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Energy absorption of vibrating sport equipment used for testing athlete performance

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Abstract. This study investigates the stability of a doubly-curved electrical shell structure under dynamic impact loads using both theoretical and analytical methods. The curved electrical shell is designed to absorb energy from deformation and is subjected to loads from a spherical impactor with various boundary conditions. The shell's behavior is mathematically modeled using von Kármán shell theory to derive the displacement field, while its electrical characteristics are described by Maxwell's equations. The mechanical model assumes linear elastic behavior, and the impactor's contact interactions are governed by Hertz's contact law. The influence of external loads and boundary conditions on the internal stress distribution of the shell is analyzed using the principle of energy conservation. In addition to the analytical approach, a finite element model is developed using the Abaqus dynamic/explicit package, allowing for a comparison between analytical and numerical results. The findings are presented through a parametric study that examines the effects of geometry, material properties, and boundary and loading conditions. This parametric analysis highlights the optimal conditions for ensuring the stability of the curved electrical shell structure.

Keywords: concrete disk; instability; nanocomposite reinforcement; non-classical boundary conditions; stability

1. Introduction

Curved composite shell structures are innovative engineering solutions that combine aesthetic appeal with exceptional structural performance (Arefi 2018, Tornabene *et al*. 2022a). These lightweight yet strong forms are characterized by their curved geometries, which efficiently distribute loads and enhance stiffness. Made from advanced composite materials, such as fiberglass or carbon fiber reinforced polymers, these shells offer advantages like corrosion resistance and reduced maintenance (Tian *et al*. 2023, Yang *et al*. 2024). Their applications range from architectural elements to aerospace components, making them a vital area of research and development in modern engineering (Ge *et al*. 2023, Habibi *et al*. 2024, He *et al*. 2024, Huang *et al*. 2024, Jin *et al*. 2024, Li *et al*. 2024, Man *et al*. 2024, Wang *et al*. 2024b, c, Zhang *et al*. 2024b, c, Zhao *et al*. 2024). The unique interplay of form and function in curved composite shells not only pushes the boundaries of design but also addresses the demands for sustainability and efficiency in various industries (Mahapatra *et al*. 2016, Mehar *et al*. 2017, Tornabene *et al*. 2022b).

Analytical modeling methods for shell structures encompass a range of approaches tailored to capture the unique geometrical and mechanical behavior of these components (Dai *et al*. 2023a, b, Gu *et al*. 2023, Li *et al*. 2023, Peng *et al*. 2023, Sabzevari *et al*. 2023, Shariati *et al*. 2023, Xiang *et al*. 2023, Yang *et al*. 2023, Zhang *et al*. 2023a, b, Zhao *et al*. 2023, Zheng *et al*. 2023). Among the most widely used is the von Kármán shell theory (Lou and He 2015, Karami *et al*. 2019, Tang and Ding 2019, Esayas and Kattimani 2021, Hasheminejad and Jamalpoor 2022, Hou *et al*. 2022), which addresses large deformations and nonlinear effects, making it suitable for analyzing the response of shells under various loading conditions. Another prominent method is the Love-Kirchhoff theory, which assumes small deflections and linear elasticity, often applied in simpler cases where computational efficiency is desired. The Reissner-Mindlin theory extends these concepts by accounting for transverse shear deformation, enhancing accuracy in thick shell applications (Li *et al*. 2020a). Additionally, variational methods, such as the Ritz method, provide powerful tools for solving complex boundary value problems by minimizing potential energy (Hamed *et al*. 2010, Qu *et al*. 2013, Carrera and Zozulya 2022, Hasheminejad and Jamalpoor 2022). Each of these methods has its advantages and limitations, with the choice

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often depending on the specific characteristics of the shell structure being analyzed, such as thickness, material properties, and loading scenarios (Habibi *et al*. 2016, 2018a, b, 2019b, d, e, Ebrahimi *et al*. 2019a, Esmailpoor Hajilak *et al*. 2019, Pourjabari *et al*. 2019, Safarpour *et al*. 2019a, Zhu *et al*. 2022, Dai *et al*. 2023b, Lu *et al*. 2023a, b, Ma *et al*. 2023, Tang *et al*. 2023, Wang *et al*. 2023, Zheng *et al*. 2023).

Curved shell structures can be subjected to impact loads under various conditions, each presenting unique challenges and responses (Khalili *et al*. 2011, Liu *et al*. 2022). These scenarios often occur in applications such as aerospace components, automotive parts, and architectural elements where dynamic forces are prevalent. For instance, during a sudden collision or a drop impact, the structure experiences localized stress concentrations that can lead to deformation or failure (Zhang and Yu 2012, Sobhani *et al*. 2023). The curvature of the shell plays a crucial role in how the load is distributed across the surface, affecting both the structural integrity and energy absorption capacity. Additionally, factors such as the material properties of the shell, the speed and mass of the impacting object, and the boundary conditions—such as whether the shell is fixed, simply supported, or free—significantly influence the impact response. Understanding these conditions is essential for designing curved shell structures that can withstand dynamic loads while maintaining performance and safety (Fazaeli *et al*. 2016, Habibi *et al*. 2017, 2019a, c, Safarpour *et al*. 2018, 2019b, 2020, Alipour *et al*. 2020, Ebrahimi *et al*. 2020a, Ghazanfari *et al*. 2020, Chen *et al*. 2022).

