INNOVATION IN WOODWORKING INDUSTRY AND ENGINEERING DESIGN, 2/2018 (14): 78–85

STATIC ANALYSIS OF A UPHOLSTERED FURNITURE FRAME MADE OF SCOTS PINE AND PB WITH STAPLE CORNER JOINTS BY FEM

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ABSTRACT

3D geometric model of one-seat frame for upholstered furniture was created by the CAD system Autodesk Inventor Pro®. A linear static analysis was carried out with CAE system Autodesk Simulation Mechanical® by the method of finite elements (FEM) simulating light-service loading of the frame.

The orthotropic material characteristics of pine solid wood (Pinus Sylvestris L.) for the rails and particleboard (PB) for the side plates are considered in the analysis. Corner joints in the frame with staples and PVA glue were considered. FEA was performed with regard to laboratory established coefficients of rotational stiffness of used staple corner joints in the frame – case butt joints and end to face butt joints.

As a result, the most loading construction part of the frame with staple corner joints is established – the rear upper rail of the seat where the maximum values for linear displacements, nodal rotations and stresses are received due to the nature of the applied loading.

Key words: upholstered furniture frame, staple joints, PB, static analysis, CAE, FEM.

INTRODUCTION

Upholstered furniture frames made of solid wood and wood based panel materials are the most widely used in manufacturing practice because of the possibilities of rationalizing the structure using the most used materials in the furniture industry. The ability to use waste strips from panel cutting significantly reduces the material cost and the corresponding cost of the product. In Bulgaria and worldwide the particle boards (PB) are widely employed panel type wood based composites for production of upholstered furniture frames.

Smardzewski (2001) has made an attempt to find an optimal solution for the construction of a supporting structure in a single-seat arm-chair made of wood and chipboard joined with staples taking into account that the problems of quality verification of supporting elements in armchairs and sofas have not been undertaken before that. Material optimization of construction has been performed only for the main component, but the materials have been considered as isotropic. The strength optimization has been carried out by a numeric method using Algor® computer program and has resulted in lower (68%) solid wood consumption, guaranteed optimal strength parameters.

Lately, Smardzewski and Prekrat (2009) have carried out laboratory and numerical investigations of two-person sofa frame with side elements from PB and beam elements from pine and beech wood and have taking account of orthotropic nature of used materials in the FEA with Algor® CAE. They have proposed new dimensions of the construction elements and in the result consumption reduction of beech wood by 36% and that of PB by 25% without significant change of the construction rigidity and strength has been established.

More information concerning the deformation and strength characteristics of case furniture made of PB is available:

Marinova and Kyuchukov (1997) and Kjutschukov and Marinova (1997) have carried out deformation and strength analyses of a case furniture with no inner fixed partition elements made of orthotropic veneered PB, subjected to the working load with the created FEM methodology by Marinova (1996) for SAP software. Three types of corner joint have been considered – rigid, hinged and semi-rigid with taking account of test established rotational stiffness of joints with dowels, joint fasteners and screw connectors. The authors have reported that the most deformable was the case with hinged sides and the maximum stresses have been localized in the front part of the case. They have recommended the point of the applied exploitation horizontal force for its displacement to be located closer to the bottom and to the back of the case. Further, Kjutschukov and Marinova (1998) have investigated the deformation of the same case furniture model by FEM applying heaviest external horizontal force in two variants: at a point 1600 mm from the floor and a point of the middle of the side and have established that the second variant is a better one for displacement of a loaded piece.

Nicholls and Crisan (2002) have analyzed the stress and strain state in corner joints made of chipboards with beech dowels, typically found in furniture box-type structures with ANSYS™ by using FEA. Although isotropic characteristics of chipboards have been considered they have found that clear areas of stress concentrations were observed in the region where the fixing components are located.

Norvydas et al. (2012) have laboratory tested the strength of the multidowel glued miter corner joints of case furniture from PB and have established that the bonding strength of the mitre joint of the wood particle board has exceed the material strength by even 10% and it is possible to achieve maximal load bearing capacity of construction.

