NUMERICAL AND ANALYTICAL MODELLING OF A PALLET USED AT A MACHINING CENTRE AT HIGH SPEED

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A workpiece manipulation assembly includes two main parts. 1. A base permanently connected to a spindle. 2. A pallet manipulated by a rotary manipulation table from working place to setting place & vice versa. This assembly is rotating around the spindle axis. The FEM analysis and analytical analysis of the workpiece exchange manipulation system of vertical turning centre, deals with stress and deformation of an assembly that is composed by the base, the palette and the other parts, at high revolution load. As requirements of customers increase – revolutions of the workpieces increase. This leads to increase the load of parts.

Keywords
machine tool, high speed, workpiece pallet exchange, FEM

1. Introduction

Machine tools represent one group of the essential machinery necessary to manufacture in principle all products which can be found in our surrounding. One machine tool typology group consists of vertical machining centres specified for rotary workpieces. These machining centres serve to manufacture rotary parts having the diameter within the range from some centimetres up to some meters. The workpiece pallet exchange system is one of the ways how to minimize non-productive time and how to obtain higher manufacturing productivity at the same time. The pallet exchange system consists of interconnected various parts which create together a complex technical system. Considering the functional layout, design and technological requirements, it is not always possible to arrange the parts symmetrically round the axis of table rotation. The particular carrying parts have complicated shapes and it is difficult to keep their symmetry to more planes than to one. Moreover, the part shape is rather divided in many cases, which causes their space unbalance. The other way how to increase productivity is to increase the cutting speed. At vertical machining centres the main motion is represented by the rotary motion of a workpiece carried by the pallet, where the workpiece is chucked.

2. Pallet exchange system at vertical turning centres

Vertical turning centres are machine tools [Weck 1979] which can be equipped also with two rail heads. This eliminates considerably idle time, if machining is performed simultaneously by the both rai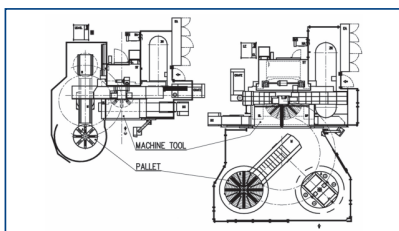l heads. The machining centre shall be understood as a machine specified for chip machining which enables to perform this machining by means of more technological operation kinds (drilling, boring, milling, threading, etc.). For these reasons, the right rail head is usually equipped with turning tools as well as with rotary tools. The left rail head is usually equipped only with turning tools. Fig. 1 shows the typical layout of the pallet exchange system used at vertical turning centres and specified for the machines having the different pallet diameters.

Figure 1. Two possibilities of vertical machining centres made by TOSHULIN

3. Balancing of parts

Balancing of rotating parts decreases the force load of bearings, i. e. the intensity of the force causing the machine vibration, which influences the machine run and its operation in a very favourable way. In dependence of the CSN ISO 1940 and NZPS 5202 standard, rotors are divided in two groups:
- rotors with one balancing plane (static balancing);
- rotors with two balancing groups (dynamic balancing).

The maximum admissible unbalance depends on the rotor size, on its mass as well as on its speed. The maximum value related to the rotor weight:

$$e_p = \frac{U_p}{m} \qquad (1)$$

e_p – specific unbalance [μm],
U_p – admissible unbalance [gmm],
m – rotor weight [kg],
Rotors are arranged in grades according to the (required) balancing quality:

$$e_p \cdot \omega = konst. = \text{balancing quality grade [mm.s}^{-1}] \quad (2)$$

$$\omega = 2 \cdot \pi \cdot n_p \qquad (3)$$

where:
ω – angular speed [s^{-1}],
n_p – operating speed [s^{-1}].

The balancing quality grades are designated G0,4; G1; G2,5; G16, etc. The numerical value in this designation represents the value of the above-mentioned constant. Typical representatives from technical practice are assigned to the particular grades. Usual machine-tool spindles belong to groups G1 and G2,5.

