

Evaluating Impact Attenuator Performance for a Formula SAE Vehicle

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ABSTRACT

Formula SAE® is one of several student design competitions organized by SAE International. In the Formula SAE events undergraduate and graduate students are required to conceive, design, fabricate and compete with a small, formula-style, race car. Formula SAE safety rules dictate a 7 m/s (or approximately 15.65 mph) frontal crash test for nose mounted impact attenuators. These rules are outlined in section B3.21 of the Formula SAE rule book. Development and testing methods of these energy absorbing devices have varied widely among teams. This paper uses real world crash sled results to research methods for predicting the performance of aluminum honeycomb impact attenuators that will comply with the Formula SAE standards. However, the resulting models used to predict attenuator performance may also have a variety of useful applications outside of Formula SAE. In this paper, various energy absorbers were mounted to a free rolling trolley sitting on top of a crash sled. The sled was launched so that the trolley with the attached attenuator was allowed to strike a rigid barrier. This resulted in a sudden deceleration measured by accelerometers attached to the trolley. The resulting deceleration from each impact attenuator was then correlated to predicted pulses from theoretical calculations. The lessons learned from extensive testing will be discussed including comparisons between size, shapes, and material properties of energy absorption devices. Additionally, a final theory will be presented describing the ideal way to predict impact attenuator performance. Ultimately it will be shown that, given a known geometry, material properties, and safety factor, the behavior of an impact attenuator can be predicted accurately enough that testing will only be needed as verification. This study will ultimately benefit all Formula SAE® teams, as it will help speed up development time and cut costs, while providing a

proven method for creating attenuators that will perform to SAE standards.

INTRODUCTION

OBJECTIVE

The Formula SAE® rules require the use of an impact attenuator mounted to the front bulkhead of the vehicle. An example of an impact attenuator is shown in [Figure 1](#).



Figure 1. Front Impact Attenuator Affixed to the Front of a Formula SAE vehicle

The objective of this paper is to validate the design of an aluminum honeycomb impact attenuator while developing the ability to predict its performance analytically. Findings will

be supported through mathematical explanations and various testing. Following a detailed description of the theory, test procedure and results, suggestions will be provided concerning the proper manner to conduct impact tests so that results are both accurate and consistent.

This paper will ultimately serve as a potential guideline for future Formula SAE® teams who wish to undertake their own testing. It also may answer questions regarding general crash safety center procedures, should any teams wish to utilize the services of crash safety facilities.

BACKGROUND

Section B3.21 in the 2009 Formula SAE® rules states that the impact attenuator should be capable of decelerating a 300 kg (661 lb) vehicle from 7.0 m/s (23.0 ft/s) such that an average deceleration of 20 g's and peak of 40 g's should not be exceeded [1]. Development and testing methods of these energy absorbing devices have varied widely, ranging from drop tests to manned barrier tests. Some of the validation test methods employed by Formula SAE teams have been decidedly unsafe. Crash centers are ideal for collecting reliable data with maximum regard to safety. The deceleration sled in Kettering University's Crash Safety Center was used for this paper to research methods for predicting the performance of various impact attenuators designed to comply with the Formula SAE® standards. The intent is to define a reliable, common way to design an attenuator that will meet Formula SAE® standards.

METHODOLOGY

ATTENUATOR THEORY

Impact attenuators are designed to reduce the peak deceleration that occupants might experience in the event of a frontal crash. A properly designed attenuator should lower the peak acceleration by managing the time variable, or more specifically, by elongating the time it takes to come to a rest so that the occupant can take advantage of “ride down”. Ride down is the effect of the vehicle structure dissipating energy through crush. Ride down is only effective if occupants are restrained properly, meaning they must be fixed to the vehicle.

How impact attenuators manage deceleration is referred to as the “crash pulse” or simply “pulse”. The shape of the pulse shows how an object decelerates over time. The basic advantage of designing an attenuator using aluminum honeycomb is that it provides a relatively constant rate of deceleration as the honeycomb crushes and dissipates energy.

As shown in [Figure 2](#), the force required to crush an aluminum honeycomb attenuator ramps up very quickly before becoming nearly constant and oscillating around an average crush force. The peak occurs just left of the dashed

blue line. Typically, the honeycomb is pre-crushed by a small amount (6 mm or so) to eliminate the initial peak during subsequent compression events. Therefore, these attenuators can be effectively modeled as constant deceleration devices, meaning they are also constant force absorbers because force is simply a scalar of acceleration.

