Dynamic Cyclic Bending, Kinetic to Strain Energy Deceleration Systems

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During the period of 2001 to 2003 an investigation has been completed at the University of Pretoria, which explored the application of a cyclic plastic bending deceleration system in a dynamic environment. The system consisted of a user-defined number of inline guide rollers, with a metal element threaded in an over and under arrangement. This metal element, when drawn through the rollers, was subjected to bending and reverse bending as the material moved from roller to roller. This action allowed for kinetic energy to be absorbed by means of the kinetic to strain energy transformation mechanism. In the past, this concept and the performance prediction thereof had been proved reliable under quasi-static conditions however, the performance prediction under dynamic conditions remained uncharted. The study performed at the University of Pretoria, produced a dynamic performance prediction model. The various predictions were compared to scale experimental simulations, as well as dynamic finite element analyses. The comparison between the predictions, (mathematical prediction, experimental model results and the finite element analyses) proved to be extremely accurate and consistent. Full-scale evaluation would be the next remaining step to evaluate the system's suitability for future application in industry.

NOMENCLATURE

a	acceleration. (m/s^2)
D	coefficient in stress strain relation. (Mild steel characteristic)
F	Force applied by deceleration system (N)
FEM	Finite Element Method
FEML	Finite Element Model
FEA	Finite Element Analysis
Gs	Gravitational acceleration. (9.8m/s^2)
M	Mass of skip. (kg)
Msec	Millisecond. (0.001sec)
n	Number of rollers in roller assembly.
p	Exponent in stress strain relation.
R	Roller radius (m)
t	Strip thickness (m)
w	Roller width (m)
σ_{yd}	Dynamic yield stress (MPa)
σ_{yo}	Static yield stress of material. (MPa)
ε'	Strain rate at the point of plastic deformation. (/s)
ε_x	Strain at distance y from neutral plane.
y	Distance from neutral plane. (m)
ρ	Curvature radius of neutral plane. (m)

Introduction

Research has been done in the past, in the field of kinetic energy dissipation, which finds application in the control of large masses moving at high velocities. In many areas of industry a safety system of this nature, capable of absorbing large quantities of energy is in high demand. The basic modes of energy absorption are numerous, such as viscous, coulomb friction, electromagnetic and plastic deformation naming only a few. Of particular interest is the technique making use of plastic deformation to facilitate kinetic energy dissipation. This technique is currently extensively incorporated into the design of automotive and aircraft structures to create a safer environment for the occupants during the impact of a collision.

The method of using dynamic cyclic plastic bending to absorb energy of masses moving at high velocity has had limited application in industry to date. The method of selecting a system of suitable size for the application has been done by means of trial and error.

At the University of Pretoria a prediction model was developed, and automated into mathematical code to deliver an estimation of performance. The output of the developed code has been compared to a series of dynamic scale model experiments, as well as a 'dynamic finite element model'. The comparison between the output of the code, the experiments performed and the finite element simulation, proved to be accurate over a range of probable application scenarios. The following article focuses on the dynamic performance prediction, of the cyclic plastic bending energy absorption system. It also investigates the comparison of the three prediction sources namely the developed code, the experimental tests and the finite element simulation.

Description of the device

The investigated cyclic plastic bending deceleration device operates in the following manner. The technique is based on the 'Strain Energy Linear Ductile Arrestor' system, 'SELDA'[1]. The method of energy absorption by means of 'kinetic to strain' energy conversion to facilitate the process, was pioneered by Jackson in 1965 [2].

The means facilitating the cyclic bending condition is created by the drawing through of a steel strip or wire, which is threaded between a set of inline rollers (Figure 1)

When the strip is bent and unbent around the rollers the kinetic energy is absorbed and transformed into low-level heat. The force required to draw the steel element through the rollers is the force that is utilised as the deceleration force. The maximum amount of energy that can possibly be absorbed by this means would be a constant force at the maximum specified level for the

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duration of the deceleration stroke.

Rosslee developed two analytical equations predicting the performance of cyclic bending energy absorbers, in his masters' thesis [3]. These models are meant for quasi-static application. Rosslee experimentally verified the equations for a variety of geometries. The predictions and the experimentally generated data have a very good correlation. In Rosslee's paper the dynamic application of this system as an energy absorber was suggested, provided extensive testing and investigation was done to verify the proposed prediction model.

