

# IMPROVING SAFETY PERFORMANCE IN FRONTAL COLLISIONS BY CHANGING THE SHAPE OF STRUCTURAL COMPONENTS

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## ABSTRACT

The design optimisation of vehicle structures for improved crashworthiness is very much a trial and error process. A method of structural optimisation is described which is a more formal and systematic approach to design improvement. The method is implemented into an existing optimisation program and is used to improve the response of structural components within a vehicle by changing their shape. A sensitivity analysis based on the elastic buckling response of a structural component drives shape changes in the optimisation process.

The program is applied to improving the crashworthiness of a box section member. The optimised component shows a substantial increase of the initial collapse load and energy absorbing capacity. Results from dynamic simulations confirm this.

## INTRODUCTION

Techniques for *analysing* the behaviour of vehicle structures in a collision are well established within the car industry, however, techniques for *improving* this behaviour are still very much in an infancy. Outcomes of recent work by Satoh *et al* (1996) and Witteman *et al* (1996) typify the current methods employed for design improvement. The authors presented a set of guidelines governing the wall thickness and section width of structural components for optimal energy absorption. While reducing the trial and error involved in design, such guidelines only provide a qualitative measure of the design changes required.

A more advanced procedure for improving the crashworthiness of vehicle structures was proposed by Hagiwara *et al* (1990). A sensitivity analysis was used to study the change in buckling response of a structural component with variations in wall thickness. While no quantitative measure of improved performance was determined, the authors found that general design studies were possible with the use of a sensitivity analysis based on the buckling load of a structure. Kitagawa *et al* (1992) used a similar concept to improve the dynamic buckling

load of a straight beam, however, the cross-sectional area was varied instead of the wall thickness.

This paper extends the use of a sensitivity analysis in relation to improving the crash safety of a vehicle. Unlike Hagiwara *et al* and Kitagawa *et al*, the sensitivity analysis is used to calculate the effect of shape changes on the buckling response of a structural component. This information is coupled with a structural optimisation program to enable the integrated redesign of components.

The program is used to change the shape of a box section member. For this case, the effects of an increased buckling load on crush characteristics such as the initial collapse load and energy absorbing capacity are investigated as a function of the impact speed. Results are correlated with a theoretical formulation governing the role of the impact speed.

## STRUCTURAL OPTIMISATION - A SYSTEMATIC METHOD FOR DESIGN IMPROVEMENT

### Background

Structural optimisation techniques are used herein to provide a systematic method for improving the crashworthiness of a vehicle structure. Numerous references are available on the topic (for example: Vanderplaats, 1984) and as such, only the key concepts and terms are defined below:

1. *Objective function*: the performance measure of a structure which is desired to be maximised or minimised.
2. *Design variables*: variables which govern the design of a structure such as material thickness and geometry.
3. *Constraint function*: a restriction on the response or design variables of a structure.
4. *Sensitivity*: a quantitative measure on how the response of a structure is affected by changes in design variables.

## Computational Basis

The computer program RESHAPE is an example of a structural optimisation program. This program is centred on the finite element method and utilises sensitivity data to search for the optimum shape of a structure (Tomas *et al.*, 1991). The objective function for this process can be chosen from performance measures such as stress, frequency and mass. Shape changes are described by selected node coordinates on a finite element model. Thus, arbitrary shape variations can be achieved in the optimisation process. Mesh integrity and boundary smoothness is retained through the use of parametric cubic geometry which overlays the finite element model. Additionally, the program enables constraints to be applied as limits on the response of a structure, as direct limits on design variables and as function constraints. The function constraints are used to maintain design requirements such as a symmetric geometry or a prismatic cross-section.

The aforementioned program will be extended to enable the design improvement of vehicle structures. This task will require the definition of a suitable objective function which characterises the crashworthiness of a vehicle structure.

