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Evaluation of the Accuracy of an Impact Load Cell Using Lumped Parameter Modeling and Analysis

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ABSTRACT. The objective of the current study is to evaluate the fidelity of load cell reading during impact testing in a drop-weight impactor using lumped parameter modeling. For the most common configuration of a moving impactor-load cell system in which dynamic load is transferred from the impactor head to the load cell, a quantitative assessment is made of the possible discrepancy that can result in load cell response. A 3-DOF (degrees-of-freedom) LPM (lumped parameter model) is considered to represent a given impact testing set-up. In this model, a test specimen in the form of a steel hat section similar to front rails of cars is represented by a nonlinear spring while the load cell is assumed to behave in a linear manner due to its high stiffness. Assuming a given load-displacement response obtained in an actual test as the true behavior of the specimen, the numerical solution of the governing differential equations following an implicit time integration scheme is shown to yield an excellent reproduction of the mechanical behavior of the specimen thereby confirming the accuracy of the numerical approach. The spring representing the load cell, however, predicts a response that qualitatively matches the assumed load-displacement response of the test specimen with a perceptibly lower magnitude of load.

Keywords : Load Cell; Impact; Lumped Parameter Model; Implicit Method

1 Introduction

Evaluation of the performance of a product and its components under impact loading is one of the key considerations in design. In order to obtain dynamic responses of objects under impact conditions, compression load cells [1] are frequently used. Authors in [1-2] have discussed the construction of such load cells. A primary consideration in the design of such a load cell is the incorporation of a high degree of stiffness to limit axial deformation to low values (in microns). Curiously, there hardly appears to be any study reported in the published literature with quantitative information on the fidelity of load cell response in a given impact testing configuration. The present investigation uses the implicit Newmark method for solving the governing differential equations of a lumped parameter model (LPM) representing a system comprising an impactor with a lightweight accelerometer and a load cell vertically striking a steel automotive body component. The nonlinear spring is approximated in a piecewise linear manner and the solution strategy adopted and implemented in the form of a MATLAB script is shown to yield excellent reproduction of the assumed load-displacement behavior of the test specimen. For the first time, to the authors' best knowledge, the load cell response is compared with the desired response highlighting the under-prediction of the dynamic load.

2 Principle of Load Cells

A load cell converts a force into an electrical signal. A common feature of these load cells is the presence of strain gages which deform under applied dynamic load momentarily giving rise to proportional electrical signal due to change in resistance. [3]. A typical column load cell utilizes four strain gages attached to a

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cylinder. The gages are connected into the bridge circuitry in such a manner as to make use of Poison's ratio i.e. the ratio between the relative contraction in the direction of force applied and the relative expansion perpendicular to the force to increase the effective gage factor [1-3].

3 Lumped Paramter Modeling of Impact Testing Set-Up

According to the approach adopted in the present study, an experimentally obtained load-time history is assumed to be the true response of an axially impacted tubular steel component which is frequently used in automotive body construction for occupant protection during collisions through crash energy absorption. In the given drop-weight impact testing system, for recording load-time history, a column-type load cell is mounted between the main body of the impactor and a rigid impactor head. Additionally, a lightweight piezo-resistive accelerometer is mounted centrally on the main body of the impactor. Locating the load cell within the impactor parts ensures versatility in testing; for example, impact perforation of plates with an indenter fitted under the impactor head can be performed in addition to axial impact testing. The impact testing system is conceptualized as shown in Fig. 1 for the purpose of creating an idealized Lumped Parameter Model (LPM) by capturing the essence of the system. Fig. 2 represents an LPM corresponding to the concept of Fig. 1 with three vertical degrees-of-freedom with Springs 1, 2 and 3 representing the specimen, the load cell and the accelerometer respectively. This model is used to simulate the impact testing of a given specimen and for predicting the load cell response.

