Proceedings in Manufacturing Systems, Volume 12, Issue 1, 2017, 3–8 ISSN 2067-9238

HORIZONTAL FEED DRIVES IN HEAVY DUTY MACHINES TOOLS

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Abstract: *This paper presents a methodology for calculating the horizontal feed drives in heavy duty machines tools. Mathematical models presented allow statically and dynamically sizing or verification for heavy duty machine tools such as lathes, vertical lathes, milling machines of Gantry, AFP, AF types, etc. The calculations herein are of a general nature but can accommodate to different constructive variants chosen by the machine tool designer. The paper presents some technological and constructive aspects coming from authors' experience. Taking into account that currently the feed drives have in most of the cases independent driving, it shows the calculation only for this variant of driving specific to CNC machine tools.*

Key words: *horizontal feed drive, heavy machine tools, mathematical models.*

1. INTRODUCTION

The basic structural diagram of a horizontal feed drive is sown in Fig. 1.

The servomotor EM (1) is driving through the gear R (2) the mechanism for transforming the rotational movement into translational movement TM (3). Usually the latter is a ball screw nut or rack-pinion mechanism, with the dimensional ratio i_D. Through this mechanism the speed v is obtained, which is the fedd speed in in machining phases or positioning spped in auxiliary phases [1]. The gear R has the transfer ratio i ($i \le 1$). In some cases, the feed drive may not have a gear. If it is considered that the servomoror speed is n_1 and that of the output of gear n_2, one can consider for steady state:

$$i = \frac{n_2}{n_1}, \tag{1}$$

$$v = n_2 \cdot i_D = n_1 \cdot i \cdot i_D. \tag{2}$$

In mechanisms for transforming rotation into translation of screw-nut type, the dimensional transfer ratio i_D has the expression:

$$i_D = p_S. \tag{3}$$

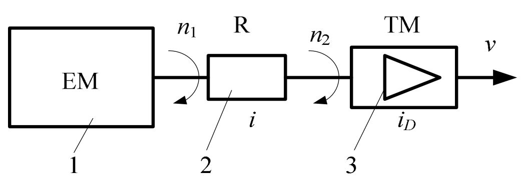

Fig. 1. Structural diagram of a horizontal feed drive.

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In case the transforming mechanism is rack-pinion type, the ratio is:

$$i_D = \pi \cdot m \cdot z. \tag{4}$$

In Eqs. (3) and (4) p_S is the pitch [mm], m – modulus [mm], and z – tooth number.

Usually, the actuators are operated in AC [5] having the possibility of adjusting the speed continuously by varying the frequency. Also, such servomotor is characterized, among others, by maximum speed and rated torque, both given by manufacturers' catalogues [5]. Electronic speed adjustment allows obtaining stable minimum speeds representing 1 / 10 000 – 1 / 8000 of the maximum speed [2]. In these circumstances, the presence of the gear R is justified by the couple amplification need and not by the need of reducing speed. Gearboxes used must be precise, with an acceptable backlasch, which will not affect the accuracy of machine tools [1]. The most used gears today in most heavy machine tools have toothed belts or planetary drives [3].

2. HORIZONTAL FEED DRIVE STATIC CALCULATION

For establishing the mathematical model, the diagram shown in Fig. 2 will be presented.

The slide 2 moves on the guides 1. The workpiece 3 is fixed on the slide and processed by the cutting tool 4. The slide moves with the feed speed v supplied by the servomotor EM (5) through the gear R (6) and rotation to translation transforming mechanism 7. The slide mass is m, and workpiece mass – M. F_H and F_V are horizontal and vertical components of the cutting force respectively. It was also noted: μ – coefficient of friction slide on guides, ω_1 – angular speed of the electric motor, equal to the angular velocity of the stage of gear R, ω_2 – angular velocity of the second gear stage and equal to the angular velocity at entry of the mechanism of transformation of movement f_1, f_2 – viscous friction coefficients proportional to angular velocity of the two gear stages;

