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SPLIT HOPKINSON PRESSURE BAR TECHNIQUE FOR DYNAMIC CALIBRATION OF FORCE TRANSDUCERS

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Abstract − The split Hopkinson pressure bar (SHPB)-technique was developed for dynamic testing of materials but can be extended to other fields. Recent research results show that this is a promising technique for dynamic calibration of force transducers. In the modern SHPB-technique incident, reflected and transmitted pulses are measured with strain gauges mounted on the bars. This paper describes a mathematical model of a principle SHPB-system. The model is based on one-dimensional wave propagation theory. The model has been used to simulate the calibration of a force transducer up to 40 kN with a loading pulse having a duration of 500 μs. With this technique it is possible to combine high forces with short impulses. The technique also allows for measurement of transducer deformation making it possible to study the dynamic characteristics of the transducer.

Keywords: dynamic calibration, SHPB.

1. INTRODUCTION

Dynamic mechanical testing is a field in progress. The techniques used are often divided into two groups depending on the type of input, impulse or harmonic excitation.

Several techniques for measuring excitation and response make use of elastic waves in bars. A cylindrical bar with circular cross-section is simple enough to permit an analytical solution of the equation of motion. Thus, stress and velocity at the end of a bar can be determined from measurements of the incident and reflected pulses at a cross section along the bar.

Kolsky [1] was the first to use this technique, with split Hopkinson pressure bar (SHPB), to investigate the mechanical properties of materials at high rates of loading. A comprehensive review of experimental techniques for high rate deformation and shock studies has been published by Field et. al. [2]. This review covers more than 400 references up to year 2001.

In the modern SHPB-technique the incident, reflected and transmitted pulses are measured with strain gauges mounted on the bars. Relatively short incident pulses are used so that there is a time interval between the end of the incident pulse and the arrival of the reflected pulse at the measurement section. A general presentation of the technique has been presented by Al-Mousawi, Reid and Deans [3].

The SHPB-technique was developed for dynamic testing of materials but can be extended to other fields. Ueda and Umeda [4] used this technique to evaluate the dynamic characteristics of force transducers. Their results showed that this is a promising technique for dynamic calibration of force transducers and the present paper deals with the simulation of a SHPB-system to make it possible to improve the technique further, e.g. by shaping the incident wave. Dynamic force measurement and the need for a traceable dynamic calibration of the force transducers is a field of research where different techniques are currently under development. Dynamic force measurement and the value of dynamic calibration have been treated by Hunt [5]. Dynamic properties of piezoelectric devices have been investigated by Kumme et. al. [6]. Force transducers have been calibrated for harmonic forces by Dixon [7] and dynamic investigation of multi-component force sensors using harmonic excitation have been performed by Park, Kang and Kumme [8]. The calibration of force transducers with impulse forces has been studied by Bruns et. al. [9] and a preliminary investigation of dynamic response of a multi-component force-moment sensor subject to impulse forces has been performed by Kobusch et. al. [10]. A literature survey on dynamic measurement of force by Hjelmgren [11] reviews recent work in this field.

2. METHOD

A mathematical model of a principle SHPB set-up, shown in Fig. 1, has been derived and studied.

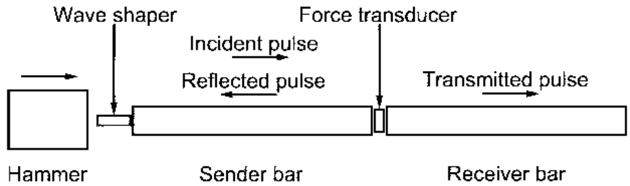

Fig. 1. Principal SHPB set-up

The model is based on one-dimensional wave propagation theory, which is accurate if the predominating wavelengths are much longer than the transverse dimensions of the bar. Each part of the system can be described by the characteristic impedance of the cross-section, Z_i, and the wave travel transit time, T_i. Wave reflection and transmission from part 1 to part 2 are governed by the reflection ratio, R_{12}

$$R_{12} = \frac{Z_1 - Z_2}{Z_1 + Z_2} . \quad (1)$$

and the transmission ratio, T_{12}

$$T_{12} = \frac{2 \cdot Z_1}{Z_1 + Z_2} . \quad (2)$$

Equations (1) and (2) are valid only as long as there is no tensile force at the intersection. The model checks for each time-step, of 1 μs, that this condition holds. If a tensile force is about to occur, the reflection ratio is altered to model full reflection and a separation of the two parts will occur. The relative displacement of the two parts is traced so that renewed contact can be properly modelled.

For each time-step the waves incident to an intersection are first calculated. As a second step the reflected and transmitted waves are calculated using the reflection and transmission ratios for parts in contact. The results are checked to ensure that no tensile forces are produced at any intersection, if that is the case full reflection at that intersection is assumed for the current time-step.

As an example, the calculation of the waves at the intersection between the positive end of part 1 and the negative end of part 2 is shown below.

