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Impact analysis on curved sandwich beam using numerical simulation

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ABSTRACT: Considering the greater resistance of curved beams compared to straight beams, the purpose of this research is to investigate the response of curved beams against low-velocity impact using numerical methods. Using ABAQUS finite element software, a curved beam with three layers in the form of a sandwich beam under the impact of a rigid body with a spherical tip is simulated. Hexagonal elements are considered for the curved beam, which has more numbers as they approach the target impact site. A simply supported boundary condition is considered for both sides of the beam. To verify the accuracy of the simulation results in this article, the impact response is compared with what is theoretically obtained in another research. The results show that with the increment in the curved beam radius, assuming that its angle is constant, the second peak related to the impact load and the absolute value of the impactor's residual velocity is decreased, but the maximum displacement of the impactor is incremented. Also, the maximum von Mises stress value is decreased with the increment in curved beam radius, but the range of maximum von Mises stress has become wider. It is seen that for a constant curved beam radius, the impactor's final velocity decreases, but no clear trend is observed for the impact load changes and the impactor's displacement.

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1- Introduction

The resistance that curved beams create as a result of their curvature can be used as a positive parameter against the impact on the beam. For this reason, in this article, the subject of curved beams against low-velocity impact is studied. In low-velocity impact problems, the contact time is so high that the entire structure feels the response, and the support conditions are important in such problems [1].

A lot of research is carried out in the field of low-velocity impact on the straight beams [2-7]. However, in the field of curved beams, considering temperature changes and using Hamilton's method, the frequency response of curved beams with circular cross-sections was done by Malekzadeh et al. [8]. The exact response for the free vibrations of curved beams was given by Wu et al. [9]. Using the differential transformation approach, static loading, and natural frequency responses of curved beams with variable cross-sections were carried out by Rajasekaran [10]. Applying the shear deformation of the trigonometric function type, Jun et al. [11] presented the free vibrations of thin curved beams. Using non-uniform rational B-splines, the natural frequency analysis of beams with various curvatures is investigated by Luu et al. [12]. Using the piecewise-linear spring method, the vibration analysis of delaminated composite curved beams was performed by Jafari-Talookolaei et al. [13]. Using the iso-geometric method, analysis of buckling, bending, and natural frequency responses of functionally graded curved beams was carried out by Huynh et al. [14]. Using the differential-integral quadrature model, vibration analysis of curved beams resting on the elastic base is proposed by Mohamed et al. [15]. Based on the various theories of curved beams, Yang et al. [16] carried out a review on the subject of curved beams. Using finite element theory, buckling, bending, and vibration analysis of curved porous beams reinforced with graphene was done by Anirudh et al. [17]. Ye et al. [18] extracted the natural frequencies of curved beams with non-linear boundary conditions. Taking into account the thermal conditions at different temperatures and using layer-wise theory, checking the vibration behavior and bending of curved beams was done by Beg and Yasin [19]. Investigation of random vibration response related to porous curved beams was done by Liu et al. [20]. Using 3D periodic arrangement and two-step homogenization model, Sadeghpour et al. [21] extracted the natural frequency responses of curved beams.

As mentioned earlier, according to the engineering requirement of the response of curved structures, in this article the investigation and numerical analysis of the resistance of a curved beam to the impact including the contact force between the beam and the impactor is discussed. Also, other responses such as displacement and velocity of the impactor are extracted.

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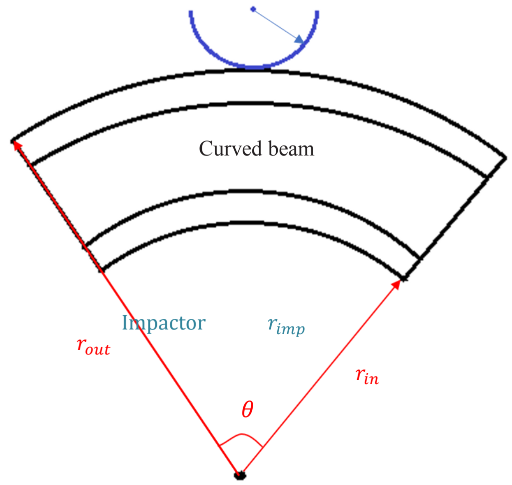


Fig. 1. Schematic of a curved sandwich beam subjected to low-velocity impact.

2- Materials and methods

In this article, a curved beam with inner radius r_{in} , outer radius r_{out} and arc angle θ is considered in Fig. 1. The desired beam is in the form of a sandwich, which has three layers with the thickness of the core t_c , the thicknesses of the upper layer t_u and lower layer t_l . The curved beam radius is calculated as the average of the inner radius and outer radius $\left(r_{cb}=\frac{r_{in}+r_{out}}{2}\right)$. A rigid projectile with a spherical tip hits the center of the curved beam. The impact between the upper surface of the beam and the impactor occurs at a low velocity.