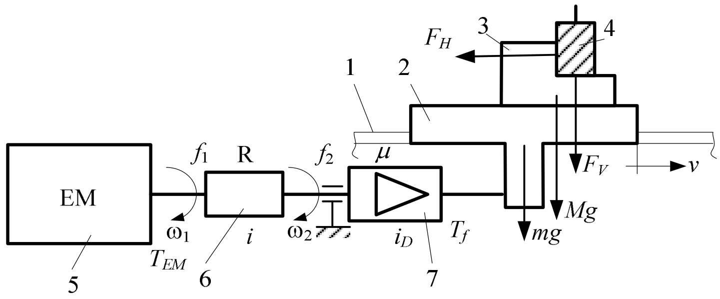

Fig. 2. Diagram used in static calculation.

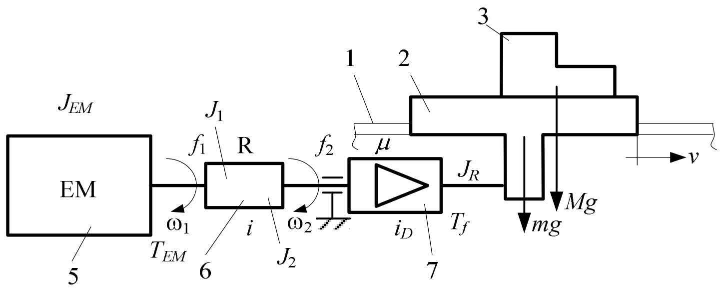

Fig. 3. Calculation diagram for static running.

T_{EM} – torque at electric motor axis; T_f – friction torque (presumed constant) of the mechanism of transformation rotational to translation motion; g – gravity value.

In steady state running, it can be considered that the horizontal force to be defeated by the motion transformation mechanism is F and has the expression:

$$F = \mu \cdot [(M + m) \cdot g + F_V] + F_H. \quad (5)$$

The static torque necessary at the electric motor axis is:

$$T_{EM} = i \cdot \left(\frac{F \cdot i_D}{2\pi} + T_f\right) + \omega_1 \cdot (f_1 + i^2 \cdot f_2) \text{ [N]}. \quad (6)$$

For achieving the feed speed v, the electric motor has the speed n_{EM}:

$$n_{EM} = \frac{v}{i \cdot i_D} \text{ [rpm]}. \quad (7)$$

The literature recommends that the electric motor to develop a torque two times greater than that resulting from Eq. (6) for the hardest working conditions [2]. If a maximum feed speed $v_{\max}$ is required, this leads to setting the maximum speed of the electric motor in feed phase:

$$n_{EMMax} = \frac{v_{Max}}{i \cdot i_D} \text{ [rpm]}. \quad (8)$$

It is recommended that maximum speed not to exceed the rated speed of the electric motor [1, 2, and 5].

3. DYNAMIC CALCULATION OF THE HORIZONTAL FEED DRIVES

For establishing the mathematical model the diagram shown in Fig. 3 is was considered.

In Fig. 3, besides the elements already defined, it was also noted: J_{EM} – moment of inertia of the electric motor; J_1, J_2 – moments of inertia of the steps 1 and 2 of the gear; J_R – reduced moment of inertia of the elements after gear.

In this case, one can consider:

$$F = \mu \cdot g \cdot (M + m), \quad (9)$$

$$T_{EM} = \frac{d\omega_1}{dt} \cdot [J_{EM} + J_1 + i^2 \cdot (J_2 + J_R)] + \omega_1 \cdot (f_1 + i^2 \cdot f_2) + i \cdot (T_f + \frac{i_D \cdot F}{2\pi}) \quad (10)$$