Imirzi and Efe (2013) have analyzed the bending strength and stiffness properties of L-type corner joints with dowels, dowels-screws and PVAc glue in cabinet type furniture made of PB with 14, 16 and 18 mm thickness laboratory and by FEM (ANSYS®). The comparisons have revealed that computer model have showed more rigid characteristics than the experimental element and have reached the fracture as the maximum difference of 12,61% have been established for dowel-screw joints and 14 mm PB.

Yuksel et al. (2014) have investigated the effects of panel thikness on moment resistance on L-type corner joints and stiffness of four-member cabinets made of PB. They have established that 16 mm PB cabinets yielded higher stiffness values than those of 18 mm.

The literature study revealed a limited number of publications on frame studies of upholstered furniture made of PB with staple joints.

The aim of this study was to define and analyze the displacements and stresses of one-seat frame of upholstered furniture with staple joints and side plates of particle boards (PB) under light-service loading by CAD/CAE using the method of finite elements (FEM).

MATERIALS AND METHODS

3D model of one-seat upholstered furniture frame with length 600 mm, width 725 mm and height 650 mm was created with Autodesk Inventor Pro® – Fig. 1. The used rails are with cross section 25x50 mm.

3D discrete model of the frame was created with plate finite elements – Fig. 1. The generated Midplane mesh has 5130 orthotropic finite elements and 33616 DOF's. A linear static analysis of 3D discrete model of the upholstered furniture frame was carried out with CAE system Autodesk Simulation Mechanical® by the Finite Elements Method (FEM).

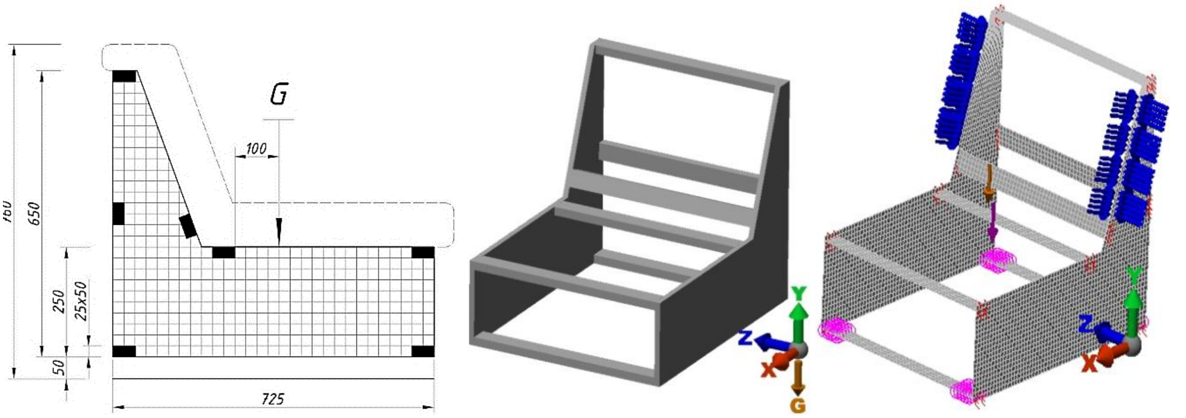

Figure 1: 3D frame discrete model and loading

Orthotropic materials type was used for construction elements of the frame:

Scots pine (*Pinus sylvestris L.*) for rails and strengthening details with measured density 430,65 kg/m³ according to BDS EN 323:2001 and elastic characteristics: $E_x=E_L=9000.10^6$ N/m^2, $E_y = E_T = 593.10^6$ N/m^2, $G_{LT}=554{,}5.10^6$N/m^2, Poisson ratios $\nu_{LR}=0{,}03$, $\nu_{LT}=0{,}027$, $\nu_{TL}=0{,}41$, $\nu_{RL}=0{,}049$.