If the pallet exchange system is simplified to one body only and if the rotor, bearings and gears are not considered, then the following is valid for the group G2,5, the speed of 1500 min^{-1} and the pallet exchange system weight of 800 kg:
Admissible specific unbalance e_p:

$$e_p = \frac{G2,5}{\omega} = \frac{2,5 \cdot 60}{2 \cdot \pi \cdot 1500} = 0,015mm = 15\,\mu m \qquad (4)$$

Admissible unbalance U_p:

$$U_p = m \cdot e_p = 800 \cdot 15 = 12000\,gmm \qquad (5)$$

The similar can be determined for the group G1 and the same maximum speed. The given model calculations show, which requirements must be fulfilled by spindles, tables and pallets at machine tools. Spindles at vertical machining centres are balanced dynamically on the basis of similar calculations, even if these calculations are more complicated. Electronic balancing devices are used in practical application cases, especially at higher manufacturing series. The electronic balancing devices accelerate considerably this specialized operation. The actual machine-tool operation brings great uncertainty to considerations about the spindle mounting design. The machines are designed to be multi-purpose ones and the workpieces which shall be machined at the machines made by

Figure 2.
Table casting

Figure 3.
Non-rotary workpieces
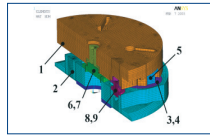
Figure 4.
Pallet exchange system example – FEM model. 1 – pallet, 2 – clamping base, 3 and 4 – cylinder and piston, 5 – clamping strip, 6 and 7 – centring pin and sleeve, 8 and 9 carrying pin and sleeve

served as supporting elements for the mesh creation. The contact surfaces between bodies are made using the following elements: TARGE170 and CONTA174 or CONTA173 [ANSYS Inc. 2002].

Considering the frame conditions, it is possible to divide the models to the variants having one symmetry plane or having two symmetry planes. Moreover, the solution was complicated by manufacturing tolerances of particular components and by conditionality of some contacts (contact pairs as observed quantities and frame conditions). It was necessary to consider the pallet exchange system like the complete system as well as like the subsystems at the same time. More than thirty calculation variants had been solved (combinations of geometric variants and load variants), but not all of them were included to the final evaluation. Tab. 3 includes a part of results obtained for pallet deformations for various calculation variants.

TOSHULIN, a. s. have generally various shapes. Fig. 3 shows the workpiece which shall be machined at the vertical turning centre. The workpiece shape creates the unbalance which causes bigger unbalancing than it is determined according to the standard for dynamic balancing as well as for static balancing. Although workpieces load the machine tool in the inconsiderable way due to their unbalance, it is not possible to undervalue the table balance itself.

4. Model and load description

In its next part this contributions deals with strength analysis and especially with deformation analysis of the pallet exchange system with the pallet diameter of 800 mm at high speed. Under the term "pallet exchange system", which is solved, we understand a technical system consisting of components which perform different functions, as e. g. position setting, manipulation, clamping, etc. Only two main parts (including their material properties) of the solved system are stated in the table. The particular system includes more than one hundred components which are not identical ones. For this manufacturing equipment type and for the given workpiece chuck diameter (pallet 1), it is possible to consider the speed higher than 600 min^{-1} to be the high speed. If there are bigger chuck diameters available, the speed is proportionally reduced so that the pallet cannot be broken due to centrifugal forces, etc. The finite element calculation was performed for the maximum operating speed. The spindle speed around the axis of rotation was entered directly by the angular speed vector. The load due to centrifugal forces seems to be the important force effect which predominates over other ones in many cases.

The data volume of particular model files is connected directly with their geometric complexity (mesh size, etc.). The significant geometric simplification was performed only at the clamping strip, most of all because this strip did not represent the main observed calculation object. Moreover, the simplification of other parts had to take into account the mass layout too. Shapes and dimensions were kept at other components. Curvature, chamfering and other details were omitted at all models, because curvature, chamfering, etc. would only complicate the creation of the mapped mesh, and their influence especially on resulting deformations can be considered to be the unimportant one. The connection of the particular components was made by contact elements. The more-detailed type execution, i. e. the connection which is more looking like the actual one (connection performed with screws and pins) was not carried out according to primary estimations (because the model complexity shall increase and this will influence the whole process).