Integrating piezoelectric materials into shell structures enhances energy absorption by enabling the conversion of mechanical energy into electrical energy (Alibeigloo and Pasha Zanoosi 2017, Arefi 2018, Eftekhar *et al*. 2018). When subjected to impacts or vibrations, these shells generate electrical charges, facilitating applications like energy harvesting and structural health monitoring. This functionality improves energy dissipation and reduces damage risk under dynamic loads (Mehralian *et al*. 2016, Ninh and Bich 2018). Additionally, the adaptability of piezoelectric materials allows for property tuning, optimizing performance across varying conditions. Overall, piezoelectric shells represent a promising advancement for improving energy efficiency and resilience in challenging environments (Ebrahimi *et al*. 2019b, c, 2020b, Hashemi *et al*. 2019, Moayedi *et al*. 2019, 2020a, b, Mohammadgholiha *et al*. 2019, Mohammadi *et al*. 2019, Habibi *et al*. 2020, Oyarhossein *et al*. 2020, Shariati *et al*. 2020a, b, Shokrgozar *et al*. 2020).

The stability of shell structures under impact loads is a critical consideration in engineering design, particularly for applications where dynamic forces are prevalent. When subjected to sudden impacts, such as collisions or projectile strikes, shell structures must withstand localized stresses that can lead to buckling, deformation, or even catastrophic failure. The inherent curvature of the shell enhances its load distribution capabilities, but it also introduces complex interactions between bending and membrane stresses that can compromise stability. Factors such as the material properties, thickness of the shell, and boundary conditions play significant roles in determining how the structure reacts to these loads. Additionally, the dynamic response of the shell is influenced by its geometry and the nature of the impact, necessitating a thorough understanding of the shell's behavior under transient conditions. Engineers often employ analytical and numerical methods to predict stability and develop design strategies that enhance the resilience of shell structures, ensuring they can absorb energy effectively while maintaining structural integrity. Habibi *et al*. (2019e) Habibi *et al*. (2019) conducted a comprehensive investigation into the critical voltage and frequency characteristics of a graphene nanoplatelets (GNP) composite cylindrical nanoshell integrated with a piezo-electric actuator. Their study delved into several critical aspects, including the effects of the piezoelectric layer, the distribution of GNPs, size effects, and the natural frequency of the system on the critical voltage required for effective operation. By systematically analyzing these variables, the authors were able to establish that each factor plays a significant role in determining the performance and efficiency of the nanoshell. The findings revealed that variations in the distribution and concentration of GNPs directly affect the electrical properties and mechanical response of the nanoshell, leading to changes in the critical voltage necessary to activate the piezoelectric actuator (Hashemi *et al*. 2019, Al-Furjan *et al*. 2020c, d, e, f, Bai *et al*. 2020, Cheshmeh *et al*. 2020, Li *et al*. 2020b, Lori *et al*. 2020, Najaafi *et al*. 2020, Shariati *et al*. 2020c, Xiong *et al*. 2020, Guo *et al*. 2021b, Liu *et al*. 2021a, Tang *et al*. 2023, Huang *et al*. 2024, Wang *et al*. 2024a, Zhiqiang *et al*. 2024). Additionally, size effects were shown to influence the natural frequency of the nanoshell, further impacting its dynamic behavior. The study highlighted the intricate interplay between mechanical and electrical properties in nanoscale materials, emphasizing the importance of optimizing these factors to enhance performance. These insights are particularly valuable for advancing the fields of materials science and nanoelectromechanical systems (NEMS), where the integration of piezoelectric components with nanomaterials is increasingly being explored for innovative applications. By providing a deeper understanding of the parameters that influence critical voltage and frequency, this research lays the groundwork for the design of more efficient nanoshells and other advanced nanostructures. Future research could build on these findings by exploring the impact of additional variables, such as environmental conditions and fabrication techniques, on the performance of GNP composite nanoshells, thereby expanding the potential applications in sensors, actuators, and energy harvesting systems.

Energy absorption (Adamian *et al*. 2020, Al-Furjan *et al*. 2020a, b, Li *et al*. 2020c, Liu *et al*. 2020a, b, 2021b, Wang *et al*. 2020, Zare *et al*. 2020, Zhou *et al*. 2020, Dai *et al*. 2021a, Dai *et al*. 2021b, Guo *et al*. 2021a, Habibi *et al*. 2021, He *et al*. 2021, Huang *et al*. 2021, Shao *et al*. 2021a, Wu and Habibi 2021, Zhang *et al*. 2021, Kong *et al*. 2022) in shell structures can be significantly enhanced through the integration of piezoelectric materials, which exhibit unique properties that allow them to convert mechanical energy into electrical energy and vice versa (Ahmadi *et al*. 2019, Guo *et al*. 2021b, Shao *et al*. 2021b). When piezoelectric shells are subjected to mechanical loads, such as impacts or vibrations, they not only deform but also generate electrical charges that can be harnessed for various applications, including energy harvesting and structural health monitoring (Shariati *et al*. 2020d, Zhang *et al*. 2022). This dual functionality makes piezoelectric shells particularly advantageous in engineering applications where both energy absorption and monitoring are critical. The presence of the piezoelectric material can improve the shell's ability to dissipate energy, mitigating the effects of dynamic loads and reducing the risk of damage. Additionally, the adaptability of piezoelectric materials allows for the tuning of their properties through changes in voltage or frequency, further optimizing their performance under varying loading conditions. Consequently, the use of piezoelectric shells represents a promising avenue for developing advanced structures that enhance energy efficiency and resilience in demanding environments (Al-Furjan *et al*. 2020g, Ali *et al*. 2022).

This study investigates the stability of a doubly-curved electrical shell structure under dynamic impact loads, employing both theoretical and analytical methods to provide a comprehensive understanding of its behavior. The design of the curved electrical shell emphasizes its capability to absorb energy through deformation, making it particularly suitable for applications subject to impact forces from a spherical impactor. The analysis considers various boundary conditions to evaluate how these constraints influence the structural response. To mathematically model the shell's behavior, von Kármán shell theory is utilized to derive the displacement field, enabling an accurate representation of large deformations. Concurrently, the electrical characteristics of the shell are articulated through Maxwell's equations, linking the mechanical and electrical responses. The mechanical model assumes linear elastic behavior to simplify calculations, while the contact interactions between the impactor and the shell are governed by Hertz's contact law, which captures the nuances of contact mechanics. The study further delves into the influence of external loads and boundary conditions on the internal stress distribution within the shell, leveraging the principle of energy conservation to understand how energy is transferred and dissipated during impacts. To validate the analytical findings, a finite element model is developed using the Abaqus dynamic/explicit package, facilitating a rigorous comparison between analytical and numerical results. This dual approach not only reinforces the reliability of the findings but also enhances the predictive capabilities of the model.

