Analysis of Blank-Die Contact Interaction in Pulsed Forming Processes

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Abstract

During recent decade, significant efforts were dedicated to increasing the amount of Aluminum Alloys in automotive parts in order to reduce the net weight of cars. Processes of pulsed forming are known to expand the capabilities of traditional stamping operations. Propagation of pulsed electromagnetic field can be defined by quasi-stationary Maxwell equations, solved numerically using a non-orthogonal Lagrangian mesh. Suggested formulation included modelling of contact interaction of the blank with deformable die. Mild contact model based on introduction of acting-in-vicinity forces repelling the surfaces to be in contact was employed. It was tested by analyzing the elastic impact of bars and then was applied to the corner filling operation. This operation was analysed as a single pulse and as a multi pulse forming process. It indicated that some compromise between the blank formability enhancement and level of contact stresses on the die surface can be found. In addition, some examples of tubular parts pulsed press fitting using tube expansion with pulsed pressure were analyzed. Specific attention was paid to the analysis of factors playing important role in residual contact pressure between the exterior and interior tubes in pulsed press fitting operation.

Keywords:

Sheet metal forming, electrical discharge, tool

1 Introduction

The idea of using the high velocity and high energy rates to form metal and to assemble parts has been known for several decades. In the literature it was established that high velocity metal forming was able to form complex components with one sided dies and also to minimize the springback effect. During recent decade, significant efforts were dedicated to increasing the amount of Aluminum Alloys in automotive parts in order to reduce the net weight of cars. Implementation of these materials into auto industry makes the issues of insufficient formability and springback even more difficult.

Processes of high rate forming, such as explosive, electromagnetic (EMF) and electro-hydraulic forming (EHF) are known to expand the capabilities of traditional stamping operations and also to provide high level short duration pulses of pressure. However, increasing the velocity of the blank leads to higher loads on the die. Taking into account deformation of the die in previously developed model of electromagnetic forming provides better understanding which forming regimes are realistic from die strength and durability point of view.

2 Theoretical Approach

Propagation of electromagnetic field within coil-blank-die-air system can defined by quazistationary Maxwell equations:

$$\nabla \times \mathbf{H} = \mathbf{j}, \tag{1}$$

$$\mu_a \frac{\partial \mathbf{H}}{\partial t} = -\nabla \times \mathbf{E}, \tag{2}$$

$$\mathbf{j} = \sigma\left(\mathbf{E} + \mu_a \mathbf{v} \times \mathbf{H}\right), \tag{3}$$

where $\mathbf{H}$ is magnetic field intensity; $\mathbf{j}$ is current density; $\mathbf{E}$ is electric field intensity; σ is electric conductivity; $\mathbf{v}$ is velocity; μ_a is magnetic permeability of medium under consideration. In EMF processes, the coil and die are almost stationary, while the blank is quickly accelerated; therefore, the equation for magnetic field intensity $\mathbf{H}$ can be transformed in Lagrangian form [1]. Dynamic elastic-plastic deformation of solid can be defined by the following equation:

$$\rho \frac{\mathrm{d}}{\mathrm{dt}} \int_V \mathbf{v} \mathrm{dV} = \oint_S \sigma \cdot \mathrm{d}\mathbf{s} + \int_V \mathbf{f} \mathrm{dV}, \tag{4}$$

where σ is stress tensor; ρ is density; $\mathbf{f}$ is EM force defined from Maxwell equations. A numerical procedure of integration (1)-(4) simulating the elastic-plastic deformation of the blank was discussed in [1]. An explicit integration procedure was considered to be suitable to simulate a high-rate deformation process. Electromagnetic (EM) part of the problem was solved using an implicit integration procedure. The integration step in EM problem was n times larger than in elastic-plastic problem. The parameter n was defined for each practical case in order to represent the changes in distribution of electromagnetic field appropriately and spend appropriate computational resources.

