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Similarity-Based Models for Testing Vehicle Roofs against Crushing during Impact Events

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Abstract. *The present work carried out finite element simulations of two vehicular roofs applying some of the boundary conditions previewed in FMVSS 216a standard. Only the roof was tested in the numerical analysis. One vehicle roof was designed based on the Similitude Theory using the Buckingham π theorem. It was called the 'Model.' The other, known as 'Prototype,' had the actual dimensions of the SUV Ford Explorer 2003. The samples were made in metallic tubes obtained from the regular stock of commercial product suppliers. The structural layout is a cage. The Model was built on a scale of 1:0.453 considering the available commercial thicknesses of tubes; it was the restrictive factor.*
The main objective of this work was to evaluate the numerical response of the Prototype through the Model's behavior under crushing phenomena. The structural response of interest was the force x displacement in column A. The FE analysis on the Model and Prototype demonstrates geometric and kinematic similarities between them, as previewed in the π values. The findings show that it is possible to carry out roof crush testing in the reduced Model and predict the Prototypes' structural behavior. Such an approach reduces costs inherent to the manufacturing process and operational difficulties related to the vehicle roof's physical installation.

Keywords: *similitude theory, vehicle roof, finite element, FMVSS 216a, roof crush*

1. INTRODUCTION

A vehicle rollover is one type of automotive accident that causes about 25% of deaths in crashes of cars and 59% in SUVs (IIHS, 2008). The excessive crushing of the vehicle's roof is the reason that leads these accidents to cause severe and fatal injuries caused by excessive intrusion into the occupant's compartment (safety cell). One approach to mitigate such a structural failure is to increase the structural performance of the vehicle's roof. However, to investigate the structural strength of vehicular roofs, we can carry out experimental tests or numerical simulations.

This performance can be analyzed by a complementary test proposed by the international standard FMVSS 216a - Roof Crush Resistance Test that evaluates rollover crashworthiness for passenger vehicles. This quasi-static test helps to determine the force required to push a metal plate into the roof at a constant velocity rate. It requires a reaction force equal to 3 times the vehicle's weight to be reached within 5 inches (127 mm) of plate displacement.

According to the FMVSS 216a, the following steps shall be performed on the crush test preparation, as described below.

- The testing vehicle must be rigidly fixed in a tie-down fixture. It should be capable of supporting the vehicle weight as well as the loads applied during the test. It should also prevent any significant movement of the vehicle during the test.
- The loading device consists of a rigid unyielding metal plate. Its lower surface which indeed applies the load is a flat rectangle whose dimensions are 762 millimeters by 1829 millimeters.
- The rigid metal plate should be tilted forward at a 5º ± 0.5º angle along its longitudinal axis (pitch angle). Besides, the plate should be rotated outward at a 25º ± 0.5º angle along its lateral axis (roll angle).
- The initial contact area is on the longitudinal centerline of the metal plate's lower surface.

- The metal plate should be positioned 254 mm forward the vehicle column A (initial contact area).

Figure 1 illustrates the set up showing an actual vehicle placed in.

