OS09F041

Experimental investigations transverse impact of free-free and free supported square duralumin beams*

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Abstract

A experimental analysis was performed to model transverse impact of free-free and free supported square duralumin beams loaded at different locations along their length. The applied impact load was obtained from tests carried out on a single Hopkinson pressure bar equipped with a high speed camera. The experimental set-up consisted of an Hopkinson measuring bar that is brought in contact with the beam. In this one-point impact experiment, a cylindrical striker, fired by the air gun, impacts the Hopkinson bar and generates stress waves that travel along the bar and impinge upon the aluminum beam. The stress waves are recorded by strain gages mounted on the Hopkinson measuring bar. These are used to calculate the applied load on the beam. Dynamical displacements of the impact zone of tested beam were recorded by the high speed camera. The dynamic experiments show that the plastic deformation, adjacent to the impact location, is due to combined dominant bending and stretching modes. Most of the plastic deformation is confined to the impact zone of tested beams. The plastic strain magnitude and distribution near the impact zone is similar for all tested impact locations, but higher for the more symmetrical impacts. The conversion of impact energy into kinetic, elastic strain energy and plastic dissipation work is characterized for various impact locations along the specimen of beam.

Key words: aluminum beam, large strain, Hopkinson bar, elastic-plastic material, transverse impact

1. Introduction

The dynamic inelastic behaviour of beams has been the subject of many investigations, as shown by Jones⁽¹⁾. Analytical solutions have been developed for transversely impacted beams. These solutions usually apply the classical beam theory which assumes small strains, displacements and rotations. The solutions are usually for a specified pressure pulse which can be step-like, rectangular or triangular. Moreover the material is usually assumed to be rigid perfectly plastic. When the beam is supported, most of the applied energy is converted to strain energy, which results in relatively large plastic deformations. By contrast, when the beam is not supported, the external work is transformed into kinetic and elastic strain energy, as well as plastic dissipation. A free-free beam subjected to central pulse loading was first studied by Lee et al.⁽²⁾, with the simplified assumption of a rigid plastic material. The authors stated that the rigid plastic assumption can be expected to

provide a satisfactory approximation to the real elastic-plastic problem if the final strains are large with respect to elastic strains. This requires that the plastic work done in the central plastic hinge should be much larger than the maximum stored elastic energy of the bent beam subjected to the limit moment. Jones and Wierzbicki⁽³⁾ developed a theoretical procedure to analyze free-free beams subjected to dynamic pressure load. Exact theoretical solutions for a uniform beam under triangular and step pressure pulse were presented. Their solution relies on the classical beam equations and assumes again that the beam is made from a rigid-perfectly plastic material. It also assumes small strains and negligible elastic effects. Jones and Wierzbicki⁽³⁾ concluded that it is more difficult to break a free-free beam, as found in space structures, than a supported beam because only ~25% of the external energy is converted into plastic work. The remaining 75% of the energy causes a rigid body acceleration of the beam. Yang et al.⁽⁴⁾ studied the small deflection response of a rigid perfectly plastic free-free beam subjected to a concentrated step load at any cross section along its span, and concluded that no less than 2/3 of the input energy is transformed into kinetic energy, meaning that the plastic energy dissipation in the plastic hinges is always less than 1/3 of the total input energy. Yu et al.⁽⁵⁾ presented a dynamic analysis of an elastic-plastic free-free beam subjected to impact by a projectile at its mid span, or to triangular distributed impulse along the beam. Their governing equations, which were based on a small deflection formulation, were solved numerically. Their results were compared with the instantaneous experimental recordings using high speed photography, and good agreement was reported. These authors concluded that an elastic-plastic analysis is necessary to provide a more accurate prediction on the plastic dissipation and possible failure of the dynamically loaded free-free beams. Yang and Xi⁽⁶⁾ conducted an experimental-theoretical study of free-free beams subjected to impact at any cross-section along its span. Their theoretical model assumes an elastic plastic material with linear strain hardening. These authors concluded that the energy partitioning is different for symmetrical and unsymmetrical impacts. The free-free beam will dissipate more plastic work for a symmetrical impact, so that the latter results in greater damage and failure. From the above-mentioned works, it appears that accurate characterization (without any simplifying assumptions, not even Kirchhoff assumptions) of the plastic damage and energy partitioning for low energy transverse impacts still needs to be carried out. Therefore, Dorogoy and Rittel⁽⁷⁾ presented the hybrid experimental-numerical investigation of free-free aluminium square beams subjected to transverse impact at four different locations along their length. The impact energy was low, less than 0.75 the elastic energy storage capacity of the beams, and the elastic strain was not neglected. Hence, the 3D transient numerical elastic-plastic analysis for large strains was performed. The impact load was applied and measured using the single Hopkinson bar apparatus. The measured experimental load was used as the input into the numerical simulations. The amount of damage and the energy partitioning for the four impact locations was investigated. In consultation with authors of above work⁽⁷⁾ we practically repeated their experiments using additionally a high speed camera to register oscillating motions of impacted beams, which are very important to validate numerical simulations. The main aim of presented experimental investigations was to register and to analyze elastic-plastic vibrations of duralumin beams under dynamic monotonic force $P(t) = \varepsilon(t) \cdot A \cdot E$, where $\varepsilon(t) = \varepsilon^I(t) - \varepsilon(t)^R$, A and E are difference of elastic strain impulses of incident and reflected waves in Hopkinson measuring bar, its cross-section area, and Young's modulus of the measuring bar material respectively, by the high speed camera. This device was used for recording fast moving objects, i. e. end of measuring bar and a centre section of the tested beam as photographic images onto a storage media. After recording, the images stored on the media were played back in slow-motion. The paper describes the experimental setup and results, followed by concluding remarks.

