Beam Sag Compensation in Single Column Heavy Duty Vertical Lathes

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Abstract: *This paper shows some of the theoretical and experimental research activities carried out by the authors during the remanufacturing of a heavy duty vertical lathe. The research focused on the diminution of the beam sag (deflection) when the vertical lathe has only one column. The paper presents some of the mathematical models and calculations used for this purpose but also the experimental results along with mechanical-hydraulic solutions enabling the compensation of the deflection caused by the cross-rail own weight but also by the weight of the saddle and rail head in different positions.*

Keywords: *Heavy duty vertical lathe, beam sag compensation, hydraulic systems for compensation*

1. Influence of cross-rail deflection on the accuracy of single column heavy duty vertical lathes

The heavy duty vertical lathes are machine tools meant for the machining of the surfaces by turning, milling and drilling operations, usually in the case of the cylindrical blanks in which the ratio of height and diameter ranges from 0.5 to 0.9 [1, 2]. The structure of such a machine-tool is shown schematically in Figure 1.

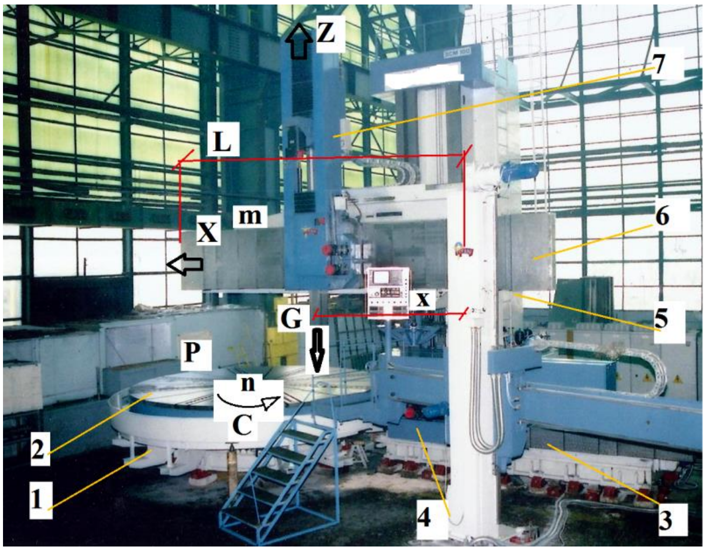

Fig. 1. Vertical lathe with 10000 mm table

The notations used in Figure 1 are the following ones: 1 - bed, 2 - table, 3 - guideways of the under-column saddle, 4 - under-column saddle, 5 - column, 6 - crossrail, 7 - slide and ram assembly (rail head), X, Z, C - CNC axes of the machine, P - plane defining the table surface (horizontal), G - weight of the rail head, n - rotational speed of table 2, m - cross-rail weight distributed along its length, L - cross-rail length, x - current position of the rail head on the cross-rail.

The main kinematic chain, completed by table 2, is assembled on the bed 1 of the machine. The table rotates at speed n in plane P (axis C). The under-column saddle 4 operates on the guideways 3 which are also placed on the bed. The column 5 is clamped on this saddle and together they can perform positioning movements, depending on the diameter of the blank. The cross-rail 6 executes a positioning movement too, this time depending on the height of the blank.

We consider that it has the weight of value m distributed along the length L. The rail head 7 has the weight G and can travel horizontally (X axis), but it also contains a ram that makes the movements vertically - Z axis. The current position of the rail head is determined by the size x.
The basic geometrical conditions required for the execution of specific machining operations can come down to:
- perpendicularity of X and Z directions (axes);
- parallelism of X axis with plane P.

The two geometrical conditions entail also the perpendicularity condition of Z axis on plane P.
Because of the weight G and the distributed weight m, the cross-rail curves and leads to the appearance of the sag f(x) and the angle φ(x), depending on the position x of the rail head, as shown in Figure 2.

