Crush Can Behaviour as an Energy Absorber in a Frontal Impact

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Abstract. The work presented is devoted to the investigation of a state-of-the-art technological solution for the design of a crush-can characterized by optimal energy absorbing properties. The work is focused on the theoretical background of the square tubes, circular tubes and inverbuck tube performance under impact with the purpose of design of a novel optimized structure. The main system under consideration is based on the patent US 2008/0185851 A1 and includes a base flange with elongated crush boxes and back straps for stabilization of the crush boxes with the purpose of improvement of the energy-absorbing functionality. The modelling of this system is carried out applying both a theoretical approach and finite element analysis concentrating on the energy absorbing abilities of the crumple zones. The optimization process is validated under dynamic and quasi-static loading conditions whilst considering various modes of deformation and stress distribution along the tubular components. Energy absorbing behaviour of the crush-cans is studied concentrating on their geometrical properties and their diamond or concertina modes of deformation. Moreover, structures made of different materials, steel, aluminium and polymer composites are considered for the material effect analysis and optimization through their combination. Optimization of the crush-can behaviour is done within the limits of the frontal impact scenario with the purpose of improvement of the structural performance in the Euro NCAP tests.

1. Introduction
The automotive industry gives the priority to crashworthiness and occupant safety during the design and manufacture of vehicles. Occupant safety is primarily characterised by vehicle deceleration pulses and deformation of the occupant compartment. Introduction of crumple zones in a vehicle is a trusted engineering solution to this problem. Thin-walled structures also known as crush cans are connected to the vehicle bumpers which absorb sufficient energy to ensure minimum damage to the passenger and critical vehicle parts during frontal impact, thereby contributing to the automobile body crash safety. The majority of research in this area is devoted to the investigation of thin-walled tubular structures with varieties of cross-sections and their behaviour [1-6].

The key concepts have been developed in the area of energy absorbers for crash safety of automobiles, as well as a number of inventions which have been made along the years. A lot of these patents have found their application in the manufacturing process. For example, Evans et al. [13] invented an energy absorber that includes a base flange and elongated crush boxes forming a continuous face surface on the energy absorber. Back straps are integrated across the back of the crush boxes to improve the energy absorbing functionality by keeping a check on unacceptable spreading of side walls of the crush boxes during an impact.

Common shapes of collapsible energy absorbers include circular tubes, square tubes, frusta, struts, honeycombs and sandwich plates [1]. Quite often the modes of deformation of these structures are studied by axial crush, lateral indentation, lateral flattening, inversion and splitting. In [7] the authors represent a study on energy absorption values of thin-walled tubes with different geometric dimensions using finite element simulation. In this case a nonlinear finite-element model is developed and analysed using LS-DYNA. A comparative study, in terms of the energy absorption properties, is performed for square, circular and elliptic tubes of steel and aluminium. The analysis concluded that the elliptic tubes absorb more energy during collisions. It was also demonstrated that the energy absorption can be increased with the width increase. Energy absorption per unit weight of steel and
aluminium is also calculated demonstrating that steel tubes have a better ability to absorb comparing to aluminium. Different material and geometry combinations thus were found to show interesting variations in performances. Another work concentrating on the design optimisation [9] introduces tapered thin-walled tubes with axisymmetric indentations. The present work attempts to go along the lines of the research mentioned in order to design a novel optimised structure for a crush-can.

Due to the superior behaviour a number of works are devoted to circular tube structures [2-6]. The buckling of a circular cylindrical shell under axial loading is a classical problem in solid mechanics. Circular tubes prove to have one of the best energy absorbing abilities under axial compression, which explains their high frequency of occurrence in automotive crash safety applications. Plastic energy can be dissipated in thin metallic (or composite) tubes in several modes of deformation, viz. inversion, splitting, lateral indentation, lateral flattening and axial crushing. Thin circular tubes thus represent the most widespread shape of collapsible impact energy absorbers, owing to their high frequency of occurrence as structural elements.

An extensive review is done on a number of related experimental investigations of crush structures under quasi-static and dynamic conditions [10]. Discussion of some innovative theoretical and analytical models can be found in the reviews similar to [1]. As a rule finite element analysis is performed using ABAQUS and LS-DYNA [14] to study certain energy absorption characteristics and crashing parameters [8].

