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# LS-DYNA CONTACT PROCEDURE ANALYSIS FOR SELECTED MECHANICAL SYSTEMS

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***Abstract***

*Finite Element Method is one of the most frequently used computational schemes in numerical analyses. A contact phenomenon is an essential issue when modelling the physical interaction between two or more bodies. Depending on the used software, the various contact algorithms are applied. In the paper, the authors present the results of evaluation of different contact methods implemented in LS-Dyna CAE software. The most common problem arising during contact modelling is the occurrence of penetration between the coupling elements, which leads to an increase of contact forces and, consequently, a local deformation of finite elements. In the introduction, a theoretical background related to mathematical aspects of contact modelling is presented, in particular search methods for contact surfaces and the penalty forces calculating method. Subsequently, a contact analysis for different variants of interaction of the elements is presented. In the first analysed case, shaft- sleeve interaction, in both rotation and translation, is presented. The variable factors in this case were a number of interacting finite elements and the type of a contact algorithm. The second case focused on the interaction forces resulting from the interaction between two sliding bodies in the harmonic motion. A sliding body was pressed against the plane using an increasing force. A variable factor in this case was the type of the implemented contact algorithm.*

***Keywords:*** *contact, Finite Element Method, LS-Dyna, shaft, sleeve*

## 1. Introduction

From the mechanical point of view, contact is interaction between the contacting bodies. In the Finite Element Method (FEM), a contact phenomenon is treated as a change in boundary conditions occurring during the analysis. Therefore, it is often referred to the boundary nonlinearities. Similarly as the other types of nonlinearities (geometric or physical), the application of a contact algorithm forces the use of incremental-iterative methods. When two separate surfaces meet, they become a contact pair. In such a pair, the forces are transferred according to physical laws, taking into consideration the mechanical properties of the interacting bodies. From the numerical point of view, the interacting bodies are represented by bonds and limiting surfaces and lines of the segments.

The very first step to solve contact problem using FEM is to determine regions than could get in touch during the analysis, In LS-Dyna software, this first step is left to the user, who has to indicate the contact regions. The penetrated (Master) and penetrating (Slave) bodies are designated. Then, a contact tolerance is determined. It is a range within which the program seeks for the exact contact pairs. In LS-Dyna solver, a bucket-sort method is used. This method involves creation of a multitude of sub-regions in the master and slave bodies (so called “buckets”) in which the searching procedure is performed [1]. The maximum distance of nodes detection is determined based on the size of the largest master segment. Determination of contact pairs ends the first stage of contact problem solving, sometimes called kinematics of contact pair. Forces and stresses generated in the contact pair are determined by procedures launched after this step [2].

In the paper, the sensitivity study on a contact kinematic change on the obtained results is presented. Previous numerical investigations done by authors showed surprisingly high results sensitivity on the chosen contact kinematics algorithm [3]. To further investigate the problem, new set of numerical examples allowing broaden previously obtained results was defined.

Different types of a contact algorithm were investigated for that purpose. The first one is a node to segment contact algorithm. The penetrated body in this scheme is described using surfaces designated and limited by the edge nodes. Those surfaces also divide the body surface into a multitude of subareas. The second contact kinematic algorithm is a segment-to-segment scheme. A visual representation for described schemes is presented in Fig. 1.

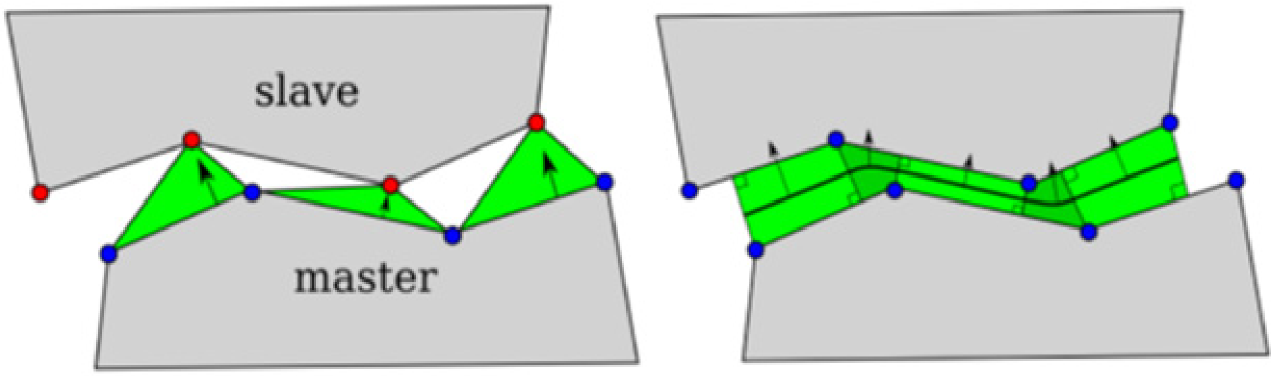



*Fig. 1. Graphic representation for nodes to segment (on the left) and segment to segment contact (on the right) [2]*