2. Formulations and models

2.1 Contact and impact law

The most well-known contact model is the Hertz model, which provides a foundational understanding of contact mechanics between deformable bodies. This model, along with its modified versions that account for the nonlinearity of loading and unloading conditions, has been extensively studied and applied in various engineering fields (Yang and Sun 1982). The Hertz model assumes that the contact surfaces are smooth and elastic, leading to a quadratic relationship between the applied load and the resulting contact deformation. However, real-world applications often present complexities that necessitate modifications to this basic model, particularly in cases involving varying load magnitudes and material properties. Building on the Hertz framework, the following formulation is utilized to calculate the contact force between the ball impactor and the shell structure during the loading phase. This formulation incorporates both the initial contact conditions and the subsequent deformations that occur as the load is applied. Specifically, the contact force $F_s$can be expressed as a function of the effective radius and the indentation depth, allowing for an accurate representation of the force distribution across the contact area.

Moreover, this model also accommodates the non-linear response of materials under varying loads, providing a more realistic depiction of how the impactor interacts with the shell structure. As the loading condition progresses, the formulation accounts for both the elastic deformation during the loading phase and the potential plastic deformation or recovery during unloading. By integrating these aspects, the modified Hertz model effectively captures the dynamic behavior of the contact interface, facilitating a deeper understanding of energy transfer and dissipation mechanisms during impact events. This comprehensive approach not only enhances the accuracy of simulations but also informs the design of shell structures that need to withstand complex loading scenarios, thereby improving their performance and durability in practical applications.

$$F_c(t) = C_i {d_\theta}^{1.5}(t), d_\theta = S_i(t) - W_c(t) \tag{1}$$

The time dependent parameters$d_\theta(t)$andC_i represents the contact indentation distance and the contact stiffness, respectively. The parameters $S_i(t)$ and $W_c(t)$ are time-dependent deflection of the mid-surface of the structure and impactor travel distance during contact. The contact stiffness is calculates using the nonlinear relation below as calculated using Young modulus and radius of impactor E_i and R_i and Poisson's ratio of ball and shell structure ν_i and ν_s, respectively (Dai *et al*. 2023b, Gu *et al*. 2023, Li *et al*. 2023, Lu *et al*. 2023a, b, Ma *et al*. 2023, Peng *et al*. 2023, Sabzevari *et al*. 2023, Shariati *et al*. 2023, Tang *et al*. 2023, Wang *et al*. 2023, Yang *et al*. 2023, Zhang *et al*. 2023a, b, c, 2024a, d, Zhao *et al*. 2023, Zheng *et al*. 2023, Guo *et al*. 2024, Liang *et al*. 2024, Song *et al*. 2024, Xiao *et al*. 2024, Yin *et al*. 2024, Yu *et al*. 2024, Zisong and Habibi 2024):

$$C_i = \frac{4}{3}\sqrt{R_i}\left(\frac{1-\nu_i^2}{E_i} + \frac{1-\nu_s^2}{E_s}\right)^{-1} \tag{2}$$

Fig. 1 A schematic view of the electrically curved screen on viscoelastic substrate exposed to a low-velocity impact

Moreover, during the period of unloading, the contact force could be obtained using the relation below (Song *et al*. 2019):

$$F_c(t) = F_s\left(\frac{d_\theta - d_{\theta_0}}{d_{\theta_m} - d_{\theta_0}}\right)^{2.5} \tag{3}$$

d_{θ_0} is set to zero in the current study.

2.2 Governed equations and BCs using principle of energy conservation

Fig. 1 shows the geometry, boundary condition and coordinate system of shell structure and the impactor position.

As mentioned in the introduction section, several models exist for mechanical behavior of shells. The displacement components of the shell according to the FSDT are given in Ref. (Arefi and Amabili 2020). Based on this displacement field assumption, the nonlinear strain components are calculated in Eq. (5) according to Von Karman formulations:

$$\begin{aligned}
\varepsilon_{xx} &= \frac{1}{1+\frac{z}{R_1}}\left(\frac{\partial u}{\partial x} + \frac{w_0}{R_1} + \frac{1}{2}\left(\frac{\partial w_0}{\partial x}\right)^2\right), \\
\varepsilon_{yy} &= \frac{1}{1+\frac{z}{R_2}}\left(\frac{\partial v}{\partial y} + \frac{w_0}{R_2} + \frac{1}{2}\left(\frac{\partial w_0}{\partial y}\right)^2\right), \\
\gamma_{xz} &= \frac{1}{1+\frac{z}{R_1}}\left(\frac{\partial w}{\partial x} - \frac{u}{R_1}\right) + \frac{\partial u}{\partial z}, \\
\gamma_{yz} &= \frac{\partial v}{\partial z} + \frac{1}{1+\frac{z}{R_2}}\left(\frac{\partial w}{\partial y} - \frac{v}{R_2}\right), \\
\gamma_{xy} &= \frac{1}{1+\frac{z}{R_1}}\frac{\partial v}{\partial x} + \frac{1}{1+\frac{z}{R_2}}\frac{\partial u}{\partial y} + \frac{1}{1+\frac{z}{R_1}}\frac{1}{1+\frac{z}{R_2}}\frac{\partial w_0}{\partial x}\frac{\partial w_0}{\partial y}
\end{aligned} \tag{3-1}$$