Particleboard (PB) for side plates with thickness 16 mm and measured density 678,06 kg/m³ according to BDS EN 323:2001. The physical and mechanical characteristics of the used PB panels are: modulus of elasticity in bending $E_x=2700.10^6$ N/m^2 and $E_y=1600.10^6$ N/m^2; bending strength 11.10^6 N/m^2; Poisson ratios $\nu_{xy}=0{,}030$ according to Bodig et al. (1982) and $\nu_{yx}=0{,}18$, calculated from equation 1:

$$\frac{\nu_{xy}}{E_x}=\frac{\nu_{yx}}{E_y}, \qquad (1)$$

Support boundary conditions were set: bottom front rail – no translation on *y* direction and bottom rear rail no translation on *x*-, *y*- and *z* direction – Fig. 1.

In order to simulate semi-rigid connections between rails and side plates of the frame two actions were performed: *First* – in the place of joints in the discrete model narrow zones were created with established via tests by FEM lower modulus of elasticity of the used materials perpendicular to the common edge of the corner joint. *Second* - the laboratory determined by Hristodorova (2018) coefficients of rotational stiffness of the corner joints with 2 staples and PVA glue, loading under compression were introduced in the nodes of the respective corner joints - case butt joints (c=1017,52 N.m/rad) and end to face butt joints (c=822,77 N.m/rad).

The discrete frame model was loaded with a total load of 800 N, distributed as follows (Fig. 1):

Seat: 80% was set as a remote force, distributed between upper rails of the seat with application point of 100 mm in front of the upper rear rail, simulating upholstery base;

Backrest: 16% set as equal nodal forces, distributed on the edges of the two sides of the backrest, simulating elastic belts.

The changed from 90^0 angle γ between the joint shoulders at upper rails of the seat was measured by vectors with the program Autodesk Simulation Mechanical®.

The more detailed the methodology has been described in our previous publication – Staneva et al. (2018).

RESULTS AND DISCUSSION

The results of static analysis for linear displacements *u*, nodal rotations θ, von Mises stresses $\sigma_{von\ Mises}$, maximum principal stresses σ_1, minimum principal stresses σ_3 and equivalent strains $\varepsilon_{von\ Mises}$, as well as the changed angle *γ* between the joint shoulders at the rails of the seat are shown in Table 1, Table 2 and in Fig.2 to Fig.7 for the frame and for the side plates of the frame respectively. The visualizations of the deformed model are shown with a scale factor 3% of model size for the frame and with a scale factor 5% for the side plates.

In Fig. 2, the distribution of resultant displacements is presented. The maximal resultant displacement of 2,68 mm (rear rail) and 1,34 mm (front rail) is received in the middle of the rails of the seat, on the inside of the rails and they are determined mainly by the *y*-displacements (u_y) – Table 1. The maximal resultant displacement in the rear rail of the seat is 2 times greater than the same in the front rail of the seat. This is due to the nature of the applied force on the seat with application point of 100 mm to the rear rail of the seat, which coincides with the application point of the weight of the model.

Table 1: Maximal displacements and strain in the model.

Parameter	Location	Value
$u_x.10^{-3}$, [m]	side plates	0.10
$u_y.10^{-3}$, [m]	front upper rail	-1,34
	rear upper rail	-2.68
$u_z.10^{-3}$,[m]	base of side plates	0.41
θ_x, [°]	rear upper rail	0.79
θ_y, [°]	side plates	0.20
θ_z, [°]	front upper rail	1,70
	rear upper rail	-1,80
$\varepsilon_{von\ Mises}$, [m/m]	side plates	0.0095
γ, [°]	front upper rail	89,79
	rear upper rail	89,66

Table 2: Maximal displacements and stresses in the side plates.

Parameter	Location	Value
$u_x.10^{-3}$, [m]	backrest	0.104
$u_y.10^{-3}$, [m]	front upper rail	-0,13
	rear upper rail	-0.16
$u_z.10^{-3}$,[m]	base of side plates	0.41
θ_{res},[°]	front upper rail	0,37
	rear upper rail	0,65
$\sigma_1.10^6$, [N/m^2]	front upper rail	3,05
	rear upper rail	2,21
$\sigma_3.10^6$, [N/m^2]	front upper rail	-4,12
	rear upper rail	-6,23

In the side plates of the frame the maximum values of the resultant displacement (0,42 mm) are received in the places of the base of the seat where dissolution of the side plates is observed (Fig. 2) - the resultant displacement is determined mainly by *z*-displacement (u_z) – Table 1. In the field of rear rail of the seat in the side plates the resultant displacement (0,18 mm) is almost 1,4 times greater than the same in the field of the front rail of the seat. In x-direction the maximum x-displacement (0,104 mm) is received in the field of the upper part of the backrest in the side plates, i.e. the backrest tilts forward – Fig. 2.

