Analysis of Contact Stresses in High Speed Sheet Metal Forming Processes*

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Abstract

In high speed metal forming, determination of contact stresses applied to forming dies is necessary in order to identify the requirements to the die material. Contact stresses greatly control the die design due to their effects on die durability. Very high contact stresses and fracture under impulsive loading have been reported in literature on contact type of high speed forming. In pulsed forming operations using electro-hydraulic forming (EHF), a work piece is often accelerated into the die cavity of a desired shape resulting in a substantial impact pressure on the die. Contact algorithm and mesh size play an essential role in providing accurate results in such high speed processes. Using the soft contact model with an appropriate control of the penetration value provided stable and consistent contact stresses.

Keywords

High Speed Forming, Die Stress, Contact

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1 Introduction

Electro-hydraulic forming (EHF) is a high speed metal forming process where the blank is forced into the die cavity by a shock wave that propagates through a chamber filled with liquid and delivers a pressure pulse to a sheet metal blank. The shockwave produces a force sufficient to deform the blank against a forming surface defined by a cavity in a die [1]. The shock wave is produced in a fluid filled chamber by the high-voltage discharge of capacitors between two electrodes positioned in the chamber shown in Figure 1. The process is extremely fast; uses lower-cost, single-sided tooling; and potentially derives significantly increased formability from many sheet metal materials because it involves elevated strain rates [2].

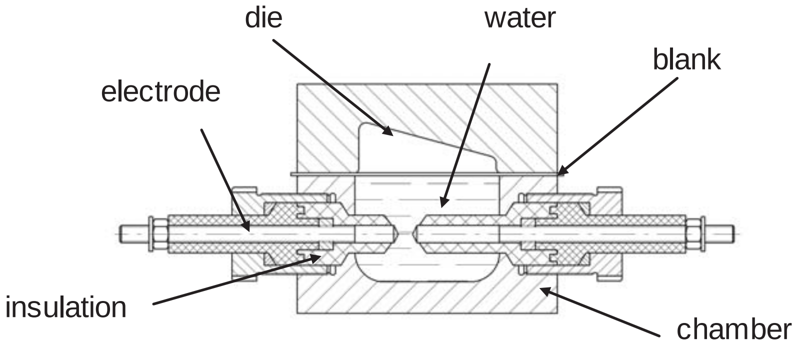

***Figure 1**: Design of the electrohydraulic forming tool.*

Beneficial for material formability, high strain rates and high blank velocity are disadvantageous from the die durability perspective. This became apparent during the experimental work accomplished in EHF project at Ford Research and Advanced Engineering, as a number of die durability issues were encountered. Examples of the failure modes observed in different die configurations are:

- Complete die failure, in an experimental die built from epoxy powder composites, the die was broken into several pieces due to low strength of selected die material.
- Appearance of a major, through the thickness, crack.
- Local cracking in the highly loaded areas.

Reduced die durability coupled with lack of existing EHF die design/durability guidelines made research and development of such guidelines mandatory. In general, failures of EHF dies, as any other stamping dies, can be attributed to wear, plastic deformation, and fatigue. Finite element analysis (FEA) is a very important part of this research, as it predicts the amount of plastic deformation in the die, and its output is used in fatigue analysis, in die wear estimation, and in the die material selection.

2 Problem Statement

Die durability is very sensitive to the contact stresses. The field of die design for high speed forming is relatively new, and modern applications are still emerging. The foundation of shock compression science is based upon observations and analysis of the mechanical responses of solid samples to shock-loading pulses [3]. Most of the work done in the area of fast, transient loading is experimental in nature. This is either due to complexities of geometry, the nonlinearity of the material behaviour or both. Closed form analytical solutions are generally rare and apply only to some small subset of the overall problem. Numerical solutions, in the form of finite difference and finite element codes, have been successfully used in the past. However, the computer codes available for dynamic analysis are quite complex [3]. Considerable experience with both the codes and the physical problems they are intended to solve is vital. There are several occurrences where different users with different backgrounds get quite different results. Codes have a number of free parameters that must be set by users. This is a major requirement that if not met properly, incorrect physics is being computed and correlation between calculations and experiments is rare.

