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IMPACT OF FINITE BEAMS OF DUCTILE METAL

by

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BROWN UNIVERSITY

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IMPACT OF FINITE BEAMS OF DUCTILE METAL

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Abstract

An analysis is here presented of the large plastic deformations of a beam under impact such as that due to a blow of a massive hammer, in which one cross-section is suddenly forced to move with a given velocity. The analysis treats both the case in which the velocity of the struck section is maintained constant until the total permanent deformation is acquired and cases of interrupted impacts in which the force at the struck section is suddenly removed after an arbitrary contact time. A complete solution in general non-dimensional form is obtained in a simple manner by basing the analysis on the assumption of "plastic-rigid" behavior, and results can be expected to be valid when the plastic deformations are large enough. Criteria for the validity of the present results are discussed, based on the major assumptions of the analysis.

1. The results in this paper were obtained in the course of research conducted under Contract N7onr-35810 between the Office of Naval Research and Brown University.
1. **Introduction**

We consider the problem of a uniform beam of arbitrary length, initially at rest, which is subjected to a concentrated impact load at its mid-point such that the mid-section of the beam instantaneously acquires a velocity $V$ which is then maintained constant. Bohnenblust [1] treated the corresponding problem for the case of an infinitely long beam, in general terms for an arbitrary moment-curvature relation. It was recently discussed again by Conroy [2], who investigated the simplifications obtainable by neglecting the elastic part of the deformations, and took, in particular, moment-curvature relations of the two types shown in Fig. 1; again only infinitely long beams were considered.

The solution presented here is another example of the analysis of large plastic deformations in a finite structure subjected to a dynamic load, on the basis of the moment-curvature relation of Fig. 1(a). In this "plastic-rigid" type of analysis it is assumed that infinitely large curvatures can occur at cross-sections where the bending moment maintains the magnitude $M_o$, the "limit moment" or "fully plastic moment" used in limit analysis of structures under static loads. The concept of localization of deformations at such "plastic hinge" sections is assumed to be applicable to problems of dynamic loading of structures of ductile metal provided the energy absorbed in plastic deformations greatly exceeds that which could be absorbed in a wholly elastic manner. Based on this hypothesis criteria were given in a previous paper [3] for the validity of the solutions obtained, and similar criteria are given in the present paper for the new

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*Numbers in square brackets refer to the Bibliography at the end of the paper.*
solution presented here.

Earlier papers [3, 4, 5] dealt with finite beams subjected to force pulses with specified magnitudes and shape. Our main purpose in treating the present problem was to obtain a solution which could be treated more easily experimentally than the previous solutions obtained; the imposition of a known velocity involves fewer experimental difficulties than the application and measurement of a specified force-time curve. The present problem was emphasized by Vigness [6] in discussion of [3] as of interest from the point of view of experimental verification of the general method.

2. Analysis

For simplicity we treat the problem of a beam moving initially with velocity V normal to its length. At time \( t = 0 \) the mid-section is instantaneously brought to rest by contact with a rigid stop. It is obvious that the subsequent deformations will be identical with those which would be produced if the bar, initially at rest, were struck so that the mid-section suddenly acquires and maintains the velocity V. The beam is taken to have uniform mass per unit length \( m \), limit moment \( M_0 \), and length \( 2L \). A typical cross-section will be specified by its distance \( x \) from the center of the beam, (Fig. 2).

For a sufficiently small time interval after the bar strikes the stop, its deformation will be indistinguishable from that of the infinite beam treated by Conroy [2]. Hence it will be expected that the force on the beam will vary initially as \( 1/\sqrt{t} \), and that plastic hinges will occur at the mid-section and
at two cross-sections each at a distance \(x_h\) from the center; \(x_h\) will be expected to increase initially in proportion to \(\sqrt{t}\).

The analysis of the present problem will therefore be based on the diagrams of Fig. 3. From the symmetry of the problem it is obviously enough to consider either half of the beam. The segments OH and HB are subjected to end moments as shown. At the center of the beam there is a reaction force \(R\) exerted by the stop so that a shear force \(R/2\) acts as shown on the one segment. However, there is no shear force at cross-section \(x = x_h\), where the travelling hinge is located at a given instant, since this is a cross-section where the bending moment has a relative maximum; thus \(dM/dx = Q = 0\). The loading on the segments includes also the distributed inertia forces due to the accelerations. It can be verified that although they are changing in length, at any instant the two segments move as rigid bodies (hinged together as shown) since at any instant the complete set of loads is such that the bending moment reaches the magnitude \(M_o\) only at the sections \(x = 0\) and \(x = x_h\). The lateral hinge moves outward (i.e., \(x_h\) increases with time) so that the segment OH acquires a permanent deformation, but since the moment in the interior of the segment HB is always less than \(M_o\) in magnitude the segment to the right of the travelling hinge at any instant is undeformed.