The pallet exchange system models are created in approx. 96% using the mapped mesh. The models have been created using the elements SOLID45, SOLID95 and SOLID92. The MESH200 elements

5. Calculation results and their analysis

We selected the standard calculation method – combination of numerical simulations together with analytical calculations. The conclusions obtained from the numerical analysis, i. e. especially reaction forces were used for a great number of analytical calculations. The FEM model existence enabled to determine the forces, which must be transferred by connecting material (especially by screws and pins). However, the main target was to specify the influence of big load on the behaviour of this technical system. The next sections also state the comparison of forces obtained by means of analytical calculations with the results obtained using the ANSYS program at the selected components. The results from the selected variants are

Table 2. Maximum pallet displacements in the Z-axis direction			
Model variant	ΔUZ max [mm]	Model variant	ΔUZ max [mm]
V12	0,172	V11	0,060
V13	0,102	V14	0,044
V16	0,106	V15	0,041

Table 3. Force load of observed components				
Observed place	F_X [N]	F_Y [N]	F_Z [N]	F_{oFEM} [N]
cylinder/clamping base	114762,6	−114762,0	0.170.10^{-1}	162297,9
piston/clamping strip	90920,8	15723,8	0,55·10^{-1}	92270,3

mentioned in the following text part.

The important matter for designing is to compare centrifugal forces acting on the clamping cylinder and on the piston. These centrifugal forces are determined as the sum of forces in the contact surface, calculated analytically using Pythagorean theorem for the forces in the XY plane and analytically according to $F = m\omega^2 r$. The forces are determined with regard to the global coordinate system.
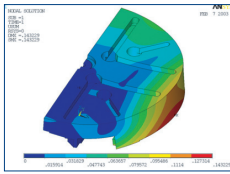
Cylinder – clamping base:
• Centrifugal force, determined analytically (spot body): 138258,3 N
• Force determined by means of FEM calculation (real body model): 162297,9 N
• Difference: 24039,6 N

The piston and the cylinder are considered to be spot bodies in the numerical calculation. In the second case (FEM), the body is considered to be the real one and therefore the increase gradient can be recorded for the centrifugal force. The resulting force can be influenced e. g. also by body deformations.
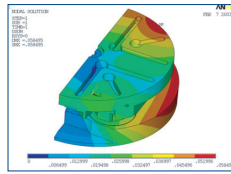
The centrifugal force acting on the clamping cylinder and on the piston was again determined as the sum of forces in the contact surface and the forces are stated with regard to the global coordinate system.

Piston – clamping strip:
• Centrifugal force, determined analytically (spot body): 90306,8 N
• Force determined by means of FEM calculation (real body model): 92270,3 N
• Difference: 1963,5 N

Table 1. Permissible stress [CSN 41 4220, CSN 41 5240]						
No.	Com-ponent	Material	R_e [MPa]	R_m [MPa]	$\sigma_{DII} = R_e/k_{pt}$ [MPa]	$\sigma_{DIII} = 0,6\sigma_I$ [MPa]
1	Pallet	14220.4 or 15240.5 15240.6	588 or 314 588	785 or 588 785	294 or 157 294	176,4 or 94,2 176,4
2	Base					

Figure 5. *Deformation of the non-relieved pallet, USUM (total structural displacement) [mm]*
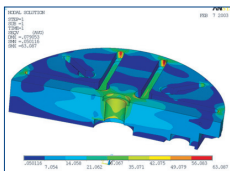

Figure 6. *Deformation of the clamping base and of the pallet, USUM (total structural displacement) [mm]*
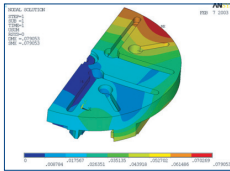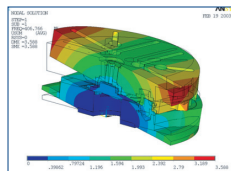

Figure 7. *Deformation of the relieved pallet, USUM (total structural displacement) [mm]*
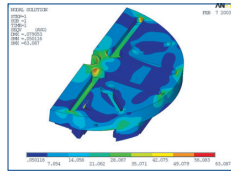
Figure 8. *Effective stress of the relieved pallet [MPa] (V11)*

Figure 9. *Effective stress of the relieved pallet [MPa] (V11)*

Figure 10. *Example of frequency shape of the clamping base and of the pallet (407Hz)*

In this case, the numerical calculation and the analytical calculation do not differ from each other very much.