The transfer ratio is $i = 1$ in case of direct drive or has subunitary values. According to Eq. (10), it is obtained:

$$i \in (0{,}1], \quad (11)$$

$$T_{EM(1)} = \frac{d\omega_1}{dt} \cdot (J_{EM} + J_R) + T_f + \frac{i_D \cdot F}{2\pi}. \quad (12)$$

For a gear with the ratio i very small ($i \rightarrow 0$), one obtain:

$$T_{EM(0)} = \lim_{i \to 0} T_{EM} = \frac{d\omega_1}{dt} \cdot (J_{EM} + J_1) + \omega_1 \cdot f_1. \quad (13)$$

From constructive considerations, when using gear units with toothed belts, the transfer ratio i is included in the range [1/4; 1/2]. The planetary gearboxes usually used can have the transfer ratio in the range [1/9; 1/2].

If it is considered that the angular acceleration of the crankshaft is ε_1 and the time required to accelerate from 0 to the angular velocity ω_1 is t_a, one the acceleration can be written as:

$$\varepsilon_1 = \frac{d\omega_1}{dt} \cong \frac{\omega_1}{t_a}. \quad (14)$$

Under these conditions, the Eq. (10) becomes:

$$T_{EM(\varepsilon_1)} = \varepsilon_1 \cdot [J_{EM} + J_1 + i^2 \cdot (J_2 + J_R)] + \omega_1 \cdot (f_1 + i^2 \cdot f_2) + i \cdot (T_f + \frac{i_D \cdot F}{2\pi}). \quad (15)$$

The torque $T_{EM}(\varepsilon_1)$ dependency of the angular acceleration value ε_1 is shown in Fig. 4. The torque T_{EM} depends on the acceleration time value t_a according to the diagram in Fig. 5.

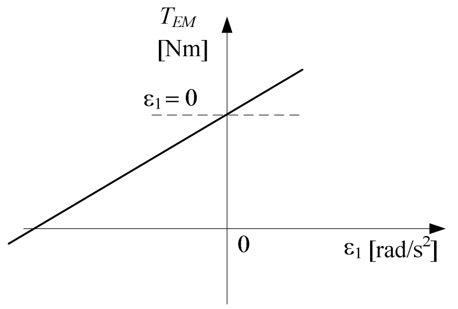

Fig. 4. Torque $T_{EM}(\varepsilon_1)$ dependency of the angular acceleration value ε_1.

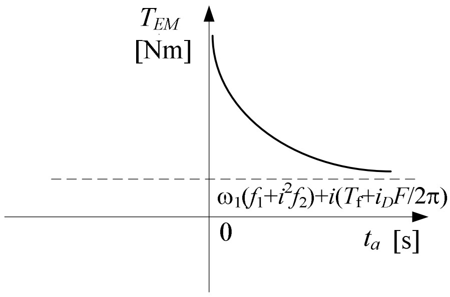

Fig. 5. Torque $T_{EM}(\varepsilon_1)$ dependency on the acceleration time value t_a.

The reduced moment of inertia at the motor axis of the elements after gear is:

$$J_R = J_{TM} + (M + m) \cdot \left(\frac{i_D}{2\pi}\right)^2. \quad (16)$$

In Eq. (16), besides the already known measurements, it was noted J_{TM} – moment of inertia of the mechanism for transforming rotation into translation motion.

Usually, the acceleration time t_a is chosen according to the manufacturer's recommendations for servomotors [5]. Further, the total moment of inertia J_T and total viscous coefficient of friction proportional to the total angular velocity of the motor will be considered according to the expressions:

$$J_T = J_{EM} + J_1 + i^2 \cdot (J_2 + J_R) \text{ [kg m}^2\text{]}, \quad (17)$$

$$f_T = f_1 + i^2 \cdot f_2. \quad (18)$$

The characteristics of servomotors commonly used most are like those shown in Fig. 6.