A metal strip being pulled over a roller was considered by Alexander [4], and an analytical model was developed to predict the difference in magnitude of tip forces, required to pull the strip over the roller while conforming to the arc. According to Alexander the device delivered nearly constant resistive force and this was increased with each unit of width and thickness. These findings were also under quasi-static conditions.

Johnson and Mamalis [5] developed an approximate theory for the force required to bend a metallic strip over a roller while forcing it to conform to the roller radius. It was suggested that this method of energy dissipation should be used to absorb the kinetic energy of a moving conveyance in order to protect the occupants. The magnitude of the force was also a function of width and thickness of the strip that was bent and unbent over the radius. This mechanism was applied to a moving mass of 23 tonnes, but the equation does not make use of strain rate sensitivity.

Important performance parameters

In the equations that were used as prediction mechanisms the following factors played an important role:

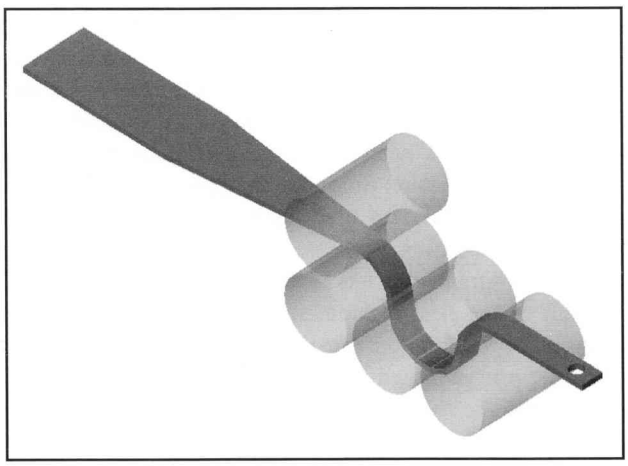

Figure 1: Schematic of cyclic bending device

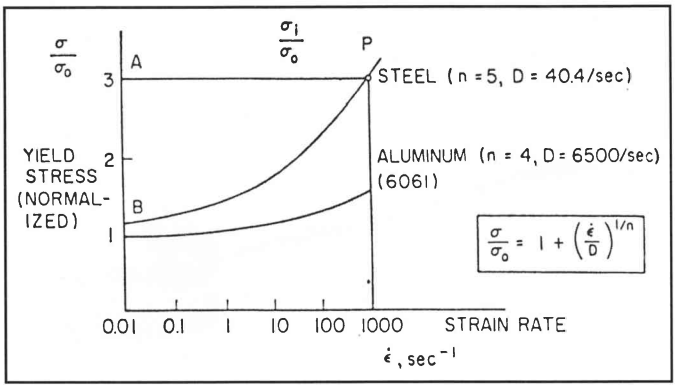

Figure 2: Yield stress vs. strain rate. Perrone

The *Baushinger effect*, which is commonly ignored in plastic theory [6]. This is the case in the development of the analytical equations that have been sourced and used as performance predictors [3]. The yield stress is thus accepted as being the same in both directions of bending.

H.D. Nine found during his experiments [7], that the total energy absorbed by a strip which was pulled through a series of rollers, was 'far' greater than the amount of energy that the strip could store elastically. This would then support the assumption that *elasticity effects* could be neglected in the cyclic plastic bending energy absorber, given that the ratio of roller radius to strip thickness is such that almost the whole cross section is plastically deformed [3].

Material strain hardening has a small influence on the performance of the system at hand for two reasons. The first reason would be that strain hardening decreases with the increase of strain rate. The second reason would be that mild steel, as used in the cyclic plastic bending energy absorber, has moderate strain hardening characteristics, and the influence of this on the results would be negligible [8]. For these reasons, the four authors who developed the analytical prediction equations, which have been used, ignored the strain hardening effects.

It has been noted that materials that have a tendency to be strain rate sensitive should be modelled accordingly, under high deformation rate conditions. The 'dynamic yield stress' of a material has been known to increase by 300% from the 'static yield stress', at high rates of deformation (Figure 2). Perrone's paper [10] referred to the well-known Cowper Symonds model for strain rate sensitive materials [9]. This model was developed for the problem of impact loading of cantilever beams, which exhibit either strain hardening or strain rate sensitivity. This model is adopted as the basis of applying the strain rate effects in the cyclic plastic bending retardation system.

Strain Rate Sensitivity Compensation.

The strain rate sensitivity is appended to the prediction equations in the form of the Cowper Symonds formulation [9]. This is combined with a deduction made from the research of H.D. Nine[7], concerning the period in which the straining of the material takes place. When this distance is known and the velocity of the moving material is also known, the strain rate can be calculated and thus the dynamic yield stress at the point of strain deformation.