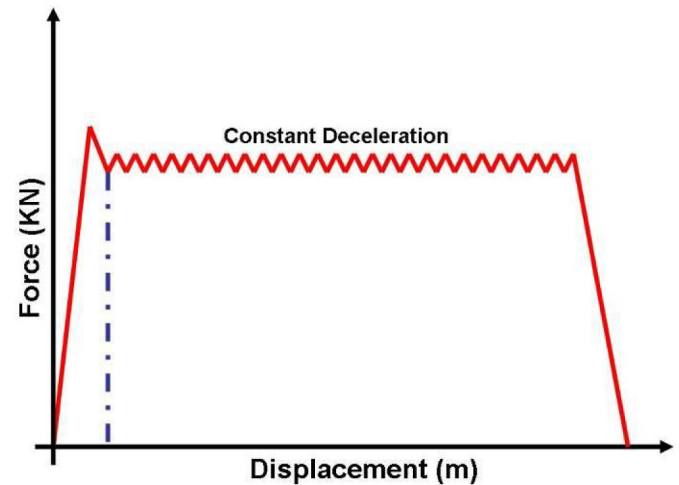


Figure 2. Typical Crush Characteristics of an Aluminum Honeycomb Impact Attenuator

The crush strength (P_{CR}) for the honeycomb is found by taking the average crush strength (F_{CR}) and dividing by the cross-sectional area (A_{CS}) of the test specimen. Basic variable are standard so that mass is symbolized as m , force as f , gravity as g , and weight as W . Since the crush strength is essentially constant, the deceleration resulting from crushing a section of honeycomb with a uniform cross-sectional area is essentially constant and can be found from:

$$F = (m)(a) = (P_{CR})(A_{CS}) \quad (1)$$

$$a_{avg} = \frac{(P_{CR})(A_{CS})}{m} = \frac{(P_{CR})(A_{CS})g}{W} \quad (2)$$

The equation shows that the average deceleration rate can be controlled by properly selecting the honeycomb materials crush strength and cross-sectional area.

By applying the well-known constant rate of acceleration formulas, other relationships can be determined. For instance, the nominal time for the vehicle to be brought to a stop can be found from:

$$v = v_0 + (-a_{CR})(t_{CR}) \quad (3)$$

$$t_{CR} = \frac{v_0}{a_{avg}} = \frac{(W)(v_0)}{(P_{CR})(A_{CS})(g)} \quad (4)$$

$$a_{avg} = \frac{v_0}{t_{CR}} \quad (5)$$

The resulting equations show that a longer duration (higher t_{CR}) pulse will reduce the average rate of deceleration. As shown in a different manner, the crush strength and cross-sectional area are the primary variables for controlling the pulse duration.

Equation 7 is another constant acceleration formula that may be used to determine the length of material that must be crushed to absorb the impact energy. Note that v_i stands for initial velocity, v_f for final velocity, and S_{CR} denotes the difference in attenuator length after it has been impacted.

$$v_f^2 = v_i^2 + 2(-a_{CR})(S_{CR}) \quad (6)$$

$$S_{CR} = \frac{v_i^2}{2(a_{CR})} = \frac{(W)(v_i^2)}{2(g)(P_{CR})(A_{CR})} \quad (7)$$

An alternative approach to determining the crush stroke is to apply conservation of energy, which can be expressed as:

$$E_{Incoming} = E_{Crush} + E_{Rebound} \quad (8)$$

$$\frac{1}{2}(m)(v_i)^2 = (P_{CR})(A_{CS})(S_{CR}) + \frac{1}{2}(m)(v_f)^2 \quad (9)$$

Performing an energy balance can predict the amount of energy in the system before impact, the energy absorbed by the attenuator, and any energy left over after impact. The energy absorbed by the attenuator should always be less than the energy before impact because of losses due to heat and friction. Additionally, aluminum honeycomb is not an “ideal” attenuator, so not all of the energy will be absorbed. The energy not dissipated will cause the trolley to bounce off the barrier. Therefore, Equation 9 says that the kinetic energy of the trolley moving down the rails and into the barrier is equal to the energy dissipated by the attenuator plus the kinetic energy left over when the trolley rebounds. The velocity is known to be zero immediately after the impact. The velocity was positive as the trolley was heading toward the barrier,

decelerated to zero, and then began moving backward in the negative direction. The quantity v_f is the peak velocity after impact when the attenuator has rebounded off of the barrier.

