CALCULUS METHODOLOGY OF ROTARY TABLES INCLUDED IN POSITIONING AND FEEDING MACHINE TOOLS KINEMATICS CHAINS

Miron ZAPCIU, Adrian GHIONEA, Ionuț GHIONEA, Florea Dorel ANANIA
University POLITEHNICA of Bucharest,
zapcium@sun.cfic.pub.ro, ghionea@imst.msp.pub.ro,
mecosoft@k.ro, dorel.anania@sun.cfic.pub.ro

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Abstract: To realize feeding and positioning in circular motion, many types of machine tools use rotary tables in the structure of boring and milling machine and metal cutting center. Applications are referring at positioning or circular feed kinematics chains. Main stages of calculus for rotary tables that have coupling with face teeth are presented. On the basis of specific data are determined power of the drive motor, design and verification of teeth coupling, hydraulic motor and verification of DC motor included. The methodology offers to mechanical engineers an algorithm to define and dimensioning kinematics chain with rotary table.

1. INTRODUCTION

The rotary tables (M_r) are used frequently in the structures of boring and drilling machine, milling machine and metal cutting center. These are realized in different constructive variants with different drive possibilities. Applications are referring at positioning or circular feed kinematics chains. The circular motion of the mobile elements, which support workpiece, is accomplished on horizontal or vertical plane in both senses. Ensemble of M_r is fixed on the longitudinal saddle of machine tool or it is integrated in this one [1].

From constructive point of view, M_r are composed by circular casting, carcass, drive kinematics chain (electric motor, belt transmissions, gears), blocking and unblocking mechanisms, coupling, simple joints for circular or straight motion, elements for clearance compensation, position encoder. Some of M_r have the surfaces which allow guidance and workpiece pallet fixation, when machine tool is integrated into FMS.

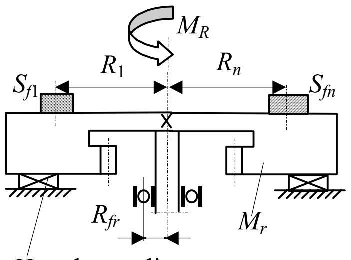
DC motors drive most of rotary tables used in the structure of CNC machine tool. The speed of rotary table is n_{Mr} = 0,003 … 2 *rpm* in circular feed motion and $n_{Mr} \approx$ 4 *rpm* in positioning. The coupling with face teeth is used to realize rigidity and precision of positioning for the rotary table. Angular precision is ±1″… ±3″ for special cases and ±3″… ±12″ for high precision domain.

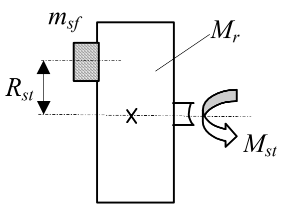
Blocking mechanism has mechanical action (helical or disc springs) and unblocking linear motion is realized with a hydraulic motor.

2. CALCULUS METHODOLOGY

The stages of calculus presented in the table 1 are considering the dimensioning and verification activities of main constructive elements of rotary tables.

Table 1 Calculus methodology

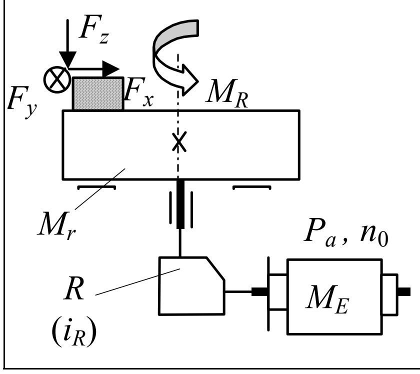
<table>
<tr><th>Stages of calculus</th><th>Calculus relations</th><th>Recommendations</th></tr>
<tr><td>1</td><td>2</td><td>3</td></tr>
<tr>
<td>1. Total torsion moment on the rotary table shaft MM_R,
in N·m;
M_{din} – dynamic moment do it by inertial forces, in N·m;
M_{fr} –friction moment into bearing of rotary table, in N·m;
M_{st} –static moment do it by unbalanced masses, in N·m;