ABAQUS finite element code is applied to simulate the low-velocity impact process on the curved sandwich beam. To simulate the impactor as a rigid object, the analytical rigid part is used in the part module and the deformable mode is used for the curved beam (Fig. 2).

As it is clear from Fig. b, the beam is divided into three layers. The mechanical and geometrical properties of different components of the impact issue are presented in Table 1. The width of the beam is constant and equal to 30 mm.

The problem solver is considered dynamics explicit. As seen in Fig. 3, the surface-to-surface contact is assumed between the impactor and the surface of the beam. In this model, the tangential behavior is assumed as the contact property option and the kinematic contact method is assumed as mechanical constraint formulation. A simply supported boundary condition is considered so that the displacements on both sides of the beam along the *r*-axis are zero. Meshing related to curved sandwich beams is presented in Fig. 4. As it is clear from this Fig., meshes are incremented as they approach the point of collision. To control the meshes, the shape of the element is assumed to be hexagonal and the technique is structured. The element library in ABAQUS software includes two standard and explicit options. In the case of nonlinear problems, it is superior to ABAQUS/Standard; in contrast, for high-velocity dynamics analysis such as crash analysis, ABAQUS/Explicit is the preferred choice. Considering that low velocity is considered in this article, the standard option is chosen. From the options of the element family, three-dimensional stress (3D-stress) is selected. The geometric order of elements in ABAQUS software includes two linear and quadratic options. In this research, the linear option is used for simulation.

3- Validation

In this article, the results of simulation in ABAQUS finite element software are compared with the results of Euler–Bernoulli theory (EBT) and layer-wise theory (LWT) [22]. The geometrical parameters $r_{cb}=573\ mm$ and $\theta=30^0$ are considered. A rigid impactor with a mass of 1 kg is included in this problem. For case EBT, the displacement field is written as follows [22]:

$$
\begin{aligned}
&w_j\left(z_j,\theta\right)=w_{0j}\left(\theta\right)\\
&u_j\left(z_j,\theta\right)=u_{0j}\left(\theta\right)+z_j\beta_j\left(\theta\right)\\
&\beta_j\left(\theta\right)=\frac{u_{0j}\left(\theta\right)-\frac{dw_j\left(\theta\right)}{dx}}{r_j}
\end{aligned}
\tag{1}
$$

Where u_{0j} is circumferential displacement and w_{0j} is radial displacement for midplane. In this Eq., *j=t* shows the top face sheet and *j=b* shows the bottom face sheet. But for the case, LWT, the displacement field is written as follows [22-27]:

$$
\begin{aligned}
&u\left(\theta,z,t\right)=U_k\left(\theta,t\right)\ \varphi_k\left(z\right)\\
&v\left(\theta,z,t\right)=V_k\left(\theta,t\right)\ \varphi_k\left(z\right)\\
&w\left(\theta,z,t\right)=W_k\left(\theta,t\right)\ \varphi_k\left(z\right)
\end{aligned}
\tag{2}
$$