COMPRESSION TESTING

The attenuator testing conducted for this paper utilized aluminum honeycomb from Plascore rated at a crush strength average of 1689 kPa (245 psi) $\pm 10\%$. It is important to verify these ratings to quantify the relative variation during the crash testing between attenuators. If crash test results contain any anomalies, prior verification of the material's average crush strength will make it easier to rule out possible inconsistencies in the material.

The compression tests conducted followed the ASTM D 7336 entitled “Standard Test Method for Static Energy Absorption Properties of Honeycomb Sandwich Core Materials” [2]. In summary, it consists of crushing samples of honeycomb between two flat faces at a constant change in displacement. The resulting force to crush the sample is measured by a load cell, in this case in pound-force. Additionally, ASTM D 7336 described the proper sized area of the crushed face and depth of the sample as a function of the honeycomb cell size [2]. A 6000lb (approximately 27 kN) load cell was used to collect the data for these tests.

When the aluminum honeycomb is cut there are partial cells left behind which may alter the crush properties. Therefore, the largest square cross-sectional area that would not exceed the capacity of the load cell was selected because a larger area lessens the effect of the edges. Additionally, sectional area was also influenced by the amount of force it will take to crush. A size of 103.225 square centimeters (4 in $\times$ 4 in) was chosen so at an average crush pressure of 1689 kPa the sample should hold approximately 17.4 kN before crushing. It was imperative that the expected load required to crush the attenuator was large enough to accurately read without exceeding the load cell limit of 27 kN. A $\pm 15\%$ margin of error was used because it was uncertain whether or not the sample would truly crush at 1689 kPa. If the samples were within the specified manufacturer tolerances they should fall somewhere between 14.8-20 kN ($\pm 15\%$).

Plascore noted in their brochures that the material properties of their aluminum honeycomb material are consistent until about 70% of the length of the sample is used up. Following this logic, the samples utilized for these tests were 10 in (25.4 cm) long so they could have been crushed up to 7 in (17.78 cm). After this point, the sample will start to experience “stack-up” which is the point when the material becomes overly compressed. When stack up occurs the material has no more room to buckle, forcing the resistance to increase. Once stack up occurs, the material will be outside of the acceptable range for which it is designed. Therefore, the samples should