## BUCKLING LOAD AS AN OBJECTIVE FUNCTION

### The Role of Buckling in Crash Safety

In a vehicle, there are Key Structural Components (KSC's) which absorb a significant amount of crash energy and influence the behaviour of a vehicle structure

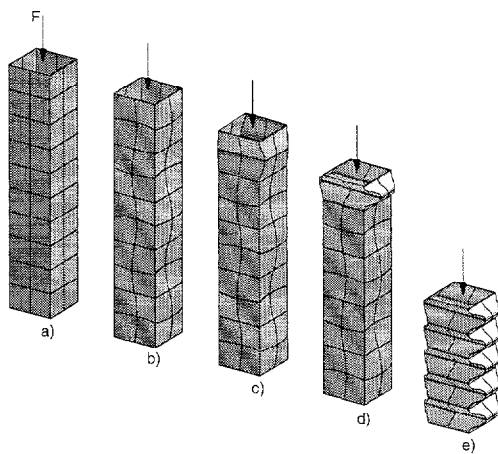


Figure 1. Sequence of axial column crush a) loading, b) buckling, c) edge yielding, d) folding and e) continuation.

during a collision. Improving the performance of these components will therefore improve the crash safety of a vehicle as a whole.

In the initial stages of a collision, predominantly axial loads are transferred to the KSC's. The following transpires as the collision advances (see Figure 1 and Figure 2):

- The compressive stresses in the KSC's begin to increase rapidly as the applied axial load increases.
- A critical load is reached where the walls buckle on the weakest KSC.
- The edges of this KSC carry the increasing axial load as a result of the buckled walls. Edge yielding eventuates. In general, this occurs directly after buckling.
- The load carrying capacity of the KSC drops as the walls fold.
- Continuing (secondary) buckling, edge yielding and folding occurs as the energy from the collision is absorbed.

The onset of buckling triggers the initial collapse of a KSC. However, the actual load  $F_{max}$  at which this occurs depends not only on the elastic buckling load, but on the impact velocity and yield load as well. This dependence can be quantified by the following equations (see Appendix for derivation):

$$\text{When } F_b < F_y: F_{max} = \begin{cases} f(F_b, V), & \text{for } V \leq V_c \\ F_y, & \text{for } V > V_c \end{cases} \quad (1.)$$

$$\text{When } F_b \geq F_y: F_{max} = F_y, \text{ for all } V \quad (2.)$$

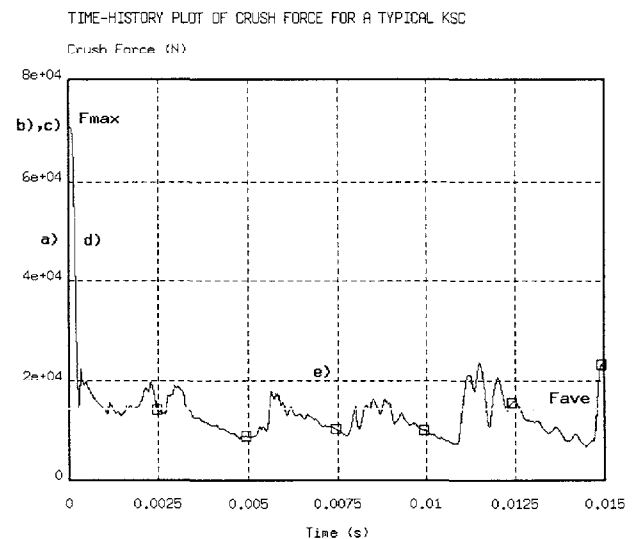


Figure 2. Typical time-history plot for the crush of a KSC.

where  $V$  is the impact velocity,  $F_b$  is the buckling load,  $F_y$  is the yield load and  $V_c$  is the critical impact velocity.

With reference to equation (1.), in a high speed impact ( $V > V_c$ ) the initial collapse load is characterised by the yield load because stresses generated from the impact are sufficient to cause plastic deformation. Buckling, therefore, occurs after the onset of yielding and does not influence the initial collapse load. In the case of a low velocity impact ( $V \leq V_c$ ), buckling transpires before stresses induced from the collision have time to reach the elastic limit. Hence, the elastic buckling load influences the initial collapse load in this instance.