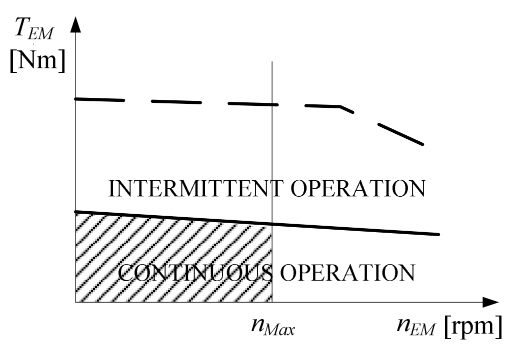

Fig. 6. Servomotors characteristics used in feed drives.

For most of the feed drives of the heavy machine tools, the servomotor torque versis maximum speeed developed in continuous running should be found in the hatched area in Fig. 6.

The necessary torque for accelerating from 0 to maximum speed controlled is

$$T_a = \frac{2\pi}{60} \cdot \frac{n_{EM}}{t_a} \cdot J_T. \quad (19)$$

In these circumstances, it can be considered for the servomotor necessary torque working without acceleration control, as Fig. 5 shows, the expression [5]:

$$T_{EM} = \frac{2\pi}{60} \cdot \frac{n_{EM}}{t_a} \cdot J_T + \frac{2\pi \cdot n_{EM}}{60} \cdot f_T + i \cdot T_f + i \cdot \frac{i_D \cdot F}{2\pi}. \quad (20)$$

The diagram in Fig. 7 shows the servomotor speed progress in time, and the evolution of accelerating torque versus servomotor speed is shown in Fig. 8.

If the actuator works in control loop with position transducer [5], it is considered that the evolution of the rotational speed in the acceleration phase of the actuator has the chracteristic in Fig. 9.

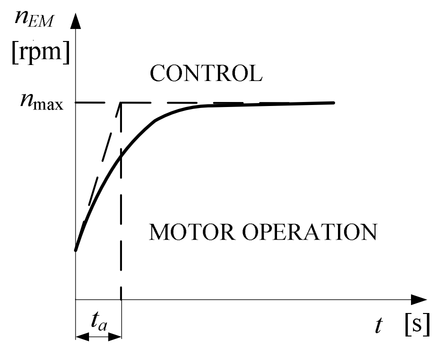

Fig.7. Servomotor speed versus time in running without control loop.

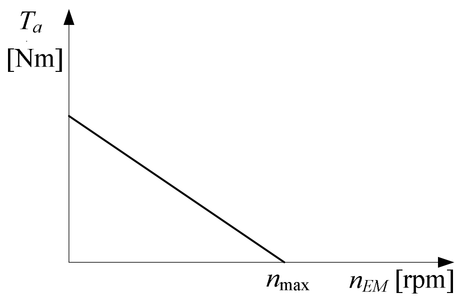

Fig. 8. Accelerating torque evolution versus speed in running without control loop.

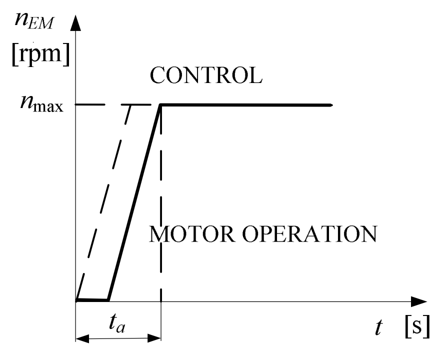

Fig. 9. Servomotor speed progress versus time in horizontal feed drive with using position transducers.

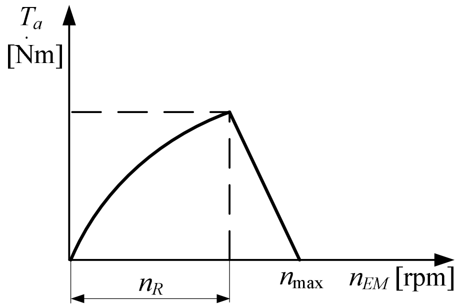

Fig. 10. Servomotor acceleration torque in running with controled loop.