3. Strain-Stress of the structure:

The most general linear constitutive model for elastic anisotropic elastic structures are given below (Reddy 2003):

$$\begin{aligned}
\sigma_{rq} &= c_{rqkl}\varepsilon_{kl} - q_{nrq}H_n - e_{srq}E_s - \beta_{rq}\Delta T \\
D_r &= e_{rkl}\varepsilon_{kl} + d_{rn}H_n + s_{rs}E_s + p_r T \\
B_r &= q_{rkl}\varepsilon_{kl} + r_{rn}H_n + d_{rs}E_s + \lambda_r \Delta T
\end{aligned} \tag{4}$$

We neglect the effects of temperature in the current study. The parameter E_i could be calculated based on the potential of electric energy Φ as follows:

$$\begin{aligned}
&E_1 = -\frac{1}{1+\frac{z}{R_1}}\frac{\partial \Phi}{\partial x}, \qquad E_2 = -\frac{1}{1+\frac{z}{R_2}}\frac{\partial \Phi}{\partial y}, \\
&E_3 = -\frac{\partial \Phi}{\partial z}, \qquad H_1 = -\frac{1}{1+\frac{z}{R_1}}\frac{\partial \Psi}{\partial x}, \\
&H_2 = -\frac{1}{1+\frac{z}{R_2}}\frac{\partial \Psi}{\partial y}, \quad H_3 = -\frac{\partial \Psi}{\partial z}
\end{aligned} \tag{5}$$

Based the relation given in Ref. (Wang 2002) the parameter $\Phi(x,y,z,t)$ is as below:

$$\begin{aligned}
\Phi &= \tfrac{2z\phi_0}{h} - cos(\pi z/h)\phi(x,y,t), \\
\Psi &= \tfrac{2z\psi_0}{h} - cos(\pi z/h)\psi(x,y,t)
\end{aligned} \tag{6}$$

where ϕ_0 is the external initial electric and $\beta = \pi/h$.

The variation of total energy of the structure composed of variation of kinetic energy δT, potential energy δU and external energies δW_i:

$$\int_{t_1}^{t_2} (\delta T - \delta U + \delta W_1 + \delta W_2 - \delta W_3)dt = 0 \tag{7}$$

The variation kinetic energy could be given as: in Eq. (8).

$$\begin{gathered}
\delta T = \int_V \rho\left(\frac{\partial u}{\partial t}\frac{\partial \delta u}{\partial t} + \frac{\partial v}{\partial t}\frac{\partial \delta v}{\partial t} + \frac{\partial w}{\partial t}\frac{\partial \delta w}{\partial t}\right)dV: \\
\int_A \left\{ \begin{array}{c}
\left[I_0^{\gamma^3\lambda}\frac{\partial^2 u_0}{\partial t^2} + I_1^{\gamma^2\lambda}\frac{\partial^2 u_1}{\partial t^2}\right]\delta u_0 \\
+\left[I_0^{\gamma\lambda^3}\frac{\partial^2 v_0}{\partial t^2} + I_1^{\gamma\lambda^2}\frac{\partial^2 v_1}{\partial t^2}\right]\delta v_0 \\
+\left[I_0^{\gamma\lambda}\frac{\partial^2 w_0}{\partial t^2}\right]\delta w_0 \\
+\left[I_1^{\gamma^2\lambda}\frac{\partial^2 u_0}{\partial t^2} + I_2^{\gamma\lambda}\frac{\partial^2 u_1}{\partial t^2}\right]\delta u_1 \\
+\left[I_1^{\gamma\lambda^2}\frac{\partial^2 v_0}{\partial t^2} + I_2^{\gamma\lambda}\frac{\partial^2 v_1}{\partial t^2}\right]\delta v_1
\end{array} \right\} dxdy
\end{gathered} \tag{8}$$

where the parameter $I_i^{\gamma^\eta\lambda^\Delta}$ is:

$$I_i^{\gamma^\eta\lambda^\Delta} = \int_z \rho(1+\frac{z}{R_1})^\eta(1+\frac{z}{R_2})^\Delta z^i\,dz \tag{9}$$

4. Galerkin's method

Galerkin's method is a powerful technique used for solving differential equations, particularly in the context of structural analysis and finite element methods. This approach involves selecting a set of trial functions that approximate the solution, often chosen to satisfy the boundary conditions of the problem. By weighting the residuals of the differential equations with these trial functions and integrating over the domain, Galerkin's method transforms the original problem into a system of algebraic equations. Compared to other methods, such as finite difference methods or traditional perturbation techniques, Galerkin's method offers greater flexibility in handling complex geometries and boundary conditions. While finite difference methods discretize the domain into a grid and approximate derivatives, potentially limiting accuracy and applicability, Galerkin's method can utilize polynomial or other functional forms that provide higher-order approximations. Additionally, methods like the Rayleigh-Ritz approach, which also employs trial functions, focus on energy minimization and are generally used for linear problems. In contrast, Galerkin's method is applicable to a broader range of nonlinear and time-dependent problems, making it a versatile choice in computational mechanics and applied mathematics.

5. Results' verification with a FE package:

For shape mode analysis and further validation of smart shells, a finite element (FE) investigation has been conducted using an ABAQUS model. This model employs the C3D8R solid element, which features an 8-node element. A robust bonding interlayer is incorporated to ensure effective load transfer between the layers of the smart sandwich shell, enhancing its overall structural integrity. Additionally, boundary conditions (BCs) are applied to both ends of the shell to simulate real-world constraints and loading scenarios. To accurately determine the associated eigenvalues and buckling modes, the Lanczos eigensolver is utilized, which is known for its efficiency in solving large eigenvalue problems. The displacement vector can be calculated using specific formulations that account for the shell's geometry and material properties. This comprehensive approach not only facilitates a deeper understanding of the shell's vibrational characteristics but also assists in optimizing its design for various applications, including aerospace and civil engineering. Through this FE analysis, insights can be gained regarding the impact of geometric and material parameters on the shell's performance, contributing to the development of more resilient and efficient smart shell structures. Then, the harmonic vibration relation would be as:

$$([K] - \omega^2[M])\{\delta\} = 0 \tag{10}$$

We have to care about mesh convergency, if we are looking for a precise FE simulation. Thereby, the elements' number would be raised, if the structure's frequency does not alter and the optimum elements' number is chosen. the convergency occurs, while there are more than 53,144 elements.