Based on the preceding discussion, it is postulated that design changes based on improving the elastic buckling load of a KSC will also improve its initial collapse load in a low speed impact. This is an important aspect of crashworthiness because an elevated initial collapse load reduces the possibility of any permanent structural damage in low speed collisions.

### Secondary Influences of Design Changes Based on the Elastic Buckling Load

Apart from the initial collapse load, another important aspect of crashworthiness is the energy absorbing capacity of a KSC. This reflects on the potential of vehicle structure to absorb crash energy. Therefore, improving the energy absorbing capacity of a KSC will enhance the safety performance of a vehicle as a whole.

The energy absorbing capacity of a KSC is primarily characterised by the average force  $F_{ave}$  at which secondary buckling, edge yielding and folding occurs (see Figure 1 and Figure 2). It is predicted that design changes which improve the elastic buckling load of a KSC will also improve the average force required to crush the KSC. This is because the walls of an optimised KSC have a high resistance to bending, even after initial collapse. Hence:

1. a higher compressive load will be required to cause secondary buckling of the walls,
2. bending deformations will be reduced and therefore edge yielding will take place at a higher load and
3. folding of the walls will require more work due to their increased bending stiffness.

### Calculating the Elastic Buckling Load and Related Sensitivity Data

To implement an elastic buckling objective in the optimisation program requires calculation of the buckling load and related sensitivity data. The load at which

elastic buckling occurs is calculated from the following characteristic eigenvalue equation by the finite element method (Cook *et al*, 1989):

$$(\mathbf{K} + \lambda \mathbf{K}_\sigma) \mathbf{u} = 0 \quad (3.)$$

where  $\mathbf{K}$  is the stiffness matrix,  $\mathbf{K}_\sigma$  is the initial stress matrix,  $\lambda$  is the eigenvalue and  $\mathbf{u}$  is the eigenvector. The eigenvalue  $\lambda$  is proportional to the buckling load.

By differentiating equation (3.) with respect to the design variables  $\mathbf{x}$ , an expression relating the variation of buckling load with respect to changes in the shape of a structure is obtained:

$$\frac{d\lambda}{d\mathbf{x}} = \frac{\mathbf{u}^T \left( \frac{d\mathbf{K}}{d\mathbf{x}} + \lambda \frac{d\mathbf{K}_\sigma}{d\mathbf{x}} \right) \mathbf{u}}{\mathbf{u}^T \mathbf{K}_\sigma \mathbf{u}} \quad (4.)$$

This equation is used for calculation of the sensitivity data.

### The Order of Elastic Buckling

An eigenvalue analysis based on equation (3.) yields many buckling loads and related mode shapes. Only the first buckling load is considered in static applications as this is always the critical value. However, in dynamic situations such as the collapse of a KSC, higher order buckling modes have been found to influence the collapse behaviour (Kitagawa *et al*, 1992).

While the capability exists to optimise higher order modes, the approach adopted in this paper is to use only the first buckling load as the objective function in the optimisation program. This is for two reasons. Firstly, an improvement in the buckling response for the first mode will inevitably improve the response in higher order modes as well. Secondly, the theoretical lower bound for the initial collapse load is the first buckling load (in cases where  $F_b < F_y$ ). Therefore, increasing this lower bound will ensure that there is no possibility of failure at a lower load.

### Limitations of an Elastic Buckling Analysis

The foregoing discussion was primarily related to the axial collapse of a straight KSC with a uniform cross-section. While the design optimisation of such members will lead to the improved safety performance of a general vehicle structure, in some cases it may be desirable to enhance the collapse behaviour of KSC's with a varying cross-section or curved profile. Optimisation based on the elastic buckling load must be approached with caution

in these cases. This is because the solution of equation (3.) requires the existence of a bifurcation point. In addition, equation (3.) is derived on the basis that any bending deformations prior to buckling are negligible. Hence, an elastic buckling analysis may not properly characterise the failure of a KSC's with an arbitrary geometry.