Further, for each calculation the user must be sure to use an appropriate mesh that should provide converging solution. Once the mesh is chosen, an appropriate artificial viscosity should be selected to avoid damping the solution and artificial ringing. In dynamic calculations, the location of the stress or pressure gradient is a function of time as well as the space. In many instances wave propagation and their reflection from material interfaces and geometric boundaries control the response.

3 Limiting Stress in High Rate Forming Impact

3.1 Determination of Correct Impact Stresses

The most popular method of modeling contact interaction is based on the geometrical analysis of mutual position of boundary nodes of each mesh [2]. 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 and logical gaps in the contact algorithm. As soon as the blank's node penetrates into the die's surface, it is unable to return back, and further calculations are erroneous.

Several simulations were run for the test problems for which the analytical solutions exist: elastic impact of steel bar to steel bar and steel bar to steel sheet plate impact were simulated according to the schematic in Figure 2. In this elastic impact the contact stress is $\sigma = \rho c \Delta v$ where ρ is the material density, c is the elastic wave speed, and Δv is the change in particle velocity [4].

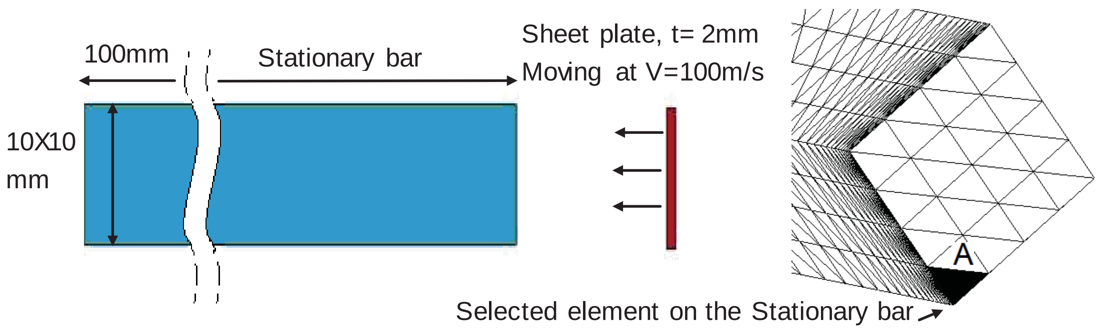

***Figure 2**: Sheet plate, meshed with shell elements impacting a bar meshed with tetrahedral elements.*

For common steel at impact velocity of 100m/s this stress value is around 2GPa. Initial attempts to analyze these problems showed excessive contact stresses. Figure 3 shows that unphysical stress values can substantially exceed 2GPa.