The maximal resultant nodal rotations θ_{res}=1.80^o for rear rail of the seat and θ_{res}=1.70^o for front rail of the seat are located in the middle of the upper rails with insignificant difference between them and are determined by the rotation around z-axis – Table 1 and Fig. 3.

In the side plates maximum resultant nodal rotation (0,65^0) is received in the field of joining of rear rail of the seat and is almost 1,75 times greater than the same in the field of front rail of the seat – Fig. 3.

Expectedly, the change in the angle γ between the joint shoulders at the rear rail of the seat is also greater than the same in the front rail of the seat – Table 1 and Fig 4.

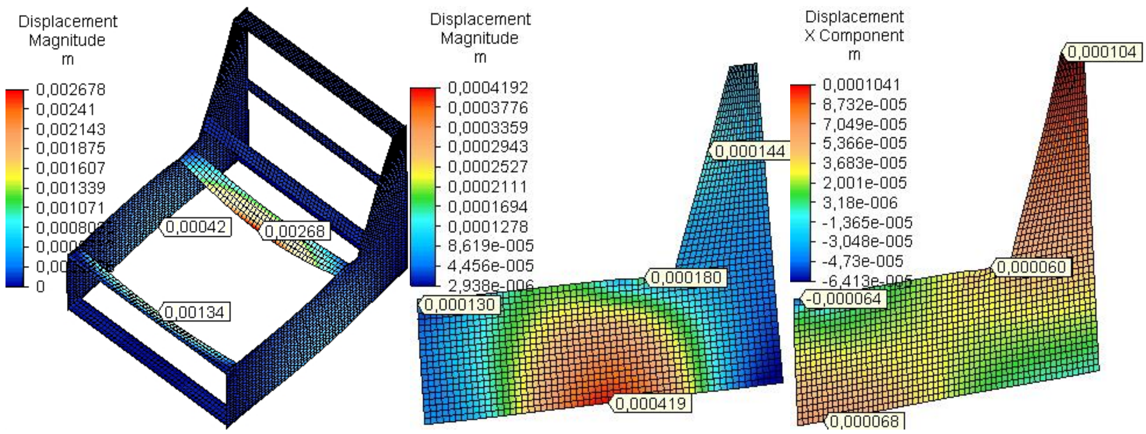

Figure 2: Distribution of resultant displacements in the model and in the side plates

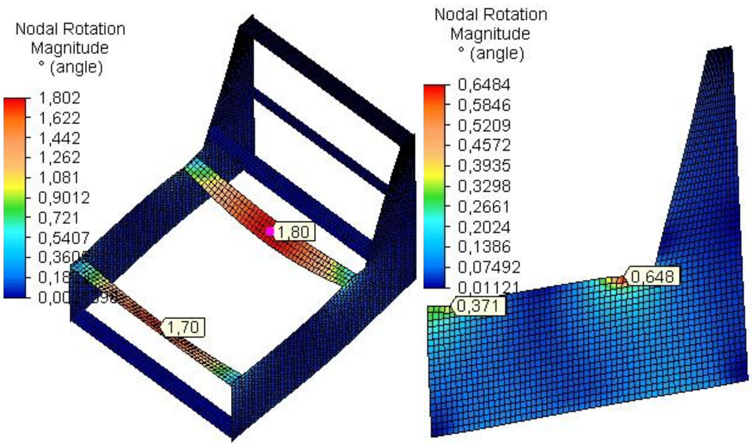

Figure 3: Distribution of resultant rotational displacements in the model and in the side plates

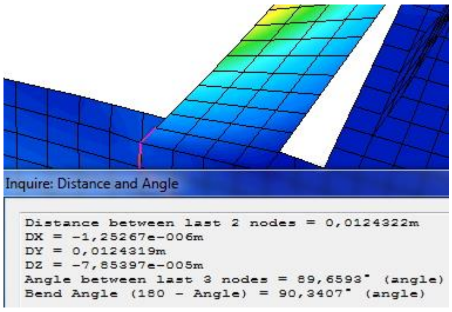

Figure 4: Angle γ between the joint shoulders at the rear rail of the seat

The distribution of von Mises stresses, maximum principal and minimum principal stresses in the model and in the side plates is presented in Fig. 5, Fig. 6 and Fig. 7.