</td>
<td>$M_R = M_{din} + M_{fr} + M_{st}$;
$M_{din} = \left(\frac{m_{Mr} \cdot R_{Mr}^2}{2} + \sum_{i=1}^{n} m_{sfi} \cdot R_i^2 \right) \varepsilon_a$,
where m_{Mr} -mass of rotary table, in kg;
R_{Mr} -rotary table radius, in m;
m_{sfi} -workpieces masses turns by M_r,
in kg;
R_i- position of the workpieces from turn axes, in m;
ε_a –angular acceleration or deceleration (the biggest value) of rotary table, in rad/s^2;
$M_{fr} = \mu\ (m_{Mr} + \sum_{i=1}^{n} m_{sfi}) g \cdot R_{fr}$,
where μ -friction coefficient;
g –gravity acceleration, in m/s^2;
R_{fr} –friction radius, in m;

$M_{st} = m_{sf} \cdot g \cdot R_{st}$,
where m_{sf} –workpiece mass in unbalanced position, in kg;

R_{st} –length between workpiece and rotary axis, in m;</td>
<td>$\varepsilon_a = \frac{\omega_r}{t_a}$;
ω_r –angular stable speed, in rad/s;
ω_r =0,5…1 rad/s;
t_a –acceleration time, in s;
t_a =0,2…0,4 s;

μ =0,008…0,01 for friction bearing case;
μ =0,1…0,2 for slide friction case;

M_{st} is considered only for rotary table with horizontal axis;</td>
</tr>
<tr>
<td>2. Power of the motor drive P_a, in kW;

</td>
<td>$P_a = \frac{M_R \cdot n_{Mr}}{9550 \cdot \eta}$,
where n_{Mr} -stable speed of the rotary table, in rot/min;
$n_{Mr} = 30\omega_r / \pi$;

η -output of the kinematics chain for rotary table turning;</td>
<td>η =0,78…0,92 (product of the outputs of the kinematics chain mechanisms).</td>
</tr>
</table>

Table 1 (continuation)

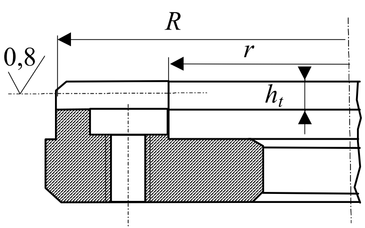
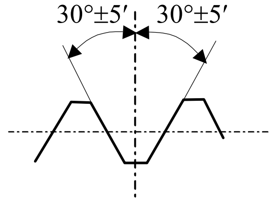
1	2	3
3. Hearth coupling calculus -Outside diameter has the maximal value normalized up to outside diameter of the rotary table; -Verifying the Hearth coupling: -*Torsion stress calculus*; -*Initial force calculus*; -*Verification of screws coupling fixing*; -*Verification of contact pressure on the teeth*; -*Verification of the base tooth bending*.  Reference teeth angle (α): 	-***Tangential force calculus***: $F_t = M_{cp} / R_m$, where F_t is tangential force according M_{cp} moment, in N; M_{cp} –total moment of cutting process, in $N{\cdot}m$; R_m –average radius of the coupling, in m; $f_t = F_t / z$, where f_t is tangential force transmitted by one tooth, in N; z –teeth number of the coupling; -***Initial force calculus***: $F_a = (1{,}5 \ldots 2)\, F_t \dfrac{\sin 2\alpha}{2}$, where F_a is initial axial force, in N; α -inclination angle of teeth, in *deg*; $f_a = F_a / z$, where f_a is axial force transmitted by one tooth, in N; -***Verification of screws coupling fixing***: $\tau_{fs} = F_t /(n \cdot A_1)$; $\tau_{fs} \le \tau_a$; where τ_{fs} is screw cutting stress, in N/mm^2; n –number of screws; A_1 –single screw surface section, in mm^2; τ_a -accepted stress, in N/mm^2; -***Verification of contact pressure on the teeth***: $p = \dfrac{\dfrac{F_a}{2 \cdot \tan\alpha} + \dfrac{M_{cp}}{2 \cdot R_m}}{2(R-r) \cdot 0{,}68\, h_t \cdot c_p} \le p_a$, where p –contact pressure, in N/mm^2; R, r –outside, inside radius of teeth, in mm; h_t –high of tooth, in mm; c_p –real contact spot, in %; p_a –accepted pressure, in N/mm^2; -***Verification of the base tooth bending***: $\sigma_i = \dfrac{6 \cdot f_t \cdot h_t \cdot z^2}{\pi \cdot (R+r) \cdot (R-r)^2} \le \sigma_{ai}$; where σ_i is real bending stress, in N/mm^2; σ_{ai} –accepted bending stress, in N/mm^2;	Diameter series: 20, 25, 28, 32, 36, 40, 45, 48, 50, 56, 63, 71; z =24…240; n = 8…18; τ_a=40…60 N/mm^2; c_p=40…60%; p_a =80…120 N/mm^2; σ_{ai} = 40…50 N/mm^2 for carbon steel; σ_{ai} =90…140 N/mm^2 for alloy steel case;