Let the angular velocities of the segments OH and HB be \(\omega_0\) and \(\omega_1\), respectively. Writing moment-angular acceleration equations with respect to the fixed point 0 in one case and the
center of gravity of the segment in the other, we obtain

\[
\frac{1}{3} mx^3 \frac{d\omega_o}{dt} = -2M_o
\]  \hspace{1cm} (1)

\[
\frac{1}{12} m(\ell - x_h)^3 \frac{d\omega_1}{dt} = M_o.
\]  \hspace{1cm} (2)

The following equation expresses the fact that the change of moment of momentum of the half-beam OB with respect to an axis at 0 is equal to the angular impulse of the moment \(M_o\) acting during the time \(t\):

\[
-M_o t = \left[ \frac{1}{6} m(\omega_0 - \omega_1)(3x_h^2 - x_h^3) + \frac{1}{3} m\ell^3 \omega_1 \right] - \frac{1}{2} m\ell^2 V. \]  \hspace{1cm} (3)

The expression in brackets is the moment of momentum at any time \(t\), while the last term is the initial moment of momentum of the half-beam.

Equations (1) - (3) are the basic equations of the analysis from which, together with appropriate initial conditions the three unknowns \(x_h\), \(\omega_o\), \(\omega_1\) are to be found. It is now convenient to rewrite the equations in terms of new dimensionless variables defined as follows:

1. These equations are correct despite the fact that \(x_h\) is a function of time, because the velocities of elements just to the left and just to the right of the travelling hinge are equal at any instant. Thus, although the segment of length \(x_h\) is increasing in length, the element joining it in time \(dt\) comes in with the same velocity as the end of the segment, and there is no impulsive contribution to the momentum.
\[ \Omega = \frac{l}{V} \omega_0; \quad \psi = \frac{l}{V} \omega_1, \]  
\[ \eta = \frac{M_0 t}{ml^2 V}; \quad \xi = \frac{x_n}{l}. \]  

Equations (1) - (3) now take the following forms:

\[ \Omega' = \frac{d\Omega}{d\eta} = -\frac{6}{\xi^3}, \]  
\[ \psi' = \frac{d\psi}{d\eta} = \frac{12}{(1 - \xi)^3}, \]  
\[ 3 - 6\eta = (\Omega - \psi)(3\xi - \xi^3) + 2\psi. \]

We will henceforth use a prime to denote differentiation with respect to \( \eta \). The initial conditions may be taken as

\[ \eta = 0; \quad \xi = 0; \quad \psi = 0. \]  

From Eq. (6c) this implies that \((\Omega\xi)_0 = 1\), so that in this type of impact there is initially a singularity in the angular velocity at the point of impact.

We begin the solution of Eqs. (6) by differentiating Eq. (6c) with respect to \( \eta \), making use of Eqs. (6a) and (6b) and simplifying to obtain the relation

\[ \Omega - \psi = \frac{6}{\xi} \left[ \frac{1}{\xi^2} - \frac{1}{(1 - \xi)^2} \right]. \]  

For completeness, we note that the above result may be derived from a quite different viewpoint, namely by differentiating the equation which expresses the fact (as shown in [3]) that velocities are continuous at the hinge section. Equation (8) can thus be identified as expressing the fact that accelerations are
discontinuous across the moving hinge section.

A second differentiation of Eq. (8), substitution again of \( \Omega' \) and \( \psi' \) from Eqs. (6a) and (6b), and rearrangement leads to the following equation for \( \xi \):

\[
\left[ \frac{1 - 2\xi}{(1 - \xi)^2} \right] \xi'' + \xi' = 0, \tag{9}
\]

which is to be solved subject to the initial conditions (7). A first integration of Eq. (9) can easily be performed since the independent variable does not appear explicitly. Let \( \xi' = p(\xi) \); then

\[
\xi'' = \frac{dp(\xi)}{d\xi} \frac{d\xi}{d\eta} = \xi' \frac{dp}{d\xi} = p \frac{dp}{d\xi}.
\]