### IMPROVING THE CRUSH RESPONSE OF A BOX SECTION MEMBER

The following example is presented to illustrate the capabilities of the shape optimisation program with the elastic buckling load as an objective. It also serves to substantiate the previous inferences made on the relation between the elastic buckling load and dynamic response of a KSC.

#### Optimising the Design

Details of the box section member are shown in Figure 3. The first step in improving the crush response of the member was to create a finite element model for the optimisation program. This was meshed with approximately 3000 quadrilateral elements. The base of the member was rigidly fixed and a compressive axial load was applied to the top of the tube in order to simulate the force from the impact. Design variables for the optimisation process were based on the cross-sectional shape of the member. Constraints were imposed to keep the cross-section prismatic and of a constant area.

Figure 4 shows the cross-section of the optimised member. The elastic buckling load for this design is 310kN as compared to 27kN for the original design. The yield load in both cases is 67kN (assuming a uniform stress distribution and ignoring strain hardening effects).

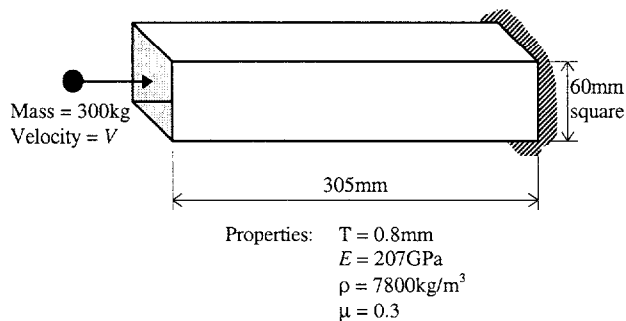


Figure 3. Details of axial crush example.

### Analysis of Collapse Behaviour

The original and optimised models of the box section member were transferred to an explicit finite element program to enable a comparison of the crush characteristics. An elastic-plastic material model was used which had a yield stress of 350MPa, hardening modulus of 450MPa and hardening exponent of 0.5. Results from the analysis are presented in the ensuing sections.

#### Improvements in the Initial Collapse Load

Figure 5 shows a plot of the initial collapse load versus impact velocity for both the original and optimised designs. A number of observations can be made from this:

1. The initial collapse load of the original design varies linearly with the impact velocity below approximately 8km/h. The initial collapse load approaches the elastic buckling load of 27kN as the impact velocity approaches zero.
2. Above an impact velocity of 8km/h, the initial collapse load of the original design is equal to its yield load of 67kN.
3. The initial collapse load of the optimised design is equal to its yield load of 67kN, irrespective of the impact velocity.
4. The optimised design has a higher initial collapse load than the original design for impact velocities below 8km/h. Hence, design changes based on the elastic buckling load are effective in improving the initial collapse load of the box section member.

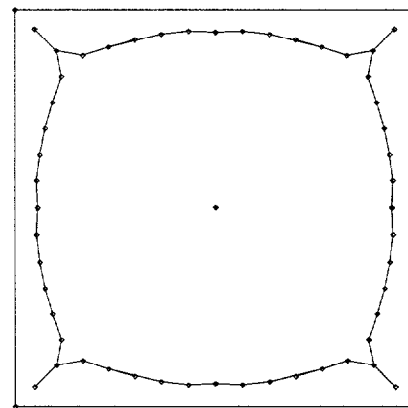


Figure 4. Optimised cross-section in comparison to original square cross-section.

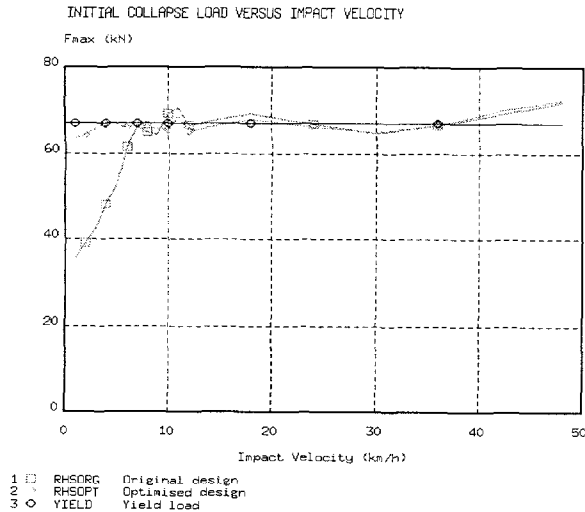


Figure 5. Initial collapse load as a function of impact velocity for the original and optimised designs.