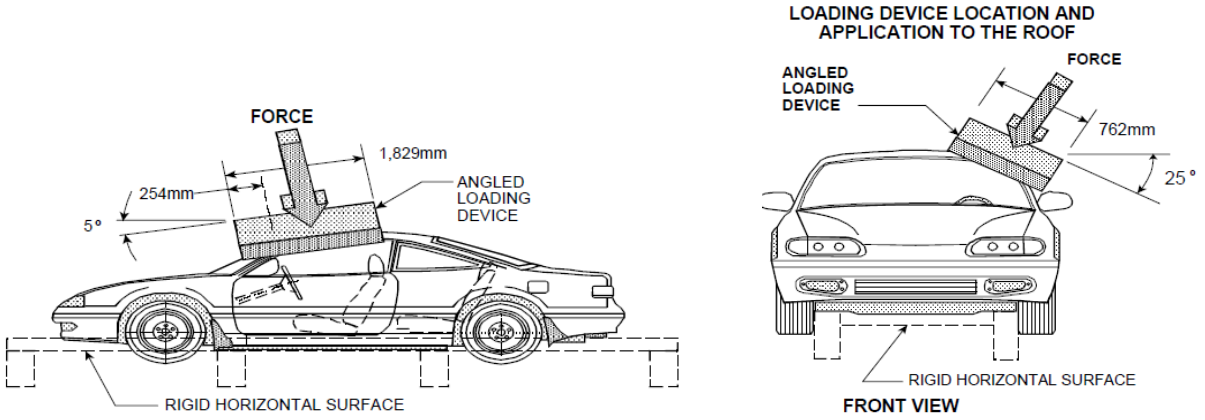


Figure 1. Loading device for crushing roof vehicle according to FMVSS 216 (source: FMVSS 216, 2006).

The (FMVSS 216a, 2009) standard requires the roof structure to endure an applied loading. For vehicles weighing less than 2.722 kg, this required load equals 3.0 times the unloaded vehicle weight. For heavier vehicles, the required load undergoes an increase of 50% of the unloaded vehicle weight.

The researchers have been approached the passengers' safety in different ways. Some of them are dedicated to studying active and passive systems incorporated into the vehicles. Others are interested in reinforcing some vehicle structures either by using high-performance materials or by modifying the structural arrangement (Rittner, 2015). One of the essential structural systems for passenger safety is the roof.

Albrodt *et al.*, 2014 investigated the vehicle roof strength considering the (FMVSS 216a, 2009) requirements and their capacity to well-perform in rollover accidents under different pitch angles. The findings showed, among other things, that a reinforcement in pillar B was necessary.

Dzerkelis *et al*., 2015 investigated the influence of A-pillar on the strength of vehicle roof structures repaired. The analysis was carried out using the finite element method under a quasi-static compression loading representing a vehicle rollover. The findings pointed out that A-pillar had the greatest impact on the vehicle's roof structure and contributed about 78% of crushing strength.

Zhao *et al*., 2017 studied the optimization of pillars A, B, and the roof rail using preliminary roof compression results. The authors increased the thickness of A and B pillars by 10% and 20%. The results showed that changes in the thickness of pillar A improved its strength while the same improvement in pillar B had a minor effect on the vehicle roof compressive strength. However, these structural changes led to an increase less than 10% in the peak force computed during the Roof Crush Testing.

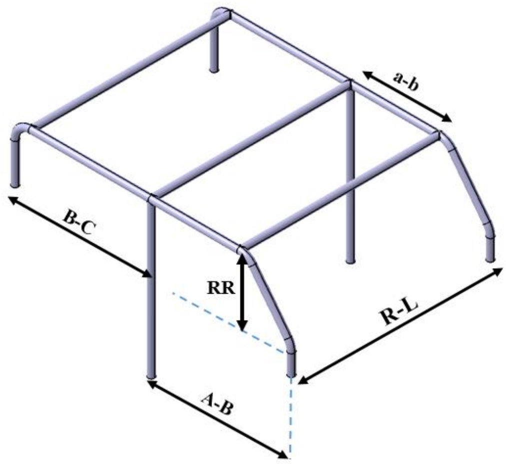
The present work aims to establish similarity-based models for testing vehicle roofs against impact events. The loading was quasi-static crushing, and the reference method was based on the FMVSS 216a, 2009 standard. The Buckingham π theorem was used to establish the similitude relations between the Model (reduced structure) and the Prototype (actual structure). Three classes of parameters were considered geometric, kinematic, and dynamic and, the Theorem leads to scale factors (geometric scale factors) that clearly relate to the parameters concerning Model and Prototype.

The results showed that the similitude between reaction forces from the Model and Prototype was achieved. Also, the roof vehicle's cages supported the minimum load required by the standard during the crushing phenomena, achieving a strength weight ratio (SWR) of about 3. The findings outlined that the metal plate's size used to load the vehicle's cage is not relevant for analyzing the phenomenon. In fact, its size did not affect the curve Force x displacement of the reduced model.