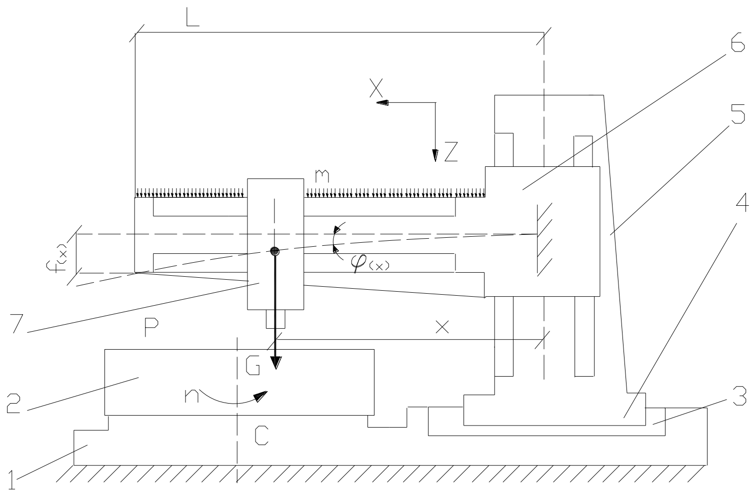

Fig. 2. Block diagram of the vertical lathe

This figure uses the same notations as in the previous figure, plus the following notations: f(x) - cross-rail deflection (sag) depending on the variable x, φ(x) - angle of inclination of the cross-rail related to the horizontal due to weight G and load m.

2. Compensation of the errors entailed by the cross-rail deflection

It is impossible to eliminate totally the errors entailed by cross-rail deflection. There are different methods to reduce these errors. Some of the most frequently used ones are shown below:
– Machining of the guideways that ensure the parallelism with the plane P under a suitable angle for deflection compensation.
– Compensation of the deflections by means of CNC equipment [3].
– Initial sloping of the cross-rail on column guideways with a certain angle.
– Compensation of cross-rail deflection by stressing it with a hydraulic cylinder.

This paper presents the fourth solution. Usually, one or several of the solutions mentioned above can be applied to this type of machines. Figure 3 shows the operating principle of the discharge by means of hydraulic cylinders.
In Figure 3, besides the elements defined in Figures 1 and 2, it is also noted: p(x)- supply pressure, T - tank of the hydraulic unit, F(x) - developed force parallel to X axis. On the cross-rail 6 there is a hydraulic cylinder supplied with oil at p(x) pressure on the active surface. The cylinder draws the cross-rail extension 9 by means of the rod 10. Cross-rail extension has the height a. The bending moment related to the point O, created by the weight G with the arm x and the distributed weight m with the arm L is compensated by the moment given by the force F(x) having the arm a.
For a null resultant moment it is possible to take into consideration:

$$F_{(x)}a = Gx + \frac{mL^2}{2} \tag{1}$$

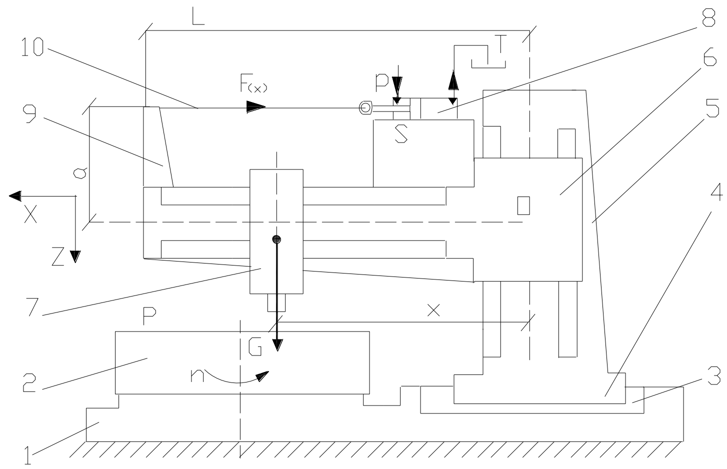

Fig. 3. Beam sag compensation by means of a hydraulic unit

The relation (1) helps to determine the value of the force F(x) depending on the rail head position (size **x**). The necessary pressure too is an **x** function and has the expression below:

$$p_{(x)} = \frac{F_{(x)}}{\frac{\pi(D^2-d^2)}{4}} \qquad (2)$$

The numerical control equipment allows measuring the size x in real time; this size determines the amount of pressure required for the compensation. The pressure is adjusted by means of a proportional reducing valve [3, 4]. Even if this size is continuously measured, in practice it is sent towards the hydraulic equipment as discrete signals with an imposed pitch (100 mm usually). Therefore, the cylinder supply pressure increases as the size x increases. For such a drive it is recommended to use the proportional type hydraulic devices. The method is effective and solves also the X and Z axes perpendicularity issue. This method is also applied in the case of the horizontal boring and milling machines (HBMs) [4, 5]. The major disadvantage of the method is the price of the necessary hydraulic devices.
A hydraulic unit as shown in Figure 4 was built in order to compensate the deflection of the cross-rail by stressing it.
Notations used in Figure 4: P - pump, EM - electric motor for pump actuation, F_1, F_2, F_3 - filters, M_1, M_2, M_3 - manometers, PV - pressure valve, D_1, D_2 - directional valves, PS_1, PS_2, PS_3 - pressure switches, RVP - proportional reducing valve, C - cylinder, PT - pressure transducer, Ac - accumulator, SB - safety block, R - valve for accumulator discharge, Amp. - Electronic amplifier, 1S, 2S, 3S - electromagnets, ST - thermal probe, NI - level gauge.
The pump P, driven by the electric motor EM, sucks the oil from the tank T through the suction filter F_1. The filtration required by such units (3-5 μm) is performed by the filters F_2 and F_3. The maximum operating pressure of the unit is adjusted by means of the pressure valve PV and can be read on the manometer M_1. The hydraulic unit includes – after the check valve CV - a circuit which contains the accumulator Ac [6] assembled on the safety block SB. The accumulator is charged up to the pressure adjusted by the pressure switch PS_1 and is discharged up to the pressure adjusted by the pressure switch PS_2. The discharge of the accumulator at the tank is made by actuating the valve R. The accumulator is coupled to the charging circuit by commanding the electromagnet 1S from the directional valve D_1. The pressure in the accumulator circuit can be viewed any time by means of the manometer M_3. The proportional reducing valve RVP is supplied from the accumulator circuit. This valve is controlled by the machine equipment so that the adjusted pressure increases in the same time with the increase of size x. The control signal originates from the position transducer of X axis from where it is sent to the amplifier block Amp. When the directional valve D_2 is set on the position shown in the figure above, after the actuation of the electromagnet 3S the oil with the pressure adjusted by the proportional reducing valve is sent to

the active surface S of the cylinder C. The value of this pressure is viewed by means of the manometer M_2 and is confirmed electronically by the pressure transducer PT. This one makes possible the view of the pressure on the machine display too. The force developed by the cylinder C, by means of a bar, stresses the cross-rail. The circuit can be discharged by switching the directional valve D_2 by the electromagnet 2S actuation. This status is confirmed by the pressure switch PS_3. The unit also includes a thermal probe ST for controlling the oil temperature and an electric level gauge NI for checking the quantity of oil in the tank.