Table 1 Validation of computational results with FE ones

b/a	Numerical results for $\Phi = 0$	FEM results for $\Phi = 0$	Numerical results for $\Phi = 0.5$	FEM results for $\Phi = 0.5$
3	300.702	308.209	214.204	212.903
4	141.345	140.036	94.729	94.454
5	79.385	79.698	54.141	54.510
6	52.314	50.650	35.391	35.252
7	36.569	35.879	24.605	24.739

The verification analysis between the computational and FE models has been reported in Table 1. Since it can be observed, the maximum pertained nuances between the FEM and numerical outcome is less than 4%.

5.1 Validation study

To validate the results obtained in the current study, the dimensionless frequency of a plate—essentially a shell structure with infinite radii ($R_1 = R_2 = \infty$) is analyzed across various modes of vibration, as presented in Table 2. This comparison serves as a benchmark, allowing for an assessment of the accuracy and reliability of the numerical models employed. The findings indicate that the maximum difference between the results from this study and the established data is approximately 3%. This level of discrepancy is considered satisfactory, suggesting that the numerical methods and assumptions made in the analysis are robust. Such validation is crucial, as it reinforces the credibility of the findings and provides confidence in the applicability of the model to more complex geometries and loading conditions. Furthermore, this small variance highlights the effectiveness of the chosen methodologies in capturing the dynamic behavior of shell structures. Overall, this validation not only supports the current study's conclusions but also lays a solid foundation for future investigations into the vibrational characteristics of more intricate smart shell designs.

5.2 Parametric results

The effects of Young's module of the impactor to principle young module ratio (E_i/c_{11}) on the energy of impactor during the process of impact is provided in Fig. 2. Accordingly, as the E_i/c_{11} factor raises, the time of contact is reduced. Furthermore, reducing the impactor energy would be swifter due to raising E_i/c_{11} factor. Moreover, before time contact, energy of impactor and E_i/c_{11} element would have an indirect relation, Although, the relation would be a direct relation, after time contact.

The indentation of the electrically curved screen during the impact process is analyzed over time and is depicted in Fig. 3 for various values of the E_i/c_{11} parameter. Initially, the results indicate that there are no noticeable effects from the E_i/c_{11} parameter on the indentation of the electrically curved screen. This observation suggests that, at the outset

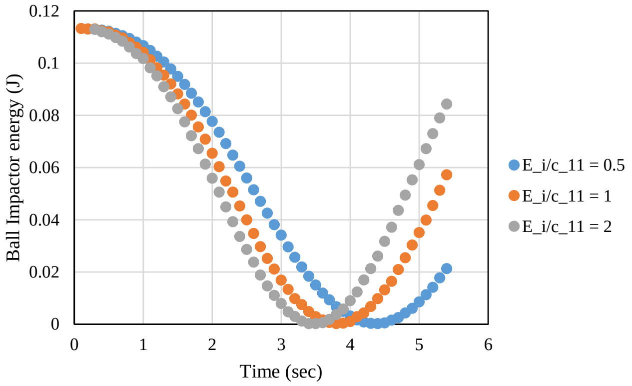

Fig. 2 Effects of E_{im}/c_{11} factor on the impactor energy

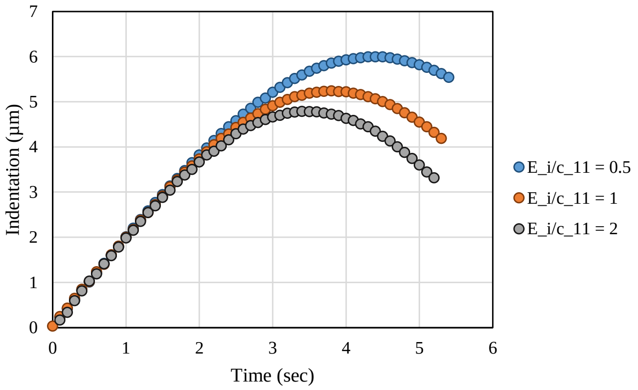

Fig. 3 Effects of E_{im}/c_{11} parameter on the indentation of the electrically curved screen

of the impact event, the material's stiffness and the interaction dynamics are not significantly influenced by this specific parameter, allowing the system to respond predominantly to the initial impact forces. As the impact process progresses, however, a distinct trend emerges. By the end of the impact event, it becomes evident that as the E_i/c_{11} parameter decreases, the indentation of the screen increases significantly. This increase in indentation correlates with a reduction in the stiffness of the material represented by the E_i/c_{11} ratio, indicating that softer materials are more susceptible to greater deformation under the same impact conditions. Consequently, the maximum indentation observed occurs later in the impact duration as the E_i/c_{11} factor diminishes. This delayed response highlights the dynamic behavior of the system, where the effects of material properties become more pronounced as time progresses. The correlation between reduced stiffness and increased indentation emphasizes the importance of material selection in designing electrically curved screens, particularly in applications where impact resistance is critical. By understanding how the E_i/c_{11} parameter influences indentation over time, engineers can optimize material properties to enhance the performance and durability of such structures under dynamic loading conditions. This analysis not only aids in predicting deformation patterns but also informs the development of advanced materials tailored for specific impact scenarios, ensuring better energy absorption and structural integrity.