2. METHODOLOGY

The vehicle roof design used in the present study was based on the research of (Rittner, 2015), which investigated the characteristics of the forces in some vehicle roof shapes considering the Jordan rollover tests. His study was essentially numerical. The vehicle roof design in the present work refers to the SUV Ford Explorer 2003, which weighs approximately 2,000 kg. This vehicle model was validated experimental and numerically by the NCAC (*National Crash Analysis Center*) of the George Washington University in a frontal crash collision. The vehicle roof in real dimensions, called Prototype, is depicted in Figure 2.

The vehicle roof in scale, called Model, was designed according to the concepts of the Similitude Theory and, it was established as 1:0.453. This choice was guided by the possible relations between the thickness of the tubes used in the roof construction of both roofs, i. e., Prototype and Model. The tubes were acquired from the regular stock of commercial product suppliers, having marketable dimensions. In fact, as the present work progressed, the authors realized that the thickness of the thin-walled tubes was of great influence and, a limiting issue considering material suppliers.



Main dimensions	Prototype cage (mm)
Base width (**R-L**)	1536.4
A-B-pillar distance at roof rail (**a-b**)	683.8
A-B-pillar distance at beltline (**A-B**)	1078.6
Roof rail height (**RR**)	476.4
B-C-pillar distance (**B-C**)	1050.7

Figure 2. Prototype cage (real size vehicle roof) and the real dimensions.

The technical literature consulted to design the Model was (Zohuri, 2015); (Nicoreac *et al*., 2010); (Shehadeh *et al*., 2015); (Tan, 2011) e (Coutinho *et al*., 2016). According to (Tan, 2011) using prototype material as model material is the easiest way to obtain perfect simulation results. (Nicoreac *et al*., 2010) states that direct models can be used even for analysis in the post-elastic range.

In addition, both roofs were statically charged what did not constitute an impact loading but a crushing loading with a constant velocity of 13 mm/s, as prescribed in the (FMVSS 216a, 2009). (Booth *et al*., 1983) and (Jones, 1997) discussed the effects of impact loading in reduced models and, they concluded that in impact events, sometimes similarity results are negatively influenced by the strain rate effects. The strain rate is classified as low in the present work, being about 10^{-1} s^{-1}. Thus, strain rate and inertia effects were not considered on similarity scale relationships. The principles of the Buckinghan Theorem were applied to define the dimensions of roof Model. The variables treated concern three classes: geometry, kinematics and dynamics.

Most standard's recommendations were followed in the numerical simulation carried out in a finite element commercial package, as discussed in section 1. The major change was not fixing the entire vehicle for the test but only the vehicle roof. Previously, the authors checked this feature and concluded that there was no significant change in force and displacement comparing both situations, the whole vehicle and only the cage. Nevertheless, there is an important save in time processing. (Toczyski *et al*., 2013) used the same numerical strategy in his work.

The metal plate shown in Figure 1 (section 1), responsible for provoking an intrusion of about 127 mm in the Prototype and 57.5 mm in the Model, was treated as a rigid body (undeformable) at the simulations. This feature avoids deformations in the plate, concentrating the stress in the cage only, as prescribed in the (FMVSS 216a, 2009). Figure 3 illustrates the set Prototype and metal plate used in the numerical simulation.

The boundary conditions were fixed support at the basis of columns A, B, and C. The element type was mostly second order tetrahedral TET10. The contact between the longitudinal centerline of the metal plate's lower surface and the cage was considered frictional. A sensitivity study determined the mesh refinement for the Prototype and Model.

The main numerical result, according to the standard (FMVSS 216a, 2009) was the Force x displacement curve (section 3) which showed the intrusion on the passengers' compartment and, thus, the safety of them against serious or fatal injuries. In addition, the analysis of such a curve allows monitoring the level of force, which needs to reach the standard limit (58,8 kN) since the intrusion remains below the kinematic limit.