The maximal value of von Mises stress $\sigma_{vonMises}$=7,99.10^6 N/m^2 is concentrated in the middle of the rear rail of the seat and is 1,95 times greater than the value of von Mises stress in the front upper rail ($\sigma_{vonMises}$=4,09.10^6 N/m^2), which is localized near by the corner joints with the side plates due to the loading type and location of the biggest bending moments **(**Fig. 5).

In the side plates von Misses stress has maximum value in the field of rear rail contact area of the seat, but it is 1,45 times smaller than the maximum value in the rear rail of the seat – Fig. 5.

The maximum principal stress (tension stress) has maximum value (7,79.10^6 N/m^2) located in the middle of the rails of the seat - at the bottom side. The value in the rear rail (3,65.10^6 N/m^2) is 2,13 times greater than this in the front rail – Fig. 6.

In the side plates the maximum tension stress has localized in the area of the joining of the front rail – Fig.6.

The maximum of minimum principal stress (compression stress) in absolute value of 8,01.10^6 N/m^2 is located on the top of the rails of the seat, in the middle of the rails of the seat, as in the rear rail it is 2,11 times greater than this in the front rail –- Fig. 7.

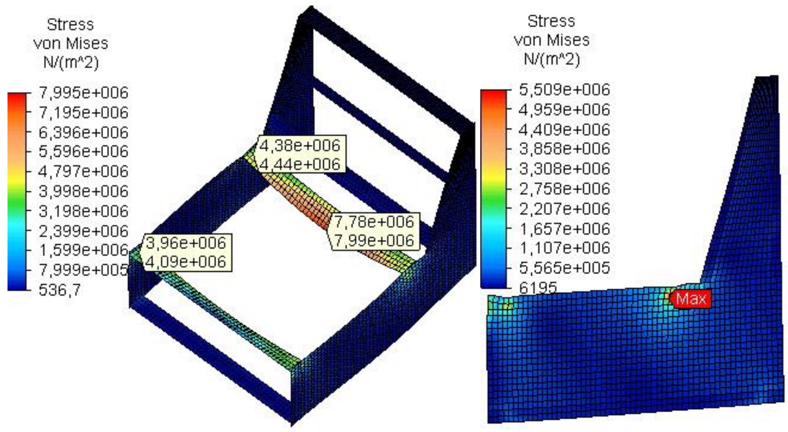

Figure 5: Distribution of von Mises stresses in the model and in the side plates

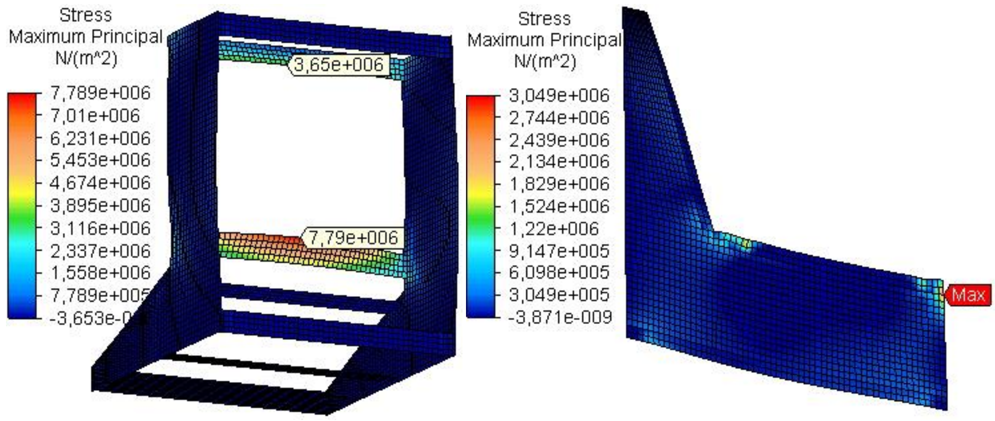

Figure 6: Distribution of maximum principal stresses in the model and in the side plates