The Cowper Symonds formulation is the following:

$$\frac{\sigma_{yd}}{\sigma_{yo}} = 1 + \left(\frac{\varepsilon'}{D}\right)^{\frac{1}{p}} \qquad (1)$$

σ_{yd} = Dynamic yield stress approximation.
σ_{yo} = Static yield stress of material.
ε' = Strain rate at the point of plastic deformation.
D = Coefficient in stress strain relation. (Mild steel characteristic)
p = Exponent in stress strain relation.

The Cowper Symonds equation is then appended with the prediction of strain rate relevant to the specific mode of operation at hand, yielding Equation 2. The following equation is used to predict the dynamic yield stress of the strip element, in the

areas of deformation, as it travels between the rollers.

$$\sigma_{yd} = \left(1+\left(\frac{\left(\frac{v}{12R+6t}\right)}{D}\right)^{\frac{1}{p}}\right)\times\sigma_{yo} \qquad (2)$$

Models used as predicting mechanisms

The theoretical approach used to solve the presented prediction problem, is not one of developing new models. It is based on the manipulation of existing models and verifying whether the dynamic aspect applied to them delivers a prediction that coincides with the experimental readings gathered, as well as the FEA results. When a model is found to deliver comparable answers, the range of accuracy is to be evaluated.

In this investigation, four analytical equations have been gathered, based on work from various authors. The quasi-static prediction models have then been automated into mathematical code (MATLAB), and supplemented with the dynamic strain rate effect equations applicable to high strain rate situations. Predictions were then made of specific roller strip systems.

The first analytical model used as a foundation for the dynamic prediction model presented in this section has been developed by Rosslee [3]. The model was based on bending moment theory. When appended with the dynamic yield stress prediction (Equation 2), the following model is presented:

$$F = 2\left(\left(1+\left(\frac{\left(\frac{v}{12R+6t}\right)}{D}\right)^{\frac{1}{p}}\right)\times\sigma_{yo}\right)w\left[(2R+t)-\sqrt{(2R+t)^2-t^2}\right] \qquad (3)$$

The second quasi-static analytical model used as a prediction foundation, was also developed by Rosslee [3]. Deformation energy theory was used to approximate the solution for the cyclic plastic bending problem. When appended with the dynamic yield stress prediction (Equation 2), the following model is presented:

$$F = \frac{2\cdot w\cdot t\cdot\left[\left(1+\left(\frac{\left(\frac{v}{12R+6t}\right)}{D}\right)^{\frac{1}{p}}\right)\times\sigma_{yo}\right]\cdot\left[\ln\left(\frac{R+t}{R+\frac{t}{2}}\right)-\ln\left(\frac{R+t}{R+\frac{t}{2}}\right)\right]}{2\cdot\sqrt{3}-\left[\ln\left(\frac{R+t}{R+\frac{t}{2}}\right)+\ln\left(\frac{R+t}{R+\frac{t}{2}}\right)\right]} \qquad (4)$$

The third dynamic prediction model was based on the analytical model developed by W.Johnson and A.G.Mamalis [5]. This was based on plastic work principles. Once appended with the dynamic yield stress parameter (Equation 2), the following model is presented:

$$F = \frac{w\times t^2\times\left(1+\left(\frac{\left(\frac{v}{12R+6t}\right)}{D}\right)^{\frac{1}{p}}\right)\times\sigma_{yo}}{2\times R\times t} \qquad (5)$$

The final model used to predict the resistance force generated by a metallic strip being drawn between inline rollers was developed by J.M. Alexander [4], and then published by W. Johnson and S.R. Reid [5]. This prediction was also appended with the dynamic parameter to yield the following:

$$\Delta T = \frac{4\times\left(\left(1+\left(\frac{\left(\frac{v}{12R+6t}\right)}{D}\right)^{\frac{1}{p}}\right)\times\sigma_{yo}\right)\times w}{\sqrt{3}}\times\left(t-2\times R\sqrt{1+\frac{t}{R}}+2R\right) \qquad (6)$$

The four models presented have been programmed into MATLAB code and allowed to deliver dynamic performance predictions of both, experimental and probable scenarios.