Table 1 **(continuation)**

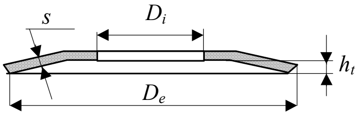
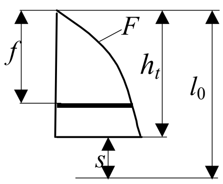
1	2	3
	-Total assurance axial force (F_{ts}); $F_{ts} = F_a \cdot c_a$, where c_a –safety coefficient;	c_a =1,5…2;
4. Fixing system calculus with STAS 8216-80 considering.	-Disc springs are fixed initial at the deformation f_1 = (0,3…0,5)·h, where h is maximal deformation of the single spring; -Load on single disc spring, that corresponding of the deformation f_1 is: $F_1 = \frac{4E}{1-\mu^2} \cdot \frac{s^4}{\alpha D_e^2} \cdot \frac{f_1}{s} \cdot \left[\left(\frac{h}{s} - \frac{3f_1}{2s} \right) + 1 \right]$,	E=2,1·10^5 N/mm^2; μ =0,3; $\frac{4E}{1-\mu^2} = 9{,}23 \cdot 10^5$ N/mm^2;
Disc springs; Main dimensions: 	where E is Young modulus, in N/mm^2; μ –Poisson coefficient; s –thickness of disc spring, in mm; $\alpha = \frac{[(\delta-1)/\delta\ ]^2}{\pi\left(\frac{\delta+1}{\delta-1} - \frac{2}{\ln\delta}\right)}$ where $\delta = \frac{D_e}{D_i}$; D_e –outside diameter of spring, in mm; D_i –inside diameter of spring, in mm; -Number of springs in group is according to relation $n_a \geq F_{ts}/F_1$, where n_a is rounded as first integer value;	
Charge diagram: 	-Number of spring groups to be able that assure linear displacement of hydraulic motor and coupling opening: $i = (h_t + h_s) / (f - f_1)$, where h_t is high of the teeth Hearth coupling, in mm; h_s –security high for coupling opening, in mm; f –displacement of spring that corresponding at 0,75·h compression, in mm;	h_s =1…3 mm; f is according with STAS 8216-80
	-Free length of spring groups: $L_0 = i\ [l_0 + (n_a - 1)s]$, where l_0 is free length for single spring, in mm; -Length of the spring groups when the springs are compressed to 0,75·h is: $L = L_0 - i \cdot f$; -Length of the spring groups after hydraulic motor assembly: $L_2 = L_0 - i \cdot f_1$;	i is superior rounded as integer value;

Table 1 (continuation)

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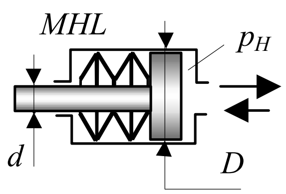
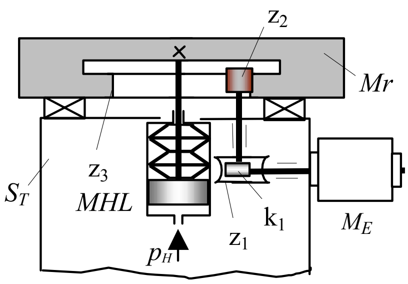
<td>

5. Hydraulic motor drive calculus

Main dimensions:



Using example:



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<td>

-Force for Hearth coupling opening (F_H), in *N*:

$F_H \geq (m_{Mr} + m_{sf})g + F_{fr} + n_a \cdot F$,

where F_{fr} is friction force into hydraulic motor, in *N*; *F* –standardized force when the spring is compressed to 0,75·*h*;

-Diameter of the hydraulic cylinder (*D*), in *mm*:

$D = \sqrt{\frac{4F_H}{p_H \cdot \pi}}$, where p_H is recommended work pressure for auxiliary chain drive, in N/mm^2.