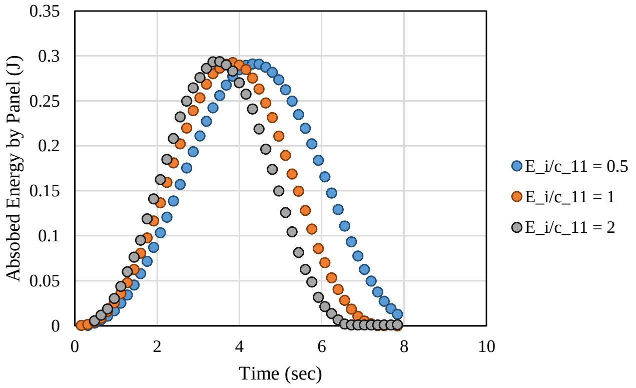

Fig. 4 Effects of E_{im}/c_{11} parameter on the absorbed energy of the electrically curved screen

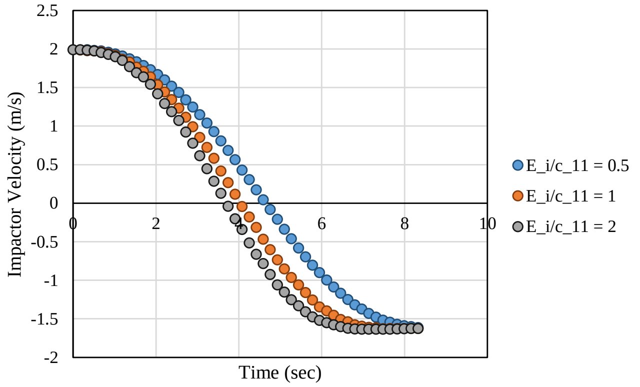

Fig. 5 Effects of E_{im}/c_{11} parameter on the indenter velocity

The panel's absorbed energy respect to time domain for a range of E_i/c_{11} parameter is analyzed in Figure 4. Then, the time of contact and E_i/c_{11} parameter are directly related to each other. Next, as the E_i/c_{11} parameter is reduced, the highest absorbed energy would not alter, however it occurs at the greater time.

Indenter speed respect to time domain is provided in Fig. 5 for multifarious E_i/c_{11}. Since it would be observed, at the ultimate and primary impact process time, it doesn't alter in the indenter speed due to raising the E_i/c_{11} ratio, however, at the middle time, the indenter speed could be reduced due to soaring the E_i/c_{11} ratio. Basically, by spending the process time, the indenter speed would be decreased instantly. Moreover, the relation between indenter speed and time becomes remarkable due to raising E_{im}/c_{11} parameter.

The contact force as a function of time for a wide range of the E_i/c_{11} parameter is thoroughly analyzed in Fig. 5. The results reveal that the time of contact and the E_i/c_{11} parameter exhibit an indirect relationship. Specifically, as the E_i/c_{11} parameter decreases, the maximum contact force experienced by the system also declines, however, this peak force occurs later in the impact process. This delay signifies that softer materials, represented by lower E_i/c_{11} values, take longer to reach their maximum contact force due to their increased deformation under load. A closer examination of Fig. 5 indicates a nuanced relationship

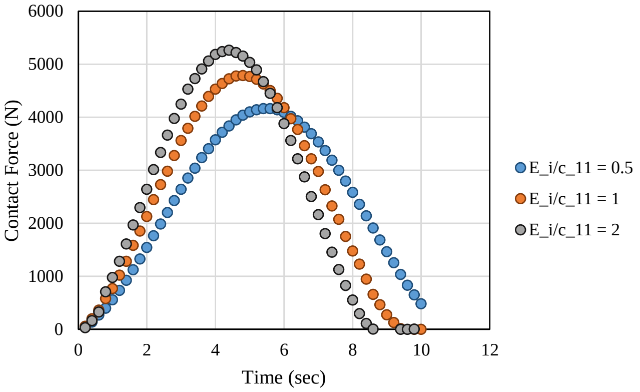

Fig. 6 Effects of E_i/c_{11} parameter on the contact force

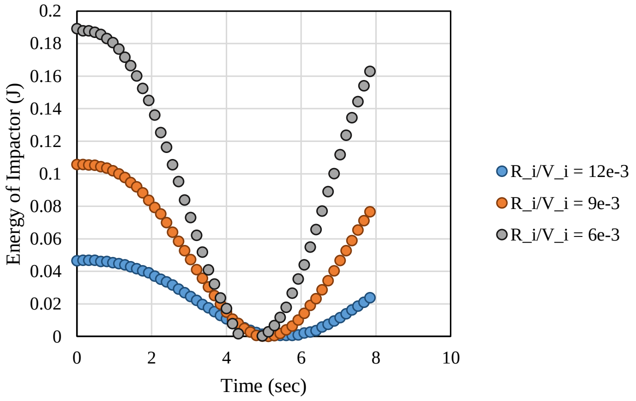

Fig. 7 Effects of R_i/V_i factor on the impactor energy during the process of impact

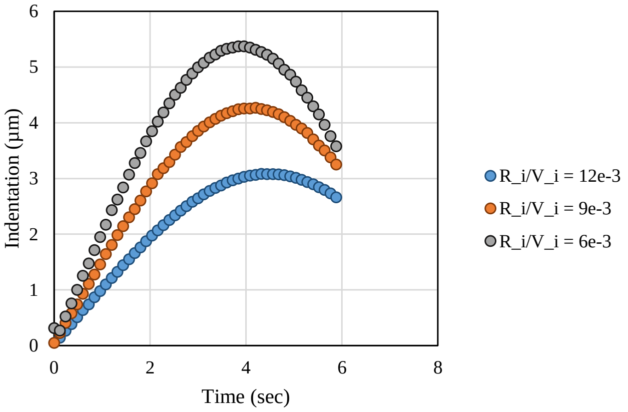

Fig. 8 Effects of R_i/V_i factor on the indentation of the electrically curved screen during the process of impact