To make a viable analysis on crush-can behaviour it is important to set all the required load and boundary conditions. In a frontal crash, the impact load is experienced by the bumper assembly under different angles based on the barrier offset, and the varieties of oblique load. Reyes et al. [10] perform quasi-static experiments and numerical simulations to study the behaviour of aluminium columns in alloy AA6060 subjected to oblique loading. The force was applied under three different angles to the centreline of the column thereby being the oblique load [11,12]. In this case LS-DYNA was used to validate the numerical model and describe the force-displacement curve. It was demonstrated that the deformation modes are dependant on both load angle and thickness of the structure. The energy absorption was proved to drop with an increase of the loading angle after a drastic drop under the load angle of 5°.

The main objective of the present work is to investigate the state-of-the-art technological solution for the design of a crush-can characterised by optimal energy absorbing properties and to propose a novel design of a circular crush-can optimising its shape and behaviour. The main system under consideration is based on the patent US 2008/0185851 A1 [13] and includes a base flange with elongated crush boxes and back straps for stabilization of the crush boxes with a purpose of improvement of the energy-absorbing functionality. The modelling of this system is carried out applying both theoretical approach and finite element analysis concentrating on the energy absorbing abilities of the circular crush-cans.

2. Mathematical formulation

The crush-cans geometry can be represented as a thin-walled tube with a circular or square cross-section. In our case, a cylindrical tube of diameter \( D \), length \( L \) and thickness \( t \) is considered, as shown in Figure 1. The longitudinal axis of the geometry is set along the \( z \) axis. Going along the lines of the pre-existing patent [13], one end of the crush-can is clamped and the other end is fixed to the frontal bumper cover. This makes the non-clamped end free to respond to the impact but constrained in terms of rotation.

Alexander in his work [2] assumes a simple mode of collapse of the tube and determines the work required to achieve this mode. This theoretical analysis was presented for an axisymmetric crush of a thin-walled cylindrical shell subjected to a static axial loading. The final solution for the mean crush load for this mode was derived empirically:

\[
P_{\text{av}} = K \sigma_f t (Dt)^{1/2}
\]  

(1)
where \( P_{av} \) is the mean crushing load, \( \sigma_y \) is the yield stress and \( K \) is a mathematically and experimentally derived constant and is also a function of the material’s Poisson’s ratio \( \nu \) (\( K = 6.08 \) for \( \nu = 0.25 \)).

Figure 1. Geometry of a circular crush-can

The study for an energy absorber is incomplete without a consideration of the energy analysis. In a front vehicle crash, to be specific, there are two types of impact: elastic and plastic; necessarily a combination in the actual case. The elastic energy (\( E_{\text{Plastic}} \)) primarily includes the energy absorbed by the crush cans while a small fraction of it is dissipated as heat to the surrounding (the latter can be neglected for simulation purposes). Hence,

\[
E_{\text{plastic}} = E_{\text{Deformation}} = E_a
\]

(2)

where the term \( E_a \) is introduced as the energy absorbed by the deforming structures. This energy is a consequence of elastic-plastic deformation of the structures.

With this basic information, the absorption characteristics are to be assessed and for this purpose we lay down some crashing parameters that have been accepted as standard in similar studies. The parameters under consideration are average failure load/mean crush load (\( P_{av} \)), Specific Energy Absorption (SEA) and Crush Force Efficiency (CFE). The mean crush load is used as the response parameter for energy absorption capability and can be defined as:

\[
P_{av} = \frac{E_a}{\delta}
\]

(3)

where \( E_a \) is the energy absorbed during a collapse and \( \delta \) represents displacement.

The SEA characterises the ability of the structure to absorb more energy with the least possible weight. It is therefore the energy absorbed per unit mass, and can be represented as:

\[
SEA = \frac{E_a}{m}
\]

(4)

where \( m \) is the mass of the individual crush can/thin-walled structure.