3 Soft Contact Algorithm

The most popular method of modelling contact interaction is based on the geometrical analysis of mutual position of boundary nodes of each mesh. At every integration step, it is being verified whether a boundary node of the blank has penetrated through the certain element of the surface mesh of the die. If it happens, it is necessary to make certain corrections bringing the node back on the surface of the die and let the node slide along the die's surface instead of penetrating through this surface. A significant downside of this approach is in occasional penetration of the node through the surface. It usually happens due to insufficient accuracy of calculations. As soon as the blank's node penetrates into the die's surface, it is unable to return back and further calculations are useless. Therefore, a different approach based on the idea of a mild contact was employed. This approach is very popular in molecular dynamics where it is employed to model thousands of atoms colliding and springing back from each other. The contact force is in inverse proportion to the distance between the interacting surfaces. From mathematical point of view, mild contact is some variable boundary of an unknown shape where we have to satisfy the non-penetration condition and equilibrium of forces between both surfaces. This idea is based on the introduction of acting-in-vicinity forces repelling the surfaces to be in contact. As a result, the surfaces do not come in contact, but stay at some very small distance from each other. The force is localized in a small neighbourhood of mesh elements, and it increases to infinity when the distance between them is approaching zero. In other words, when the blank and the die are in geometrical contact, the force between them is localized on their joint surface. Mathematically it can be considered as continuous functions where the absolute value of the force F is given by:

$$F = \begin{cases} k\left(\frac{1}{h} - \frac{1}{h_0}\right) & \text{at} \quad h < h_0 \\ 0 \quad \text{at} \quad h \geq h_0 \end{cases}, \qquad (5)$$

where h is the actual distance from the node of the blank to the die surface, and h_0 is the width of the layer where the force is different from zero. The direction of the force F is aligned along the local normal to the die surface.

The described algorithm allows taking into account the Coulomb friction and other contact effects. Indeed, when we compute the interaction of the node with an object of the other mesh, we know their mutual positions and velocities. It is sufficient to find the friction directed along the tangent to the surface and opposite to the tangential part of the relative velocity vector.

4 Testing of the Contact Algorithm

The developed contact algorithm was tested for contact interaction of two elastic bodies with the following characteristics: ρ = 2800 kg/m³, G = 26.5 GPa, K = 65.1 GPa, the stress-strain curve was approximated as $\sigma_s = 0.24+0.56\varepsilon$,GPa. For simplicity, the dimensions of the colliding bodies were 2 m x 4 m. The initial velocity of the first body (shown as left in Figure1) was set equal 10 m/sec, while the velocity of the second body (right in Figure 1) was set to be zero. For one-dimensional case of bar impact, the contact stress can be calculated using the following formula (assuming that the material of both bodies is identical).

$$\sigma = \frac{\rho c v}{2}, \qquad (6)$$

where c is speed of sound equal to 5000 m/sec. For the selected material and speed of impact, the contact stress σ is 70 MPa, which is below the yield stress for chosen material. The duration of the impact is equal to 0.0016sec. Parameters of the contact force from formula (6) were set k = 0.1GN and h_0 = 0.01 m.

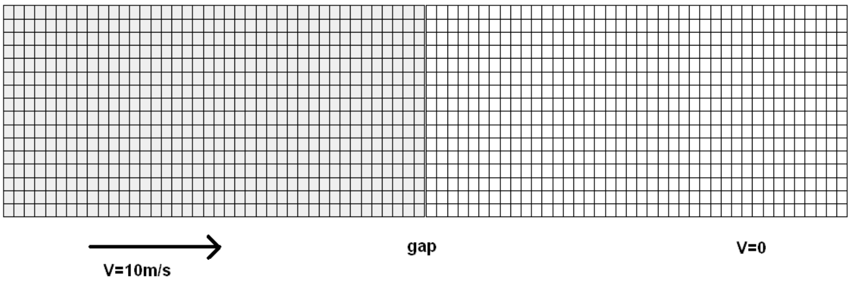

Figure 1: *Numerical meshes in bar impact test.*

In Figure 2, the stress history at the central point of the contact surface is shown with the red solid line using mild contact defined by equation (5), and the result of the analytical solution is illustrated with the dashed line. Since we are modelling the impact of two identical solids, the impact plane also serves as a symmetry plane. From this perspective, it is possible to represent the problem as an impact of one solid into the rigid wall with the velocity of 5m/sec. For the rigid wall, the boundary condition can be formulated as follows: if(x < x0) then {x = x0, v = 0}, corresponding to the simplest algorithm providing non-penetration of the nodes through the rigid contact surface. The numerical results obtained with the use of this approach are shown in Figure 2 with blue solid line. In this particular case the contact surface has a coordinate of x_0 = 4.005.