The incident waves, expressed with their particle velocities, at the intersection are

$$v_{1pp}(t) = v_{1np}(t - T_1) \quad (3)$$

$$v_{2nn}(t) = v_{2pn}(t - T_2) \quad (4)$$

where v_{1np} is the particle velocity in the positive going wave, p, at the negative end, n, of part 1.
If

$$v_{1pn}(t) = v_{1pp}(t) \cdot R_{12} + v_{2nn}(t) \cdot T_{21} < v_{1pp}(t) \quad (5)$$

and

$$v_{2np}(t) = v_{1pp}(t) \cdot T_{12} + v_{2nn}(t) \cdot R_{21} > v_{2nn}(t) \quad (6)$$

then, the force at the intersection, F_{12}

$$\begin{aligned} F_{12}(t) &= Z_1 \cdot [v_{1pn}(t) - v_{1pp}(t)] \\ &= Z_2 \cdot [v_{2nn}(t) - v_{2np}(t)] \end{aligned} \quad (7)$$

will be a compressive force. If the conditions are not fulfilled, the reflected waves are calculated as

$$v_{1pn}(t) = v_{1pp}(t) \quad (8)$$

and

$$v_{2np}(t) = v_{2nn}(t) \quad (9)$$

thus, ensuring free end conditions.

When the system depicted in Fig. 1, being initially at rest, is impacted by the hammer with a velocity v_0 the resulting positive and negative going waves can be calculated for the entire system. From these results, forces, F_i, velocities, v_i, and displacements, d_i, at any cross-section can be calculated using Eqs. (10) to (12).

$$F_i(t) = Z_i \cdot [v_{in}(t) - v_{ip}(t)] \quad (10)$$

$$v_i(t) = v_{in}(t) + v_{ip}(t) \quad (11)$$

$$d_i(t) = \int_0^t v_i(\tau) d\tau \quad (12)$$

3. RESULTS

The model has been used to simulate the calibration of a force transducer up to 40 kN with a loading pulse having a duration of approximately 500 μs. In the system a wave shaper is introduced to shape the incident pulse to a shape more appropriate for calibration purposes. A sine pulse has been the target because of the symmetry during loading/unloading. Theoretically a sine pulse can be achieved if the wave shaper can be modelled as a spring with zero wave travel time and the hammer can be modelled as a rigid mass. Additionally the characteristic impedance of the sender bar must be infinite compared to the impedance of the wave shaper. In practice a shape close to a sine pulse can be achieved if: *i*) The characteristic impedance of the wave shaper is much smaller than that of both the hammer and the sender bar, *ii*) The wave travel transit times for the hammer and the wave shaper are much shorter than the pulse duration. Fig. 2 shows the strain pulses in the sender bar for the simulated system, with an impact speed of 3 m/s.

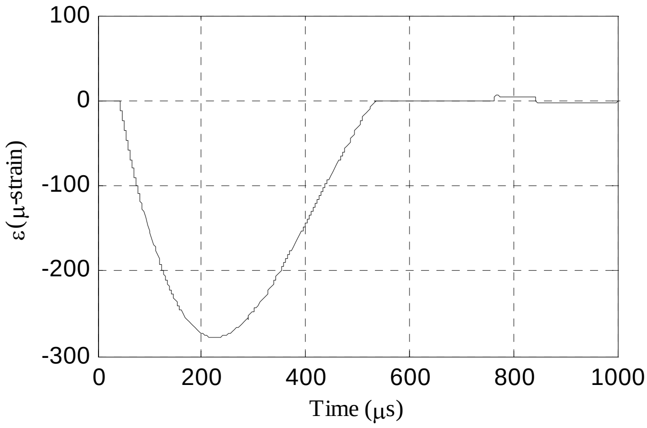

Fig. 2. Calculated strain pulse in sender bar

In Fig. 2 the incident and reflected waves are clearly separated in time, which improves the measurement accuracy considerably. The length of the sender bar should be chosen long enough to achieve this. The incident pulse resembles a sine pulse but has slightly different slopes during loading and unloading. This is due to the fact that the sender bar has finite impedance. The effect of the wave travel transit time of the wave shaper can be seen in the stepwise behaviour of especially the loading part of the strain curve. The rather small amplitude of the reflected wave is caused by a small impedance difference between test object and bars. The transmitted wave will thus be rather similar to the incident wave as can be seen in Fig. 3.

The strain and stress in the test object can be calculated from the incident, reflected and transmitted waves. The reflected wave is important especially for evaluating the strain whereas the transmitted wave is important for the evaluation of the stress. In this application (calibration) focus is on reducing the uncertainty in the stress and, hence, force in the test object so an impedance match and a small resulting reflected wave is in this specific application an advantage.