Designation of two different surfaces is necessary in order to determine which nodes are considered in the contact procedure. During the analysis, the distance between them is checked and if it is smaller than the tolerance value, the process of contact point and contact forces designation occurs. Contact point designation starts from determination which nodes are closest to the node that exceeds the tolerance. The assumed interacting bodies are described as follows: penetrating body (slave) $B^{(1)}$, penetrated body (master) $B^{(2)}$. The interacting surfaces, are indicated as $\Gamma^{(i)}$ which belongs to interacting body $B^{(i)}$, in such a way that all points of the body, which may contact at any time t, are taken into account [4] (Fig. 2).

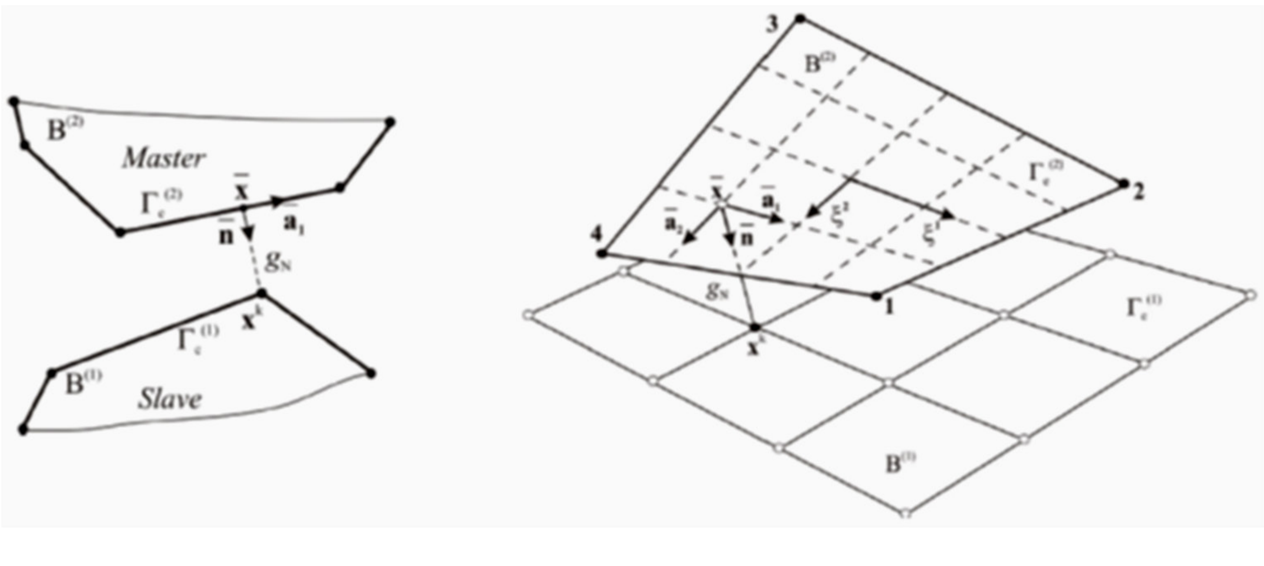



*Fig. 2. Determination of the contact point S in 2D solution (on the left) and 3D solution (on the right) [4]*

A projecting point of an actual position of a slave node on the master surface $\Gamma^{(2)}$ can be defined as:

$$\frac{x^k - \bar{x}(\bar{\zeta^1}, \bar{\xi^2})}{\|x^k - \bar{x}(\bar{\zeta^1}, \bar{\zeta^2})\|} \cdot \overline{a_\alpha}(\bar{\zeta^1}, \bar{\xi^2}) = 0, \tag{1}$$

where:

$x_k$ – slave surface node,

$\bar{x}$ – projecting point of an actual position of a slave node on the master surface,

$\overline{a_\alpha}(\bar{\zeta^1}, \bar{\xi^2})$ – base tangential vector in point $\bar{x}$.

Designation of the projection point allows for determination of a distance between the interacting node and the interacting surface. A gap or penetration $g_N$ is determined, for node $x^k$, on the normal direction as a distance between this node and master surface $\Gamma^{(2)}$:

$$g_n = (x^k - x) \cdot \bar{n}, \tag{2}$$

where:

$\bar{n}$ – normal vector to master surface $\Gamma^{(2)}$in point$\bar{x}$.