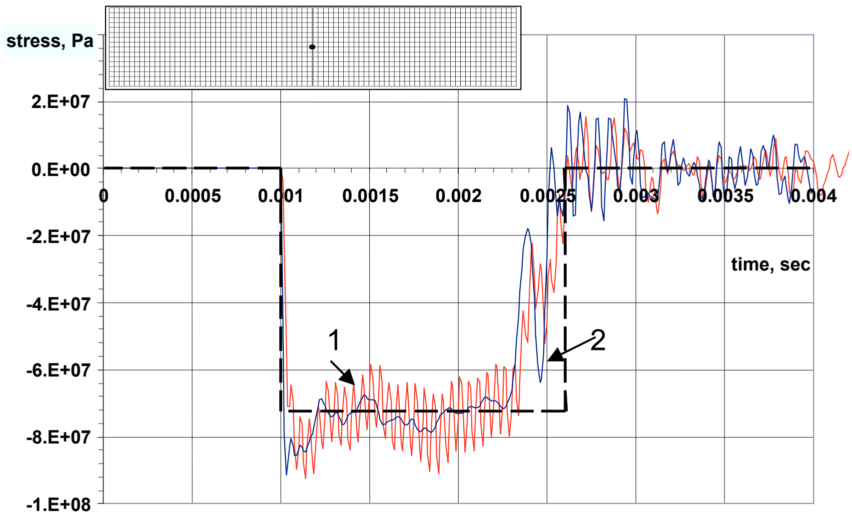

Figure 2: *Comparison of numerical results on mild contact of two elastic solids (1-red solid line), contact of elastic solid with the rigid surface with 50% less impact velocity (2-blue solid line) and analytical results (dashed line) on contact stress of two elastic bars. Contact stress is registered at the middle point of the contact zone (•).*

As it is shown in Figure 2, the duration of the impact obtained both numerically and from analytical solution (as a ratio of the double length of the bar and sound velocity in this material) is equal to 0.0016 sec. The average value of contact stress σ in selected control point • also is identical and equal 70 MPa. The contact stress σ has "shortwave" and "longwave" variation: the "longwave" variation is related to 2D character of the problem; the "shortwave" variation is related to the formulation of the mild contact, and frequency of this variation is defined by the stiffness and width of the contact layer.

In order to reduce the amplitude of "high frequency" variation of contact stress, the artificial viscosity can be incorporated into equation (5) in such a fashion that viscous component of the contact stress acts against the direction of movement of the contact node and is equal to the value of $\nu|\mathbf{v}|^2$, where ν is coefficient of artificial viscosity. Second order of velocity module allows to affect the behaviour of the contact nodes with higher velocity more efficiently. The influence of coefficient ν on variation of the contact stress was studied: $\nu = 10^4$ reduces the variation of the contact stress; however, its further increase up to 10^6 makes it unacceptable, which indicates that an optimal range of ν exists.