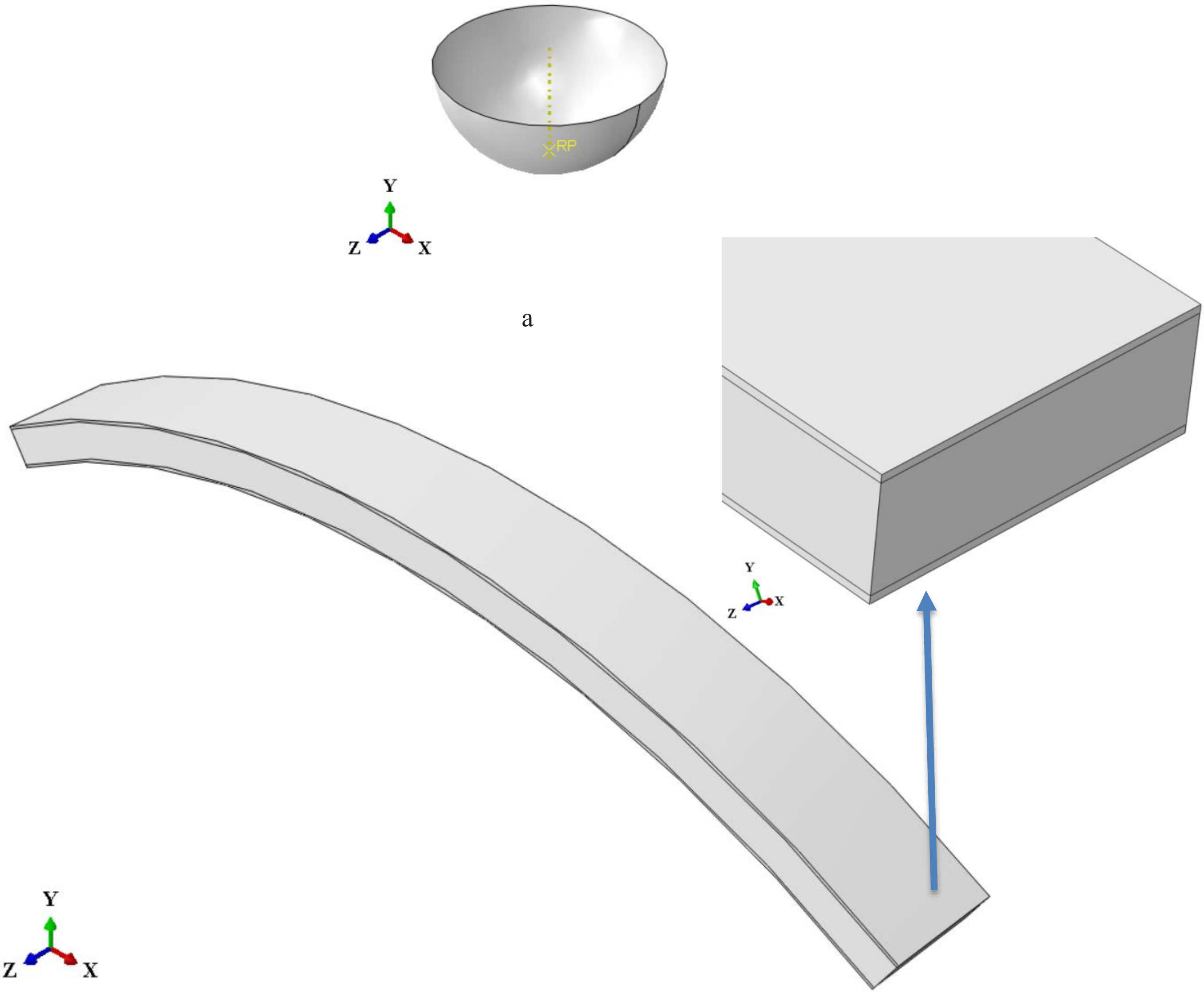


Fig. 2. Designing different components in ABAQUS software a) analytical rigid impactor and b) deformable curved sandwich beam.

Table 1. Geometrical and mechanical constants of different components

Components	Geometrical and mechanical constants	
Top and bottom layers	Young's modulus	36 GPa
	Density	$1800\ \frac{\mathrm{Kg}}{\mathrm{m^3}}$
	Poisson's ratio	0.3
	Thickness	1.5 mm
Core	Young's modulus	0.05 GPa
	Density	$50\ \frac{\mathrm{Kg}}{\mathrm{m^3}}$
	Poisson's ratio	0.25
	Thickness	20 mm
Impactor	Initial velocity	2 m/s
	Impact location	Middle of the upper surface of the beam
	Material	Rigid

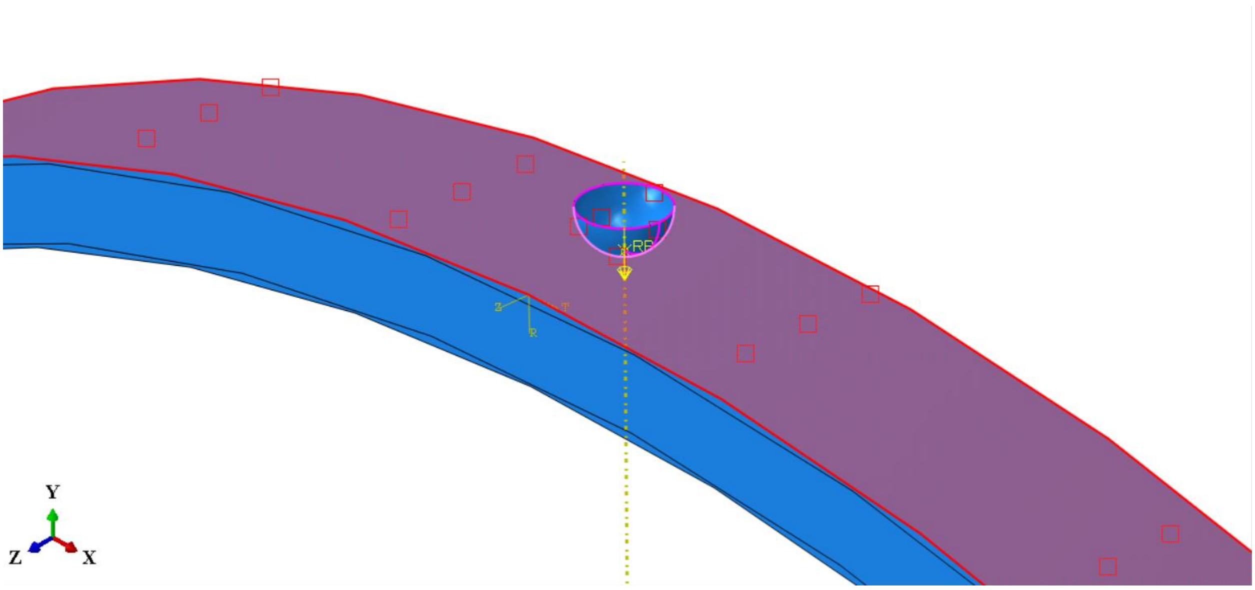
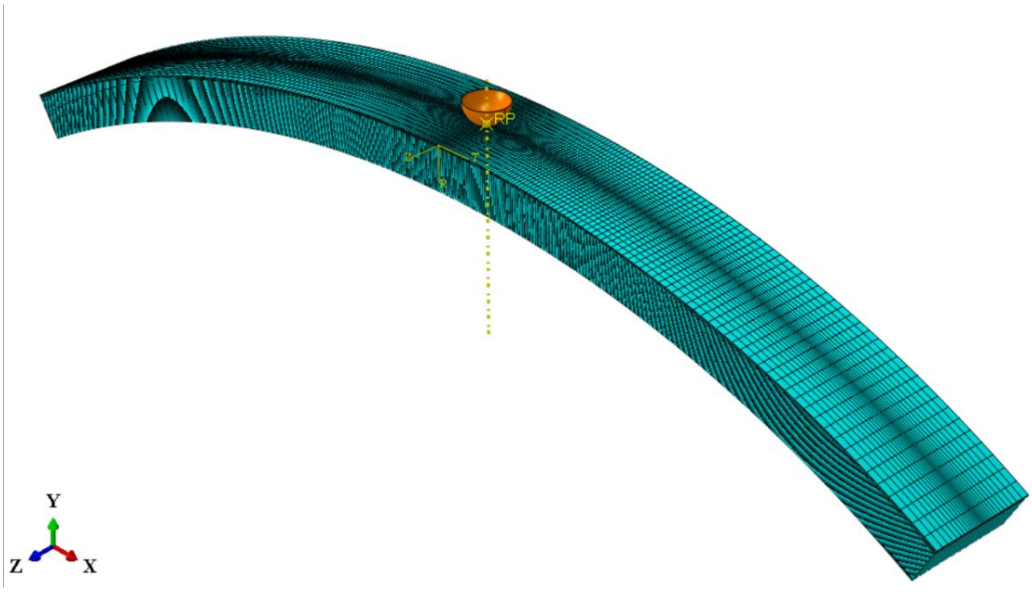