Programming strategy

A common characteristic is identifiable in most approximation programs. This characteristic is the need for a realistic estimation of the workable solution by the user. Usually the hindrance is that the user has no idea of what the expected solution's parameters are going to be, thus has no starting point from which the simulation program can work. For this reason the author decided to use two versions of a prediction program. The first version is 'spreadsheet' based, and has all the allowable ratios of geometry inserted in envelope form (*limit<X<limit*), for the user to simply check whether the chosen values are of correct proportion. This version produces an appropriate force, based on a simplistic calculation, ignoring the mass's change in speed during the retardation process. The second version of the simulation program, which is MATLAB code based, is then fed the result of the first version, which would be a reasonable estimation of the final outcome's proportion. The second version of the program then delivers an accurate approximation, based on realistic scenario estimations, allowing for the change in speed during the deceleration process. The second prediction version of the program also accounts for the taper section of the strip element that has been adopted for smooth application of the deceleration force (Figure 1).

Experimental procedure

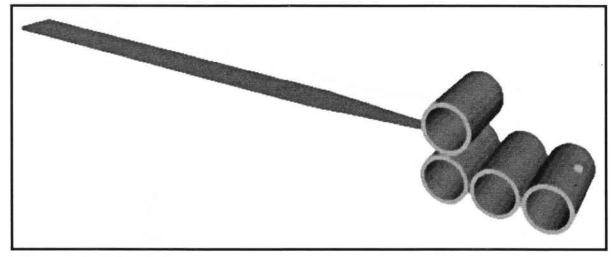

Figure 3: Mass impacting deceleration strip

The experimental verification has been carried out on a scale model of a mineshaft and conveyance, which was constructed for past research at the University of Pretoria. This tenth scale model has been used to simulate dynamic, scaled down conditions, similar to real-life applications. The existing model has been adapted to accommodate the dynamic bending deceleration systems. A piezoelectric accelerometer was attached to the model skip, and acceleration readings were captured during the experimental procedures. This recorded data has then been compared to the condition estimation of the 'design program', as well as the 'FEA', based on the inputs of a similar condition. Various parameters were identified as variables such as strip element thickness, width as well taper profile and conveyance free fall height, thus the velocity of the decelerated mass. The experiments were performed for a range of these variables.

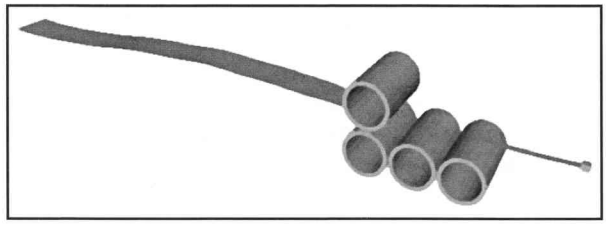

Figure 4: Taper section of strip moves between rollers

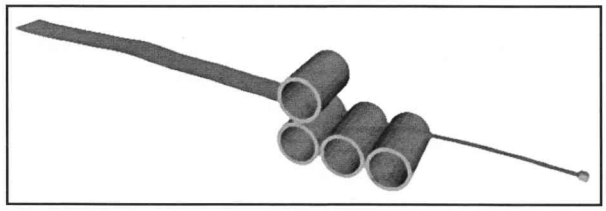

Figure 5: Parallel section of strip moves between rollers

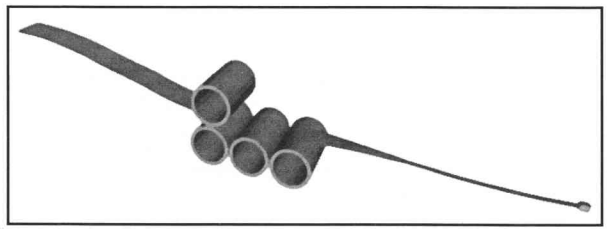

Figure 6: Mass is brought to rest

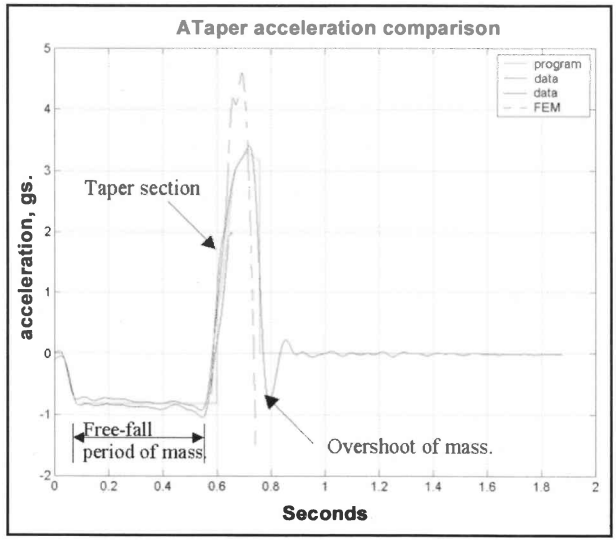

Figure 7: Comparison of accelerations, MATLAB program, FEA simulation and experimental data