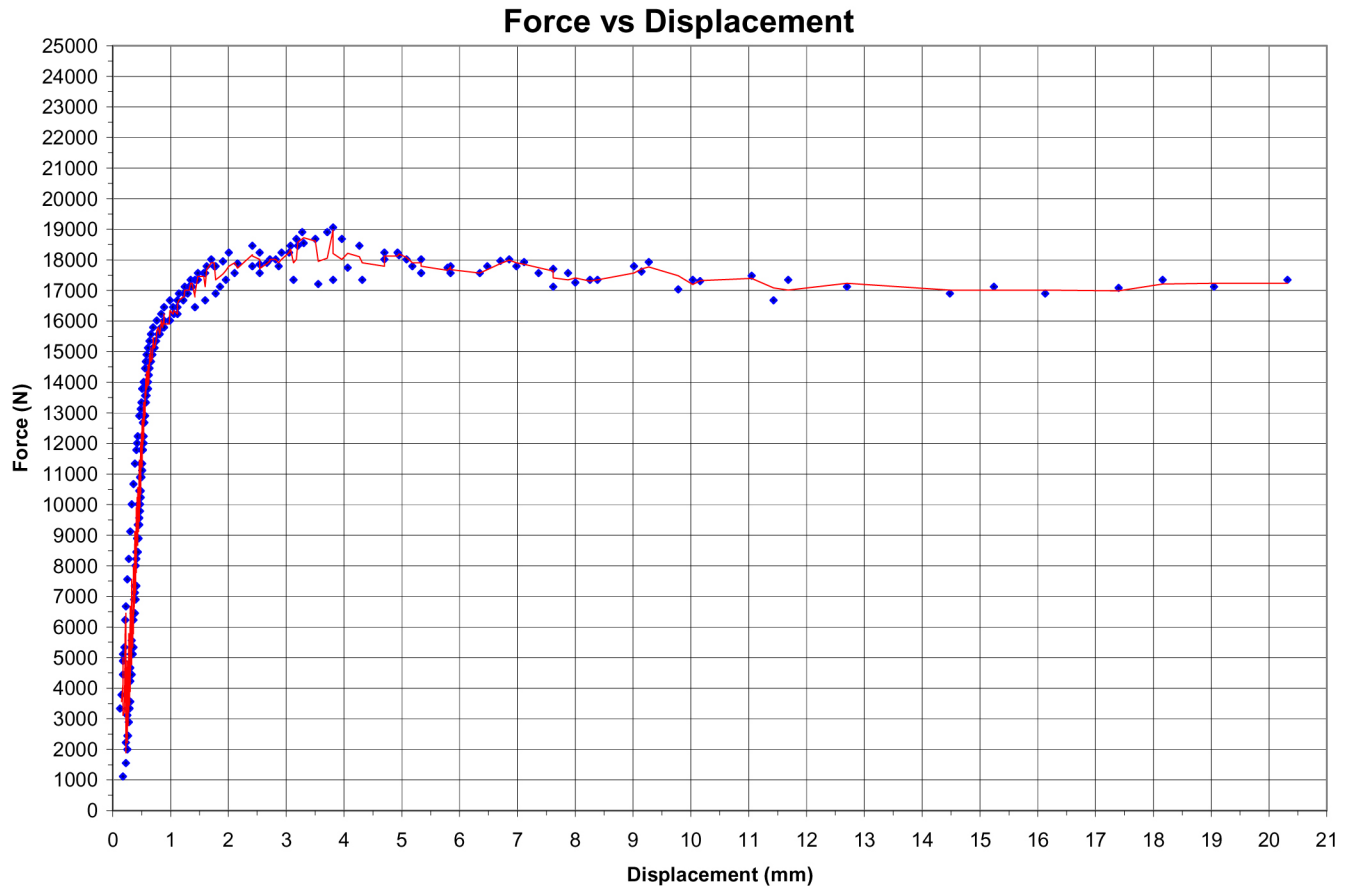


Figure 3. Compression Test Data: Average Load versus Displacement

not be crushed far enough to risk stack-up, or the results of the compression test will not be accurate.

Figure 3 depicts the average load versus displacement of four samples. The samples continue to bear the load to a given point before crushing at a nearly constant and uniform rate. During the compression testing it was observed that none of the samples began to fail until about 17.8 kN, after which, each sample began to crush very consistently. The aluminum folded over in tiny little buckles on the pre-crushed end causing the load to fluctuate between 16.9 and 18.2 kN. Like an accordion, as each small buckle gave way, the load cell reading would abruptly drop and then begin to climb again until the next buckle appeared. As can be seen in Figure 3, after reaching an initial peak, the load that was required to continually compress the honeycomb became relatively constant.

All the compression tests yielded results within the manufacturer tolerance of $\pm 10\%$ of 1689 kPa, as shown in Table 1 where P_{CR} denotes the average crush force required to crush the samples. All the samples held a little more than 17.4 kN, especially Test Sample #2 which held 9.4% more than expected. Test sample #3 held the least, at just 2.4% higher than expected. From this data it can be concluded that

Plascore's rating of 1689 kPa with an error of $\pm 10\%$ is accurate. Note that the material tolerance should be considered when designing an impact absorber so that all attenuators of the same design, using the same material, will pass Formula SAE® rules if tested.

Table 1. Compression Test Results

Test	P_{CR} , expected (kPa)	P_{CR} , measured (kPa)	Difference %
1	1689	1788	5.9
2	1689	1849	9.4
3	1689	1730	2.4
4	1689	1831	8.4
Measured Average:		1800	6.1

To summarize the compression results, the average crushing force or the force required to *continue* crushing the attenuator, is close to the rated 1689 kPa. However, the average of the compression tests was slightly higher at 1800 kPa. Therefore, 1800 kPa will be used instead of Plascore's recommended 1689 kPa for the calculations throughout the rest of this paper. Also note that this test is a static test, because the honeycomb was slowly loaded to the point of failure. This usually results in lower average crush force