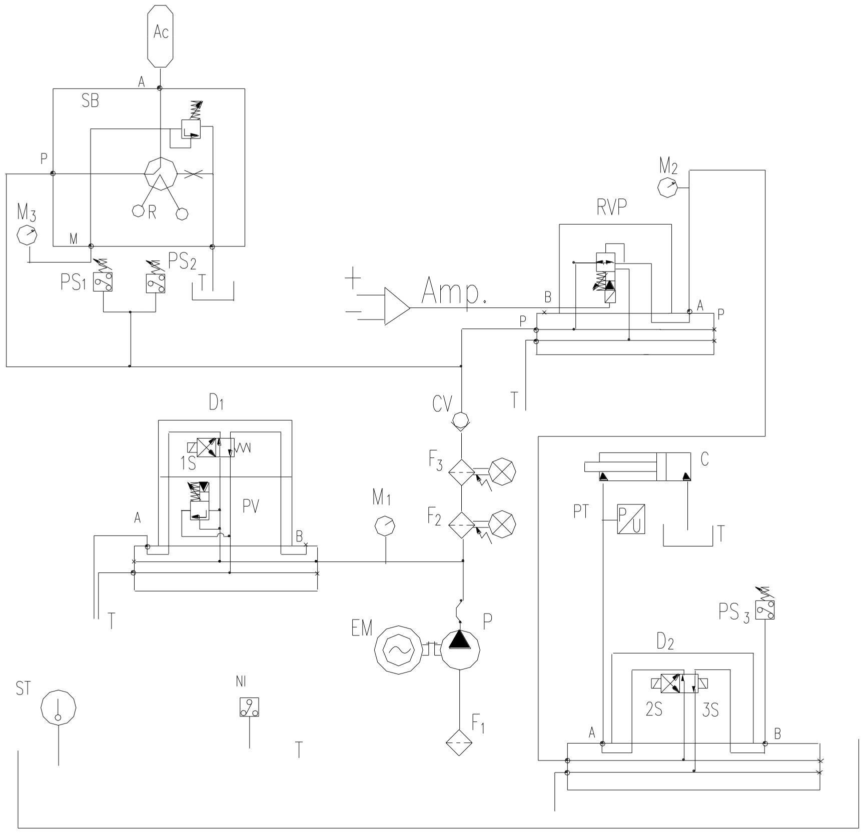

Fig. 4. Hydraulic unit for cross-rail deflection

3. Simulation of the discharge hydraulic system operation

Before the creation of the hydraulic unit – according to the diagram in Figure 4 – we made several simulations using specialized programs. The electric drive motor EM has the power P = 3 KW and the synchronism speed n = 1500 RPM. The used pump P has the capacity q = 4 cm^3. The pressure valve PV is adjusted at p_{MAX} = 230 bar, while the proportional reducing valve RVP can be adjusted continuously [3] in the range of 15 - 200 bar. The accumulator A_C has the volume V_0 = 2.5 l and is charged with nitrogen at the pressure p_0 = 140 bar. The tractate force developed by the cylinder C, at the pressure p = 200bar, is F_{Max} = 11 000 daN.
The accumulator was used in order to reduce the pump operating time under load and to prevent the heating of the unit and the noise as well.
Initially it is considered that the proportional reducing valve is adjusted at a pressure of 25 bar and the pump is started without a pre-control (1S –). When the electromagnet 1S (1S+) is coupled, the circuit charging begins, as shown in Figure 5.
The simulation was meant to monitor the evolution of the pressures in the circuit. Thus the pressures shown by the manometers can be viewed as follows: M_1- supply circuit, M_2 - circuit after the proportional reducing valve and M_3 shows the pressure in the accumulator circuit up to the proportional reducing valve location.
If the pressure regulated at the pressure switch PS_1 is reached, then the (1S-) circuit charging will

be stopped. In this case the pump will be put off, the pressure of the supply circuit decreases and the pressure in the compensation circuit is not affected due to the presence of the accumulator– as shown in Figure 6.