A normal unit vector is determined by vectors tangential in point $\bar{x}$:

$$\bar{n} = \frac{\bar{a_1} \times \overline{a_2}}{\|\bar{a_1} \times \overline{a_2}\|}, \tag{3}$$

The following situations [variants] are possible:

$g_N$= 0 – contact, $g_N > 0$no contact, $g_N < 0$ - penetration

The equations presented above, describe the contact kinematics in a case of when no friction between the interacting elements occurs. Therefore, if friction is considered, a relative tangential displacement is introduced.

In following examples, to determine forces acting between the interacting elements, a penalty function method is applied. In this scheme, normal force depends on penetration in a given time step. Virtual one-way spring elements are created; therefore, when one of the bodies penetrates through the other it is pushed by the force proportional to the penetration depth:

$$F_N = g_N \cdot k, \tag{4}$$

where:

$F_N$ – normal force vector,

$g_N$ – penetration depth,

k – scaling factor (contact stiffness).

The value of the force $F_N$ is estimated by including work done by it in the global energy balance (5) [5].

$$\int_\Omega \delta\varepsilon^T \sigma d\Omega - \int_\Omega \delta u^T b d\Omega - \int_\Omega \delta u^T f d\Gamma + \int_\Omega \rho\delta u^T u d\Omega - C_c = 0, \tag{5}$$

where:

*Ω* – volume of the body,

*Γ* – surface of the body,

f – externall forces,

b – mass forces,

u – displacement,

ε – strains,

σ – stresses,
$C_c$ – energy generated by contact forces.

Application of FEM to this balance and assumption that there is no damping in the system leads to the following matrix equation:

$$M\ddot{U} + (K + K_{TN})U = F - F_N, \tag{6}$$

where:
M – global mass matrix,
K – stiffness matrix,
$K_{TN}$ – work done by contact force,
U – displacements vector,
F – external forces vector,
$F_N$ – contact forces vector.

Additionally, we want to make sure, that found force $F_N$ value is optimal, i.e. gives penetration value $g_N = 0$. Therefore, force $F_N$ has to fulfil so-called Kuhn-Tucker conditions [6]:

$$g_N \geq 0; \;\; F_N \leq 0; \;\; g_N F_N = 0, \tag{7}$$

## 2. Contact analysis for shaft- sleeve rotational interaction

Numerical methods based on a discretization procedure which transforms a continuous medium (with an infinite number of degrees of freedom) into a discrete problem described by a system of equations with a finite number of variables.

Because of a decreasing number of finite elements used for shaft and sleeve modelling, geometry simplification occurs. A cross-sectional view of a discrete cylinder is a polygon, therefore the greater the number of elements used in modelling is, the closest to a circle the representation is. (Fig. 3).

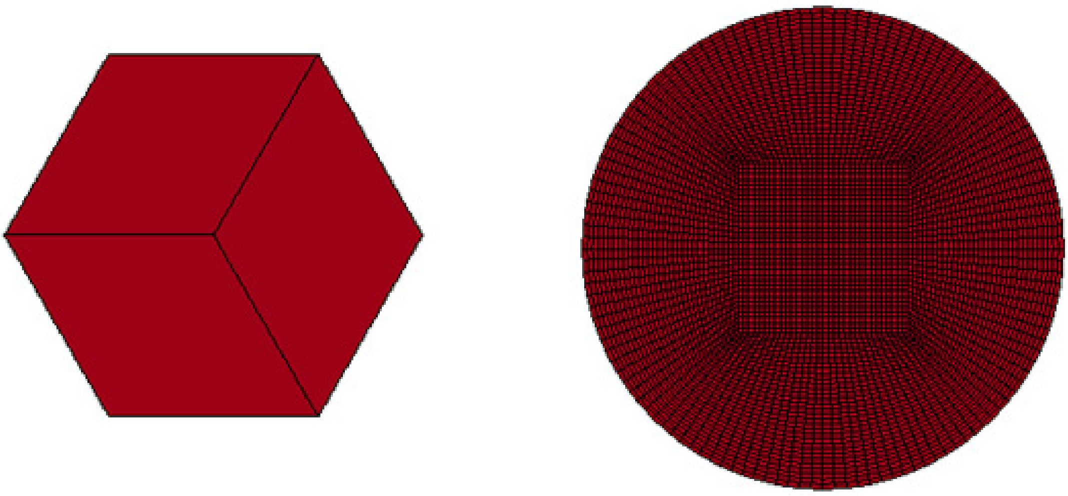


*Fig. 3. Shaft numerical model with 6 elements on edge (on the left) and 128 elements on edge (on the right)*

A numerical model for a sensitivity study in terms of a number of finite elements was developed as a parametric model (shaft diameter, sleeve diameter, a number of shaft and sleeve finite elements and initial gap). A schematic view of the model is presented in Fig. 4. In order to simulate shaft rotation in the sleeve, all translational and rotational degrees of freedom for the sleeve finite elements were constrained. Shaft rotation was obtained by prescribed motion of the beam elements normal to shaft axis resulting in the shaft torque. The model was discretized using solid elements with a constitutive model equivalent to steel (elastic formulation according to

Hooke law). A contact mechanism was defined with solver default parameters. Two contacting bodies were defined: a shaft as a slave interface and a sleeve as a master interface. The coefficient of static friction was set to 0.2 [-]. The gap between the shaft and the sleeve was set to 0.01 [mm].