Dynamic finite element analysis

The Finite Element Analysis (FEA) is a numerical method for solving problems of engineering and mathematical physics. Typical areas of application are structural analysis, heat transfer, fluid flow, mass transport and electromagnetic potential problems. The finite element analysis package used was MSC Dytran. The pre and post processing of the model was done with MSC Patran. The condition encountered in this FEA case is classified as 'structural impact'. The finite element model was constructed of 'shell' elements for their computational efficiency. The reason for the finite element analysis is to explore the geometrical limitations of the experimental scale model. The FEA results have been compared to the tested readings and the comparison is of such accuracy that the FEA output can be accepted as reliable and accurate. The maximum deviation of 25% between the FEA model and the test setup can be explained as follows: In the FEA only the strip and rollers, see figure 1, were modelled and not the complete system. If only the cyclic bending device is considered, the FEA is much more accurate. It is therefore acceptable to evaluate the application of the MATLAB prediction program against these results.

Having established the reliability of the FEM analysis's output, the next step was to vary the roller radii, strip thickness and taper profile in the FEM for different situations and compare the data readings to the output predictions of the MATLAB design program. This simulation has been used to further verify the accuracy of the design program. An extreme situation involving a 40 tonne skip moving at 100 km/h was also simulated and compared to the output of the MATLAB design program. The only difference between the two sets of results was the initial deceleration profiles. The reason for this is that the FEA takes the strip inertial impact into account and the MATLAB design program does not. The mass of the strips that would be able to absorb 15,5MJ, are 900 kg each, of which there would be 'four', proving to be a substantial impact factor. This impact influence has been explored to determine exactly what limitations can be expected.

The following sequential figures show a visual play-back of the Dytran FEA simulation. The mass being decelerated by the strip is specified as 'extremely' high-density material, allowing for the mass to be small in size and simplifying the model.

Experimental, predicted and simulated performance comparison

Three areas were dealt with concerning comparisons. Firstly the experimental data captured was compared to the FEA models generated, which represented the experimental conditions. Figure 7 shows the comparison of experimental acceleration levels of 'one data set', with the MATLAB prediction and the FEA simulation. This particular FEA comparison is approximately 20% un-conservative, with the MATLAB prediction being on the mark. This tendency has been observed throughout the investigation. The FEA acceleration prediction is un-conservative, with the MATALB prediction being conservative.

Secondly, once the integrity of the FEA was established due to the good correlation with the experimental data, the MATLAB prediction program was further evaluated by means of generating probable scenarios as 'finite element models', and again comparing the resulting predictions. Figure 8 shows a scenario comparison of the acceleration levels, and as previously ob-

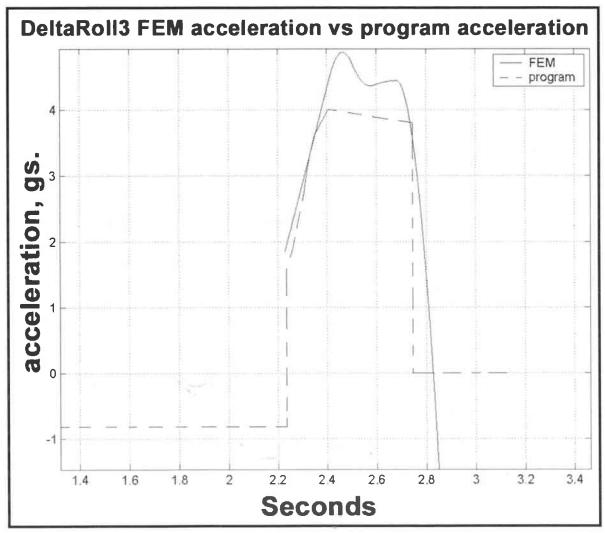

Figure 8: Comparison of accelerations, MATLAB program, FEA simulation

served, the FEA prediction being un-conservative compared to the MATLAB program. Further a 'real life' scenario was simulated and compared to the MATLAB prediction entailing a 40tonne mass moving at 100km/h and being retarded at 20m/s^2, or 2Gs.