The servomotor necessary torque expression is [5]:

$$T_{EM} = \frac{2\pi}{60} \cdot \frac{n_{EM}}{t_a} \cdot J_T \cdot \left(1 - \frac{1}{e^{k_S \cdot t_a}}\right) + \frac{2\pi \cdot n_{EM}}{60} \cdot f_T + + i \cdot T_f + i \cdot \frac{i_D \cdot F}{2\pi}. \quad (21)$$

In Eq. (21) k_S is the loop motor gain coefficient [s^{-1}] specific to the machine tool serviced.

The dependency of the acceleration torque on motor speed is shown in Fig. 10.

The necessary torque only for accelerating from 0 to maximum speed controled is achieved at speed n_R ingferior to the maximum programed speed n_{Max}. Thi shas the expression [5]:

$$n_R = n_{Max} \cdot \left[1 - \frac{1}{k_S \cdot t_a} \cdot \left(1 - \frac{1}{e^{k_S \cdot t_a}}\right)\right]. \quad (22)$$

For modern machine tools, especially in CNC ones, the accelerating time t_a and system frequency f should satisfy the following relations [2]:

$$t_a < 200\, ms, \quad (23)$$

$$f = \frac{1}{2 \cdot \pi} \sqrt{\frac{K}{M+m}} > 40 Hz. \quad (24)$$

In Eq. (24) K is the feed drive stiffness. This depends on the type of the rotation to translation transformation mechanism and bearings type [2].

4. HORIZONTAL FEED DRIVE TYPES IN HEAVY DUTY MACHINE TOOLS

In heavy duty machine tools the ball screw pitches has the values 10 mm, 12 mm or 20 mm. For using the pinion-rack mechanisms the minimum tooth number of the rack is Z = 20 teeth, and the modulus m is l is at least an order of magnitude higher in-rack pinion gear units. That is why in case of long strokes (c > 6000–8000 mm) the mechanism of this type are preferred.

Obtaining higher speeds in this case meets the demands of productivity imposed. Another reason for preferring pinion-rack mechanisms for big strokes is that resulting from the technology of fabrication of ball screws, which have a maximum length of 6000–8000 mm. Racks are made of sections, which by joining can cover virtually unlimited strokes. There are exceptions in which the ball screws achieved in several sections provide strokes higher than 20,000 mm [1, 5]. In such cases the screw is fixed with intermediate support, the nut having both rotary and linear motions [2].

Further, some real applications of mechanisms screw-nut and pinion-rack are described. For ball screw having rotation motion, the nut is fixed on the slide, which has a translation motion. This is the most common case for strokes less than 5000–6000 mm.

Figure 11 shows such feed drive used by Pietro Carnaghi company [5].

The servomotor 1 drives through the gear 2 the ball screw 4. The spindle is supported by specific bearings 3 and 6 and moves the ball nut 5. The servomotor has a rated speed n_{rated} = 2000 RPM and torque T_{rated} = 38 Nm, for which the motor power is P = 8 kW. The spindle pitch is p = 20 mm and the gear transfer ratio is i = 1/2.5. The maximum motor speed electrically limited is n_{EMMax} = 1875 RPM. In these conditions, the feed drive supplies, according to Eq. (8), a maximum linear speed v_{Max} = 15000 mm/min. The force developed in the guide plane is F = 30000 N along all stroke c = 4000 mm. The used ball screws have double nuts preloaded [2, 4] for reducing the backlash to values less than 2/100 mm.

For a similar machine tool, the company GPM International uses a gear with toothed belts, its constructive solution being presented in Fig. 12.

The electric motor 1 rotates the ball screw 3 via gear reducer 2 having the ration i = 1/2. The spindle has an nominal diameter D = 80 mm and the pitch P = 10 mm. The toothed belt used enables minimum backlash. The electric motor characteristics are like those of the electric motor in Fig. 11. The maximum speed electronically limited is v_{Max} = 10000 mm/min.