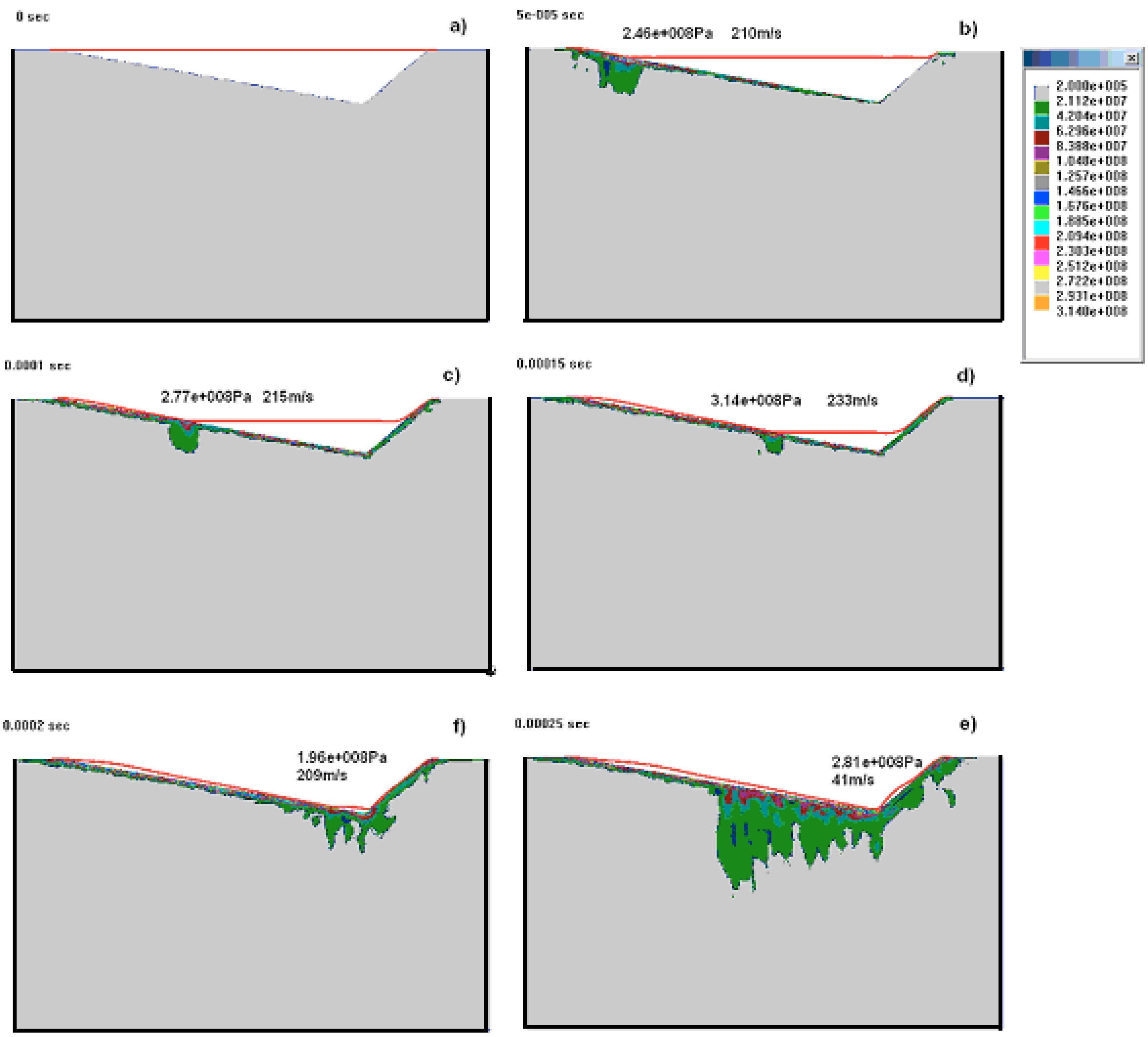

Figure 3: *Values of $J_2(\sigma)$ and velocity of the blank vs. time (P_0 = 30MPa) for a single step corner filling operation.*

5 Dynamic Contact Interaction between the Blank and the Die

The problem of dynamic contact interaction of the blank and deformable die shown in Figure 3 is under consideration. Aluminum blank is 1 mm thick and 280 mm long.