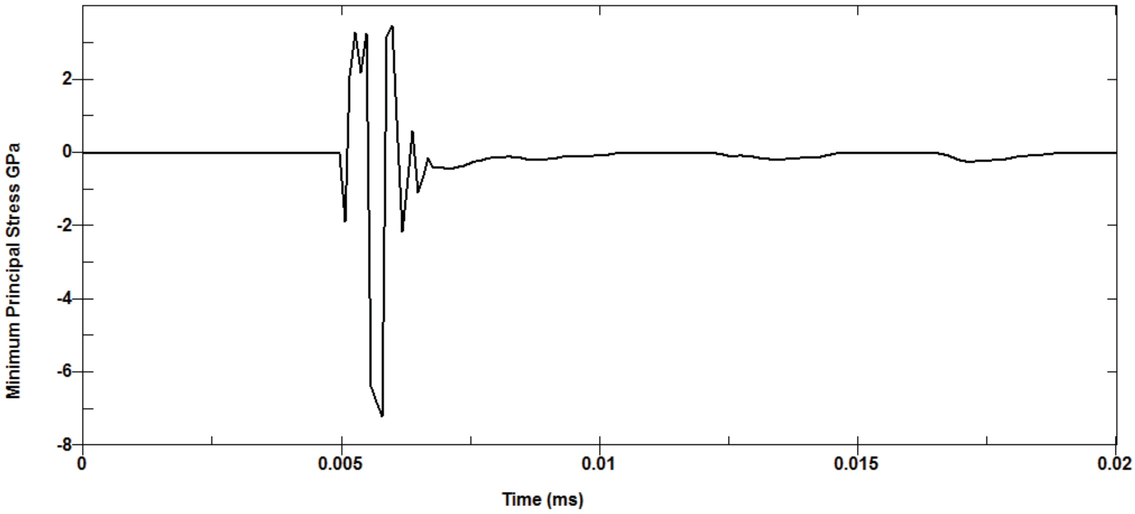

***Figure 3**: Unphysical peak in contact pressure in element A of the impacted bar*

Those incorrect reported values cannot be carried over for further design steps. Simulation parameters and model formulation must be revised, and solution correctness judged. The analytical solution for this impact test is presented in the following section.

3.2 Modelling of Dynamic Problems

Electro-hydraulic forming of sheet metals is a dynamic problem. In simulations of such physical problems it becomes necessary to recognize two important factors:

- The rate at which our observed phenomenon changes;
- The fact that information is propagated at a finite speed

In mechanical systems, this means taking into account both strain rate and wave propagation effects, [3].

Waves in rods and rod-like structures have been considered to create a state of uniaxial stress. In this configuration, however it is impossible to reach very high tri-axiality stress states. With velocity increases, 2D and 3D effects begin to dominate the rod deformation. Plasticity and material failure govern the magnitude of the stress the rod can carry. Typical idealized stress-strain curves used in computations for materials in such configurations are derived from typical uniaxial stress experiments routinely performed under quasi-static loading conditions.

In order to examine other possible states of die material in pulsed forming conditions, we need to achieve higher level of stresses: we study a thin plate, known as the flyer striking a thicker plate. Waves will radiate into the stationary plate and the flyer plate in the thickness direction as well as in the transverse direction. However, until these reflect from the lateral boundary and return to the centre, a state of uniaxial strain (but 3D stress) will exist there. This change in geometry can achieve hydrodynamic stress state (or pressures) substantially higher than strength of the material at high strain rates.

3.3 Shock Waves in Solids

Uniaxial strain state can be visualized where the deformation is restricted to one dimension such as in the case of plane waves propagating through a material dimensions, and constraints are such that the lateral strains are equal to zero, see [5], [6], and [7], for more detailed description.

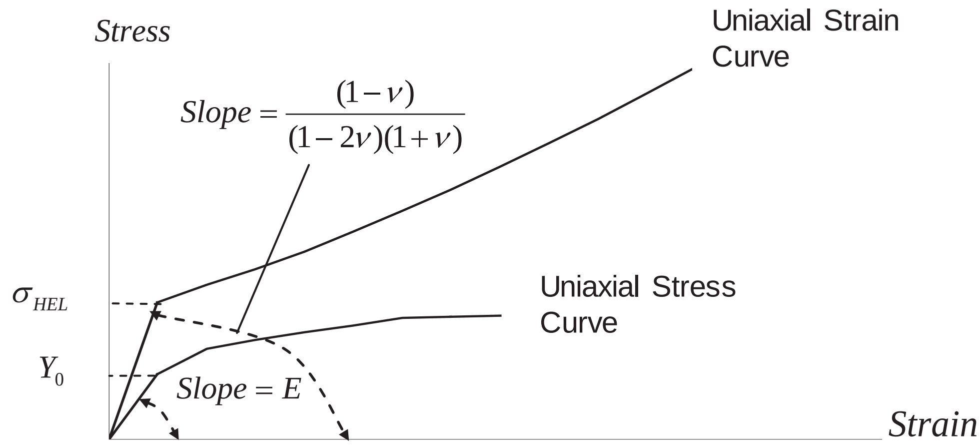

Figure 4*: Schematic of Stress-strain curve for uniaxial tensile test (upper) vs. Stress-strain curve for uniaxial strain states (lower).*