2. Experimental setup

The PA-38 duralumin⁽⁸⁾⁽⁹⁾ beams were transversely impacted using a single Hopkinson pressure bar apparatus⁽¹⁰⁾. Chemical constitution and mechanical properties of this material are listed in Tables 1 and 2. The duralumin beams had a square cross section of $b \times h = 20 \times 20 \text{ mm}^2$, and a length of $2L = 520 \text{ mm}$. The beam was impacted at four different locations (see Fig. 1): $0.0, 0.25, 0.5$ and $0.75 \cdot L \cdot X$, respectively. The experimental setup consisted of an Hopkinson measuring bar that is brought in contact with the tested beam at $(X, Y, Z) = (0, 0, h/2)$ – Fig. 2 and 3.

Table 1 Chemical constitution of PA-38 duralumin⁽⁸⁾

Mg, [%]	Si, [%]	Fe, [%]	Ti, [%]	Zn, [%]	Cr, [%]	Mn, [%]	Cu, [%]	Others total, [%]	Others
0,45-0,9	0,20-0,6	0,35 max	0,1 max	0,10 max	0,10 max	0,10 max	0,10 max	0,15 max	others

Table 2 Mechanical properties of PA-38 duralumin⁽⁹⁾

Strength in tension	Elongation ($A_{50\text{mm}}$)	Elongation (A)	Yield limit (0,2%)	Hardness	Impact resistance (DVM)
215 N/mm^2 <i>min.</i>	8%	6%	170 N/mm^2 <i>min.</i>	65 HB	24 J min.

Fig. 1 The duralumin beam with coordinate system.

Fig. 2 Sketch of the experimental setup. (1) – tested beam, (2) – Hopkinson measuring bar, (3) – striker, (4) – high speed camera, (5) – lighting, (6) –measuring device of striker velocity v_0 , (7) – two pairs of strain gages, (8) – air gun, A – digital oscilloscope, B – impulse generator, C – power supply, D – dynamic strain amplifier.

In this one-point impact experiment, a cylindrical striker (4), fired by the air gun (8), impacts the Hopkinson bar (2) and generates stress waves that travel along the bar and impinge upon the beam. The stress waves are recorded by strain gages (7) mounted on the Hopkinson measuring bar. These are used to calculate the applied load on the specimen as well as the striker velocity. The striker diameter and length are: $d^0 \times L_s = 20 \times 200 \text{ mm}$, respectively. The incident bar diameter and length are: $d^0 \times L_H = 20 \times 1000 \text{ mm}$.

Fig. 3 End (a) and front (b) views of the experimental setup.

The striker and the measuring bar are made of Maraging hardened steel type BÖHLER Z-V720-VMR, grade 1.6354⁽¹¹⁾. Chemical constitution and mechanical properties of this material are listed in Tables 3 and 4. Two pairs of strain gages are cemented on the incident bar (3), 200 mm and 500 mm from the edge that is in contact with the tested beam. This experimental setup is equipped additionally with the high speed camera (4) to record an oscillating motion of the impact loaded beam (1) using the “shadow” method. Measurement of beam motion was performed with using the Tema Motion 2D software (Fig. 4).