Based on the first two observations, a relation describing the initial collapse load of the original design as a function of impact velocity can be derived (based on equation (1.)):

$$F_{max} = \begin{cases} (27 + 5V)\text{kN}, & \text{for } V \leq 8\text{km/h} \\ 67\text{kN}, & \text{for } V > 8\text{km/h} \end{cases} \quad (5.)$$

Similarly, with reference to the third observation, the initial collapse load of the optimised design can be written as:

$$F_{max} = 67\text{kN}, \text{ for all } V \quad (6.)$$

using equation (2.).

### Improvements in the Energy Absorbing Capacity

A time-history plot of the crush force for an impact velocity of 48km/h is shown in Figure 6. The average crush force for the optimised design is 91% higher than for the original design. Hence, secondary effects of design changes based on the elastic buckling load are immediately obvious. The deformed shapes of the original and optimised designs are depicted in Figure 7 and Figure 8 respectively. The higher frequency of folding and reduced bending deformations substantiates the increased bending stiffness of the walls on the optimised KSC.

Table 1 lists the average crush force (and thereby a measure of the energy absorbing capacity) for additional impact velocities. In all cases, the optimised design is significantly better.

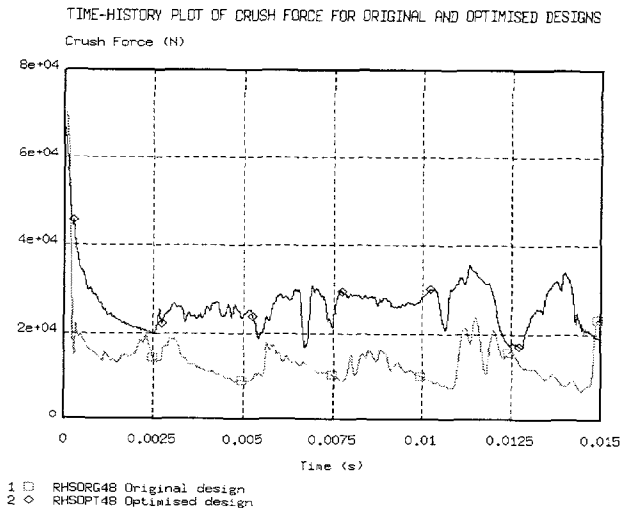


Figure 6. Time-history plot of crush force for original and optimised designs at an impact velocity of 48km/h.

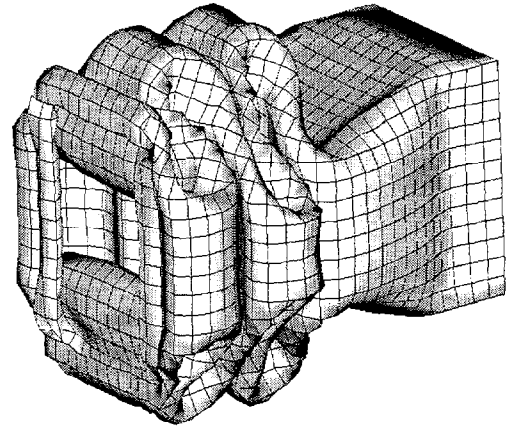


Figure 7. Crushed shape of original design due to impact at 48km/h.

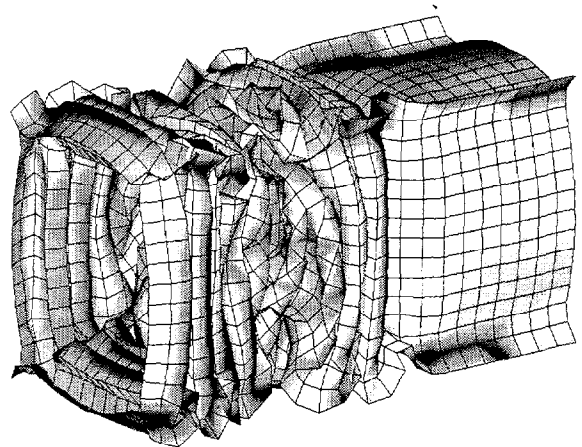


Figure 8. Crushed shape of optimised design due to impact at 48km/h.