With this substitution Eq. (9) becomes

\[
\left[ \frac{1 - 2\xi}{(1 - \xi)^2} \right] \xi \frac{dp}{d\xi} + p = 0
\]

if it is assumed that \( p \neq 0 \). Integration of the above yields

\[
p = \frac{d\xi}{d\eta} = \frac{1}{A} e^{\xi/2} \frac{(1 - 2\xi)^{1/4}}{\xi}.
\]

Thus the general solution of Eq. (9) can be written in the form

\[
\eta = A \int_{\xi}^{\xi'} e^{-z/2} \frac{dz}{(1 - 2z)^{1/4}} + B \tag{10}
\]

where \( A \) and \( B \) are constants to be evaluated by means of the initial conditions Eq. (7). The numerical evaluation of the integral in Eq. (10) is speeded by making the substitution \( (1 - 2z) = 4s \). Integration by parts and rearrangement then leads to the following form:

\[
\eta = A [e^{-\xi/2} (1 - 2\xi)^{3/4} + 2F(\xi)] + B \tag{11}
\]
where
\[ F(\xi) = \sqrt{2} e^{-1/4} \int_{1/4}^{1/4} s^{-1/4} e^{s} ds - \frac{1}{2}. \]

The advantage of this form appears when the integrand in Eq. (11) is expanded in a power series; term by term integration then leads to the following rapidly converging series for \( F(\xi) \):

\[ F(\xi) = 0.07952 - 2e^{-1/4}(1 - 2\xi)^{3/4} \sum_{n=0}^{\infty} \frac{(1 - 2\xi)^n}{n!(3 + 4n)^4}. \]

Equations (6) and (8) can now be integrated to give the dimensionless angular velocities as follows:

\[ \Omega = \frac{3}{2} + 6A \left[ \frac{\left( 1 - 2\xi \right)^{3/4} e^{-\xi/2}}{\xi} \right] - 3B \quad (12a) \]

\[ \psi = \frac{3}{2} - 6A \left[ \frac{(2 - \xi)(1 - 2\xi)^{3/4} e^{-\xi/2}}{(1 - \xi)^2} + F \right] - 3B. \quad (12b) \]

After applying the initial conditions (7) to Eqs. (10) and (12b) we find that \( A \) and \( B \) must have values \( B = 0, A = 1/6 \). Thus \( \xi \) and \( \eta \) are related by

\[ 6\eta = e^{-\xi/2} (1 - 2\xi)^{3/4} + 2\sqrt{2} e^{-1/4} \int_{1/4}^{1/4} s^{-1/4} e^{s} ds - 1 \quad (13) \]

It can be seen from the denominator in Eq. (10) that the above solution is valid only for \( 0 \leq \xi < \frac{1}{2} \). This is the range we are interested in, as will be seen shortly.
3. Numerical results

The deformations of main interest are the angles \( \theta_0 \) and \( \theta_1 \) at mid-point and tip, respectively, and the permanent curvature \( \kappa \) of the part of the bar through which the lateral hinge has travelled. We have the following general formulas for the angles

\[
\theta_0 = \int_0^t \omega_0 dt = \frac{mLV^2}{M_0} \left( \frac{\Omega}{\xi} \right) d\xi
\]

and

\[
\theta_1 = \int_0^t \omega_1 dt = \frac{mLV^2}{M_0} \left( \frac{\psi}{\xi} \right) d\xi.
\]

The change \( d\theta \) of the angle of the beam across the lateral hinge has the value

\[
d\theta = (\omega_0 - \omega_1) dt.
\]

Then

\[
\kappa = \frac{d\theta}{dx} = (\omega_0 - \omega_1) \frac{dt}{dx} = \frac{mLV^2}{M_0} \frac{\Omega - \psi}{\xi}
\]

and

\[
\kappa = \frac{mLV^2}{6M_0} \frac{(1 - 2\xi)^{1/2} e^{-\xi}}{(1 - \xi)^2}.
\]

Numerical results are tabulated in Table I.