CFE which is defined as the ratio of the mean crush load to the peak crush load (\( P_{\text{max}} \)) with the most desirable value being unity, can be written as

\[
CFE = \frac{P_{av}}{P_{\text{max}}}
\]

(5)

3. Novel design discussion

To study the behaviour of a crush-can in a front impact, under axial impact loading an energy absorption comparison is done for circular tubes with different geometric dimensions. For investigation of the effect of geometry and material properties, parameters presented in [7], are chosen for the thin-walled tube under consideration. Load conditions are the 50% frontal offset impact at 55 km/h [15]. The total crush mass of the vehicle (curb mass set at 1350 kg) is calculated according to Table 1.
With regard to the information presented for various front crash impact tests, the impact is considered to be lasting for 100 milliseconds. This is an approximated value chosen to maintain simplicity in the calculations. The crash impact load is thus applied over the unclamped end of the structure and is treated as a distributed force over the section area of the crush tube.

### 3.1. Numerical parameters and pre-simulation calculations

The input load \( P_i \) is calculated using the concept of Newton’s second law which relates to the change of momentum of the vehicle during the crush. The crush-cans undergoing deformation don’t change the overall mass of the vehicle as the pre-crash speed of 55 km/h is brought to a stop within the typically assumed 100 ms duration. Thus the impact force is calculated as:

\[
P_i = TCM \frac{V_A - 0}{\Delta t}
\]

where \( TCM \) is the total crash mass from Table 1, \( V_A \) is the equivalent pre-crash speed and \( \Delta t \) is the crash duration. Therefore \( P_i = 2.4627 \times 10^5 \) N. This can be considered as the peak crush load \( P_{\text{max}} \). The tube dimensions under consideration are presented in Table 2.

### Table 2. Tube parameters

| Thickness (mm) | Length (mm) | Width (mm) |
|----------------|-------------|------------|
| 1.25           | 150         | 30         |
| 1.5            | 150         | 30         |
| 1.25           | 150         | 50         |
| 1.5            | 150         | 50         |

The metals used in the current work are steel (AISI 1045) and aluminium (alloy; Al 6063), the properties of which are listed in Table 3.

### Table 3. Mechanical properties

| Material   | E (GPa) | \( \sigma_Y \) (MPa) | \( \nu \) | \( \rho \) (kg/m³) |
|------------|---------|----------------------|-------|------------------|
| Steel      | 205     | 530                  | 0.30  | 7850             |
| Aluminium  | 99      | 214                  | 0.33  | 2700             |

### 3.2. Comparative geometrical analysis

A parametric study of tube geometry facilitates the model verification as FEM is introduced into the analysis. According to the Table 2, we have four cases of circular tube parameters, i.e. two values of section diameters, 30 mm and 50 mm, and two values of tube section thickness, 1.25 mm and 1.5 mm. Four different geometries are investigated on their energy absorbing performance studied by reviewing the stress distribution and performance curves as load vs. displacement and total energy vs. time graphs. The crush-can behaviour is analysed with regard to the work presented in [7]. For a steel circular tube of geometrical dimensions presented in Table 1, the mean crush load \( P_{\text{av}} \) is calculated using Eq(2), CFE is obtained according to Eq(5), (Table 4).

### Table 4. Calculating \( P_{\text{av}} \) and CFE for different tube geometry

| Thickness (mm) | Length (mm) | Width (mm) | \( P_{\text{av}} \) (N) | CFE (%) |
|----------------|-------------|------------|------------------------|--------|
| 1.25           | 150         | 30         | \( 2.46 \times 10^4 \) | 10.01  |
| 1.5            | 150         | 30         | \( 3.24 \times 10^4 \) | 13.17  |
| 1.25           | 150         | 50         | \( 3.18 \times 10^4 \) | 12.93  |
| 1.5            | 150         | 50         | \( 4.18 \times 10^4 \) | 16.99  |
3.3. Comparative analysis of material performance
From Table 4, it is obvious that the last geometry shows the best CFE, thus demonstrating optimum energy absorbing characteristics which is in agreement with the work [7].
At this stage, Aluminium is introduced for a comparative material study. Since Aluminium has lower mass density, it may be worthwhile to explore its energy absorbing capability per weight performance. Thus circular tubes of 50 mm wide, 150 mm long and 1.5 mm thick are investigated with Aluminium as the structure material and compared with the corresponding steel model. Results of the structural performance of Aluminium and Steel, for tubes of $t = 1.5$ mm, $L = 150$ mm and $D = 50$ mm are presented in Table 5.