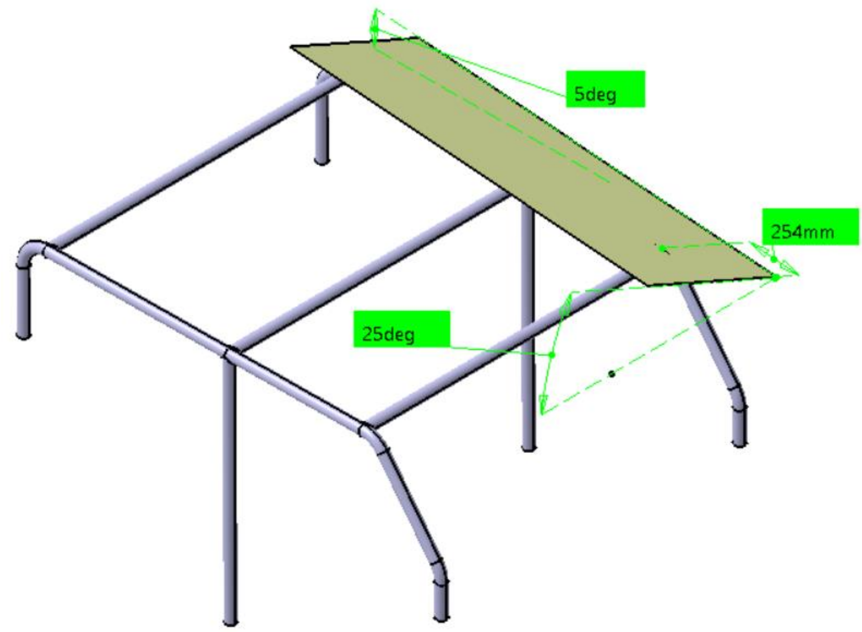
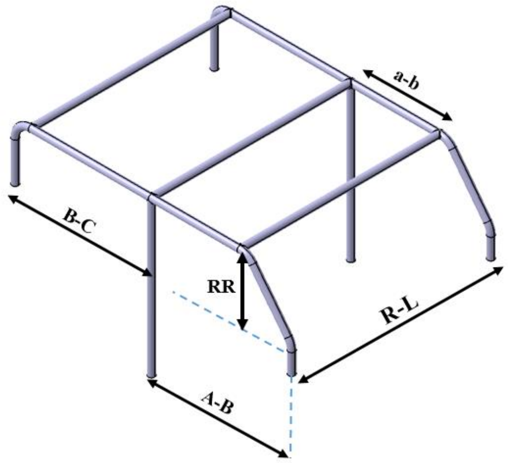


Figure 3 – Set metal plate and Prototype used in the numerical simulation.

Three variables classes were adopted for the similitude study. The first class is the geometry of the Model cage that includes the thin-walled tube thickness, external diameter, among other dimensions, as shown in Figure. 4. These dimensions' indications are as shown in Figure. 2. The second class formed by the kinematics variables is the maximum intrusion value toward the cab's interior (displacement) and the velocity established for the quasi-static experiment. The force applied to the Prototype and Model composed the third group of variables which depends on the actual vehicle mass as described in section 1. According to the (FMVSS 216a, 2009), the force was estimated using the mass, gravity acceleration and, a 3.0 strength-to-weight ratio (SWR).

It is worth highlighting that the metal plate's (Fig. 3) dimensions are predefined by the standard (FMVSS 216a, 2009), and they were part of the geometry class.



Main dimensions	Model cage (mm)
Base width (**R-L**)	696.0
A-B-pillar distance at roof rail (**a-b**)	309.8
A-B-pillar distance at beltline (**A-B**)	488.6
Roof rail height (**RR**)	215.8
B-C-pillar distance (**B-C**)	476.0

Figure 4. Reduced model of the vehicle roof respecting the established scale.

The list of all parameters involved in the study led to 14 variables (**n = 14**). All the expected parameters which are relevant to the problem are included Then, the primary dimensions selected were the mass (M), the length (L) and the time (T), thus **r = 3**. In this problem (kg, m, s). The number of restrictions (**r**) is equal to the number of primary dimensions.

The next step consists of assigning the primary dimensions to the parameters. As n = 14, the restrictions r = 3, the number of π groups is 11 π groups. The dimensions for each parameter is *mm* (L); *kg* (M), *m/s* (LT^{-1}) and *N* (MLT^{-2}).

For each variable class an independent variable was set up thus, for the geometry, the tube thickness (t); for the kinematic class, the speed (v) and for the dynamic class, the force applied to the roof (F). These three variables were not considered in the π functions.