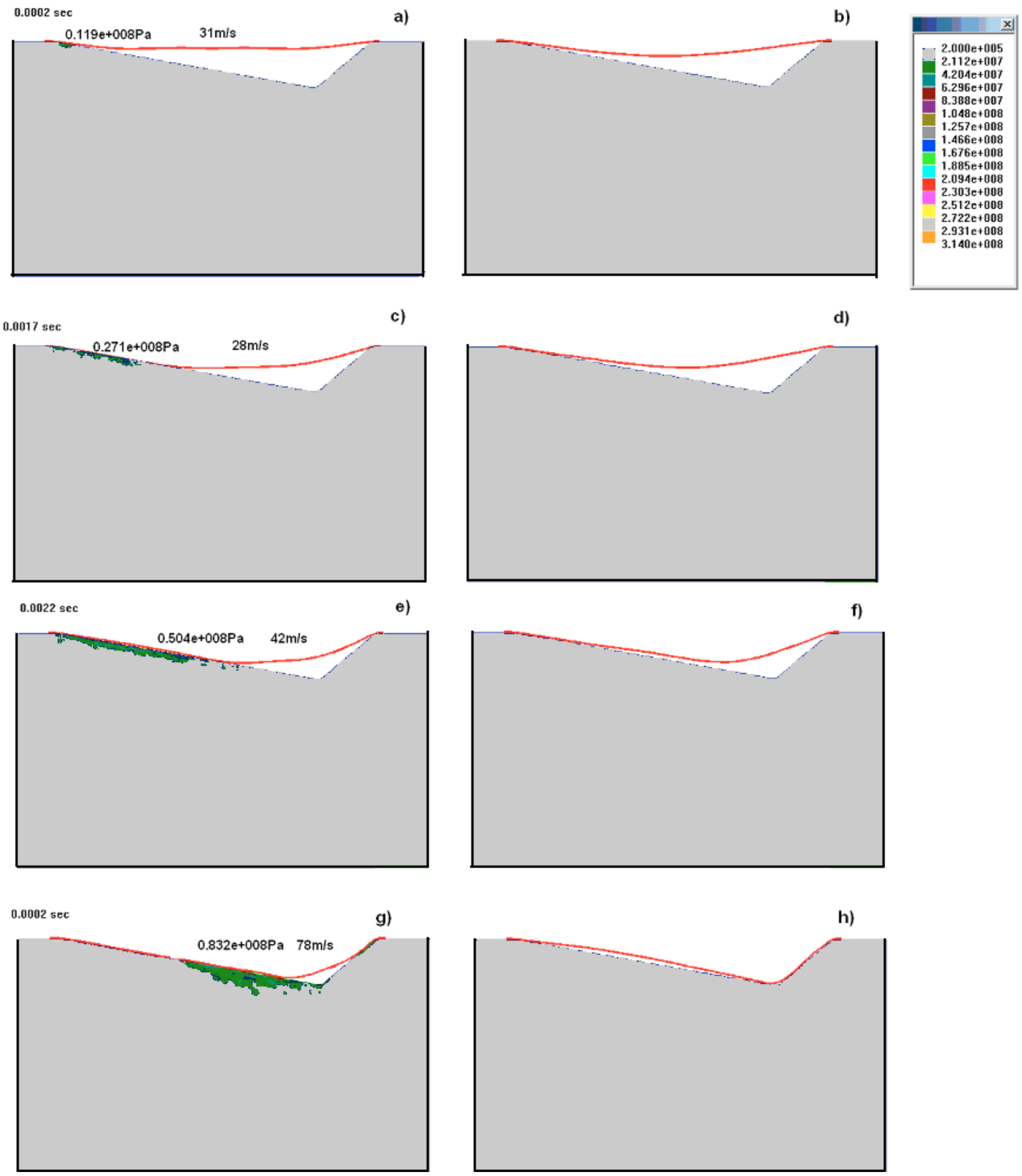

Figure 4: *Values of* $J_2(\sigma)$ *and velocity of the blank vs. time.*
a,b) - P_0 *= 5MPa; c,d) -* P_0 *= 10MPa; e,f) -* P_0 *= 15MPa; g,h) -* P_0 *= 25MPa.*

The die is made out of steel and has an angle of 130 degrees. Material properties for the die are as follows: density ρ = 7800 kg/m3; shear modulus G = 76.9 GPa, bulk modulus

K = 166 GPa, the stress-strain curve $\sigma_s = \sigma_0 + A\varepsilon$, σ_s = 620 MPa, work hardening modulus A = 1020 MPa. For the blank these properties are: ρ = 2735 kg/m³, G = 26.5 GPa, K=65.1 GPa, σ_0 = 195 MPa, and A = 520 MPa. In order to reduce the computational time, electromagnetic forces were modelled as a mechanical pressure (7) uniformly applied to the blank surface.

$$P = P_0 e^{-\beta t} \sin^2(\omega t)\,, \tag{7}$$

where $\beta = 10^4\,\mathrm{sec}^{-1}$; $\omega = 10^5\,\mathrm{sec}^{-1}$; P_0 was varied to define the appropriate value to fill the die cavity with one pulse. Results of simulation are shown in Figure 3. The stress state of the die material was characterized using second invariant of the stress tensor $J_2(\sigma)$. Maximum value of $J_2(\sigma)$ in Figure 4 is coupled with maximum velocity of the blank for different time intervals at P_0 = 30MPa. For smaller values of pulsed pressure, the blank does not fill the die completely.