<table>
<thead>
<tr>
<th>( \xi )</th>
<th>( \eta )</th>
<th>( \Omega )</th>
<th>( \psi )</th>
<th>( \frac{M_0}{mLV^2} \theta_0 )</th>
<th>( \frac{M_0}{mLV^2} \theta_1 )</th>
<th>( \frac{M_0}{mLV^2} \kappa )</th>
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<tr>
<td>0</td>
<td>0</td>
<td>( \infty )</td>
<td>0</td>
<td>0</td>
<td>0</td>
<td>0.1667</td>
</tr>
<tr>
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<td>0.0002</td>
<td>19.524</td>
<td>0.003</td>
<td>0.0083</td>
<td>0.0000</td>
<td>0.1667</td>
</tr>
<tr>
<td>0.10</td>
<td>0.0008</td>
<td>9.045</td>
<td>0.012</td>
<td>0.0167</td>
<td>0.0000</td>
<td>0.1665</td>
</tr>
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<td>0.0019</td>
<td>6.583</td>
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<td>0.0250</td>
<td>0.0000</td>
<td>0.1661</td>
</tr>
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<td>0.063</td>
<td>0.0334</td>
<td>0.0001</td>
<td>0.1652</td>
</tr>
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<td>0.786</td>
<td>0.0769</td>
<td>0.0059</td>
<td>0.1111</td>
</tr>
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<td>0.50</td>
<td>0.0265</td>
<td>1.421</td>
<td>1.421</td>
<td>0.0888</td>
<td>0.0138</td>
<td>0.0000</td>
</tr>
</tbody>
</table>
The previous analysis is based on the configuration of Fig. 3. This type of motion continues until the angular velocities of the inner and outer segments become equal, i.e., until \( \Omega = \psi \). From Eq. (8) this happens when

\[
\frac{1}{\xi_s^2} \frac{1}{(1 - \xi_s)^2} = 0; \text{ therefore } \xi_s = \frac{1}{2}
\]

(17a)

where \( \xi_s \) denotes the final value of the hinge coordinate. The corresponding value of the dimensionless time is

\[
\eta_s = 0.0265. \tag{17b}
\]

For later times \( \eta \geq \eta_s \) there is a plastic hinge at the mid-section only, and the two halves rotate as rigid bars pinned at one end. The equation of angular acceleration of the right-hand half, in dimensionless variables, is

\[
\frac{d\Omega}{d\eta} = -3. \tag{18}
\]

Hence in this final phase of the motion we have

\[
\Omega = -3\eta + c_1
\]

\[
\frac{M_o}{mLV^2} \Phi \eta = -\frac{3}{2} \eta^2 + c_1 \eta + c_2.
\]

Making use of the conditions \( \eta_s = 0.0265, \Omega_s = 1.421, \)

\[
\frac{M_o}{mLV^2} \Phi \eta_s = 0.0888, \text{ as given in Table I, we obtain}
\]

\[
\Omega = 1.500 - 3\eta \tag{19a}
\]

\[
\frac{M_o}{mLV^2} \Phi \eta = 0.0500 + 1.5\eta - 1.5\eta^2. \tag{19b}
\]

These hold until \( \Omega = 0 \), or, from Eq. (19a), until \( \eta = \eta_f \), where
The corresponding final value of the angle $\theta_0$ is

$$\theta_0 = 0.425 \frac{m^2v^2}{M_0}.$$  \hfill (20b)

The growth of the deformations with time is shown in Fig. 4.

Some simple checks on the above results are of interest. First, we note that the momentum relation Eq. (6c), evaluated at the time $\eta_s$, becomes

$$3 - 6\eta_s = 2\Psi_s.$$  \hfill (20d)

Taking values from the last line of Table I the two sides of the equation have values

$$3 - 0.1590 = 2.841; \quad 2 \times 1.421 = 2.842$$

which is a satisfactory check. Again, from Eq. (6c) the time $\eta_f$ at which the deformation is completed is given by

$$3 - 6\eta_f = 0; \quad \eta_f = \frac{1}{2}$$

which verifies the value of Eq. (20a) computed from the tabulated values.

Finally, the energy relations may be examined. Before the beam hits the rigid stop it has the kinetic energy $E$

$$E = \frac{1}{2}(m^2t)v^2 = mlv^2.$$  \hfill (20e)

When the motion has ceased, this energy has been spent in work at plastic hinges. The total energy absorbed in plastic deformations is $W$, where
\[ W = 2M_0\Theta_{of} + 2(\Theta_{of} - \Theta_{lf})M_0. \]

Inserting values from Eq. (20b) and Table I this becomes

\[ W = 2M_0(0.425 \frac{mlv^2}{M_0}) + 2M_0(0.0750 \frac{mlv^2}{M_0}) = 1.000 mlv^2. \]

Thus the necessary energy balance is checked.

