Experimental testing of selected energy absorbers in the context of their use in rear underrun protective devices

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Abstract. The energy-absorbing structures have been being developed for many years by various research centres all over the world. The article presents experimental test results obtained for selected structures designed for the absorbing of kinetic energy in order to reduce the loads on occupants of a motorcar when hitting the rear of a motor truck. The differences between selected energy-absorbing structures have been analysed, based on impact resistance indicators having been determined. The main objective of this research work was to verify the applicability of selected energy absorbers to the rear underrun protective devices as the truck parts accountable for vehicles’ passive safety.

Keywords: safety, vehicle accidents, crash test

1. Introduction

The searching for effective impact energy absorbers makes a challenge for various industries, e.g. automotive industry, aviation, or armed forces. The elements accountable for the absorption of kinetic energy should meet definite requirements and be characterised by appropriate parameters [1, 2, 3], depending on their intended use. In aviation, for example, the parameter of critical importance is the ratio of the energy absorbed to the absorber mass. The materials intended for military applications should be characterised by good effectiveness at high impact velocity values. In the automotive industry, the above characteristics are certainly very important, but a critical matter is the absorber size, especially when the energy-absorbing elements are to be applied to small L-category vehicles. Another issue that cannot be considered unimportant is the absorber cost, because such an element installed in a vehicle should not noticeably increase the vehicle price. Moreover, the absorber installed in a motor vehicle must ensure comparable effectiveness in varying operational conditions (especially temperature and humidity), which additionally raises the difficulty of selecting the most effective absorber design.

An analysis of the commercially available engineering solutions of energy-absorbing structures shows that in the automotive industry, three-dimensional structures are commonly used, having the form of hollow profiles with diversified cross-sections and additionally embossed areas, made of steel or aluminium (less frequently, composite material can also be met). As another solution, structures of the multi-cell type are used, having the form of a pack of many thin-walled pipes with different cross-sections. The large number of pipes connected together make a structure in the form of e.g. a honeycomb, a good point of which is low mass and relatively low production cost. As a similar solution, materials referred to as “foams” are also used [4].
In consideration of their intended use, motor trucks are designed for the cargo space volume and load capacity to be as big as possible. The questions of road traffic safety are dominated by the issue of active safety [5]. The elements related to the passive safety are limited to the vehicle cabin (driver’s safety) and to the front, side, and rear underrun protective devices (to ensure safety for the other road users). The basic function of the rear underrun protective devices (RUPDs) is to prevent passenger cars from running under the rear of a motor truck [6]. Despite the fact that the RUPD is to ensure safety for the occupants of a passenger car when hitting the rear of a motor truck, the impact energy absorption issue is generally ignored (see e.g. the UN ECE Regulations) or marginalised (the US or Canadian regulations) [7, 8, 9, 10]. In principle, such a device is only to be a rigid barrier for the car, while this may cause in some cases an increase in the loads acting on the car [11].

In consideration of the growth in social consciousness as well as the authorities’ pursuits of improvement in the road traffic safety (the “Vision Zero” strategy etc.), it is indispensable to develop modern engineering designs for motor trucks as well, the more so that the rear-end collisions make a significant percentage in all the road accidents [12, 13].

2. Research methods
To determine the energy-absorption characteristics of various engineering designs, dimensions, and materials adopted to build energy absorbers, some absorber samples characterised in Table 1 were prepared. The sample denoted as HC1.71_x2 was made of aluminium sheet and had a honeycomb structure. The absorber denoted as HC1.71box_x2, unlike the one above, was additionally enclosed with steel sheet 1.2 mm thick. The absorbers denoted as ST-KMC, ST-OAG and ST-RNO were structures based on rectangular hollow steel profiles. The sample denoted as AL-TMC was a rectangular hollow profile, too, but made of aluminium. It was assumed that the sample length could not exceed 0.2 m because of the limited space available for the mounting of a large-size RUPD.