Occasionally, the Prototype should be constructed for an experiment considering infrastructural, technological and, financial restrictions. It leads to a Model in reduced scale based on geometric scale factor (λ), representing the ratio between the Model parameter and the Prototype parameter, as shown in Eq.1. The subscript (t) represents the thickness in the present work.

$$\lambda_t = \frac{P_{model}}{P_{Prototype}} \tag{1}$$

Table 1 outlines the parameters considered in the calculation of the functions π. The scale factors from 1 to 7 corresponds to the ratio between the thin-walled thickness of the Model and Prototype. The same comes up to the scale factors from 9 to 11. Moreover, the ratio between the Model velocity and Prototype velocity (V_M/V_P) equals 1.

Table 1 – Parameters considered in the calculation of the functions π

Parameter	Symbol	Function π	Function values	Scale factor
Base width	RL	π_1	$\pi_1 = \frac{RL}{t}$	λ_t
A-B pillar distance at roof rail	a-b	π_2	$\pi_2 = \frac{a-b}{t}$	λ_t
Roof rail height	RR	π_3	$\pi_3 = \frac{RR}{t}$	λ_t
Center line height	CL	π_4	$\pi_4 = \frac{CL}{t}$	λ_t
B-C pillar distance	B-C	π_5	$\pi_5 = \frac{B-C}{t}$	λ_t
External diameter	Dext	π_6	$\pi_6 = \frac{dext}{t}$	λ_t
Radius of the machine to bend tubes of the cage	R_{mach}	π_7	$\pi_7 = \frac{R_{mach}}{t}$	λ_t
Mass	Mass	π_8	$\pi_8 = \frac{MV^2}{tF}$	λ_t^3
Width of the metal plate	WP	π_9	$\pi_9 = \frac{WP}{t}$	λ_t
Length of the metal plate	LP	π_{10}	$\pi_{10} = \frac{LP}{t}$	λ_t
Maximum intrusion	δ_{max}	π_{11}	$\pi_{11} = \frac{\delta_{max}}{t}$	λ_t

The scale factor for the vehicle mass pointed out in Tab. 1 was deduced from Eq. 2 to 5, considering that the Model and Prototype are made up of the same material. Thus $\rho_M = \rho_P$, knowing that density is the ratio between the mass and the volume we have:

$$\rho_m = \frac{m_m}{V_m} = \frac{m_m}{L_{1m} * L_{2m} * L_{3m}} \tag{2}$$

$$\rho_p = \frac{m_p}{V_p} = \frac{m_p}{L_{1p} * L_{2p} * L_{3p}} \tag{3}$$

$$\frac{m_m}{L_{1m} * L_{2m} * L_{3m}} = \frac{m_p}{L_{1p} * L_{2p} * L_{3p}} \tag{4}$$

$$\frac{m_m}{m_p} = \frac{L_{1m} * L_{2m} * L_{3m}}{L_{1p} * L_{2p} * L_{3p}} = \lambda_t * \lambda_t * \lambda_t = \lambda_t{}^3 \tag{5}$$

where 'L' is a geometric parameter; 'm' is the mass; 'ρ' the density and 'λ' the geometric scale factor. The subscript 'M' represents the Model and 'P' the Prototype. The roll and pitch angles applied to the metal plate were the same in the Model and Prototype, as discussed in section 1.

Table 2 outlines examples of how to obtain *function values* for π functions from 1 to 7, from 9 to 11 and, for π_8. The calculation method was based on the aspects discussed above and in the work (Shehadeh *et al*., 2015).

Table 2 – Examples to show how to calculate the *functions values*

$\pi_1 = RL\ (t\ V\ F)$
$\pi_1 = RL(L^a (L\ T^{-1})^b\ (M\ L\ T^{-2})^c) = M^0 L^0 T^0$
$\pi_1 = mm\ (mm^a (mm\ s^{-1})^b)\ (kg\ mm\ s^{-2})^c = kg^0\ mm^0 s^0$
Solving the system:
L: 1+a+b+c=0 → a= -1
M: c=0
T:-b-2c=0 → b=0
$\pi_1 = RL\ t^{-1}$
$\pi_8 = Mass\ (t^a\ V^b\ F^c)$
$\pi_8 = kg\ (mm^a (mm\ s^{-1})^b\ (kg\ mm\ s^{-2})^c) = kg^0 mm^0 s^0$
Solving the system:
$L: a + b + c = 0 \therefore a + 2 - 1 = 0 \therefore a = -1$
$M: 1 + c = 0 \therefore c = -1$
$T: -b - 2c = 0 \therefore -b - 2(-1) = 0 \therefore b = 2$
$\pi_8 = t^{-1} V^2 F^{-1}$
$\pi_8 = \frac{MV^2}{t\ F}$