The third area of comparison was the influence of inertial load under impact conditions. The investigation was done by comparing the outputs of the MATLAB program predictions versus the FEA with modified material densities. By these means a FEA system could be simulated with no material inertial properties, similar to the assumptions made in the MATLAB program. In Figure 9 the various results can be observed. The density of the material was reduced by the percentage of the full mass. The impact-induced accelerations can be seen to reduce as the density of the strip is decreased to 'zero' strip density on the 'FEM0' curve. This tendency is expected.

Conclusion

This investigation shows that the MATLAB prediction program promises to be a useful tool, that shows scope to be applied in industry in various fields and not only mining or conveyance applications. It proves to be capable of accurately predicting the dynamic performance of the deceleration systems under a wide range of scenarios. The computational time for the MATLAB program to make a prediction is also extremely short compared to the FEM analysis. This is a great advantage. The difference in calculation time is in the order of 100 times faster, and the reliability based on the experiments conducted and the comparisons drawn, is good. An important note is that similarly with all computer packages delivering prediction results, the user must be aware of the limitations of the program, before it is applied to a problem situation. Making the wrong assumptions would lead to dangerous consequences.

Before this system is applied in industry, full-scale tests need to be performed according to the predictions of the MATLAB program. The experimental data of these full-scale experiments then need to be closely analysed to determine the degree of comparison between the prediction and the data captured.

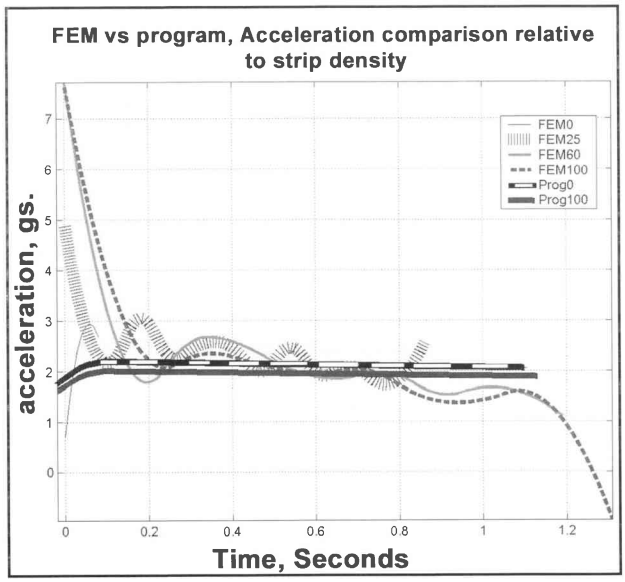

Figure 9: Comparison of acceleration, MATLAB program and FEA simulation for various strip densities

Reference:

1. Seltrust Engineering 'Strain Energy Linear Ductile Arrestor', England. Bellamble Mining, ID9085806, 11 June 1999.

2. Van Zelm 'Patent on an Energy Associated Inc. Absorbing Device'. United States Patent No. 3/211/260, Jackson M.A. 12 October 1965

3. Rosslee F. 'Design and analysis of an energy absorbing mechanism for mine cages.' Coetzee G. Pretorius L., MIng. thesis, Rand Afrikaans University, Johannesburg 1996.

4. Alexander J.M 'Metallic Energy Dissipating Systems' Published by Johnson W. and Reid S.R, Applied Mechanics Review vol. 31 No. 3 1978

5. Johnson W. 'Crash worthiness of vehicles', Mamalis A.G. Mechanical Engineering Publications, London, 1978, pg. 13-125

6. Dieter G.E. 'Mechanical Metallurgy' McGraw-Hill Book Company, London, 1988.

7. Nine H.D. 'Drawbead Forces in Sheet Metal Forming' Mechanics of Sheet Metal Forming General Motors Research Labs, Warren, Michigan. Plenum Press, New York, 1978

8. Jones N. 'Recent Studies on the Dynamic Plastic Behaviour of Structures'. Applied Mechanics, Rev. Vol. 42 no 4 1989.

9. Cowper G.R. 'Strain hardening and strain-rate effects in the impact loading of cantilever beams,' Symonds P.S Division of Applied Mathematics, Technical report 28. Brown University, Providence, USA. September 1957.

10. Perrone N. 'On a Simplified Method for Solving Impulsively Loaded Structures of Rate Sensitive Materials', Journal of Applied Mechanics, vol. 32, 1965.