The dynamic tests of the samples were carried out on a test stand at the Łukasiewicz Research Network – Automotive Industry Institute (Łukasiewicz – PIMOT). The test stand has been shown in Figure 1. The absorber sample under test was installed on an impact trolley with a mass of 325 kg, which struck an undeformable impact barrier with a velocity of 6.67 m/s. During the experiment (impact test), the trolley deceleration was measured. Additionally, the course of each test was filmed by means of a high-speed camera.

| Sample       | Height [m] | Width [m] | Length [m] | Mass [kg] |
|--------------|------------|-----------|------------|-----------|
| HC1.71box_x2 | 0.125      | 0.245     | 0.15       | 1.56      |
| HC1.71x2     | 0.125      | 0.245     | 0.15       | 0.34      |
| ST-KMC       | 0.060      | 0.115     | 0.135      | 1.20      |
| ST-OAG       | 0.073      | 0.070     | 0.187      | 1.14      |
| ST-RNO       | 0.100      | 0.072     | 0.158      | 1.34      |
| AL-TMC       | 0.100      | 0.060     | 0.132      | 0.66      |
3. Test results

The experimental test results have been shown in Figures 2-7 and in Table 2. The curves in Figure 2 represent impact trolley accelerations as functions of the absorber deformation. It can be noticed that for all the absorber materials chosen, apart from that of the AL-TMC sample, the maximum deformation values remained at a similar level. The extreme acceleration values for the ST-OAG and HC1.71box_x2 samples were $-28$ g and $-30$ g, respectively; for the ST-KMC and AL-TMC samples, they varied between $-35$ g and $-37$ g. In the test with the HC1.71_x2 sample, the trolley acceleration curve did not
reach −20 g; in contrast, the highest deceleration values were reached for the ST-RNO sample (the acceleration was almost −65 g).

To analyse the experimental test results, the indicators denoted as EA, SEA, MCF, PCF, and CLE were used.

The EA indicator is defined as the total deformation energy absorbed during the plastic deformation. It is represented by a formula:

\[ EA = \int F(x) \, dx \]  
(1)

where: F(x) is the instantaneous crushing force and x is the deformation value (see Figure 3).

SEA is the ratio of EA to the mass of the structure under test, according to the formula:

\[ SEA = \frac{EA}{m} \]  
(2)

where: m is the total mass of the structure under test (in our case, it is the mass of the test sample).

MCF is the mean crushing force; for the known deformation value, it was determined from the formula:

\[ MCF = \frac{1}{x} \int_{0}^{x} F(x) \, dx \]  
(3)

The peak crushing force (PCF) was determined as the maximum value of the crushing force recorded during the test. It has a significant influence on the safety of the impacting car occupants [14]. For the loads acting on the car occupants during a collision to be minimised, the value of this indicator should also be as low as possible. By comparing the PCF and MCF values with each other, we may determine the CLE indicator value [3], defining the uniformity of the load acting on the structure:

\[ CLE = \frac{MCF}{PCF} \]  
(4)

The results of the above calculations have been presented in Table 2 and in the graphs below.
Table 2. Results of dynamic strength tests.

| Sample                | Absorbed Energy [J] | PCF [kN] | SEA [kJ/kg] | MCF [kJ/m] | CLE [m/s²] |
|-----------------------|----------------------|----------|-------------|------------|-------------|
| HC1.71box_x2          | 7 763                | 106      | 4.97        | 52         | 0.49        | 302         |
| HC1.71x2              | 7 755                | 75       | 22.81       | 52         | 0.69        | 215         |
| ST-KMC                | 7 713                | 132      | 6.43        | 57         | 0.43        | 377         |
| ST-OAG                | 7 701                | 99       | 6.76        | 41         | 0.42        | 283         |
| ST-RNO                | 7 696                | 227      | 5.74        | 49         | 0.21        | 649         |
| AL-TMC                | 7 696                | 125      | 11.66       | 58         | 0.47        | 358         |

Figure 4. SEA for different samples

Figure 4 shows the SEA values determined for individual samples. It can be noticed that the highest SEA value (almost 23 [kJ/kg]) was achieved for the HC1.71x2 sample. Noteworthy is also the result obtained for the HC1.71box_x2 sample, which was made of the same material as the one used for the sample mentioned above, but was additionally enclosed with steel sheet. For this absorber, the SEA value was reduced to only somewhat more than one-sixth of that for the former sample.
According to an analysis of the MCF calculation results, the lowest values of this indicator were obtained for two samples, i.e. ST-OAG and ST-RNO. For the other four samples, the MCF values varied between 50 kN and 60 kN.