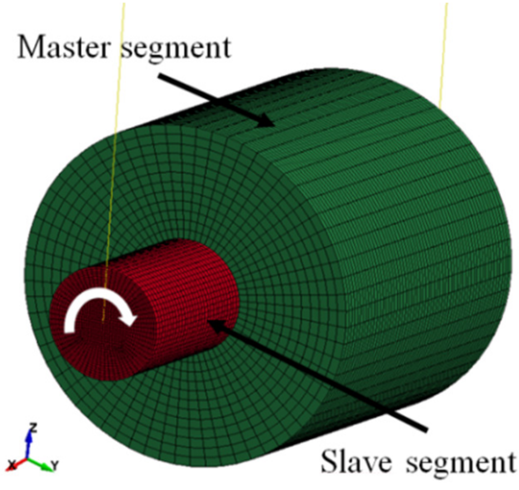
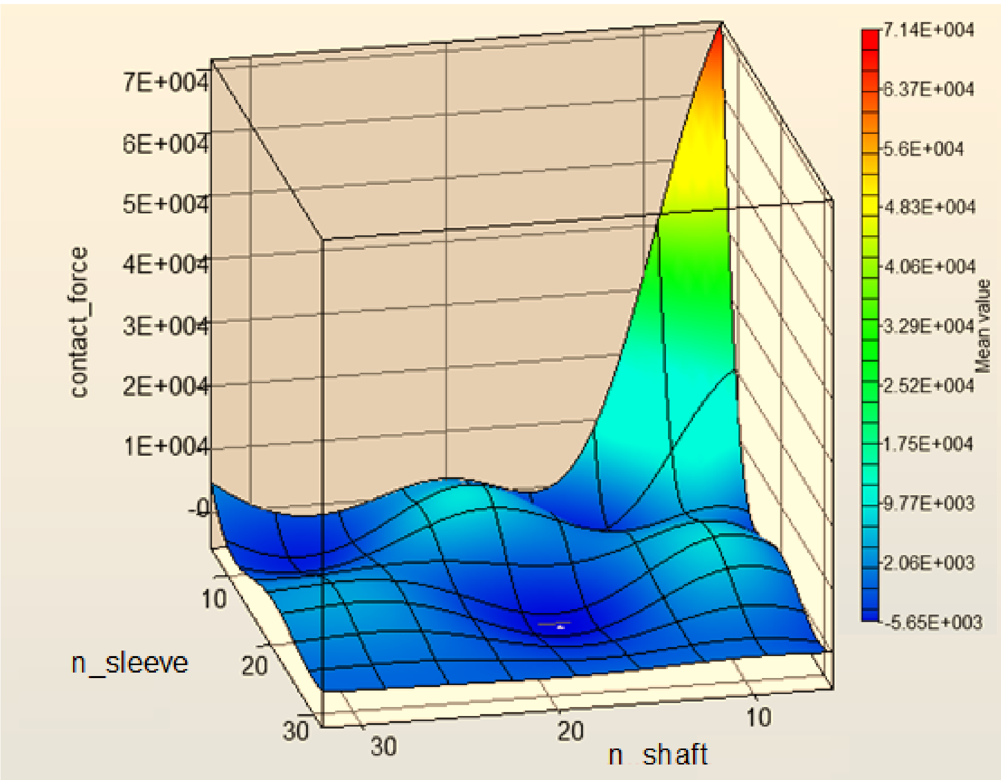



*Fig. 4. Model schematic view for rotational shaft - sleeve connection*

The analysis was performed using LS-Opt software with the following variables:

- number of finite elements on the shaft circumference: 6, 8, 12, 16, 24, 32;
- number of finite elements on the sleeve circumference: 6,8,12,16,24,32;
- two types of contact procedures: node to surface and surface-to-surface.

Because of the performed analyses, meta-models of the beam resultant forces for a various number of elements of the shaft and the sleeve were obtained (Fig. 5). Based on those results, a significant influence of the elements quantity on the resultant force at the end of the beam element can be noted. A contact force calculation algorithm is the same for the considered contact types; the significant divergence arises from the different contact kinematics mechanisms.