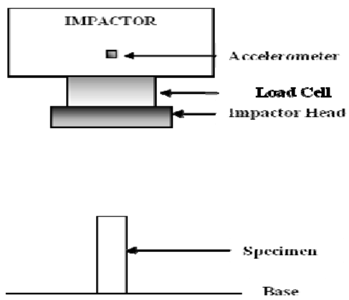

Fig. 1 Conceptual model of impact testing system

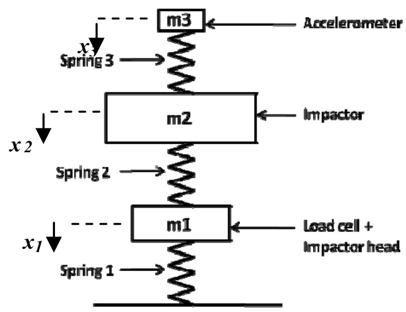

Fig. 2 A lumped parameter representation of conceptual impact testing system

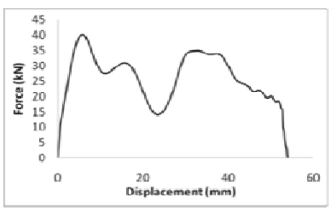

Fig. 3 Load-displacement behavior of specimen (Spring 1) from an impact test

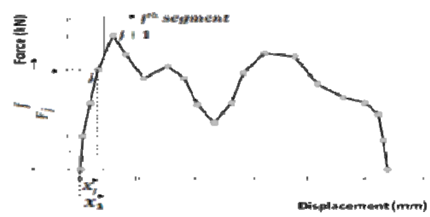

Fig. 4 Piecewise linear representation of the force-displacement curve of Fig. 3

The force-displacement curve obtained in an experiment for a given specimen geometry (i.e. cross-section and length), impactor mass and drop height is shown in Fig. 3 and is nonlinear because of its varying slope i.e. instantaneous stiffness k_1 which is a function of displacement x_1, although k_2 and k_3 can be assumed as constant slopes of linear force-displacement responses of load cell and accelerometer respectively. The governing equations can be simplified without sacrificing accuracy by assuming a piecewise linear representation of the force-displacement response of Spring 1 as shown in Fig. 4 and increasing the number of linear segments. It is noted from Fig. 4 that the force f in the spring 1 (i.e. the specimen) for any displacement x_1 is:

$$f = F_j + k_1^j (x_1 - X_j) \qquad (1)$$

where, F_j is the force in Spring 1 and X_j is its shortening at the beginning of the j^{th} linear segment in Fig. 4, and k_1^j is the constant slope of the j^{th} segment.

With the above assumption, the governing differential equation of the 3-DOF spring-mass system (i.e. the LPM) of Fig. 2, in matrix form, is obtained as:

$$\widetilde{M}\,\ddot{\bar{x}} + \widetilde{K}_j\,\bar{x} = \bar{R}_j \tag{2}$$

The solution of Eq. (2) can be initiated by imparting an initial common downward velocity (i.e. $\dot{x}_1$, $\dot{x}_2$ and $\dot{x}_3$ being equal to the impact velocity at time $t = 0$) to m_1, m_2 and m_3. The following are noted from Eq. (2):

$$\widetilde{M} = mass\ matrix = \begin{bmatrix} m_1 & 0 & 0 \\ 0 & m_2 & 0 \\ 0 & 0 & m_3 \end{bmatrix}, \quad \ddot{\bar{x}} = acceleration\ vector = \begin{Bmatrix} \ddot{x}_1 \\ \ddot{x}_2 \\ \ddot{x}_3 \end{Bmatrix},$$

$$\widetilde{K}_j = stiffness\ matrix = \begin{bmatrix} k_1^j + k_2 & -k_2 & 0 \\ -k_2 & k_2 + k_3 & -k_3 \\ 0 & -k_3 & k_3 \end{bmatrix},$$

$$\bar{x} = Displacment\ Vector = \begin{Bmatrix} x_1 \\ x_2 \\ x_3 \end{Bmatrix}, \quad \bar{R}_j = Force\ Vector = \begin{Bmatrix} k_1^j X_j - F_j \\ 0 \\ 0 \end{Bmatrix}; \text{ where,}$$

$m_1 = mass\ of\ impactor\ head\ and\ load\ cell$
$m_2 = mass\ of\ main\ body\ of\ the\ impactor$
$m_3 = mass\ of\ accelerometer$
$\ddot{x}_1, \ddot{x}_2\ and\ \ddot{x}_3\ are\ accelerations\ of\ m_1, m_2\ and\ m_3\ respectively$
$x_1, x_2\ and\ x_3\ are\ displacements\ of\ Springs\ 1, 2\ and\ 3\ respectively$

4 Numerical Simulation Procedure

In the present study, Eq. (2) is solved by numerical integration by marching forward in time with a small uniform time increment Δt. An implicit time integration scheme in the form of Newmark method [4-5] is adopted here. The advantage of such a method lies in its being unconditionally stable although the magnitude of error in computation can be reduced by decreasing Δt. In the Newmark method, the following approximations are commonly made:

$$\dot{\bar{x}}_{t+\Delta t} = \dot{\bar{x}}_t + \frac{1}{2}(\ddot{\bar{x}}_t + \ddot{\bar{x}}_{t+\Delta t})\,\Delta t \tag{3}$$

$$\bar{x}_{t+\Delta t} = \bar{x}_t + \dot{\bar{x}}_t\,\Delta t + \frac{1}{4}(\ddot{\bar{x}}_t + \ddot{\bar{x}}_{t+\Delta t})\,\Delta t^2 \tag{4}$$

Eq. (4) can be re-written as:

$$\ddot{\bar{x}}_{t+\Delta t} = \frac{4}{\Delta t^2}\,\bar{x}_{t+\Delta t} - \frac{4}{\Delta t^2}\,\bar{x}_t - \frac{4}{\Delta t}\,\dot{\bar{x}}_t - \ddot{\bar{x}}_t \tag{5}$$