In Figure 4 the process of multiple loading of the blank with four pulses of pressure is described: the left column of figures (a,e,c,g) illustrates the middle portion of the process, while the right column of figures shows the final shape of the blank after corresponding forming step of the blank is completed. The pressure pulses with the following pressure amplitudes were applied: 5, 10, 15, 25 MPa. Maximum value of $J_2(\sigma)$ for the die and the impact velocity of the blank in this case are significantly lower than for a single pulse loading when the blank filled the die cavity as a result of only one forming step.

6 Dynamic Contact Interaction in Pulsed Tube Press Fitting Operation

Press fitting of tubes is another application of high-rate technologies where the advantage of pulsed pressure can be taken. In order to accomplish a tight press fit joint, the pressure pulse is applied inside the internal tube, as shown in Figure 5. Appropriate pressure pulse parameters and initial clearance between the inner and outer tubes should be identified to provide good quality joint. A practical example of joining tubes from stainless steel is discussed below. The internal tube with the external diameter of 25 mm and thickness of 2 m was expanded into the external tube from the same material with the external diameter of 50 m. The internal diameter of the external tube was varied in the range between 25.0 m and 26.25 mm in order to investigate the influence of the initial radial clearance on the residual contact pressure. This analysis is very important to formulate the requirements for tubes' accuracy. The dynamic yield stress of both inner and outer tubes material was 300 MPa , E=2x10^{11} Pa, ρ=7800 kg/m^3, Poisson's ratio was μ=0.3. The pressure pulse for such type forming processes [2] can be approximated as:

$$p(t) = p_o\, e^{-t/T}\,, \tag{8}$$

where P_o is the pressure amplitude, which depends on the energy stored in the EHF machine capacitors; T is a decreasing period, which depends on machine characteristics. In current analysis P_o=350 MPa and T = 8.3 x 10^{-6}sec.

The numerical model, taking into account elastio-plastic deformation of both the inner and outer tubes, was used to investigate the contact interaction mechanism. During the initial stage, the internal tube is being expanded driven by the internal pressure pulse until the radial displacement reaches the initial radial clearance between the tubes. The simulation results of the impact dynamics are given in Figures 6 and 7 for rather small initial clearance of 0.006 of inner tube internal radius. Radial displacements ΔR of the tubes contact surfaces are given in dimensionless form $\Delta R / R_o$. Real time is related to the expression $R_0 / \sqrt{\sigma_{s0}/\rho}$, where R_o is the initial internal radius of the inner tube. Stresses are related to the tubes' initial yield stress σ_{so}.

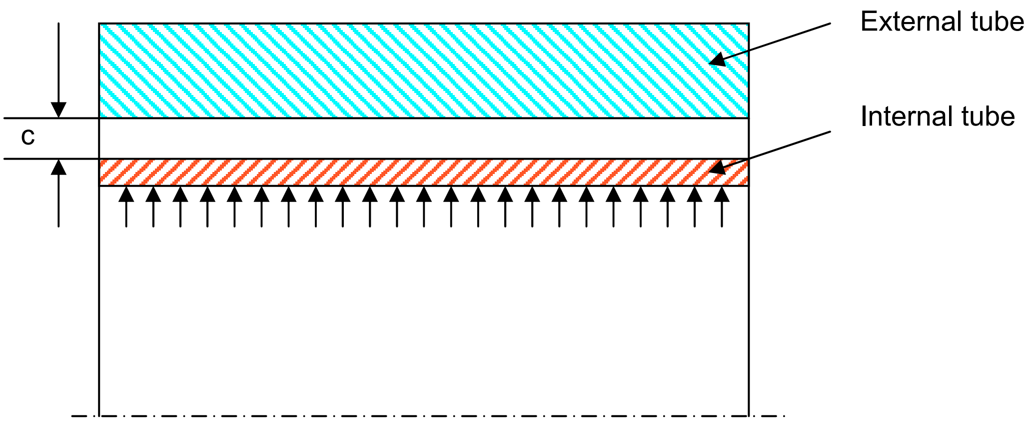

Figure 5: *Schematic of tube press fitting operation by internal tube pulsed expansion.*