We may also compare the results obtained here with those obtained by Conroy [2]. It is there shown by equations (2) and (20) that \(12\eta = \xi^2\) for an infinitely long beam. Our result should converge to this value for small \(\xi\). We expand the right-hand side of Eq. (13) as a power series in \(\xi\), as follows:

\[ e^{-\xi/2}(1 - 2\xi)^{3/4} = 1 - 2\xi + \frac{1}{2} \xi^2 + o(\xi^3)\]

\[ \int_{1/(1-2\xi)}^{1/4} \frac{e^{s\xi}}{s^{1/4}} = \frac{\sqrt{2}}{2} e^{1/4} \xi + o(\xi^3). \]

Hence Eq. (13) reduces, for small \(\xi\), to

\[ \eta = \frac{1}{12} \xi^2 + o(\xi^3). \] (21)

Furthermore Conroy shows by Eqs. (20) and (26) that the shear force at the center varies as \(\frac{1}{2} \sqrt{\frac{3}{\eta}} \frac{M_0}{\xi}\). The moment equilibrium for the inner part of the beam around its center of gravity requires

\[ \frac{1}{4} R_{xh} - 2M_0 = \frac{m x^3}{12} \omega_0. \]

Transforming this into dimensionless coordinates we get

\[ \mu \xi = 8 + \frac{1}{3} \xi^3 \Omega' \] (22)

where \(\mu = \frac{R_{xh}}{M_0}\). With the use of equation (6a) we get
In the limit for small $\xi$ we have $\xi = \sqrt{12\eta}$. Hence in the limit $\mu = \sqrt{3/\eta}$, which agrees with the result in [2].

For completeness we finish the analysis with the calculation for the motion of the beam if the support is removed at a time $t = \tau < \frac{ML^2V}{MO} \eta_\tau$. The subsequent motion ($t > \tau$) is to be determined by analysis appropriate to a beam acted on by no external loads. Such an analysis is described in [4]. There it is shown that when lateral hinges are present at the instant the central force is reduced to zero the lateral hinges then move with constant velocity until the angular velocities become equalized. By simple calculations, using Eqs. (20) - (29) of Ref [4], one can find the increments in angular velocities and displacements which occur after removal of the force at arbitrary times $\tau$, corresponding to dimensionless times $\eta_\tau = \frac{M_0\tau}{ML^2V} < 0.5$.

Fig. 6 shows a curve of final deformation angles resulting from "interrupted impacts" of various durations, plotting $M_0Q\tau/mLV^2$ against $M_0\tau/ml^2V$. The shape of the force-time curve concerned here is indicated in Fig. 5.

4. Criteria for Validity of Results

We consider now the implications for experimental comparisons of two major assumptions involved, namely the neglect of all elastic deformations by comparison with plastic deformations, and the neglect of shape changes throughout the analysis.

The first assumption can be expected to be valid [3] if the total energy absorbed in plastic deformations greatly exceeds the maximum possible amount of elastic strain energy that could
be stored in the beam. In the present case this implies that

\[ 2M_0 \theta_{of} = m\ell V^2 >> \frac{M_0^2}{EI} \]

where EI is the elastic flexural rigidity. Thus for the present results to be good approximations one requirement is that

\[ V^2 >> \frac{M_0^2}{mEI} \]  (24a)

Alternatively the above requirement may be expressed as

\[ V^2 > n \frac{M_0^2}{mEI} \]  (24b)

where n is a number which presumably is of the order of 10.

A further requirement is set by the assumption of negligible shape changes, i.e., the use of coordinates referring to the undeformed rather than to the actual beam. This assumption actually restricts the magnitude of the central angle \( \theta_0 \) attained only during that part of the response in which deformations are occurring at lateral hinges; when only the central hinge is present the equations apply to deformation angles of unrestricted magnitude. From Table I the maximum value of \( \theta_0 \) obtained while the lateral hinges are present is \( 0.0888 \frac{m\ell V^2}{M_0} \); if we assume that this does not exceed about 0.15 radians, we have the additional inequality:

\[ 0.0888 \frac{m\ell V^2}{M_0} < 0.15. \]  (25)

This imposes a limitation on the length-depth ratio of the beam, if the requirement of inequalities (24) is taken into account. Let (24b) be taken as an equation and combined with the inequality (25). We obtain
But the limit moment $M_o$ can be computed \cite{7} as

$$M_o = \alpha \sigma_y Z = \alpha \sigma_y \left(\frac{2I}{h}\right) \quad (27)$$

where the "shape factor" $\alpha$ is a number of the order of unity which depends on the shape of the cross-section and has the value $1.5$ for a rectangular cross-section; $\sigma_y$ is the yield stress; and $Z = \frac{2I}{h}$ is the elastic section modulus, i.e., quotient of moment of inertia $I$ and half-depth $h/2$. Using this formula for $M_o$ the inequality (26a) can be written as

$$\frac{h}{2L} > \frac{1}{1.7} \frac{\alpha \sigma_y}{E}. \quad (26b)$$

Since $\sigma_y/E$ is of the order of $10^{-3}$ for steel, and $\alpha$ will probably be about $1.5$, it is seen that the restriction of small shape changes requires only that $h/2L$ exceed about $1/100$, for $n = 10$. Hence it is clear that this restriction would introduce no experimental difficulties.