The enclosed results (the figures show only a selection from the solved variants) show the pallet behaviour in particular FEM models at the considered load condition. V02 represents the alternative with one symmetry plane to the variant V01. The variants V01, V03, V04 and V09 served to compare the influence of the clamping base relieving on its deformations. The variants V05, V06 and V07 compare the influences of particular components on deformations and on stress at the pallet exchange system. These represent 1/4 of the pallet exchange system, so they omit the influence of unsymmetry. Under the considered geometric and load conditions the model V11 shows that the pallet exchange system deformations are transferred on the positioning pin. The variants V11 to V16 (Fig. 6 to Fig. 9) compare the influence of pallet relieving and of T-slots on the pallet deformation, if one symmetry plane is respected. The models do not presume any contact between the piston and the clamping strip. The model V10 shows e. g. the minimum influence of the T-slot on the stress increase in the body. The models V17 and V18 are modal analysis models. The results proved that prestressing inside the body (in the continuum) caused by the pallet exchange system rotation is not big enough to have an important influence on the frequency and shapes of oscillations. Therefore, the oscillation shapes at the variant V18 are similar to the oscillation shapes at the variant V17 (it is true that at a different frequency which increases together with prestressing intensity).

6. Conclusions

This contribution shows the comparison of forces obtained using the analytical calculations and the results obtained by means of the ANSYS program. The finite element method plays the important role at the design process of vertical turning centres. As it can be seen from the previous sections, the centrifugal force represents the dominant force at high speed used at machines equipped with the pallet exchange system. It is necessary to underline the fact, that using the

FEM analysis it was possible to eliminate successfully some possible undesirable effects after the manufacture and assembly of the pallet exchange system. Moreover, it was also possible to optimize technologically the pallet shape as well as the clamping base shape.

Critical points result from the analysis of stress and forces in the particular variants at the given load condition. The maximum stress and deformations are given in the table and the courses of effective stress and feeds for the interesting variants (interesting with regard to the results) are shown in figures. The deformation in the Z-axis direction was selected to be used as the criterion, whether the analysed system is useful to be used in technical practice (or to compare the manufactured body with given geometry) or not. The maximum stress originates in contact surfaces, no other important stress peaks were determined inside the pallet or inside the clamping base. It can be seen from the calculation that each reduction of the mass, which shall rotate, results in the considerable reduction of deformation and stress values in all directions, especially in the Z-axis.

Some increased values of stress or forces in the contact surfaces are caused by the fact that the surfaces transfer centrifugal load from the particular components in the slipping way, i. e. by means of the model. If there is a real connection made using the screws, it is possible to expect that stress will be considerably bigger near the holes than in the homogenous material (notch effect, load from screw prestressing, etc.). In spite of this, thanks to the material used to manufacture pallets (steel 14220.4 or 15240.5 or cast steel), the stress values in notch places were in the sufficient distance from inadmissible values. The design and the strength check of the screw were made simultaneously with the calculations mentioned here.

It is possible to see these main critical points on the calculation results:
- sharp corners and edges in the T-slots on the inside pallet diameter;
- end of the T-slots on the outside pallet diameter.

In the cases, where it is possible to perform the analytical calculation, FEM corresponds very well to these results. It can be seen from the results that centrifugal forces seem to be the essential ones for the load of the analysed system. Not only for this reason it was recommended for designing e. g. to evaluate very thoroughly, if it is useful and safe to use pins for setting and levelling the hydraulic cylinder in the clamping base body, when this is compared with the cylinder location in the pressed hole. The authors were not surprised by the conformity extent of the analytical calculation in comparison to the calculation performed using FEM. This only confirms the designer rule that if a design shall be made, in many cases it is possible to perform the analytical calculation in the first phase and then to specify it in more details using FEM, or it is possible to aim at searching the problematic places which cannot be found by analytical calculations.

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