between contact force variations and the E_i/c_{11} parameter. Initially, there is a direct correlation, where increases in E_i/c_{11} lead to higher contact forces at earlier time intervals. However, this relationship shifts to an indirect correlation at a specific time point, which extends as the E_i/c_{11} factor improves. This transition suggests that while stiffer materials can withstand higher forces initially, their capacity to maintain this force diminishes as the impact progresses, especially when softer materials continue to deform. This dynamic interplay highlights the importance of considering the E_i/c_{11} parameter in the design of structures intended to absorb impacts. Understanding how variations in this parameter influence the timing and magnitude of contact forces can aid engineers in optimizing material selection and structural configurations. By strategically choosing materials with specific E_i/c_{11} ratios, designers can effectively tailor the contact behavior to enhance energy dissipation and reduce the risk of structural failure during dynamic loading events. Overall, this analysis underscores the significance of material properties in impacting the contact mechanics of electrically curved screens, offering valuable insights for future research and applications in impact-resistant design.

The effect of the radius-to-velocity ratio of the impactor (R_i/V_i) on the impactor energy during the impact process is illustrated in Fig. 7. As observed, an increase in the radius-to-velocity ratio results in a corresponding increase in contact time during the impact event. This extended contact time allows for a more gradual transfer of energy between the impactor and the target structure, leading to a more sustained interaction. Conversely, when the radius-to-velocity ratio decreases, the impactor energy dissipates more rapidly, indicating a swift reduction in the energy transferred to the structure. This rapid energy dissipation occurs because a smaller radius relative to velocity results in a sharper impact, leading to a more instantaneous transfer of energy. Notably, there exists an indirect relationship between the radius-to-velocity ratio of the impactor and the impactor energy, as the radius-to-velocity ratio increases, the impactor energy tends to stabilize, while a decrease in this ratio corresponds to an increase in energy concentration during the impact. This nuanced relationship underscores the importance of considering the geometry and motion of the impactor in the design of structures intended to withstand dynamic loads, as it directly influences both the duration of the impact and the energy absorption capabilities of the material. Understanding these dynamics can aid in optimizing materials and structural configurations for improved performance under impact conditions.

The effect of radius to velocity ratio of impactor (R_i/V_i) on the impactor energy during the impact process time is provided in Fig. 7. Then, as the radius to speed ratio of impactor raises, the contact time improves. Moreover, reducing the impactor energy becomes swifter due to decreasing the radius to velocity ratio of impactor. Next, radius to speed ratio of impactor and impactor energy have an indirect relation.

The influences of the R_i/V factor on the indentation of the electrically curved screen are thoroughly investigated in Fig. 8 throughout the impact process. A key observation from the diagram is that as the R_i/V parameter increases, there is a corresponding decrease in the indentation experienced by the electrically curved screen. This trend suggests that larger radius-to-velocity ratios contribute to a more distributed contact area, leading to less localized deformation and, consequently, reduced indentation. Moreover, the analysis indicates that the maximum value of indentation occurs at shorter time intervals when the impactor radius is decreased. This finding highlights the critical role of the impactor's geometry in determining the material response during the initial stages of the impact. A

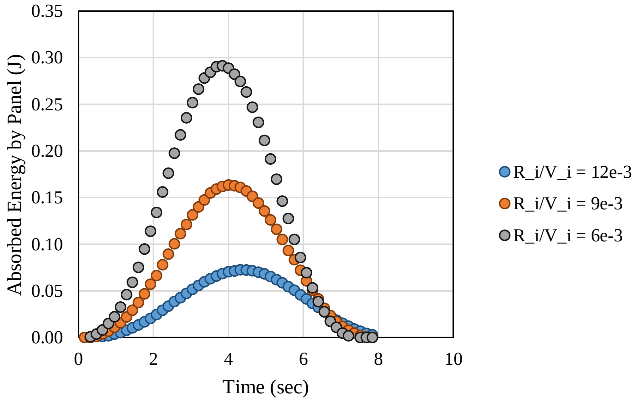

Fig. 9 Effects of R_i/V_i parameter on the indenter velocity during the impact process

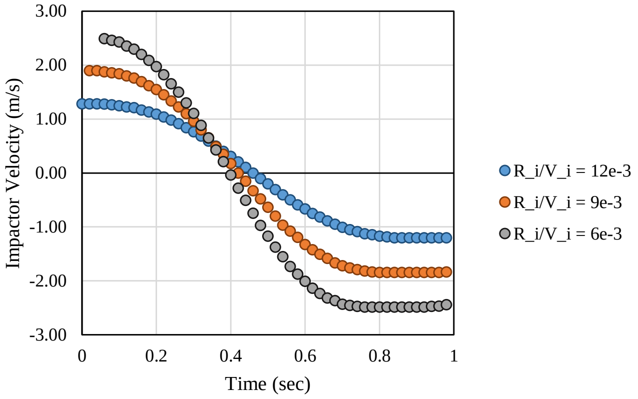

Fig. 10 Effects of R_i/V_i parameter on the indenter velocity during the impact process