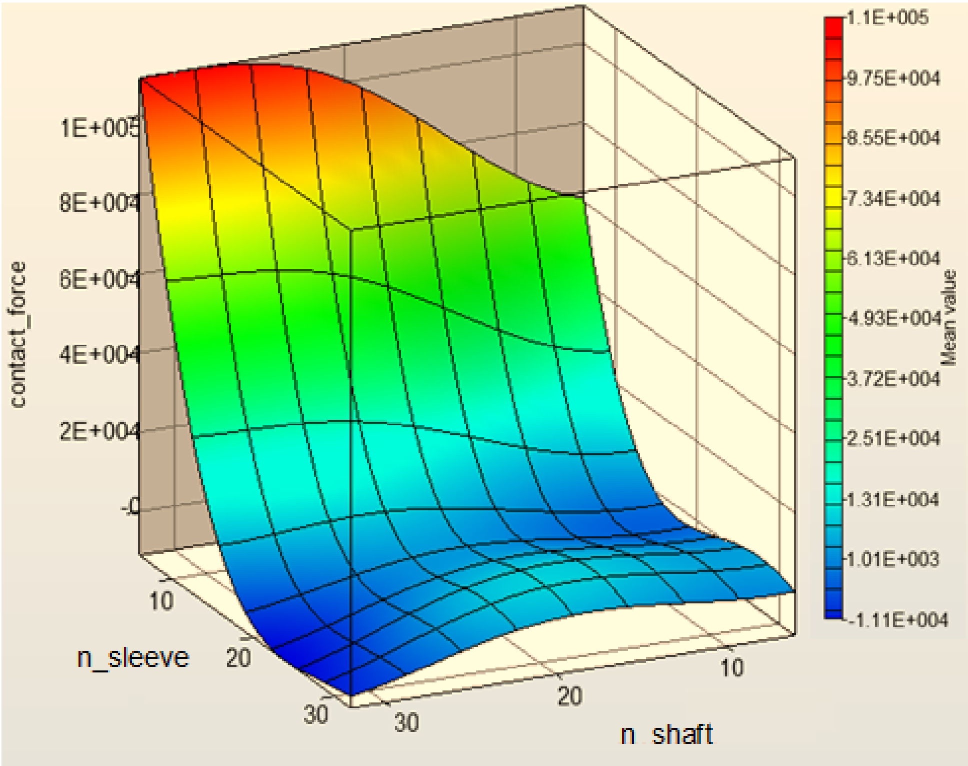



*Fig. 5. Meta-surfaces for surface-to-surface contact type (on the left) and node to surface contact type (on the right)*

Due to a large number of possible model configurations (108 cases) and hence time-consuming calculations, it was decided to reduce a number of the examined cases to 11 (Fig. 6). For those cases, other contact methods were also investigated (Fig. 7).

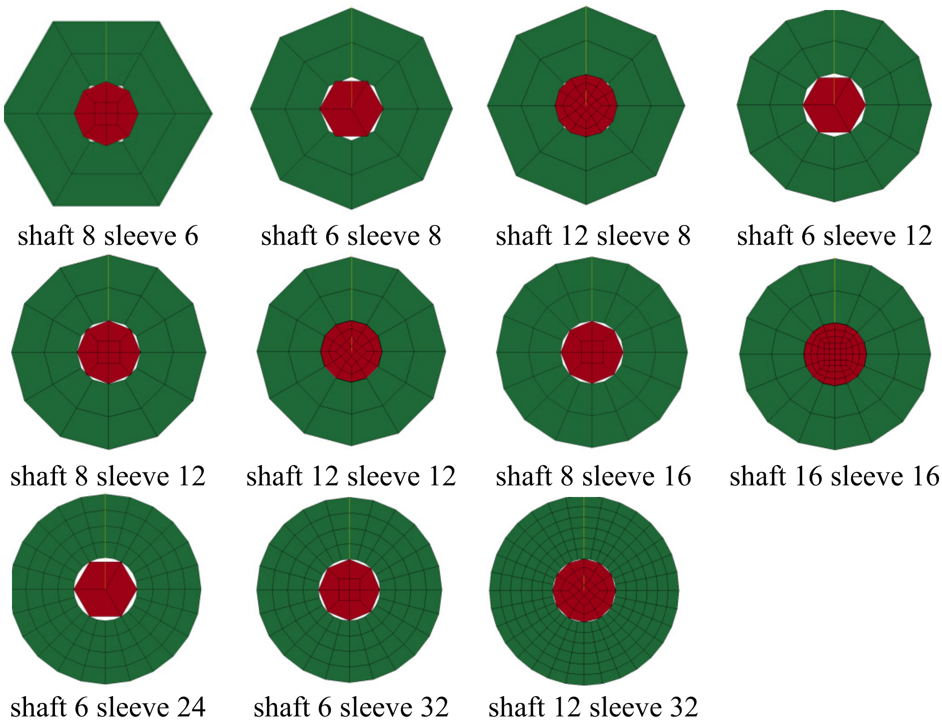



*Fig. 6. Case configurations investigated during testing of other contact methods*