The following expression for σ_1 can be obtained as in [3]:

$$\sigma_1 = \frac{E}{3(1-2\nu)}\varepsilon_1 + \frac{2}{3}Y_0 = K\varepsilon_1 + \frac{2}{3}Y_0 \tag{1}$$

where $K = \frac{E}{3(1-2\nu)}$ is the bulk modulus, σ_1 is the max principal stress, ε_1 is the corresponding principal strain and Y_0 is the static yield strength. In terms of pressure the above equation is expressed as

$$\sigma_1 = P + \frac{4}{3}Y_0 \tag{2}$$

This is the stress-strain relation for the uniaxial strain loading described in [3]. For uniaxial stress state, stress-strain relation is $\sigma_1 = E\varepsilon_1$ with the relation taking the form reported in a typical uniaxial tensile test shown in Figure 4. Thus, the most important difference between the uniaxial stress and uniaxial strain states is the bulk compressibility. In this case the stress continues to increase regardless of the yield stress or strain hardening. The stress-strain curve takes the form shown in Figure 4.

For the case of elastic 1 D strain we obtain

$$\sigma_1 = \frac{(1-\nu)}{(1-2\nu)(1+\nu)} E\varepsilon_1 , \qquad (3)$$

from which we notice an increase in modulus by $\frac{(1-\nu)}{(1-2\nu)(1+\nu)}$. In Figure 4 we also notice that the yield point for the uniaxial strain is referred to as the Hugoniot Elastic Limit, σ_{HEL}, which represents the maximum stress for 1D elastic wave propagation.

The upper curve in Figure 4 is known as the Hugoniot curve. If the material is strengthless (Y_0 = 0), it would follow a curve called the hydrostat and characterized by $P = \frac{\Delta V}{V}$, where V is the original volume. For an elastic perfectly plastic material the hydrostat curve has a constant deviation below the Hugoniot curve by $\frac{2}{3}Y_0$. If the material hardens with increasing strain, the difference between the Hugoniot and hydrostat curves increases. From such a figure we can deduce that the maximum stress in an elastic/plastic impact will have a slope less than $\frac{(1-\nu)}{(1-2\nu)(1+\nu)}$.

4 Impact Stress values in FEA simulations

The previous section sets the boundaries for the limiting stress in impact of deformable solids. This can be used to identify the unphysical high stress values which can be obtained in impact simulations. This is necessary in the die design process for EHF because using of artificially high stress values in die design can be prohibiting to this technology or lead to excessively high cost of die material and its surface treatment.

Judging the accuracy of FEA solution is not easily achieved in the impact applications. Convergence may not be noticed because of other factors, such as penetration and hour glassing associated with solid elements which can return misleading values or even worse, cause instability. The following simulation was run on elastic steel bar to bar impact at velocity of 100 m/s in LS Dyna with the standard penalty formulation [Soft = 0, *CONTACT_SURFACE_TO_SURFACE contact with the SSTYP = MSTYP = 3, SLSFAC = 0.1, no artificial viscosity applied and the default values used for all other parameters] with bars meshed into 1 mm hexahedra solid elements as shown in Figure 5.