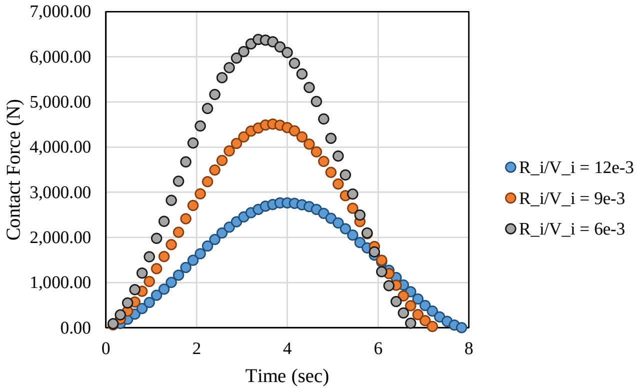

Fig. 11 Effects of R_i/V_i parameter on the contact force during the impact process

smaller radius allows for a sharper, more concentrated impact, resulting in greater deformation at the outset. As the impact progresses, the effects of the R_i/V parameter become more pronounced, illustrating how the dynamic behavior of the shell structure is sensitive to both the size and speed of the impactor. The implications of this relationship are significant for the design and optimization of electrically curved screens, particularly in applications requiring enhanced energy absorption and damage mitigation. By adjusting the radius and velocity of the impactor, engineers can strategically influence the indentation response, potentially improving the performance of the structure under dynamic loading conditions. Understanding this relationship also provides valuable insights into the material selection process, where softer materials may benefit from larger radius-to-velocity ratios to minimize indentation and maximize durability. In summary, the analysis presented in Fig. 8 not only elucidates the direct effects of the R_i/V factor on indentation but also underscores the importance of considering geometrical parameters in the design of impact-resistant structures. Further research could explore the interplay between various loading conditions and material properties to develop comprehensive guidelines for optimizing electrically curved screens in practical applications.

The electrically curved screen's absorbed energy respect to time domain for a broad range of R_i/V_i parameter is analyzed in Fig. 9. Then, as the R_i/V_i parameter decreases, the smart panel's absorbed energy boosts, and the highest absorbed energy occurs at the less time of the process of impact. Furthermore, the indirect relation between absorbed energy of the electrically curved screen and radius of the impactor could change to a direct relation at $Time > 68\mu s$, it means that at the end of the impact process, as the radius to speed ratio of the impactor increases, the curved electrically screen's absorbed energy boosts.

The indenter velocity in relation to the time domain for various radius-to-speed ratios of the impactor is analyzed in Fig. 10. The results clearly indicate a significant point in time, specifically at \(t = 30 \, \mu s\), where the relationship between the indenter velocity and the radius-to-velocity ratio of the impactor shifts from an indirect to a direct correlation. Initially, during the early stages of the impact process, a larger radius-to-velocity ratio appears to negatively influence the indenter's velocity, suggesting that the impactor's geometry plays a crucial role in determining the energy transfer dynamics. However, as the impact process progresses, this negative influence diminishes, and the relationship evolves into a direct effect. This transition implies that over time, the impactor's radius increasingly contributes to enhancing the indenter velocity, possibly due to the way energy is distributed during the impact and the interaction between the materials involved.

This shift in relationship is significant because it highlights how the impact dynamics can evolve, indicating that longer impact durations can mitigate some of the initial adverse effects of a larger impactor radius. Understanding this behavior is essential for optimizing designs in applications involving dynamic impacts, such as in automotive or aerospace engineering, where the impactor characteristics must be carefully selected to achieve desired outcomes in energy absorption and material performance. Ultimately, the insights gained from this analysis can inform future designs of impact-resistant structures by taking into account not only the static properties of the materials involved but also their dynamic responses over the duration of the impact event.

With respect to Fig. 11, the effects of R_i/V_i parameter on the contact force of the electrically curved screen is investigated. Accordingly, for $Time \leq 63\mu$s, there is an indirect relation between R_i/V_i parameter and absorbed energy of the panel, in contrast, at the grater time, the mentioned relation changes to become direct. In addition, as R_i/V_i factor decreases, the maximum absorbed energy of the electrically curved screen increases and happens at the lower time. Fig. 11 shows that due to increasing the radius of the impactor, the absorbed energy of the electrically curved screen, contact time, and maximum absorbed energy of the structure improves.

6. Conclusions

This study investigates the stability of a doubly-curved electrical shell structure under dynamic impact loads using both theoretical and analytical methods. The curved electrical shell is designed to absorb energy from deformation and is subjected to loads from a spherical impactor with various boundary conditions. The shell's behavior is mathematically modeled using von Kármán shell theory to derive the displacement field, while its electrical characteristics are described by Maxwell's equations. The mechanical model assumes linear elastic behavior, and the impactor's contact interactions are governed by Hertz's contact law. The influence of external loads and boundary conditions on the internal stress distribution of the shell is analyzed using the principle of energy conservation. In addition to the analytical approach, a finite element model is developed using the Abaqus dynamic/explicit package, allowing for a comparison between analytical and numerical results. The findings are presented through a parametric study that examines the effects of geometry, material properties, and boundary and loading conditions. This parametric analysis highlights the optimal conditions for ensuring the stability of the curved electrical shell structure. The most important conclusions would be as follows:

- The system's absorption energy and stability improve due to an increase in the orthotropic coefficient ratio and the reduces in the applied voltage.
- When the applied ampere increases, the time-domain decreases so the structure could experience a stable condition quickly.
- The resonance frequency or stability response of the electrically curved screen could improve thanks to raise in the applied ampere and a decrease in the inserted voltage
- Before time contact, E_{im}/c_{11} factor and energy of impactor indirectly related to each other. Although, the would be directly related to each other after time contact.
- As the E_i/c_{11} element is reduced, the highest absorbed energy could not change, but it occurs at the greater time.
- At the initial and final time of the process of impact, there is not any changes in the indenter speed due to an increase in the E_i/c_{11} ratio however, the indenter speed reduces due to increasing the E_i/c_{11} ratio at the middle time.
- The relation between E_{im}/c_{11} parameter and force of contact alters from direct to indirect at a particular time that the time could raise due to improve in the E_i/c_{11} factor.
- Reducing the impactor energy would be much faster due to decreasing the radius to speed ratio of impactor.
- That increasing R_i/V_i parameter provides a decrease in the indentation of the electrically curved screen and the maximum value of indentation could happen at the lower time due to decreasing the impactor radius.
- There is a special time ($Time = 30\mu$s) in which the relation between indenter velocity and of radius to velocity ratio of impactor changes from indirect to direct.

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