3. RESULTS

The material applied in the numerical simulation of the cages was SAE 4140. It is a chromium-molybdenum alloy steel that has impact resistance. Figure 5 depicts the stress-strain curve of the material, as furnished by the material supplier.

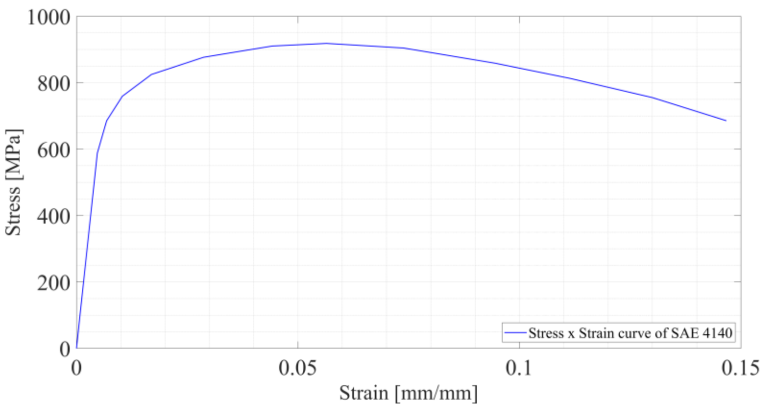
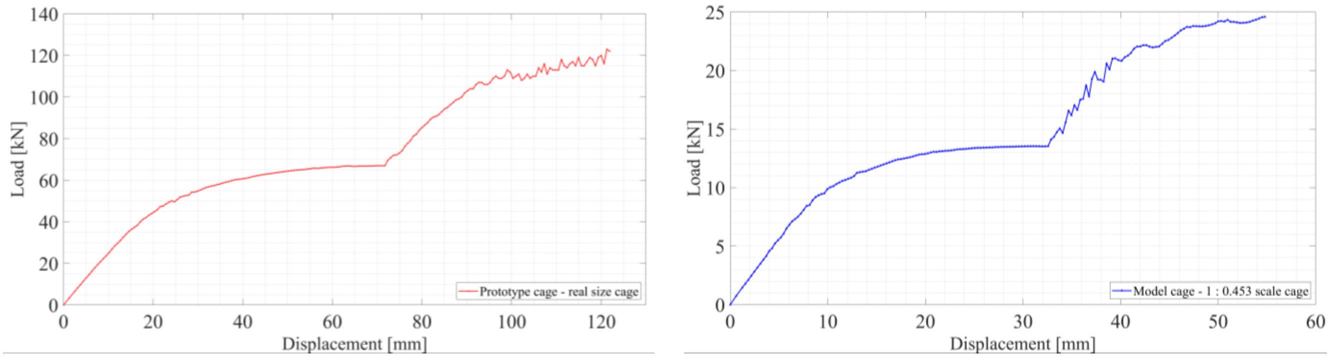


Figure 5. Stress-strain curve of SAE 4140.

The curve *Force (kN) x Displacement* (mm) represents the main result of interest after the crushing simulation of both cages, i. e., Model and Prototype. Figure 6 illustrates the behavior of both cages under a crushing force of 58,8 kN and 26,6 kN, respectively. As discussed in Section 2 the magnitude of such a force follows what is prescribed in the standard (Prototype) and the established scale 1: 0.453 (Model).



(a) (b)

Figure 6. Force x Displacement for the Prototype (a) and Model (b).

It is worth noticing that, as discussed in section 2, the strength of the Prototype cage was beyond the limit force (58,8 kN) without extrapolating the limit for the intrusion (127 mm). Force and displacement behaved as expected for the Model cage; we mean the coordinate pair (57.5 mm; 26.6 kN) has been reached.