-Thickness of hydraulic cylinder (δ_H), in *mm*:

$\delta_H \geq \frac{p_H \cdot D}{2 \cdot \sigma_{at}}$, where σ_{at} is admitted stress for traction solicitation, in N/mm^2;

-Necessary flow for hydraulic pump (Q_p), in *l/min*:

$Q_p = (1,1...1,15) \cdot 10^{-3} \cdot v_H \cdot \frac{\pi D^2}{4}$,

where v_H is speed value for hydraulic motor drive, in *m/min*;

-Diameter of pipe (d_c), in *mm*:

$d_c = \sqrt{4 \cdot 10^2 Q_p / (v_o \cdot \pi)}$, where v_o is speed of the oil into pipes system;

-Tank volume, V=4Q_p;

-Power of the electric motor for pump drive (P_{ME}), in *kW*:

$P_{ME} = \frac{1}{60} \cdot p_H \cdot Q_p$.

</td>
<td>

p_H =2…5 N/mm^2;

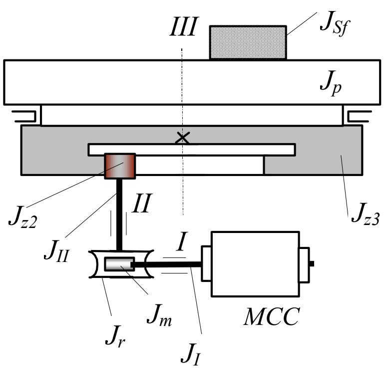
σ_{at} =110…130 N/mm^2;

v_H =2…4 *m/min*;

v_o =120…240 *m/min*;

</td>
</tr>
</table>

Table 1 (continuation)

1	2	3
6. Verification of electric motor for kinematics chain drive Kinematics chain example:  $i_1 = k / z_{rm}$, where k is number of threadlike teeth for worm gear; z_{rm} –number of tooth for worm wheel; $i_2 = z_2 / z_3$, where z_2 and z_3 are numbers of teeth for pinion and gear in spur gears.	-Kinematics values verification: $n_{\min}^{ME} \le n_{Mr} / i_R \le n_{\max}^{ME}$, where $n_{\min}^{ME}$ and $n_{\max}^{ME}$ are minimal speed of the motor drive and maximal speed respectively; i_R –kinematics transfer ratio for all mechanisms between electric motor and rotary table; -Nominal torque verification (M_n^{ME}), in *N·m*: $M_n^{ME} \ge k_s \cdot (2-\eta) \cdot M_R \cdot i_R$; where k_s is the safety coefficient; -Starting torque verification (M_p^{ME} , in *N·m*): $M_p^{ME} = \left[J_I + J_m + (J^{II} + J^{III} \cdot i_2^{2})\, i_1^2\right] \cdot \frac{\pi\, n_{ME}}{30 \cdot t_a^{ME}}$ where n_{ME} and t_a^{ME} are speed and acceleration time characteristics of electric motor; J^{II} -inertial moment of axis *II;* $J^{II} = J_{z2} + J_{rm} + J_{II}$; J_{z2} -inertial moment of z_2 pinion, J_{rm} – inertial moment of worm wheel, J_{II} - inertial moment of shaft *II*; J^{III} is inertial moment of axis *III;* $J^{III} = J_{z3} + J_p + J_{Sf}$; J_{z3} –inertial moment of z_3 wheel, J_p – inertial moment of rotary table part, J_{Sf} –inertial moment of workpiece considering axis *III* that reference.	k_s = 1,1…1,2; For an cylindrical body: $J = m \cdot R^2 / 2$, where m is the mass and R – outside radius of cylinder. M_p^{ME} must to be lowest than folder value of the motor drive.

3. CALCULUS EXAMPLE

Horizontally rotary table with following characteristics:
ω_r =1.5 *rad/s*; t_a=0,25 *s*; R_{Mr}=0.6 *m*; m_{Mr}=250 *kg*; m_{Sf}=50 *kg*; R_i=0.5 *m*;
Total torsion moment on the rotary table shaft (M_R);
M_{din}=431.2 *N·m*; μ=0,01; R_{fr}=100 *mm*; g=9,81 *m/s*^{2}; M_{fr}=2,9 *N·m*; M_R=434.1 *N·m*.
Power of the motor drive (P_a); η=0,8; $n_{Mr}=30\omega_r/\pi$=14,3 *rot/min*; P_a=0,81 *kW*;

Hearth coupling data; z=144; h_t=5,16 *mm*; α=30°; R=280 *mm*; r=225 *mm*; M_{cp}=900 $N\cdot m$;

-Tangential force calculus: R_m=(R+r)/2=252.5 *mm*; F_t=3564 *N*; f_t=24,7 *N*;

-Initial force calculus: F_a=2700 *N*; n=10 screws M10; f_a=18,7 *N*;