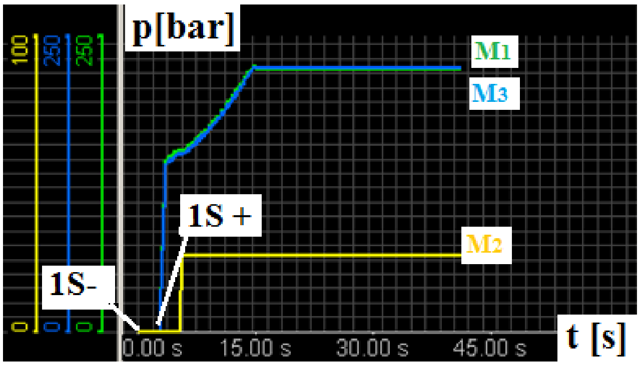

Fig. 5. Circuit charging

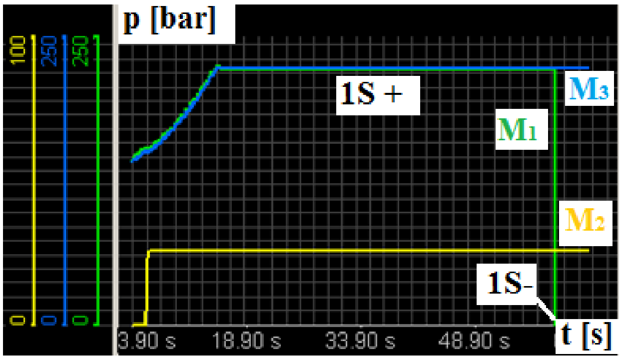

Fig. 6. Stop of circuit charging

Depending on the value of the possible losses in the circuit, the system is able to take over the pressure increases necessary for the compensation of saddle movement towards cross-rail extremity.
If the value of the pressure adjusted at the proportional reducing valve increases from 25 bar to 45 bar, the accumulator is discharged as shown in Figure 7.
In this case, the pressure decrease in the accumulator circuit is negligible. Even if higher compensation pressures are commanded, the necessary oil volume is ensured by discharging the accumulator.
Figure 8 presents the results of the simulation in the case when the following pressures are required: 85 bar, 105 bar, 165 bar and 185 bar.

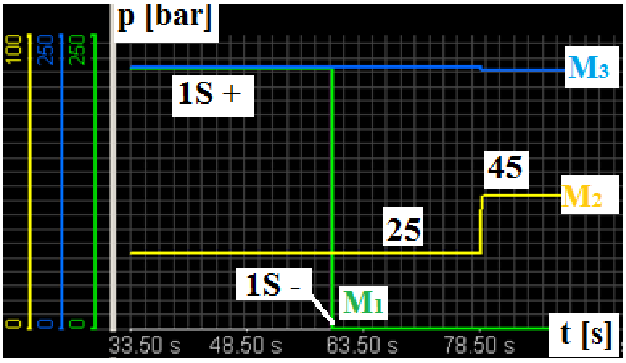

Fig. 7. Evolution of the pressures in the unit if the pressure adjusted at the proportional reducing valve increases

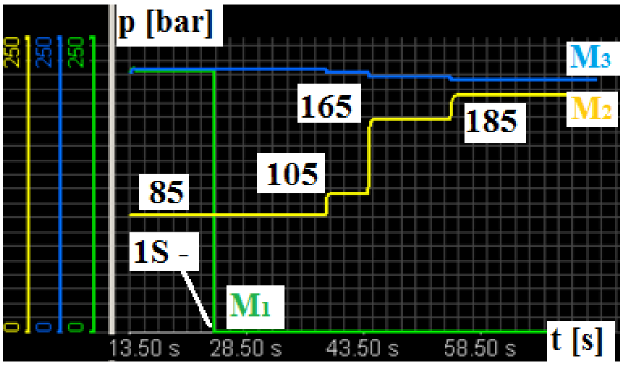

Fig. 8. Evolution of circuit pressures when the saddle moves towards the left end of the cross-rail

The proportional reducing valve receives the command depending on the position of the saddle on the cross-rail and provides the pressures required by the compensation.
The simulation enables the preliminary checks but the real values of adjustment (usually discrete, every hundred of millimeters) are determined experimentally.

4. Experimental research

On the occasion of the remanufacturing [7] of the vertical lathe SCM100, the following compensations of beam sag were made for diminishing the errors of positioning and machining:
A. Initial sloping of the cross-rail on column guideways by an angle φ as in Figure 9.