Fig. 3. The surface-to-surface contact interaction between the impactor and the surface of the beam.



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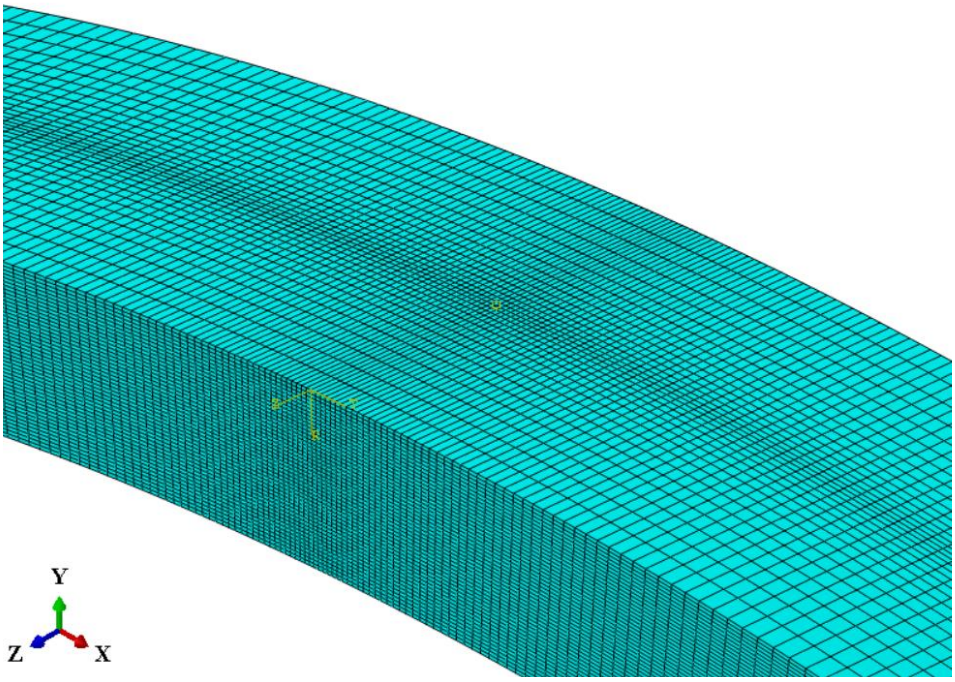


b

Fig. 4. Meshing related to curved sandwich beam, a) general view and b) close-up view.