-Verification of screws coupling fixing: τ_{fs}=3,4 N/mm^2 <40 N/mm^2;

-Verification of contact pressure on the teeth: c_p=40%; p=14,3 N/mm^2 <80 N/mm^2;

-Verification of the base tooth bending: σ_i=3,2 N/mm^2 <40 N/mm^2;

Fixing system calculus; disc springs B80 with following characteristics: D_e=80 *mm*; D_i=41*mm;* s=3,0 *mm*; h=2,3 *mm*; l_0=5,3 *mm*; F=10700 *N*;

-Total assurance axial force F_{ts}= 4725 *N*; safety coefficient, c_a=1,75; initial deformation, f_1 = 0,3·h =0,69 *mm*;

-Load on single disc spring, F_1=2430 *N*; number of springs in group, n_a=2; number of groups i=7; Lengths of spring groups: L_0=6 *mm*; L=46 *mm*; L_2=53 *mm*;

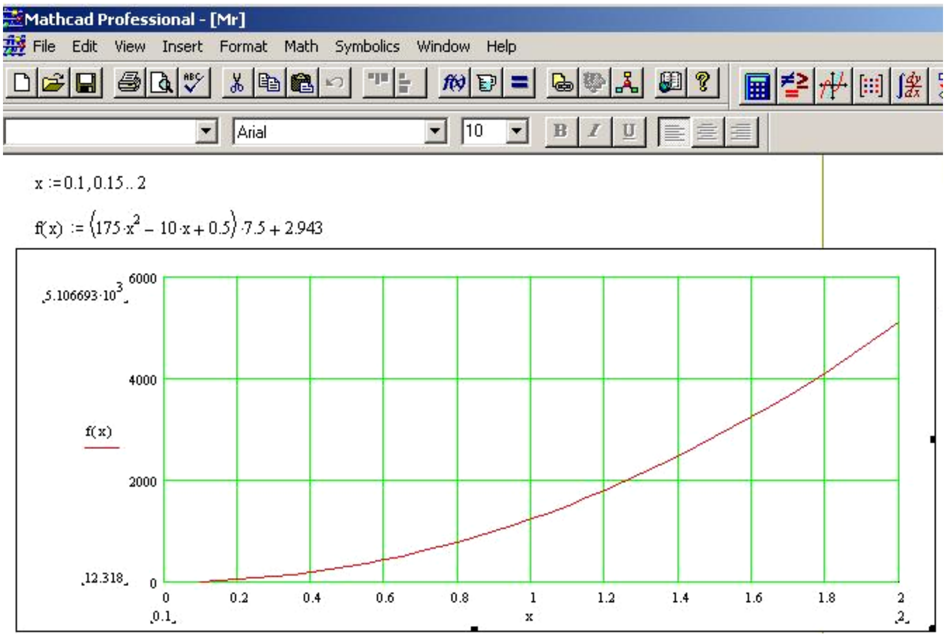
Hydraulic motor drive calculus; F_{fr}=20 *N*; p_H=2,0 N/mm^2; σ_{at}=110 N/mm^2;

-Force for Hearth coupling opening, F_H=24500 *N*;

-Diameter of the hydraulic cylinder D=120 *mm*; thickness of hydraulic cylinder δ_H=3 *mm* (≥1,4 *mm*); diameter of pipe d_c=6 *mm*; v_H=3 *mm/min*.

-Necessary flow for hydraulic pump, Q_p≥25,9 *l/min*; power of the electric motor for pump drive P_{ME}=0,86 *kW*;

DC motor drive of rotary table: $i_R=i_1\cdot i_2$=1/30·1/10=1/300; $M_n^{ME} \geq 2$ $N\cdot m$; $M_p^{ME} \geq 4$ $N\cdot m$.



For the above case studied, was defined a mathematical function for the parameter M_R in the case of using m_{Mr}=250 *kg*, R_i=R_{Mr}-0.1; ε_a =7.5 m/s^2; μ = 0.01; R_{fr} = 0.1 *m*. Using the software Mathcad Professional, in the figure 1 is presented the influence of the radius of rotary table (x scale) on the variation of the total torsion moment M_R.(function f(x)).

Fig.1 Total torsion moment

In the table 2 are presented main characteristics for an example of DC motors to drive the rotary table.