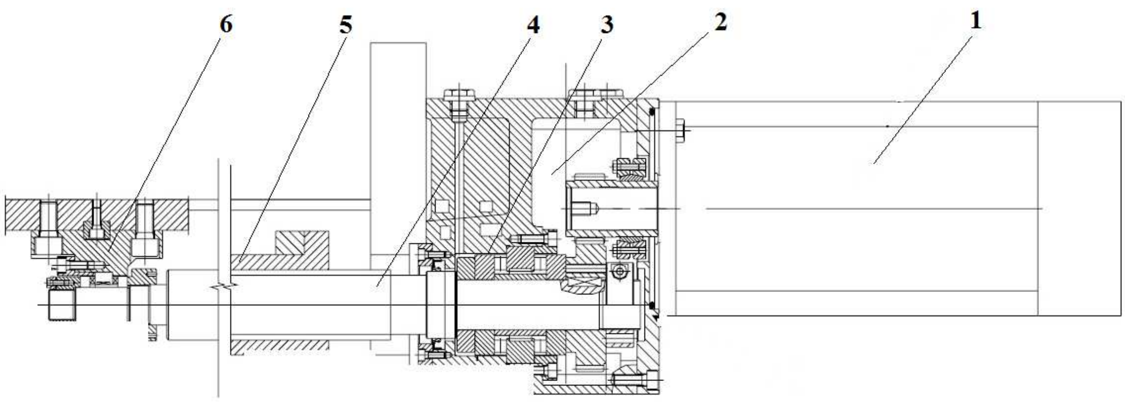

Fig. 11. Horizontal feed drive with rotary screw and gear (1 – electric motor; 2 – gear; 3 – fixed bearing support; 4 ball screw; 5 – nut; 6 – free bearing support).

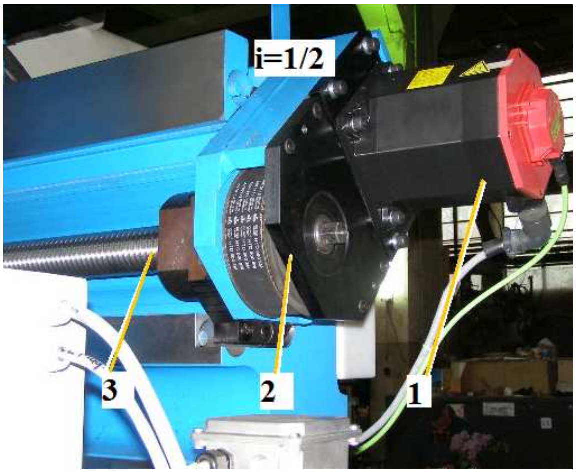

Fig. 12. Horizontal feed drive with rotary spindle and toothed belt gear (1 – servomotor; 2 – gear; 3 – ball screw).

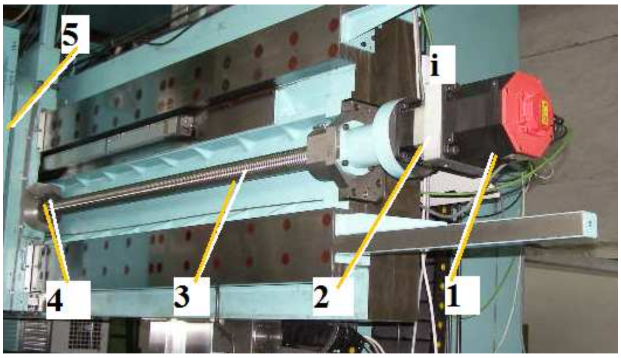

Fig. 13. Horizontal feed drive with rotary spindle and planetary gear (1 – servomotor; 2 – gear; 3 – ball screw, 4 – nut, 5 – slide).

Also for horizontal feed drives, the solution applied by the company Tehnoconsult Invest shown in Fig. 13 can be used. In this case, the electric motor 1 is coaxial with the ball screw 3 by using the planetary gear 2.