Table 5. Calculating $P_{av}$ and CFE for different materials

| Material          | $P_{av}$ (N) | CFE (%) |
|-------------------|--------------|---------|
| Steel             | $3.18 \times 10^7$ | 12.93   |
| Aluminium         | $1.69 \times 10^7$ | 6.86    |
| Aluminium ($t=2.25$ mm) | $3.10 \times 10^7$ | 12.60    |

Since steel tubes are almost twice as strong as aluminium, a higher section thickness of 2.25 mm was considered for the CFE calculations. Considering the low weight advantages of the Aluminium, different combinations of the two metals are attempted in the modelling. This contributes to proposing an optimised crush-can design as a novel energy absorbing solution.

4. Design verification applying FEM
An impact energy absorption performance of a thin-walled tube is investigated using FEM software ABAQUS/Explicit version 6.10 to study the modes of deformation of tubes under quasi-static loading conditions. The three dimensional model generated for the circular tube in ABAQUS is shown in Figure 2. The simulation is carried out using the direct matrix solver with minimum increment size of $1 \times 10^{-5}$, with structured hex-dominated mesh elements of type C3D8R.

From the observed behaviour established in Tables 4 and 5, three novel designs are proposed and investigated. These are the combinations of steel and aluminium distributed as
1) Half a tube (clamped end) is made of aluminium and the other half is made of steel;
2) Half a tube (clamped end) is made of steel and the other half is made of aluminium;
3) The stress concentration region is reinforced with steel and the rest of the tube is made of aluminium. The models and results for simulation of the crush are presented in Figures 3, 4 and 5 respectively.
For the third model, the steel material zone is proposed arbitrarily but with an instinctive decision based on the observed stress concentration pattern, and is subject to experimental verification. For simulation purposes, however the region starts 10 mm away from the base and the thickness is assigned to be of 30 mm. The three models considered are analysed based on the same crushing parameters and \( P_{av} \) is calculated independently from \( L \). Using the query tool, the mass and relative displacements are obtained, which is used to calculate the SEA from Eq(3) and Eq(4) (Table 6).

### Table 6. Comparative assessment of the proposed models

| Parameter (all in SI units) | Combination 1 | Combination 2 | Combination 3 |
|-----------------------------|---------------|---------------|---------------|
| Mass                        | 0.181         | 0.181         | 0.128         |
| Relative displacement (end to end); \( \delta \) | 19.2 x 10^{-3} | 9.05 x 10^{-3} | 19.9 x 10^{-3} |
| Relative displacement (mid to base) | 0.62 x 10^{-3} | 0.88 x 10^{-3} | 0.65 x 10^{-3} |
| \( P_{av} \)               | 2.44 x 10^4   | 2.44 x 10^4   | 1.56 x 10^4   |
| \( CFE \) (%)              | 9.91          | 9.91          | 6.34          |
| \( E_a \)                  | 468.48        | 220.82        | 310.44        |
| \( SEA \)                  | 2588.29       | 1220          | 2425.31       |

The total energy for each of the three models is presented in Figure 6. This assists in a better understanding of the performance of each design suggested.

### 5. Conclusions

The behaviour of three novel designs for circular crush-cans (Figures. 3, 4 and 5) is studied. The modelling of these systems is carried out applying both theoretical approach and FEM concentrating on the energy absorbing abilities. The optimization process is validated under quasi-static loading conditions whilst considering various modes of deformation and stress distribution along the tubular components. Moreover, structures made of different materials, steel and aluminium, are considered for the material effect analysis and optimization through their combination. Steel, being stiffer, demonstrated a better value of \( CFE \) and aluminium being lighter and more prone to crumpling results in a better \( SEA \). Considering overall characteristics for an effective energy absorber the first model (Combination 1, Figure 3) is the one exhibiting a better performance and can be recommended as an optimal option for a crush-can design. In terms of the cost, Combination 1 is also cheaper to be manufactured than Combination 3, due to the larger proportion of aluminium which is more than three times cheaper than steel. In future Combination 3 can be further optimised using a more detailed simulation or series of experiments, or a combination of both. The approach and designs presented can
be employed for further optimisation and development of other innovative design solutions. Optimization of the crush-can behaviour is done within the limits of the front impact scenario with the purpose of improvement of the structural performance in the Euro NCAP tests.

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