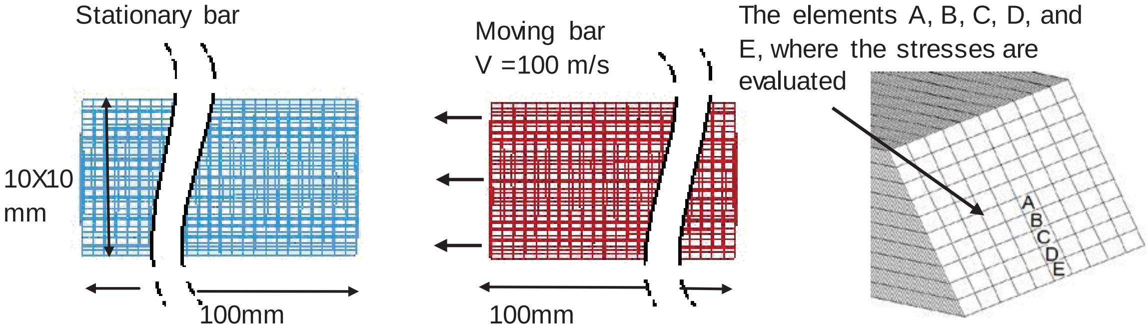

Figure 5*: Bar to bar impact model meshed with 1 mm hexahedra elements*

The pattern of the stresses doesn't agree with the analytical solution of 2 GPa contact stress. According to Figure 6 a maximum stress appeared near the edges. It should be emphasized that the solution was very sensitive to contact type used.

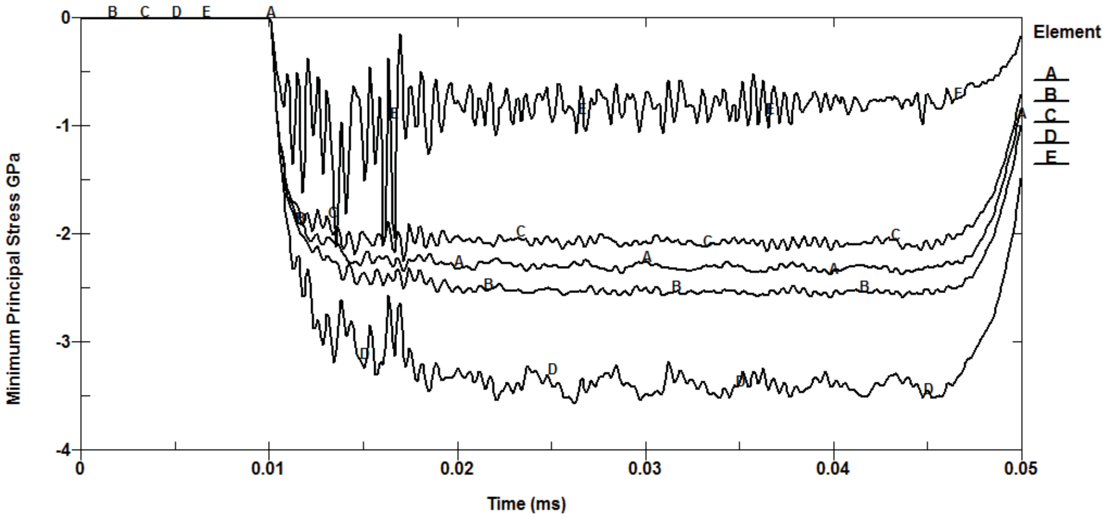

***Figure 6**: Bar to bar impact with incorrect stress values distribution at selected elements*

Consistent stress values, as shown in Figure 7, were achieved with one side of contact formulated into segments SSTYP = 0 and the other as a part type MSTYP = 3. Searching for the penetrating nodes and contact force update were required to be at high frequency. In LS Dyna bucket sorting identifies the nearest segment for each slave node. Number of cycles between bucket sorts was set to 1. For the same reason, the number of cycles between contact force update for penalty formulation was also set to 1. Those options help to keep penetration at the minimum level.