Based on the results presented in Fig 7, it can be concluded that for a number of elements less than 15, the best performance is offered by the following contact types: "nodes to surface" and "surface to surface". Contact type "surface to surface" with additional "soft 2" option, at a small number of elements, generates very large resultant forces, which may lead to incorrect modelling of interaction between the shaft and the sleeve. A very interesting solution appears to be "mortar" contact type which generates resultant force less than 700 [N] for only 13 elements (for other types of contact this value exceeds 2500 [N]). For a number of elements greater than 24, all the considered contact types generate similar resultant forces; however, "mortar" contact type is the best in this case as well.

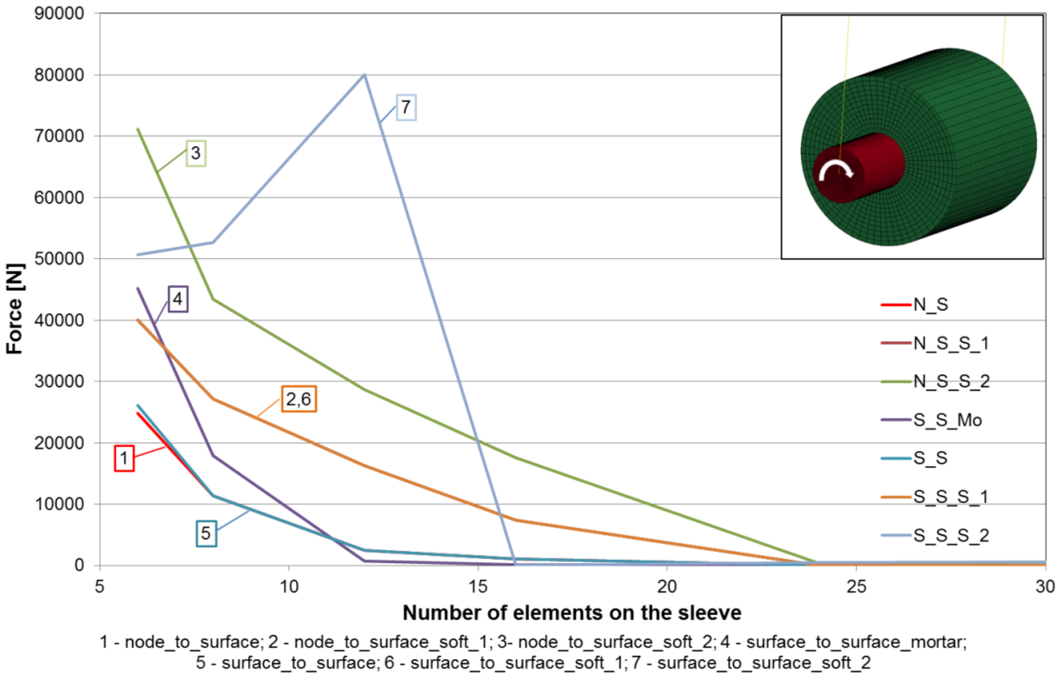



*Fig. 7. Beam resultant forces as functions of element numbers for shaft – sleeve rotational cooperation*

## 3. Contact analysis for shaft – sleeve translational interaction

The next investigated connection, which may cause serious problems in contact analyses, is a sliding (translational) joint connection. The model has been built analogously to the previous one, however, in this case the motion of the shaft was sliding and forced by a beam element orientated in X axis (Fig. 8).The gravity acceleration was also included in this case.

