpening machine allow their inclusion in a medium performance range, which is quite acceptable taking into consideration the overall mentioned goals of the project, which are balancing the development and implementation costs and the performances.

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