Table 1.

Average Crush Force for Original and Optimised Members at Different Impact Speeds

Impact Speed $V$ (km/h)	Original $F_{ave}$ (kN)	Optimised $F_{ave}$ (kN)
12.0	15.5	24.1
18.0	14.4	22.8
24.0	14.0	22.8
32.0	13.0	23.6
36.0	13.3	25.4
42.0	13.8	26.4
48.0	13.9	26.6

## CONCLUSION

A method of structural optimisation was developed for improving the elastic buckling load of a structural component by changing its shape. It was predicted that design changes based on the elastic buckling load would increase the initial collapse load and energy absorbing capacity of structural components in a collision. These postulates were confirmed through the optimisation of a box section member. The elastic buckling load of this component was increased from 27kN to 310kN by changing the shape of its cross-section while keeping the area constant. This resulted in a 91% increase in the energy absorbing capacity of the member for an impact at 48km/h. A significant increase was also seen in the initial collapse load for low impact velocities. Given these results, the method of structural optimisation was shown to be an effective design tool for improving the crashworthiness of structural components.

## FUTURE WORK

Numerous other applications exist for the shape optimisation program in relation to improving the crashworthiness of a KSC. For example, the capability exists to optimise KSC's with a varying cross-sectional area or curved profile. Other constraints can also be implemented to achieve a design which is more manufacturable. Furthermore, the optimisation problem can be restated as "minimise mass while keeping the buckling load constant" in an effort to produce lightweight components with equivalent crush characteristics. Hence, future work will be directed towards characterising a general class of problem where design changes based on the elastic buckling load will yield improvements in crashworthiness.

## ACKNOWLEDGMENTS

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## REFERENCES

- Cook, R., Malkus, D. and Plesha, M., 1989, *Concepts and Applications of Finite Element Analysis*, Wiley, New York.
- Hagiwara, I., Satoh, Y. and Tsuda, M., 1990, "Study of an analytical technique and system for conducting vehicle crash simulations", *International Journal of Vehicle Design*, volume 11, number 6, pp 564-577.
- Kitagawa, Y., Hagiwara, I. and Torigaki, T., 1992, "Development of a sensitivity analysis method for dynamic nonlinear problems", *JSME International Journal - Series I*, volume 35, number 4, pp 523-528.
- Satoh, T. and Takada, K., 1996, "Collapse and energy absorption of thin-walled frame with polygonal section", *Proceedings, 15th International Conference on the Enhanced Safety of Vehicles*, Melbourne, Australia, May 13-16, 1996, volume 1, pp 518-524.
- Tomas, J., Trinder, S. and Wille, H., 1991, "Changing the shape of vehicle components for improved performance", *Proceedings, Seminar on Numerical Methods*, University of Politecn., Torino, 1991, pp 113-139.
- Vanderplaats, G., 1984, *Numerical Optimization Techniques for Engineering Design: with Applications*, McGraw-Hill, New York.
- Witteman, W. and Kriens, R., 1996, "A cable-supported frontal car structure for offset crash situations", *Proceedings, 15th International Conference on the Enhanced Safety of Vehicles*, Melbourne, Australia, May 13-16, 1996, volume 1, pp 558-566.
- RADIOSS CRASH Documentation: Version 3.1*, Mecalog, France, 1996.
- RESHAPE User's Manual*, Advea Engineering Pty Ltd, Australia, 1998.

## APPENDIX

### An Equation for the Initial Collapse Load

An equation for the initial collapse load of a KSC can be developed by considering the stress induced in a KSC as a function of impact velocity. This problem is categorised as a wave propagation problem because the initial collapse occurs at a time shortly after impact when the stress wave effects are dominant.