The nut moves in linear motion along with the slide. The spindle pitch is P = 10 mm and the planetary gear transfer ratio in the range i = 1/9–1/2. The planetary gear has a special design enabling reduced backlash [1].
The maximum linear speed is in the range v_{Max} = 8000–10000 mm/min. The use gears having transfer small ratio allows the choice of electric motor with small dimensions. The coaxial design of the motor and screw simplifies the construction.

If on the guides moves two slides (1 and 2), as in case of the portal milling machine in Fig. 14, a fixed ball screw 3 is used and one rotary nut for each slide (4 and 5). The nuts are rotated by the electric motors 6 and 7 through the toothed belts 8 and 9.

The slides can work independently or simultaneously, independent adjustment of the electric motors 6 and 7 being possible.

In refabrication of a heavy CNC lathe, for the longitudinal stroke of 20000 mm a feed drive with fixed ball screw and rotary nut was designed. The spindle 1 shown in Fig. 15 has the outer diameter D = 200 mm and pitch P = 20 mm. For taking over the spindle deflection, five supports were provided, which retract at nut passing by operating the hydraulic cylinders 3 [9].

Fig. 14. Two speed drives with rotary nuts and fixed ball screw (1, 2 – slides moving horizontally; 3 – fixed ball screw; 4, 5 – rotary nuts; 6, 7 – feed electric motors; 8, 9 – toothed belts).

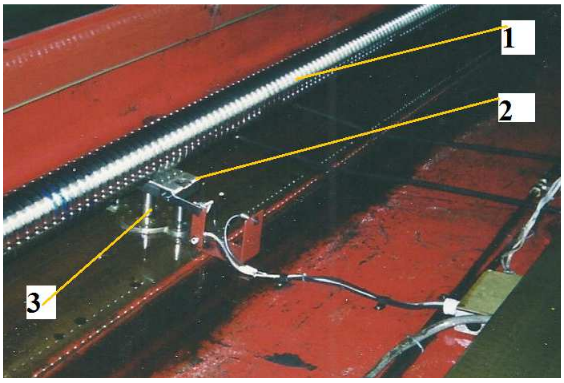

Fig. 15. Feed drive with fixed ball screw and rotary nut in a heavy duty CNC lathe (1 – fixed ball screw; 2 – retractable supports; 3 – hydraulic cylinder).

Ball screw was made of three sections that were mounted directly on the lathe. Taking over its thermal deformation is achieved with a specific hydraulic system [8].

Usually, for strokes greater than 6000–8000 mm, mechanisms for transforming the rotational movement of the rack pinion type are used. They are found in the following variants:
• fixed rack and rolling pinion driving the slide in linear motion;
• rack moving with slide on entire length, the pinion rotating and being supported in the machine frame.

The racks can be used with straight teeth or inclined teeth. For machine tools of type AFP 180–260, the rack is assembled by segments of 1300 mm, which allows obtaining strokes on X-axis having the length 5150 + $n \times$ 1300, where n is the number of sections assembled. On these machines, the rack is fixed. For taking over the backlash different systems are used, which can be mechanical, with springs, hydraulic or electrical [1, 4]. Figure 16 shows the feed drive of such a machine tool.

The electric motor 1, through the reducer 2, is driving two pinions preloaded hydraulically[1, 4], both working on the fixed rack. Given these machines sizes (over 80 t), the maximum speeds achieved on this axis are less than 10,000 mm / min.

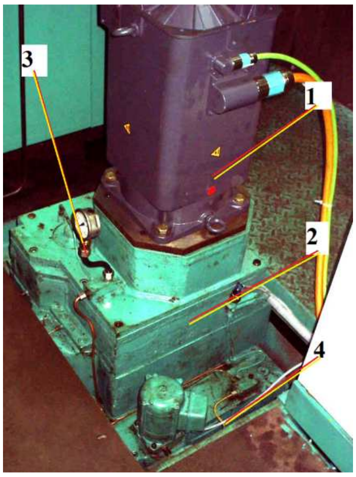

Fig. 16. Feed system on X-axis in machines of AFP type (1– electric motor, 2 – gear with backlash free system; 3 – greasing system; 4 – slide).