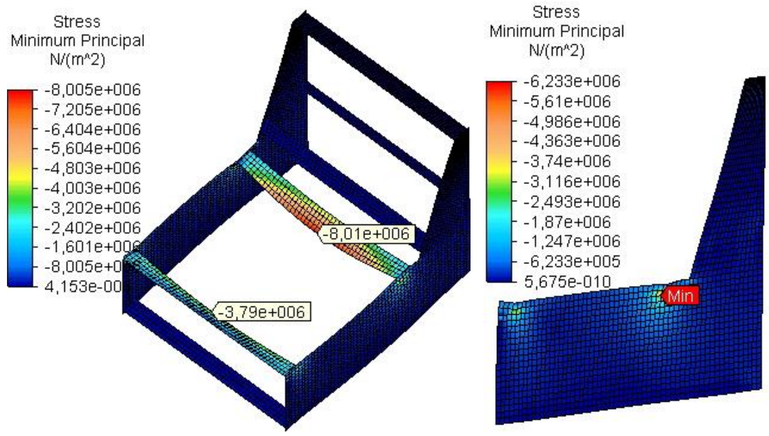

Figure 7: Distribution of minimum principal stresses in the model and in the side plates

In the side plates the compression stress has maximum in the coupling areas of the rear rail of the seat (in absolute value of $6{,}23.10^6$ N/m^2) and is almost 1,29 times smaller than this of rear rail of the seat - Fig. 7.

The maximum value of equivalent strain is localized in the side plates - in the field of joining of the front rail – Table 1.

CONCLUSIONS

From the results of this investigations of the deformations and stresses of one-seat upholstered furniture frame made of Scots pine rails and PB (thickness 16 mm) side plates with corner joints – staples and PVA glue by FEM with CAE program Autodesk Simulation Mechanical®, as well as the analysis made, several conclusions can be derived:

The type of corner joints has a determining effect on the deformations and stresses of the construction elements of the upholstered furniture frame and the joints between them under service load. Under light-service load, the critical joints in the upholstered furniture frame have been established – the corner joints between rear rail of the seat and the side plates of PB.

The most loading construction part of the upholstered furniture frame is the rear rail of the seat where the maximum values for linear displacement, nodal rotation and stresses are received. In the side plates of PB, the maximum values for linear displacement, nodal rotation and stresses are received in the field of joining of the rear rail of the seat.

It is recommended that the rails of the seat should be further strengthened in the joints with the side plates with additional strengthening construction elements in order to improve the deformation and strength characteristics of the upholstered furniture frames with side plates made of PB.

Acknowledgments

The article is a result of research made on base of the project 69/27.08.2015, sponsored by Scientific Research Sector in University of Forestry – Sofia, Bulgaria.

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UNIVERSITY OF FORESTRY

FACULTY OF FOREST INDUSTRY

INNOVATION IN WOODWORKING INDUSTRY AND ENGINEERING DESIGN

2/2018

INNO vol. VII Sofia

ISSN 1314-6149
e-ISSN 2367-6663

Indexed with and included in CABI

INNOVATION IN WOODWORKING INDUSTRY AND ENGINEERING DESIGN
Science Journal
Vol. 07/p. 1–88
Sofia 2/2018

ISSN 1314-6149
e-ISSN 2367-6663

Edition of
FACULTY OF FOREST INDUSTRY – UNIVERSITY OF FORESTRY – SOFIA

The Scientific Journal is indexed with and included in CABI.

Printed by: INTEL ENTRANCE
Publisher address: UNIVERSITY OF FORESTRY – FACULTY OF FOREST INDUSTRY
Kliment Ohridski Bul., 10, Sofia, 1797, BULGARIA
http://inno.ltu.bg
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