At the beginning of the contact interaction, the radial compressive waves propagate inside both tubes. As long as the internal tube is much thinner than the external, the stress wave propagates through its thickness significantly faster than through the external tube. During the internal tube free expansion stage, before the first impact with the external tube, the pressure pulse is partially spent to increase the internal tube. Evidently, the larger is the initial clearance the larger is the velocity of the internal tube and the smaller is technological pressure inside it. Therefore, the radial stress on the inner surface of the internal tube during contact interaction is dependent upon the initial radial clearance between the tubes.

During the first impact, the radial stress wave propagating from the contact surface is then reflected from the inner surface of the internal tube and returns back to the tubes contact surface. At this point, the contact between the tubes is usually terminated. Contact termination may be prevented if the internal pressure is still applied to the internal surface of the internal tube. The displacements of the internal and external tubes are shown in Figure 6 with the dots and thin solid line, respectively. Their joint displacement is marked with the thick solid line. If the first impact happens at the beginning of internal tube acceleration, the technological pressure is large enough, so the reflected radial stress wave from the internal tube produces compressive radial stress at the contact surface, and both tubes move together until a reflected wave comes back from the external tube's outer surface. Due to a significantly larger thickness of the external tube, the joint motion of both tubes in this case continues substantially longer. As a result, the average velocities of both tubes are much closer to each other compared to the moment when the reflected wave from the internal tube first time reaches the contact surface. After the contact termination, the internal tube is being accelerated by the technological pressure and impacts the external tube again.

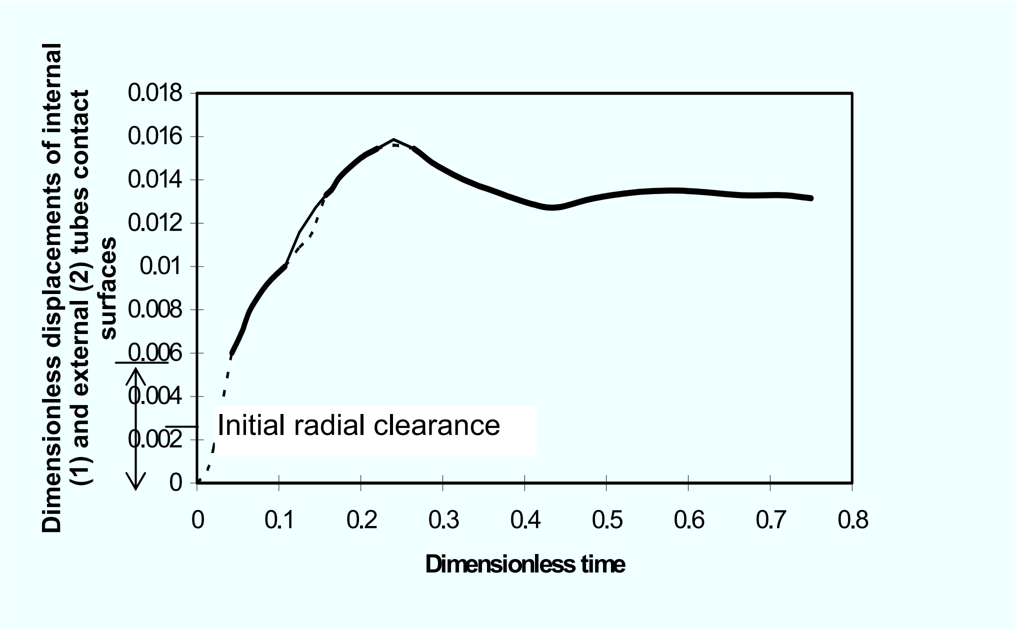

Figure 6: *Displacements of internal's tube outer surface (dotted lines), external's tube inner surface (thin solid line) and their joined displacement (thick solid line) vs. time.*

The quality of assembling process is significantly affected if both tubes start unloading from approximately the same point. In this case, residual contact pressure will be defined by the difference of elastic deformation in internal and external tubes. Evidently the elastic displacement of the external tube should prevail over the elastic displacement of the internal tube.