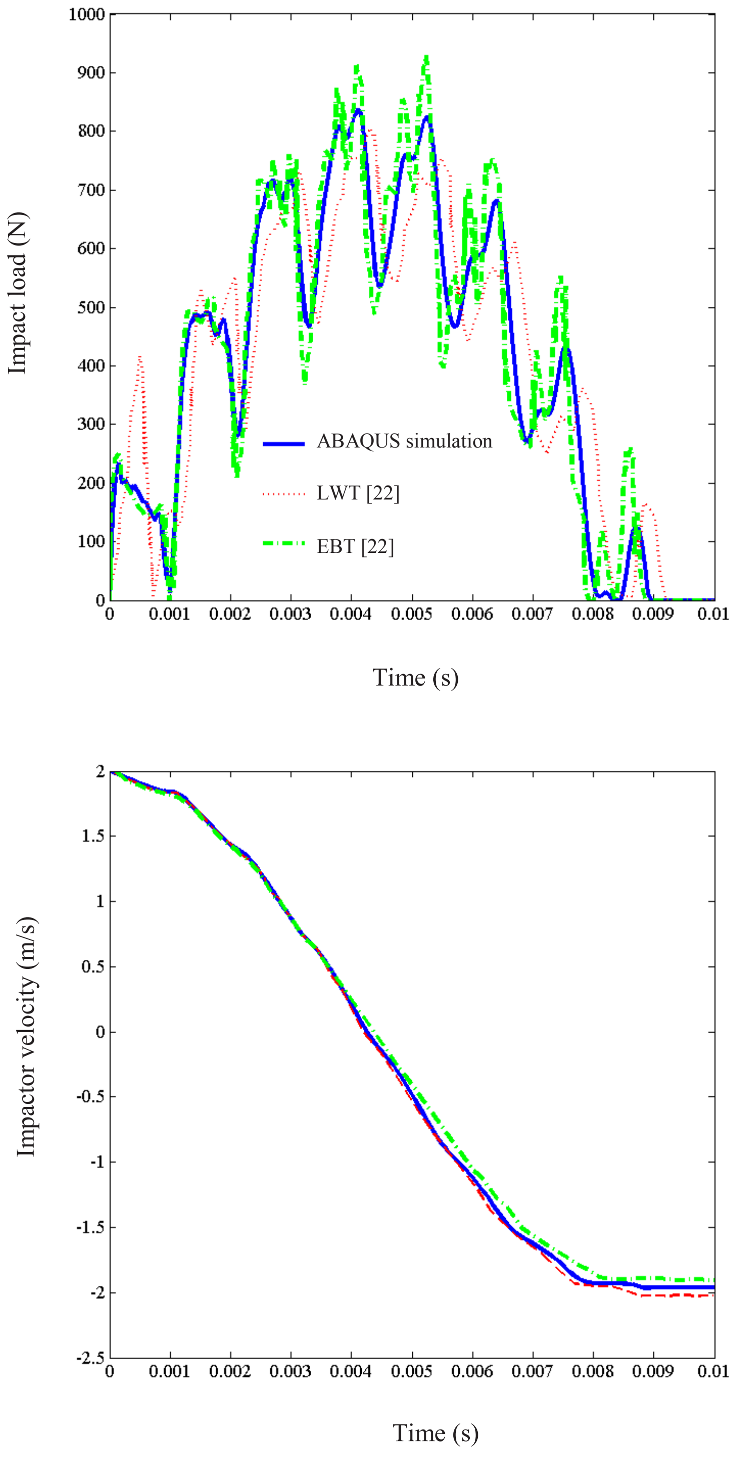


Fig. 5. Comparison between numerical simulation in this article and theoretical analysis in Ref. [22] for low-velocity impact on the curved sandwich beam, a) impact load and b) impactor velocity.

Where N is the number of numerical layers, $N+1$ is the number of numerical surfaces and φ_k is the global interpolation function. Comparison between numerical simulation in this article and theoretical analysis in Ref. [22] for low-velocity impact on the curved sandwich beam is presented in Fig. 5. At the first peak of the impact load history, the ABAQUS finite element results are more similar to the EBT method. In other impact load peaks as well as impactor velocity history, ABAQUS finite element, LWT and EBT results are compatible with each other. To conduct a convergence study in this study, it was observed that the results did not change significantly after reaching the number of about 200 thousand elements.