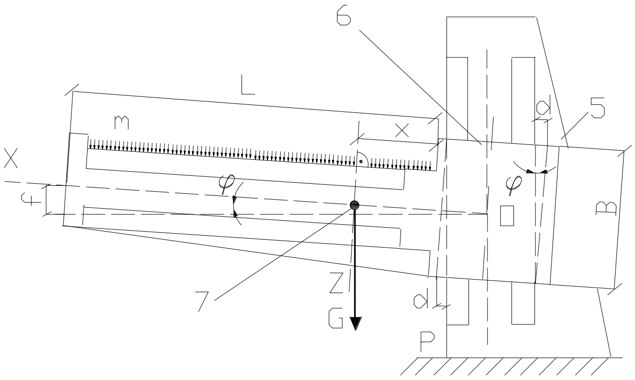

Fig. 9. Compensation of errors by assembling the cross-rail in an angle φ related to the vertical

Notations used in Figure 9: L - cross-rail length, G - weight of the rail head, m - cross-rail distributed weight, X, Z - work axes, x - position of the rail head on X axis, B - guiding width of the cross-rail on the column, f - sag (linear deflection), φ - angle in which the assembling is made, d - size of closing gibs. The other notations are the same as the notations used in the previous figures. If an angle φ is imposed (calculated or experimentally determined), the closing gibs of the guideways of the cross-rail 6 on the column 5 will be so machined to ensure the achievement of the size d as in the figure.
The value of the size d is determined by means of the relation:

$$d = B\frac{f}{L} \tag{3}$$

Keeping the same notations as above, Figure 10 presents the zones of the front guideways (G_F) and back guideways (G_B) of the cross-rail where special closing gibs were assembled. Their angle (φ) is determined experimentally. Closing gibs with φ = 0 will be assembled in the first phase. In these conditions, the sag at the extremity of the cross-rail will be measured in different positions of the rail-head.

Fig. 10. Front guideways (G_F) and back quideways (G_B) of the cross-rail on the column that are adjusted by means of the closing gibs

B. Compensation of the deflection of the cross-rail by stressing it with the help of a hydraulic cylinder

The real hydraulic unit is shown in Figure 11 which uses the same notations as Figure 4. The volume of the tank is 100 l. The accumulator keeps the pressure in circuit enough time after its charging and thus enables even the stop of the electromotor EM. This operation mode (START/STOP) prevents the excessive heating of the unit [6].

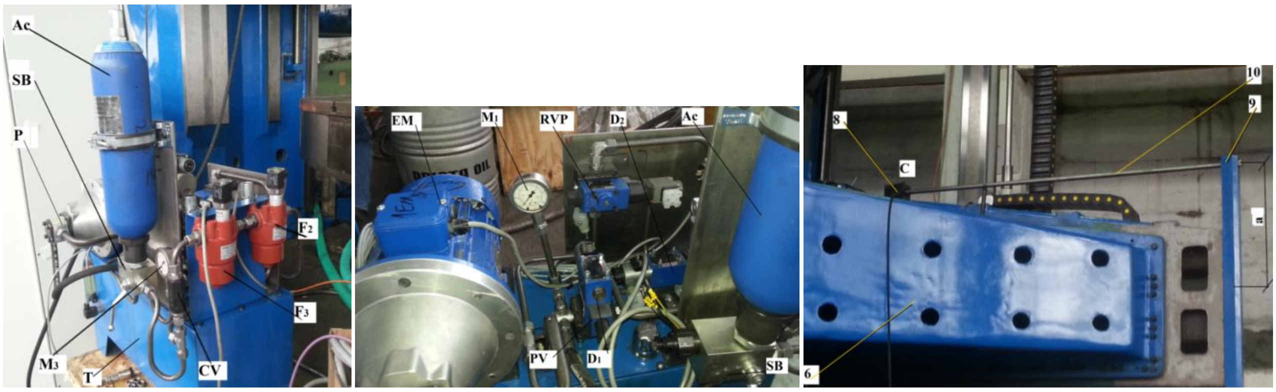

Fig. 11. Real hydraulic unit

Notations in Figure 11 are the same as the ones used in the previous figures.
This hydraulic unit can operate at a maximum pressure of 250 bar with proportional regulation in the range 0-210 bar. The active segment of the cylinder C is the one that determines (in this case) the value of the stressing force F.
In conclusion, we can say that the compensation of the errors caused by the cross-rail deflection in this machine was performed in several steps:

- After grinding the guideways of the column cross-rail (perpendicular to the reference plane P) and the guideways of the cross-rail on which operates the saddle of the rail head parallel to the plane P (X axis), we made the assembling. In the absence of the rail head, measurements were performed to determine the necessary size at the closing gibs.
- After assembling the rail head on its saddle, there were determined – by several attempts – the necessary pressures for beam sag compensation with a pitch of 100 mm on X axis. In this case, it was also measured the perpendicularity between the plane P and the axis Z.
- After finishing the building of the remanufactured machine and after making the correction mentioned above, one has checked the geometrical accuracy conditions. The values recommended by norms are the values for two-columns vertical lathes (machines in which the cross-rail deflection is more reduced thanks to their construction) [4, 7]. Some of these values are mentioned in Table 1.

Table 1: The values of the measurement

Measurement name	Value obtained /value recommended [mm/m]
Table flatness (plane P)	0.04/0.04
Horizontality of compensated cross-rail	0.05/0.04
Straightness of railhead travel (X axis)	0.04/0.04
Parallelism of X axis and plane P	0.05/0.04
Straightness of ram vertical travel (Z axis) related to table axis (plane P)	0.03/0.03
Parallelism of Z axis and table axis (perpendicular to plane P)	0.04/0.03

The table above shows that - thanks to the corrections made - the accuracy of the single column machine is not considerably smaller than the accuracy of the two-columns machines.

5. Conclusions

The heavy duty machine tools are machines suitable for the remanufacturing process. Their overall size, complexity and price justify the investments required by the remanufacturing. New and modern solutions can be applied on the occasion of the remanufacturing, solutions that did not exist or were not applied during the initial manufacture. In the case of the remanufactured heavy duty vertical lathe, intended for the machining of the work pieces up to 100 t by turning, milling, drilling etc. operations, the remanufacturing represented a variant to be preferred to the purchase of a brand new machine. The remanufacture aimed also at improving the machining and positioning accuracy, including the interpolation accuracy. Theoretical research and real measurements were made to this extent. Finally, there were chosen the possible solutions to be actually applied: initial assembling of the cross-rail so that the X axis remains parallel to plane P through deflection; compensation of the deflection of the cross-rail by stressing it with the help of a special hydraulic unit; entry of compensations in the control equipment, with a certain pitch on X axis. Taking into consideration the complexity of the construction and the specific difficulties involved by the assembly of such machine, the values obtained through mathematical models were corrected on the basis of experimental measurements.

References

[1] Joshi, P. H. *Machine tools handbook*. New Delhi, McGraw-Hill Publishing House, 2007.
[2] Perovic, Bozina. *Machine tools handbook/Handbuch Werkzeug-Maschinen*. Munchen, Carl Hanser Verlag, 2006.
[3] *** Catalogues and leaflets GE FANUC, SIEMENS, BOSCH REXROTH.
[4] Prodan, Dan. *Heavy machine tools. Mechanical and Hydraulic Systems/Maşini-unelte grele. Sisteme mecanice si hidraulice*. Bucharest, Printech Publishing House, 2010.
[5] Prodan, Dan, Adrian Ghionea and George Constantin. "Hydraulic system for quill deflection compensation in AFP 180 type machine." Paper presented at International Conference of Scientific Communications "Modern Technologies Quality, Restructuration", Chişinău, Republic of Moldova, Section 6: 145-150, ISBN 9975-9875-2, ISBN 9975-9875-4-1, May 18-21, 2005.
[6] Bucuresteanu, Anca. *Pneumatic Hydraulic Accumulators. Use and Modeling/Acumulatoare pneumohidraulice. Utilizare si modelare*. Bucharest, Printech Publishing House, 2001.
[7] Prodan, Dan and Sorin Marinescu. *Remanufacturing and Modeling of Machine Tools. Hydraulic Systems/Refabricarea şi modelarea maşinilor-unelte. Sisteme hidraulice*. Bucharest, Technical Publishing House, 2005.