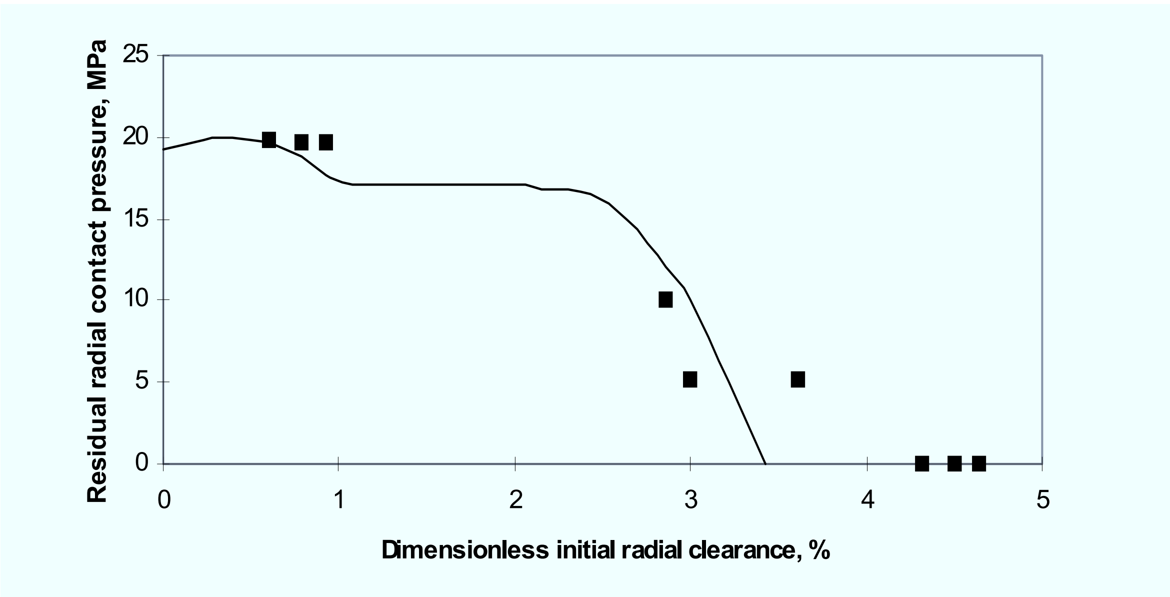

Figure 7: *Influence of initial clearance on residual contact pressure in press fitting process*

If the initial clearance between the tubes is substantial, and the pressure pulse has already diminished at the beginning of the first impact, the contact is being terminated after the reflected wave from the internal tube comes to the contact surface.

Increasing the initial clearance results in growth of the first impact velocity and descending of the technological pressure value. In this case the internal tube transfers all its velocity to the external tube, the contact between the tubes gets terminated after the reflected wave from the internal tube inner surface comes to the contact surface. In this case, the inner tube unloads after the first impact, while the external tube keeps expanding and unloads later. In such case, the unloading of internal and external tubes starts from substantially different points. This mechanism gives no residual contact pressure between the tubes and provides poor quality assembling, as indicated by the small residual contact pressure in Figure 7.

As a result of this discussion, it can be concluded that the maximum initial radial clearance between the tubes should not exceed a certain value, which is dependent on the pulse duration, geometry and material of tubes. For the discussed set of parameters the initial radial clearance should not be more than 3% of the inner tube internal radius.

7 Conclusions

1. Developed numerical model takes into account propagation of electromagnetic field, high-rate plastic deformation of the blank in contact with the deformable die, heat accumulated in the coil, and mechanical loads on the stamping die.
2. Static preforming of the blank into the die cavity combined with pulsed restrike operation or pulsed forming of the blank in several steps may substantially reduce the contact stresses on the die surface.
3. Analysis of the mechanism of contact interaction of tubes in press fitting operation indicated that initial clearance between the tubes should be minimized to provide good quality joint.

References

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