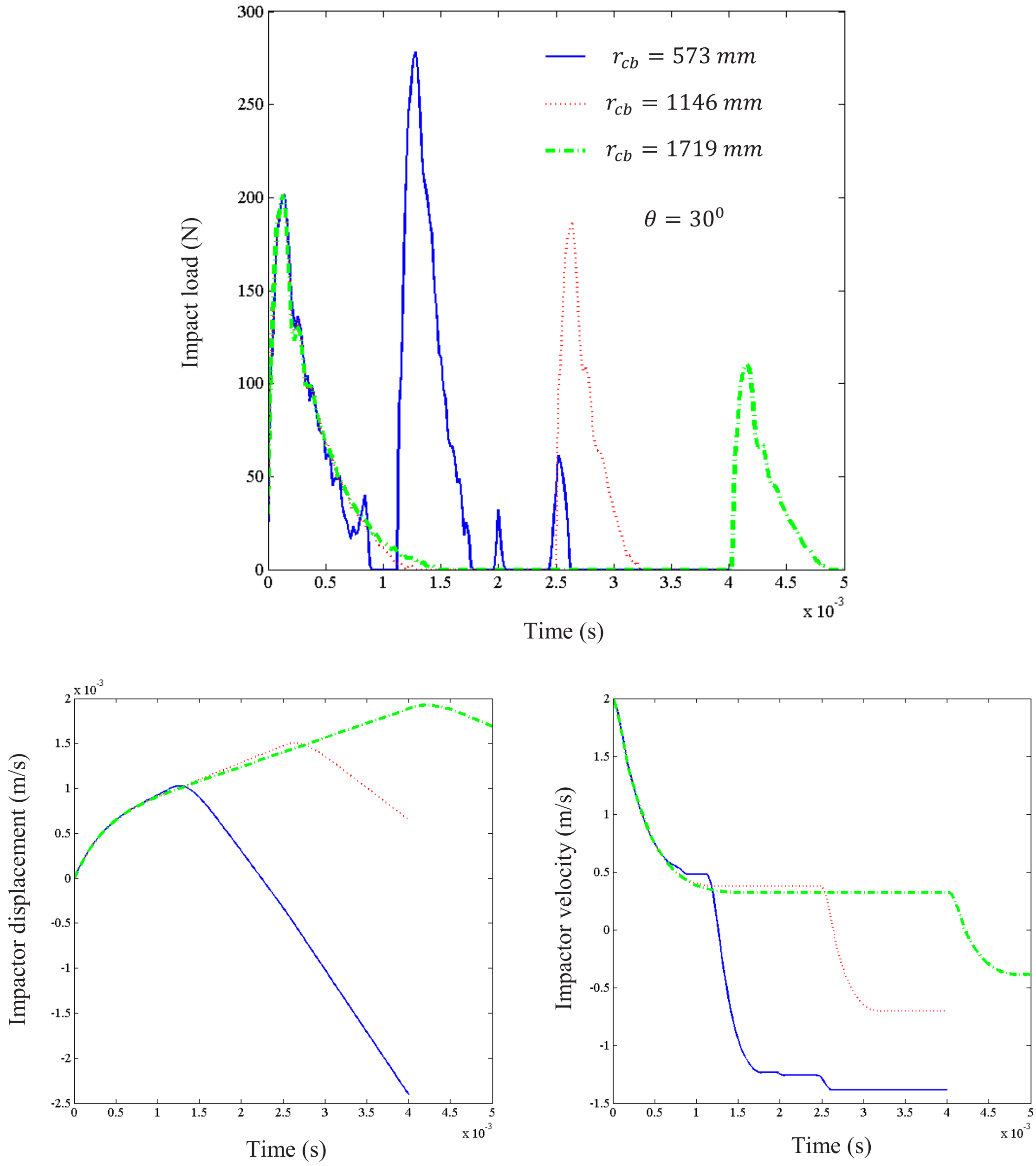
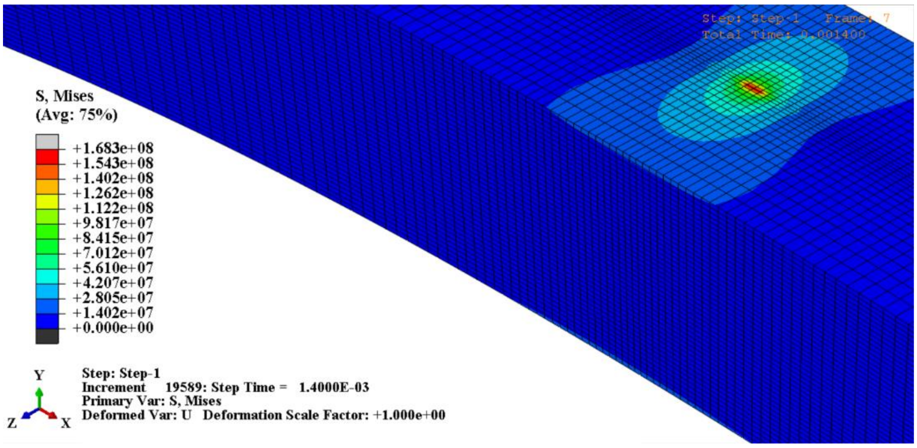


Fig. 6. The effect of different radii of the curved beam under low-velocity impact, a) impact load, b) displacement of impactor and c) velocity of impactor.

4- Results and discussions

The effect of different radii as well as different angles of the curved beam on the impact response is investigated. A rigid impactor with a mass of 50g is included in this section. In Fig. 6, the effect of different radii of the curved beam under low-velocity impact is investigated. In this Fig., the impact load, impactor displacement, and impactor velocity are illustrated. The beam curvature angle is considered to be constant and equal to $\theta = 30^0$, and the values of the radii of the curved beam are considered $r_{cb} = 573\ mm$, 1146 mm, and 1719 mm. In the impact load diagram, the first peaks have equal values for all three states of the curved beam radius. The second maximum value in the impact load history is decreased with the increment of the curved beam radius so that the curved beam radius is doubled from 573 to 1146 mm and also with the radius of the beam tripled from 573 to 1719 mm, the second maximum value in the impact load history is decreased 33.12% and 60.30%, respectively.