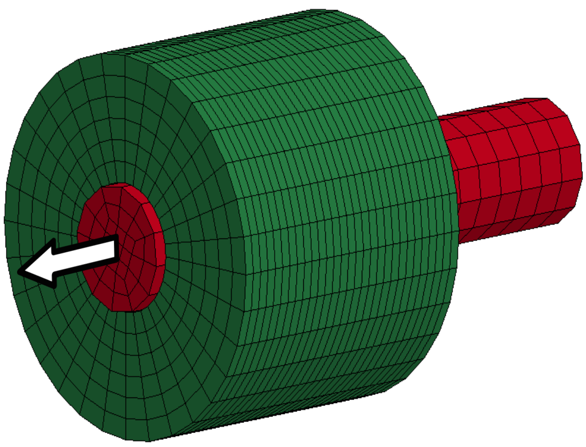
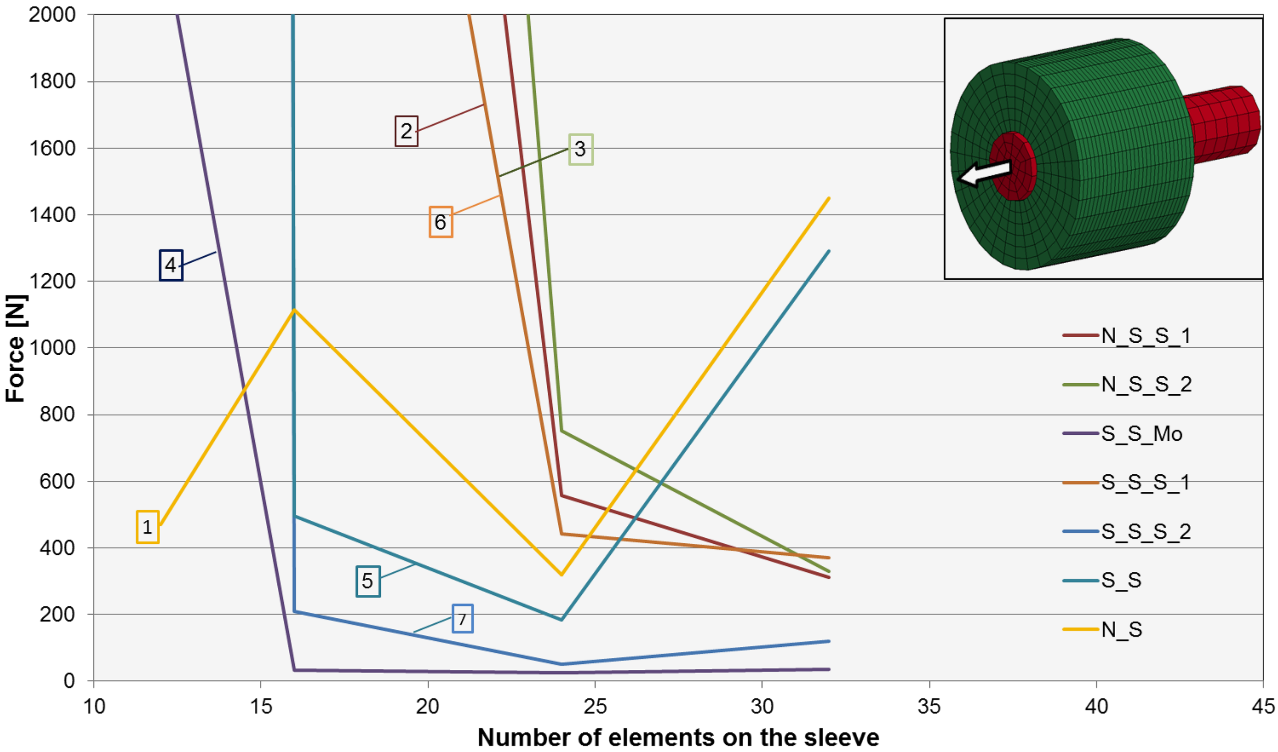


*Fig. 8. Schematic view for translational shaft – sleeve connection*

Similarly as in the previous analysis, the obtained results (Fig. 9) show that for a number of elements less than 12, the best performance is offered by the following contact types: “nodes to surface” and “surface to surface”. “Surface to surface” and “nodes to surface” contact types with additional “soft 2” option, at a small number of elements; generate very large resultant forces. “Mortar” contact type generates relatively small forces. For a number of elements greater than 24, most of the considered contact types generate small resultant forces (<10 [N]), however, “surface to surface” and “nodes to surface” contact types with additional “soft 2” option have worse performance.




1 - node_to_surface; 2 - node_to_surface_soft_1; 3- node_to_surface_soft_2; 4 - surface_to_srface_mortar; 5 - surface_to_surface; 6 - surface_to_surface_soft_1; 7 - surface_to_surface_soft_2

*Fig. 9. Beam resultant forces as functions of element numbers for shaft – sleeve translational cooperation*

## 4. Contact analysis for sliding bodies in harmonic motion

The last of the analysed cases was to investigate the contact procedure in the following configuration: a cube block is sliding over flat surface in harmonic motion (Fig. 10). An upper block is pressed against the lower block with the pressure of 2 [MPa] applied to its upper surface. 3D solid elements were defined as a fully integrated S/R. The upper block was defined as a slave interface and the lower block was defined as a master interface due to a larger size of its finite elements.

In this case, the coefficient of static friction was set to 0.4 [-], while the coefficient of dynamic friction was set to 0.3 [-]. The friction normal force, which results from the sliding, can be calculated from the known Amontons friction model, given by Eq. 5:

$$T = \mu \cdot N = 61.6\ [N], \tag{7}$$

where:

$\mu$ – coefficient of friction equals to 0.4 [-],

$N$ – pressing force equals to 154 [N].