Table 3 Chemical constitution of Maraging hardened steel type BÖHLER Z-V720-VMR⁽¹¹⁾

C	Si	Mn	Mo	Ni	Co	Ti	Al
0,003%<	0,02% <	0,02%<	4,95%	18,19%	8,64%	0,68%	0,10%

Table 4 Mechanical properties of Maraging hardened steel type BÖHLER Z-V720-VMR⁽¹¹⁾

Strength in tension	Elongation (A6)	Young's modulus	Yield limit (0,2%)	Hardness	Impact resistance (DVM)
1720 - 1870 N/mm ²	8%	210 GPa	1620 N/mm ² <i>min.</i>	51 HRC <i>average</i>	24 J <i>min.</i>

Fig. 4 Recorded movie and elaborated results of oscillating motions (displacements and velocities) of a edge of midpoint tested beam (point 1 – green line) and end face of measuring bar (point 4 – red line) given by the software of high speed camera – a view of the computer (2) monitor.

3. Experimental results

The main experimental results of three presented experiments with striker velocity $v_0 = 24.1$ (beam No C6), 30.1 (beam No A3) and 32.5 m/s (beam No B6) consist of following curves that describe for the impact location $x/L = 0.5$:

- 1) the impact force $P(t)$, as shown in Fig. 5 a) – c),
- 2) transverse displacements, velocities and accelerations of a edge of midpoint tested beams and end face of measuring bar, as shown in Fig. 6 – 8,
- 3) comparison of transverse velocities of edges of midpoint tested beams in Fig. 9.

Permanently deformed above beams after experiments are shown in Fig. 10 – 12.

Fig. 5 The impact force $P(t) = \varepsilon(t) \cdot A \cdot E$ for striker velocity $v_0 = 32.5 \text{ m/s}$ (beam No B6), where $\varepsilon(t) = \varepsilon^I(t) - \varepsilon^R(t)$, A – cross-section area of the Hopkinson measuring bar, E – Young's modulus of measuring bar material, (a) – elastic strain impulses recorded by strain gages mounted on the Hopkinson measuring bar, (b) – elastic strain impulses of incident $\varepsilon^I(t)$ and reflected $\varepsilon^R(t)$ waves in Hopkinson measuring bar, (c) – the impact force $P(t)$.

Fig. 6 a) – c). Transverse displacements (a), velocities (b) and accelerations (c) of a edge of midpoint tested beam No C6 and end face of measuring bar.

Fig. 7 a) – c). Transverse displacements (a), velocities (b) and accelerations (c) of a edge of midpoint tested beam No A3 and end face of measuring bar.

Fig. 8 a) – c). Transverse displacements (a), velocities (b) and accelerations (c) of a edge of midpoint tested beam No B6 and end face of measuring bar.

Fig. 9 Comparison of transverse velocities at the point of impact.

Fig. 10 a). Permanently deformed beam No A3 after experiment. b) Permanent deformation of two beam edges determined by VISTA coordinate measuring machine (Zeiss, Germany) points the stationary plastic hinge created at the point of impact.

Fig. 11 a). Permanently deformed beam No C6 after experiment. b) Permanent deformation of two beam edges.

Fig. 12 a). Permanently deformed beam No B6 after experiment. b) Permanent deformation of two beam edges.

4. Conclusions

The experimental analysis presents vibration behaviors of free-free square duralumin beams under impact force loaded at middle locations along their length. The applied impact load was obtained from tests carried out on the Hopkinson measuring bar. The strain gages readings during the impact were recorded and used to determine the impact time-force curve. The experimental setup was equipped with the high speed camera, which allowed to recorded oscillating motions of the impact loaded beam using the “shadow” method. Next, transverse displacements, velocities and accelerations of edges of midpoint tested beams after impact were obtained by the software of this camera. Recorded images played back in slow-motion show clearly both the vibration of tested beams and the longitudinal wave phenomenon in Hopkinson measuring bar, which influences on plane motion of this bar, periodically delay this motion.

It was observed that when the square duralumin beam is transversely impacted, a stationary plastic hinge is created at the point of impact. The experiments show that the plastic deformation, adjacent to the impact location, is due to dominant bending mode. Most of the plastic deformation is confined to the impact zone but some partial additional plastic hinges are observed to develop as was presented by Ritter and Dorogoy⁽⁷⁾.

5. Acknowledgments

The authors thank to Professors Daniel Ritter and Avraham Dorogoy of Technion – Israel Institute of Technology for the encouragement to continue their work⁽⁷⁾ expressed during 7th International Symposium on Impact Engineering in Warsaw as well as for helpful comments, and Mr Marcin Sokolnicki for his help in this work. The authors gratefully acknowledge the support by Warsaw University of Technology under Grant No N501 156038, Ministry of Science and Higher Education (Poland).

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