When the curved beam radius is incremented by 100% and 200% compared to 573 mm, the maximum displacement of the impactor has a similar behavior and increments by 46.45% and 87.54%, respectively. Considering the absolute value of the impactor's residual velocity, this value has a decrease of 49.06%, and 79.08% with an increment of 100%



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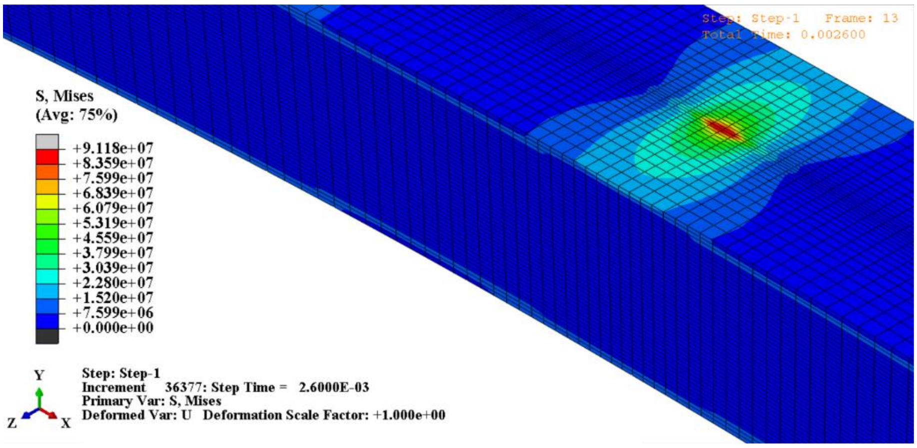

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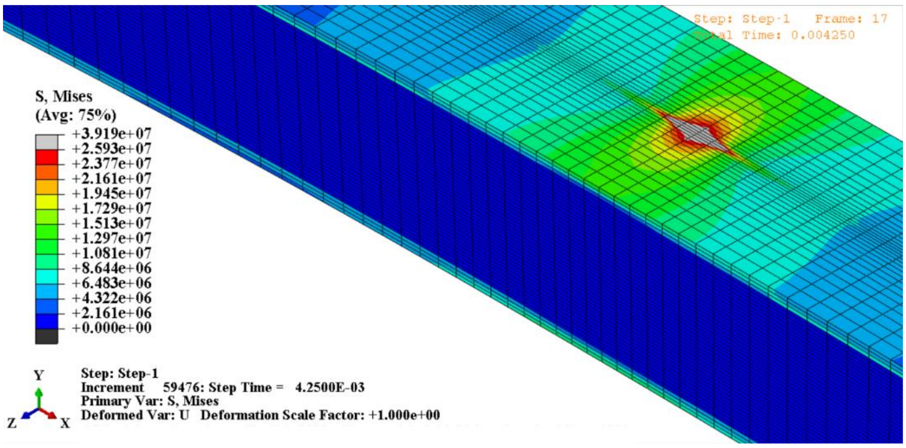

c

Fig. 7. The effect of different radii on the von Mises stress of the curved beam under low-velocity impact, a) r_{cb}=573 mm, b) r_{cb}=1146 mm and c) r_{cb}=1719 mm.

and 200% of the curved beam radius, respectively. The physical justification of the mentioned phenomenon is that by increasing the curved beam radius, in fact, the curved beam approaches a straight beam, which is why it reduces its resistance to impact, and the impact load decreases, in other words, the displacement of the beam increments. In parallel, the von Mises stress contour for the three radius values mentioned in Fig. 7 is presented. The time of drawing these