In some portal milling machines with movable table, the rack is mounted on the table. Figure 17 shows the table of a machine tool of this type.

On the slide 1 the double rack 2 is mounted. On each side of it a pinion will engage. They have the same speed but are provided with a system (mechanical or hydraulic) for relative tensioning, which allows taking over the backlash [7]. For reducing the friction, the guides 3 and 4 are hydrostatic [1, 2, and 3].

The spur gears (pinions) 2 and 3 in Fig. 18 have 20 teeth and modulus 6 mm each. The feed gear box transfer ratio i < 1/60. The electric motor 4 has the maximum speed n_{Max} = 1000 RPM and torque T_{EM} = 55 Nm.

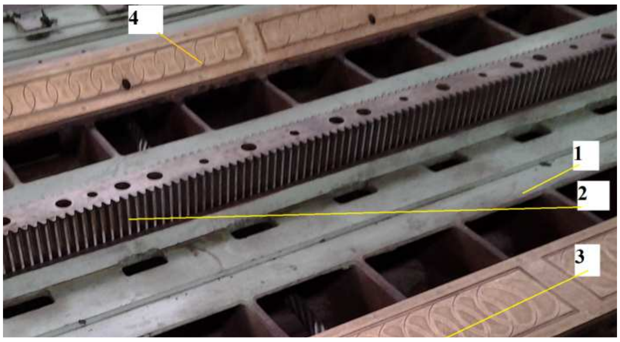

Fig. 17. Moving table of a portal milling machine.

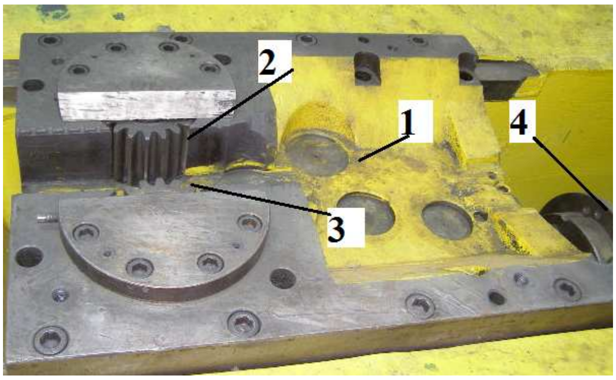

Fig. 18. Feed gear box for the variant with mobile rack.

The feed gear box 1 has the ratio i < 1/60), pinions 2 and 3 with the tooth number Z = 60, and modulus m = 6 mm. The feed drive is actuated by the electric motor 4.

In these conditions, the final element has a maximum speed (electrically limited) v_{Max} = 6000 mm/min enabling a force F = 40000 N.

5. CONCLUSIONS

In case of the heavy duty machine tools (vertical lathes, lathes, milling and boring machines as examples), the horizontal feed drives also achieves the positioning movements with rapid speeds. They use as final mechanisms for rotation-translation transformation the drives ball screw-nut or pinion-rack. In most of cases, the servomotors used currently are AC ones, having the possibility of continuous speed adjustment by frequency variation. Thus, the possible gears interposed between the servomotor and the motion transformation drive have the role of torque (force) amplifiers. Their transfer ratio is sub unitary varying in the range [1/100; 1/2].

It is recommended, after the static calculations, to make a verification dynamic calculation, taking into account the specific elements, among which we can mention: system mass, required acceleration time, maximum rapid speed, viscous friction coefficients of gear, and transfer ratio. The feed drive kinematic diagram of the heavy duty machine tool is set base on these calculations taking into account also the machine tool type.

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