Figure 9 shows the initiation of a one dimensional stress wave in a KSC. After time  $\Delta t$ , the stress wave has travelled:

$$\Delta x = c\Delta t \quad (\text{A-1.})$$

where  $c$  is the acoustic wave speed. Also after time  $\Delta t$ , the end of the KSC will have moved a distance:

$$\Delta l = V\Delta t \quad (\text{A-2.})$$

where  $V$  is the impact velocity (and is assumed constant).

Assuming a one-dimensional stress distribution and ignoring strain hardening and strain rate sensitivity effects, Hooke's law can be written for this problem as:

$$\begin{aligned} \sigma &= E\varepsilon \\ &= E \frac{\Delta l}{\Delta x} \end{aligned} \quad (\text{A-3.})$$

where  $\sigma$  is stress,  $E$  is Young's modulus and  $\varepsilon$  is strain. Substituting equations (A-1.) and (A-2.) into equation (A-3.) and taking the limit as  $\Delta t$  and  $\Delta x$  approach zero yields:

$$\sigma = \frac{EV}{c} \quad (\text{A-4.})$$

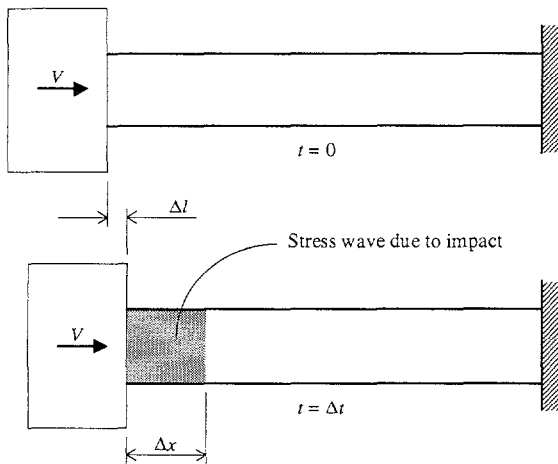


Figure 9. Stress wave propagation in a KSC due to axial impact .

Hence, the stress induced in a KSC is proportional to the impact velocity. This equation can be used to characterise the initial collapse load of a KSC depending on its buckling load.

**Case 1 -  $F_b < F_y$ :** When the buckling load of a KSC is lower than its yield load, the initial collapse load will depend on the impact speed. A high impact velocity will generate a stress wave which causes instant plastic deformation, irrespective of the buckling load. However, a low speed impact will permit buckling prior to yielding. Using equation (A-4.), the critical impact velocity below which buckling will occur is defined as:

$$V_c = \frac{c\sigma_y}{E} \quad (\text{A-5.})$$

This equation forms an upper bound to the critical impact speed. Three dimensional effects, reflection of the stress wave from boundaries and the finite time period required for a structure to buckle all affect the accuracy of equation (A-5.). A correction parameter  $\alpha$  can be introduced to account for these factors:

$$V_c = \frac{\alpha c\sigma_y}{E} \quad (\text{A-6.})$$

Hence, the initial collapse load can be written as:

$$F_{max} = \begin{cases} f(F_b, V), & \text{for } V \leq V_c \\ F_y, & \text{for } V > V_c \end{cases} \quad (\text{A-7.})$$

Note in equation (A-7.) the general function  $f(F_b, V)$  describing the initial collapse load. This is because the initial collapse load is not only a function of the buckling load below the critical impact velocity, but of the impact speed as well. The derivation of this relation is not in the scope of this paper and is a subject of future research.

**Case 2 -  $F_b \geq F_y$ :** In the case where the buckling load  $F_b$  of a KSC is greater than its yield load  $F_y$ , the stress wave generated from the impact will always reach the elastic limit before buckling occurs. This is true even for low impact speeds (assuming a constant impact velocity), because the stress wave will reflect from boundaries and compound. The yield stress will be eventually reached. Hence, the initial collapse load is defined as:

$$F_{max} = F_y, \text{ for all } V \quad (\text{A-8.})$$