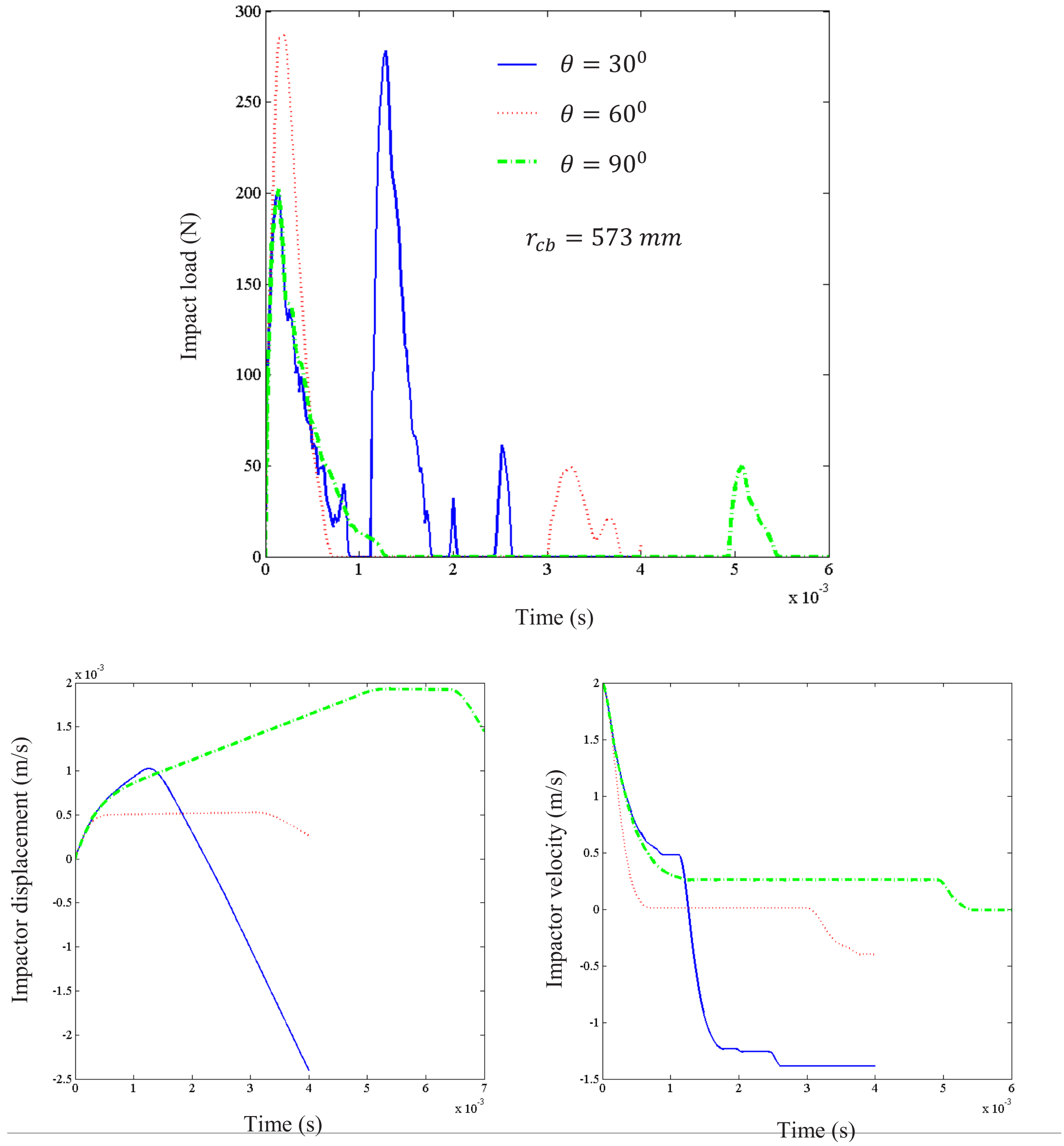

Fig. 8. The effect of different angles of the curved beam under low-velocity impact, a) impact load, b) displacement of impactor and c) velocity of impactor.

contours corresponds to the time of maximum contact force related to the radii. At 1.4 ms (r_{cb} =573 mm), 2.6 ms (r_{cb} = 1146 mm) and 4.25 ms (r_{cb} =1719 mm), the maximum von Mises stress values are 168.3 MPa, 91.18 MPa, and 39.19 MPa, respectively. The maximum von Mises stress value is decreased with the increment in curved beam radius, but the range of maximum von Mises stress contour has become wider.

The important issue is to present the effect of changes in the curvature angle on the impact response (Fig. 8). Different angles of $\theta = 30^0$, 60^0 and 90^0 are considered assuming that the curved beam radius is constant and equal to $r_{cb} = 573\ mm$. Examining the results shows that the effect of changing the angle of the curved beam on the impact load and impactor displacement does not have a special trend. Considering the absolute value of the impactor's residual velocity, this value has a decrease of 71.17%, and 99.51% with an increment of 100% and 200% of the curved angle of the beam, respectively. In fact, increasing the angle of the curved beam only increments the length of the beam, and the curved beam radius is constant in this case.

5- Conclusions

In this research, the dynamic response of curved beams against the low-velocity impact was discussed. The beam is assumed in the form of a sandwich, which bears the impact load with a low velocity on its upper surface. The mentioned analysis was carried out numerically in ABAQUS finite element software, where contact was assumed as surface-to-surface. Responses of histories of impact load, deflection, and speed of projectile were investigated with increasing radius and beam curvature angle. The important results can be summarized as follows:

Any increase in the radius of the curved beam resulted in a decrease in the value of the second maximum impact load, the absolute value of the impactor's residual velocity, and the maximum von Misses stress.

The increment in the maximum displacement of the impactor and the range of maximum von Mises stress contour was evident due to the increment in the curved beam radius.

The absolute value of the impactor's final velocity decreased due to the increment in the beam curvature angle.

Maximum impact load and displacement of the impactor due to the increment of the beam curvature angle did not present a specific trend.

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