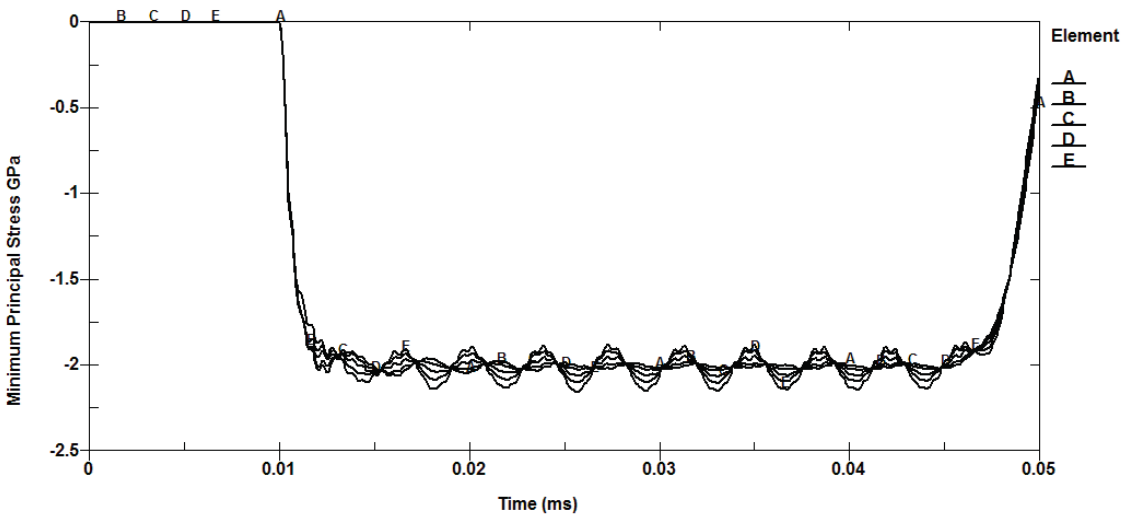

***Figure 7**: Bar to bar impact with uniform stress values distribution*

Also according to [8], bulk viscosity TYPE 1 was used with coefficients Q1= 1.5, Q2= 0.06. Soft= 1 penalty formulation was used that takes into account the nodal masses and global time step size in contact stiffness calculations to achieve better stability. Proper artificial viscosity and hour glassing control were necessary and lead to uniform solution for all contacting elements.

Artificial viscosity is included in Euler and Lagrange codes to allow the code to handle the shock waves which are mathematically discontinuous, and to provide grid stabilization for quadrilateral and hexahedral elements which use one point (reduced) evaluation element formulation as indicated in [3]. It is worth noting that higher contact stiffness helped to reduce the penetration. However, values exceeding the maximum stress introduced in the previous sections were obtained. Same was noticed with improper values for penetrating node searching parameters.

Theoretically, the ideal mesh should be uniform in all directions and get convergence for the critical values of the problem. However, it should be fine enough to give accurate results so that further refinement dramatically runs up the cost of computing with negligible improvement in accuracy [3]. To investigate this, a 38^{o} V-die was used in an EHF simulation to form a blank of DP-500 1 mm sheet as shown in Figure 8.

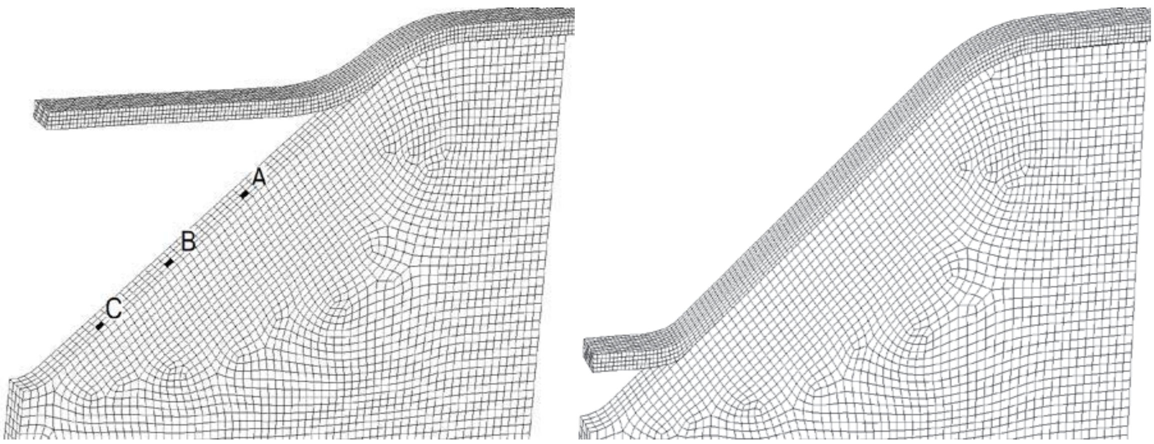

Figure 8: EHF Forming, half model, of a DP-500 blank *meshed with 0.25 mm solid elements* in a V-die *meshed with 0.5 mm solid elements*