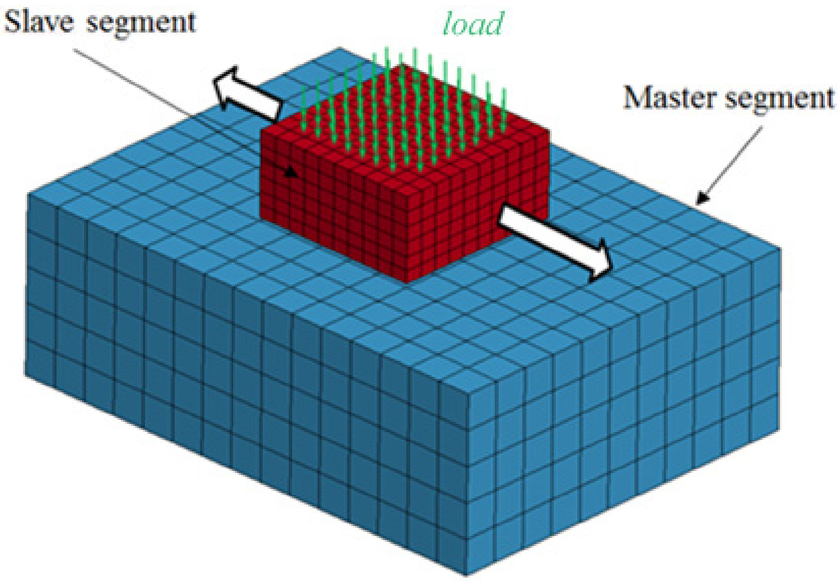



*Fig. 10. Schematic view of the model for translational flat block over block motion*

Because of the performed analyses, contact friction normal forces were obtained for different contact types. Differences between the numerical results and the analytical calculations (Eq. 5) were presented in Fig. 11. Based on this fact, it can be concluded that all the numerical results were understated in relation to the analytical value, however the difference did not exceed 1.5 [%] with the exception of "surface to surface" contact type with additional "soft 2" option (difference equals to approx. 7 [%]). Generally, "node to surface" contact type gave better results in this case, however the difference did not exceed 0.7 [%].

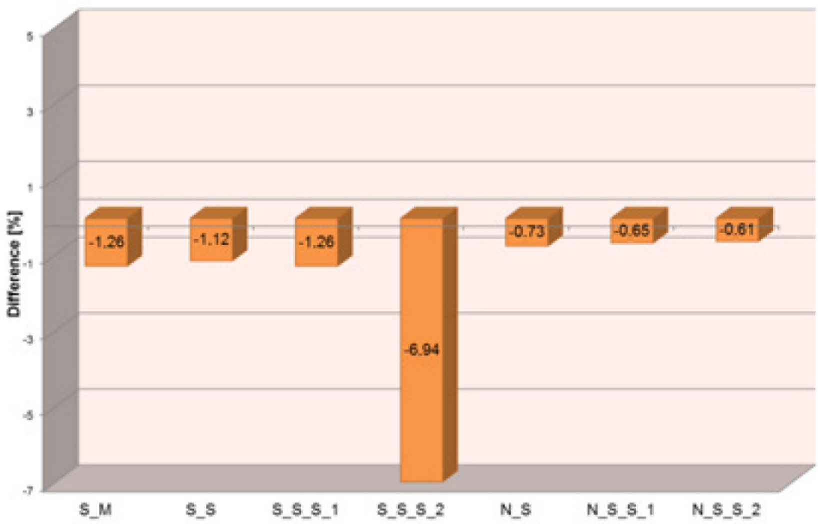



*Fig. 11. Differences between numerical results and analytical calculations for sliding bodies*

## 5. Conclusions

Contact modelling in LS-Dyna software is a complicated issue due to a large number of available contact methods and significant differences in the obtained results. The most common problem arising during the contact modelling is the occurrence of penetration between the interacting bodies, which leads to an increase of contact forces and, consequently, a local deformation of finite elements. A choice of a suitable contact method is a key issue when it comes to interaction modelling.

The obtained results show that a body with a less dense mesh should be defined as a master interface in contact analyses. "Surface to surface" contact type with additional "soft 2" option has showed worse performance than the other contact types. The increase in a number of finite elements in the connecting interfaces leads more to correct results both in rotational and translational cases.

It is worth pointing out that all the results were obtained for a penalty function method with the same parameters. All the differences arise from different methods applied to solve contact kinematics. These differences can be surprisingly high. Therefore, in contact modelling, it is important to check whether a selected contact method does not cause any additional effects, which are incorrect from the physical point of view.

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