The (FMVSS 216a, 2009) presents how to obtain the force applied in vehicle roof according to its mass, section 1. Thus, to corroborate to the methodology proposed, Figure 7 illustrates the similitude theory applied to the Prototype force's values, which show an acceptable fit. Actually, these values were adjusted by multiplying them by λ_t^2. For this, the ratio F_M/F_P between the forces of Model and Prototype and the ratio $1/\lambda_t$ to the gravity acceleration were applied (Jones, 1997). Moreover, the ratio between the metal plate displacement considering Model and Prototype (Table 1) was applied in the Prototype to obtain a similar intrusion response.

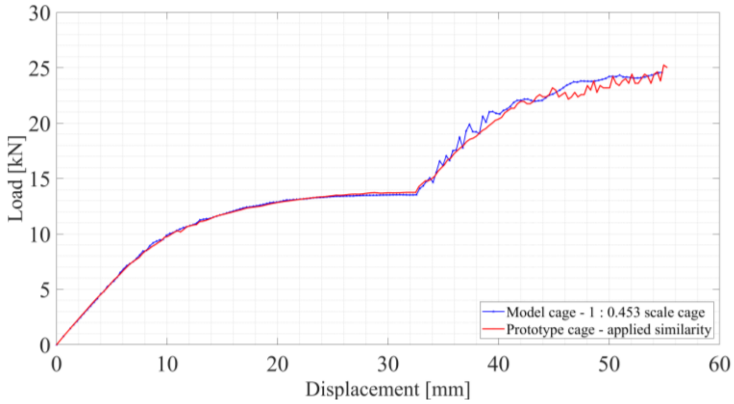


Figure 7. Force curve fitted from Prototype to Model.

The authors also checked the influence of the metal plate size in the Model cage simulations, considering the actual size of the plate according to the (FMVSS 216a, 2009) to exert force on it. Results showed that there was no influence, as depicted in Figure 8. We believe that the metal plate represents the soil that the vehicle roof impacts in a real event. The standard provides a length of 254 mm after column A (Figure 1) as a way of representing this. Based on such a result, the number of variables, **n**, in Table 1 reduces to 12.

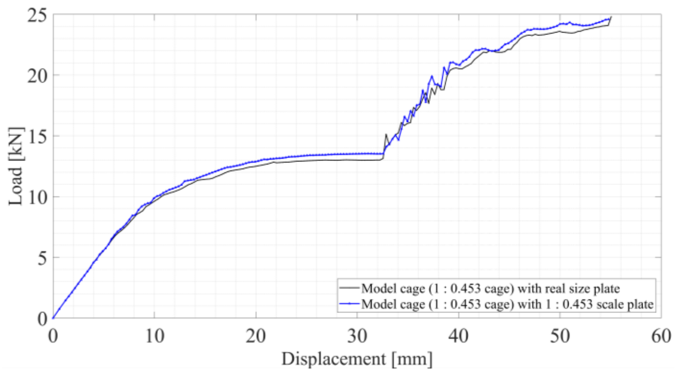


Figure 8. Force x displacement curve acting on the Model under metal plate on scale and actual dimensions.

4. CONCLUSION AND FUTURE WORKS

The use of the commercial package of finite elements to carry out the study on the similitude between vehicle roofs seems to be suitable because it considers the non-linear behavior of the cage structure given an impact event through a quasi-static simulation according to the American Standard FMVSS 216a. At these crash incidents, the intrusion is one of the most important parameters because it can cause serious injuries or even death to a vehicle's occupants.

Thus, this paper highlighted the results of Force x Displacement based on a numerical study considering most of the (FMVSS 216a, 2009) requirements in a reduced model designed according to the Buckingham π theorem. The findings depict the suitable accordance between the results of both models. This way, the methodology represents costing savings concerning the crushing strength testing of vehicle roofs.

The authors intend to treat the dynamic aspects for future works, i.e., the inertia and strain rate effects. However, under these circumstances, a more broadly and robust study should be carried on. Some researchers have shown that sometimes the results could be inconclusive due to the dynamic aspects.

5. ACKNOWLEDGEMENTS

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