Table 2. Characteristics of the Siemens DC motors 1G5

Rate speed n_N	Rated output P_N	Rated Torque M_N	Maximum speed	Order No.	Rated current I_N	Efficiency	Armature resistance at 120ºC
1170 rpm	1,09 kW	8,0 N·m	1400 rpm	1G#5 102-0 ED	5,4 A	63%	10.7 Ω

4. CAD OF ROTARY TABLE

On the basis of relations exposed in the table 1, were conceived software calculus programs to design the main elements founded in the structure of rotary table. Were considered a DC motor drive, the coupling, the disc springs and hydraulic motor.

The ensemble drawing of rotary table is presented in the figure 2 and it was obtained using AutoCAD release 14.

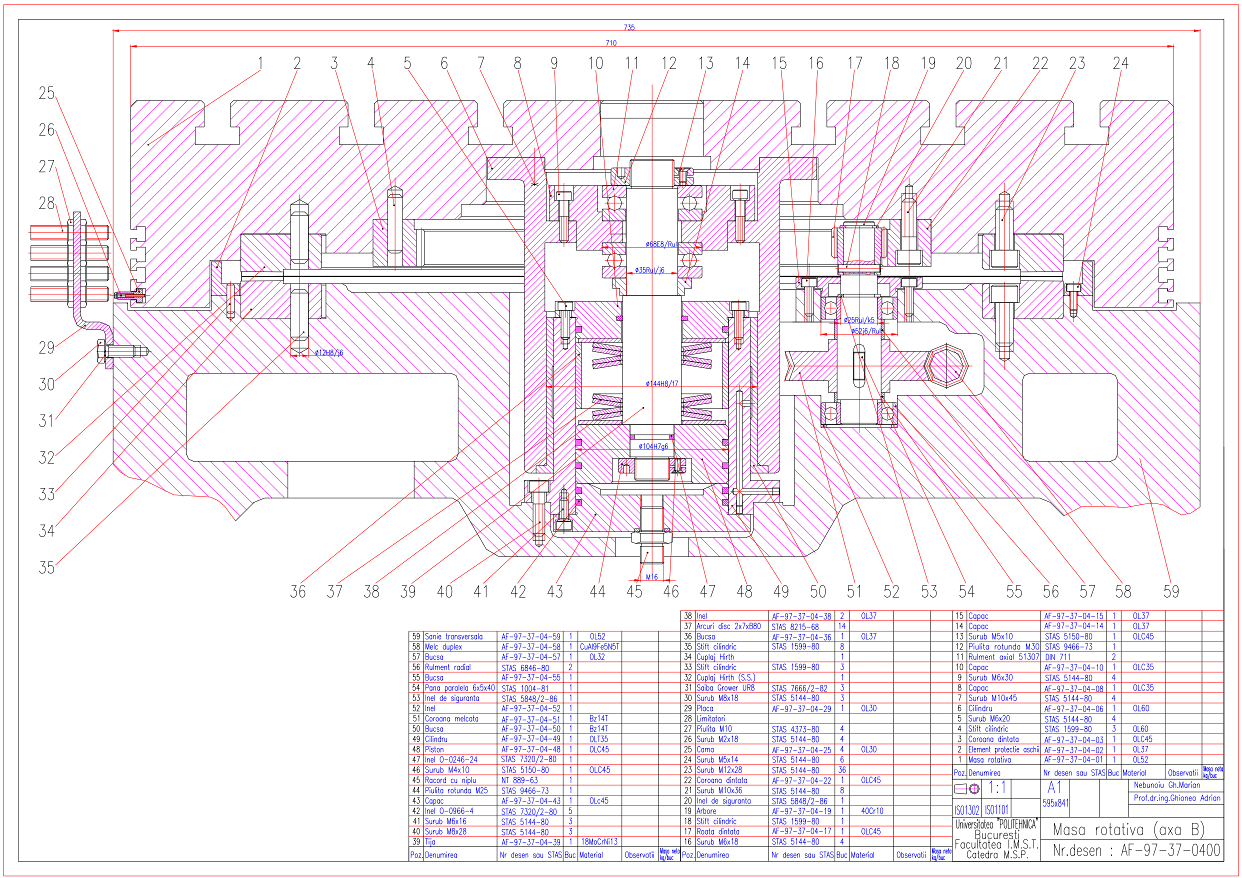


Fig. 2 Rotary table

5. CONCLUSIONS

The methodology presented offers to mechanical designers a unitary algorithm for defines positioning kinematics chain that is using rotary table. At the end of calculus example were resulted consistent results, according with those from rotary table, which equips a real machine tool. In order to obtaining final dimensions for ensemble is necessary to take into account-standardized values. These are chosen using constructive criterion or technological and fitting criteria.

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