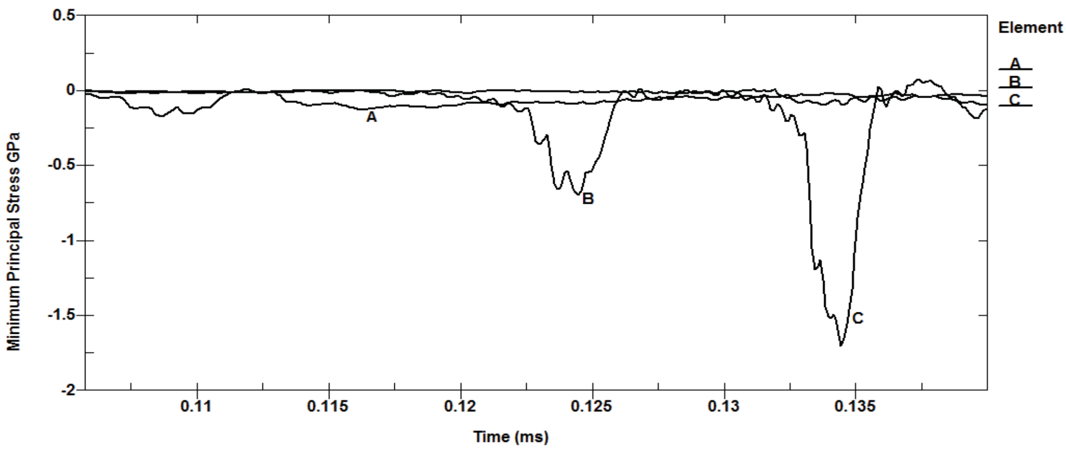

Figure 9: *Stress values in a* V-die during EHF process forming of DP-500 blank

The contact model with the suggested parameters provided stable numerical solution. Only minimized penetration was observed with reference to previously reported results. The max contact stress in the die never exceeded the max physical ceiling introduced in the previous section as it is shown in Figure 9. The maximum stresses were observed at the die corner where fracture of the die is possible.

The standard penalty formulation used in the above simulation with LS Dyna has a considerable influence on the solution stability. Equation (4) shows that the stiffness factor k_i for the master segment is given in terms of bulk modulus K_i, the volume V_i and the face area A_i for the brick element, [9]. This coupling between the element dimensions with the penalty force calculation does not lend itself to solution stability with volume and area continuously changing during impact. Penetration can be completely avoided and contact forces calculations decoupled from the element dimensions by using the contact formulation presented in [2]. Using the soft contact represented by Equation (5), the contact force is localized in a small neighbourhood of mesh elements and it increases to infinity, theoretically, when the distance between surfaces in contact is approaching zero.

$$k_i = \frac{f_{si} K_i A_i^2}{V_i} \tag{4}$$

where f_{si} is a scale factor for the interface stiffness.

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

where F is the absolute value of the contact force, h is the actual distance between the impacting entities, and h_0 is the width of the layer where the contact force is different from zero, and K is a coefficient dependent on the model simulation.

5 Conclusion

The used contact algorithm in impact simulations substantially affects the accuracy of the contact stress. This appears in the stability of the solution observed between different codes. The user has to take great care to assure that the solution reflect the true physics of the problem. Analysis of contact stresses was conducted in blank impact simulations with different contact algorithms to avoid the so called mesh sensitivity. Inter-penetration was minimized along with better control over artificial viscosity and hour glassing to assure the stresses are closest to experimental results.

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