Finally the order of magnitude of $V$ demanded by inequality (24b) is of interest. We use Eq. (27) and write also $m = \rho A$, $I = A i^2$, $E/\rho = c_0^2$, and obtain

$$V \geq \sqrt{n} c_o \frac{\sigma_y}{E} \left(\frac{4a}{h/2}\right) \quad (24c)$$

where

$$c_o = \sqrt{E/\rho} = \text{speed of longitudinal elastic waves}$$

$\rho = \text{mass density}$

$A = \text{cross-sectional area}$

$i = \text{radius of gyration}$

$h = \text{depth of beam}$

$\alpha = \text{shape factor (defined in connection with Eq. (27)).}$
Thus for steel with $c_0 = 16,000 \text{ ft/sec.}$, $\alpha_y/E = 10^{-3}$ and a rectangular cross-section, if $n = 10$ we find that $V$ must exceed about $45 \text{ ft/sec.}$

The main purpose of a laboratory test program would be to determine under what circumstances, if any, the plastic-rigid type of analysis yields satisfactory results for the major plastic deformations. The fundamental assumption is that elastic deformations are negligible. To find the range of usefulness of this assumption, a series of tests could be made with the objective of determining the value of the number $n$ used in Eq. 24b, above which the rigid-plastic analysis predicts deformation in good agreement with those observed in the tests. If the elastic deformations are the only important physical effect which is ignored then such a series of tests will yield a definite value of $n$, such that when the total energy absorbed is at least $n$ times the maximum possible elastic energy the beam could carry, the present type of analysis will be suitable. A value of $n$ determined for the present case of velocity impact would probably have significance for other types of problems of dynamic loading.

In any experiments other physical effects will occur which have been neglected here, and these might turn out to be so important that the present type of analysis does not yield accurate results even though the criterion based on elastic energy is satisfied. Among the physical phenomena which have been ignored are:

(a) **strain-hardening**, which will occur to some degree at cross-sections where plastic hinge action occurs;
(b) **speed of loading** effect on the yield-stress and post-yield properties, as, for example, reported by Manjoine [8] and Nadai and Manjoine [9];

(c) effect of **transverse shear forces** which are known to reduce the limit moment below that which can be reached in pure bending; in the present problem the shear force at the struck section has large values in the initial instants of the impact;

(d) **finite contact area** of the hammer or stop; the study in Ref. [5] of distributed as compared with concentrated loads showed that even a very small degree of spreading of the load over a finite segment may appreciably reduce the magnitude of the final deformations;

(e) **finite time of acceleration**; it was assumed that the hammer or stop was perfectly rigid and that the velocity of the struck section was instantaneously acquired or annihilated; since any physical hammer or stop has finite rigidity there is a finite time of acceleration; in fact the contact will probably be **intermittent** during the initial instants of the impact, as is known to be the case in the elastic impact of a mass on a beam.

All but one of the foregoing effects, if taken into account, would tend to reduce the deformations below those given by the present analysis. The single exception is the effect of transverse shear forces, which tend to weaken the beam and hence if properly taken into account would cause the analysis to predict larger deformation magnitudes than those of the present theory.
If the transverse shear effect is minimized by use of compact sections and fairly large ratios of span to depth, the other effects would presumably predominate, and cause the present analysis to overestimate the deformations.

Finally, it should be re-emphasized that a fundamental presupposition throughout the paper is that the material has sufficient ductility under dynamic conditions so that rupture does not occur before the predicted final deformations are attained. The present type of analysis should be regarded as a basic one, particularly attractive for its simplicity and generality, but which may have to be refined in particular cases. Carefully planned and interpreted tests are needed. They will be an invaluable aid in assessing the range of usefulness of the present type of analysis and in showing the directions in which refinements are most urgent.
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The beam at $t=0$

FIG. 2

FIG. 3
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