Eq. (2) is valid at any instant of time including at time $(t + \Delta t)$. Substituting Eqs. (4) and (5) into the Eq. (2), the following equation is obtained:

$$(\frac{4}{\Delta t^2}\widetilde{M} + \widetilde{K}_j)\bar{x}_{t+\Delta t} = \bar{R}_j + \widetilde{M}(\frac{4}{\Delta t^2}\,\bar{x}_t + \frac{4}{\Delta t}\,\dot{\bar{x}}_t + \ddot{\bar{x}}_t) \tag{6}$$

Using Eq. (6), the displacements of the springs in Fig. 2 can be obtained at any incremental (i.e. current) time $(t + \Delta t)$ knowing the responses at the previous instant of time t. As the coefficient matrix $\left(\frac{4}{\Delta t^2}\widetilde{M} + \widetilde{K}_j\right)$ in Eq. (6) is non-diagonal, the system of three linear simultaneous equations represented by (6) need to be solved at every incremental instant of time. The current time integration method is therefore called as 'implicit'. It may be noted that in the above numerical algorithm that was implemented in the form of a MATLAB script, after every increment Δt and necessary computation (starting with $t = 0$ and $j = 1$), it is checked in which segment of the force-displacement curve in Fig. 4 will x_2 lie; accordingly the value of k_1(i.e. k_1^j in Eq. (2)) for that jth segment is chosen for next incremented time.

5 Simulation Results

Solution of Eq. (2) is carried out by adopting the following input data:

(i) the behavior of Spring 1 is defined by the force-displacement curve shown in Fig. 4 with a discrete set of points; (ii) $k_2 = 1000$ kN/mm; (iii) $k_3 = 63$ kN/mm; (iv) $m_1 = 30$ kg; (v) $m_2 = 130$ kg; (vi) $m_3 = 0.02$ kg; (vii) an impact (initial) velocity of 4.5 m/s is provided in conformity with the actual test.

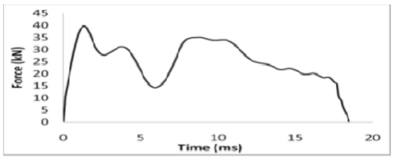

Fig. 5 Computed force time history of specimen i.e. Spring 1 (Δt = 0.0039 ms)

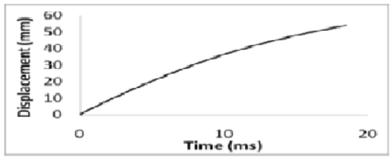

Fig. 6 Computed displacement time history of specimen i.e. Spring 1 (Δt = 0.0039 ms)

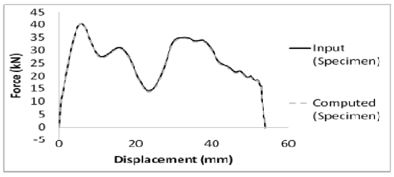

Fig. 7 Comparison between input force-displacement behavior of specimen and its computed response

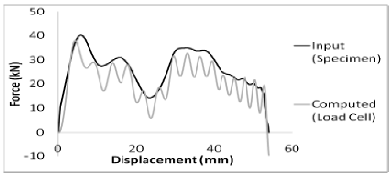

Fig. 8 Comparison between input force- displacement behavior of specimen and computed response of load cell

The computed force-time response of Spring 1 (i.e. the test specimen) is shown in Fig. 5. The displacement-time response of mass $m_1$is given in Fig. 6. By eliminating time from these two figures, the force-displacement response of Spring 1 can be plotted. This force-displacement response is compared with the input behavior of Spring 1 in Fig. 7 and excellent correspondence is observed. The ability of the present numerical simulation in closely reproducing the input nonlinear force-displacement behavior of Spring 1 (i.e. the test specimen) confirms the accuracy of the computations. The principal objective of the present study has been to assess the capability of a load cell which is part of an impactor in predicting the response of a specimen. To this end, the computed force-displacement behavior of Spring 2 representing load cell in the LPM of Fig. 2 is compared in Fig. 8 with the input force-displacement behavior of the test specimen. The following observations can be made with regard to the latter comparison: (i) the peak value in the load cell response is about 10% lower than the maximum load experienced by the specimen; (ii) the load cell response is qualitatively similar to the force-displacement behavior of the specimen provided as an input; and, (iii) load cell response contains oscillations which may be difficult to avoid completely (this behavior has been found in the load cell-based responses in actual impact tests).

6 Conclusion

The present study is an attempt at quantifying the accuracy of response of a column-type load cell under impact testing conditions when the load cell is a part of the impactor being sandwiched between the rigid impactor head and its main body. A novel lumped parameter modeling approach has been suggested for representing an impact testing set-up comprising impactor, load cell, accelerometer, and test specimen. A piecewise linear force-displacement behavior of the test specimen that is assumed to be known a-priori is adopted for efficiently solving the governing equations. The efficacy of an implicit time integration scheme is established by obtaining an extremely close correspondence between the computed force-displacement response and the input force-displacement behavior of a specimen. The response of the load cell for the same impact testing simulation is shown to be qualitatively similar to the input behavior of the specimen but with a lower magnitude of the peak load. The present simulation procedure can be a powerful tool for parametric studies and in understanding limitations of impact testing results.

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