Figure 5. MCF and deceleration for different samples

Figure 6. PCF for different samples
Results of PCF calculations for the samples under test have been presented in Figure 6. For most of the samples, the values of this indicator were at a similar level, except the standing-out value for the ST-RNO sample.

![Figure 7. CLE for different samples](image)

The CLE values shown in Figure 7 were in most cases at a similar level of about 0.45. Exceptions were the highest and the lowest values of the structure load uniformity indicator, amounting to 0.69 and 0.21 for the HC1.71x2 and ST-RNO samples, respectively.

4. Conclusion

An analysis of the impact trolley deceleration (load) values recorded as a function of the energy absorber deformation has shown that the lowest overloading values took place in the case of the sample made entirely of aluminium with a honeycomb structure and the highest values occurred, especially in the final deformation phase, for the steel profile denoted as ST-RNO. This indicates that the latter sample underwent ultimate deformation. It can also be noticed that for most of the absorber designs, the deformation values were close to each other. Moreover, significant exceedances of the 100 kN value were found to occur (for the ST-KMC, ST-RNO, and AL-TMC samples). Therefore, the reasonability of using such materials as energy absorbers in RUPDs should be reconsidered because their stiffness is close to the stiffness of structural components of the conventional RUPD [15], which in consequence may reduce the absorber’s effectiveness, especially if a reduction of the loads acting on car occupants during a collision is concerned.

A simple qualitative analysis of the test results, e.g. of the force or acceleration vs displacement curves, does not offer an unequivocal answer to the question which of the samples under test will be the most suitable for being used as an energy absorber in an RUPD. To indicate the most effective absorber, the features that should be considered the most important should be identified.

Based on the SEA indicator, the honeycomb structure (HC1.71_x2) should be considered the most appropriate. However, if the absorber is to be used in a motor truck, where the absorber mass makes merely about one-thousandth of the total vehicle mass, then the absorber mass may be deemed unimportant. Hence, the SEA indicator is actually inappropriate for evaluating the properties of the material sample under test from the point of view of using this material for an energy absorber.
If the test results are evaluated on the grounds of the MCF value, then two structures, i.e. ST-KMC and AL-TMC, for which the MCF was 57 kJ/m and 58 kJ/m, respectively, may be indicated as the most desirable. For the next two samples (HC1.71_x2 and HC1.71box_x2), this value was 52 kJ/m, i.e. by about 9% lower than the values mentioned above.

If the PCF value (detrimental to the safety of the impacting car occupants) is taken into account in the analysis, then the ST-RNO absorber should be considered the least effective in reducing the loads acting on the occupants of the car hitting the rear of the motor truck. A definitely better solution was the HC1.71_x2 structure, for which the PCF value was equal to about one-third of the value determined for the ST-RNO sample.

Attention should however be paid to the fact that if the same PCF parameter is taken into account together with the MCF indicator, then the absorbers based on the honeycomb structure show the energy absorption properties close to those of the ST-KMC and AL-TMC samples, but at lower peak deceleration values (by about 15-20%), which may be decisive for the safety of car occupants during a road accident.

Therefore, the structures of the multi-cell type based on the honeycomb structure may be recognised as capable of finding successful application in the automotive industry as elements accountable for the passive safety of motor vehicles. Moreover, their structure enables them to absorb the impact energy almost over their whole length without rapid changes (peaks) in the deceleration curves. A similar property was only showed by the structure made of the aluminium profile, but at much higher deceleration values.

The research results presented are treated by the authors as preliminary. The work on the selection of the energy-absorbing structures suitable for rear underrun protective devices will be continued.

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