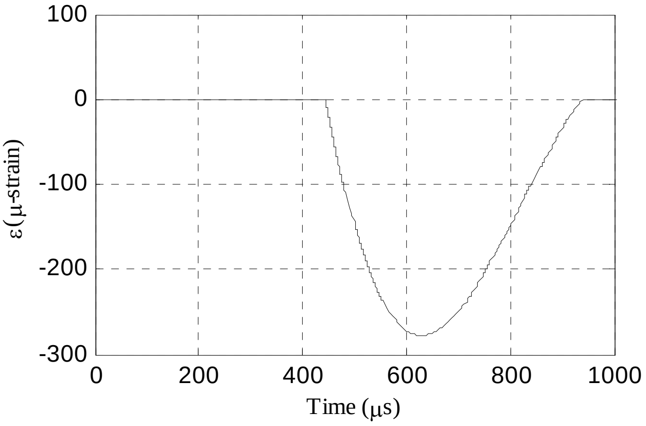

Fig. 3. Calculated strain pulse in receiver bar

The calculated results in Figs. 2 and 3 have been achieved by a proper choice of properties for the simulated system, which are shown in Table 1.

TABLE 1. Properties of SHPB-system

	Hammer	Wave Shaper	Sender Bar	Test Object	Receiver Bar
length [mm]	100	25	2000	10	2000
diameter [mm]	70	4	30	20	30
modulus [GPa]	200	290	200	200	200
density [kg/m^3]	8000	1850	8000	8000	8000
impedance [kNs/m]	154	0.29	28.3	12.6	28.3
travel time [μs]	20	2	400	2	400

The main part of the incident pulse is transmitted through the force transducer (test object), which is modelled as a homogeneous steel cylinder in this simulation. The true force in the transducer will vary along the length due to the dynamic loading and the wave travel time for the transducer. In a real test the force has to be estimated from the forces evaluated at the ends of the sender and receiver bars. Thus, the force in the force transducer is calculated as the average of the forces on both sides whereas the deformation is calculated as the difference between the two sides respective displacements. The compressive force is shown in Fig. 4,

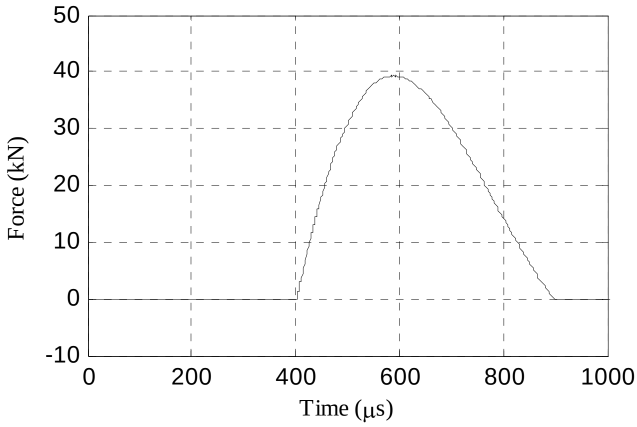

Fig. 4. Calculated compressive force in force transducer

whereas the relative force difference between the two sides of the force transducer is shown in Fig. 5.

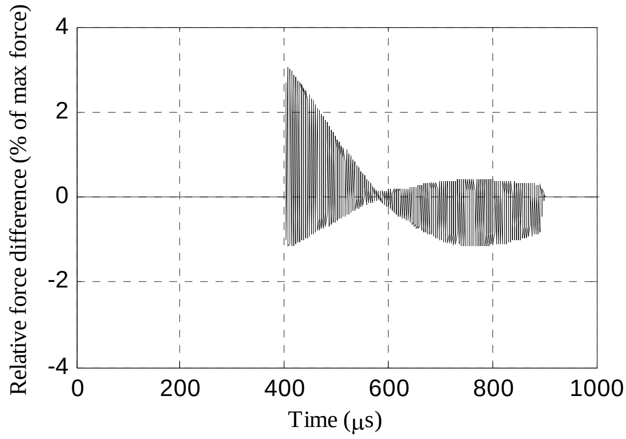

Fig. 5. Relative force difference between ends of force transducer

A zoomed image of Fig 5 is shown in Fig 6. The relative force difference is in this example as much as 3% of the maximum force and it is caused by wave reflections at the interfaces. With a real transducer the wave propagation through the object would show dispersion due to non-homogenous impedance and the force difference would be smaller.

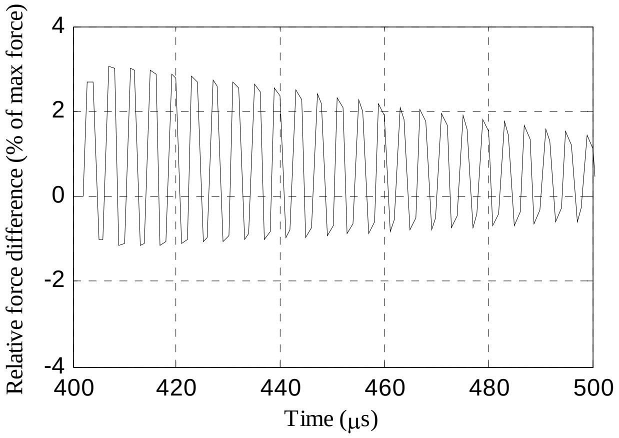

Fig. 6. Relative force difference between ends of force transducer

A force-deformation curve for a linear test object can look rather non-linear if the travel time for the test object is long or if the loading is not smooth enough. Using a linear test object in the simulation can reveal such problems with the set-up. Fig. 7 shows the deformation and Fig. 8 shows the force-deformation curve for this simulation.

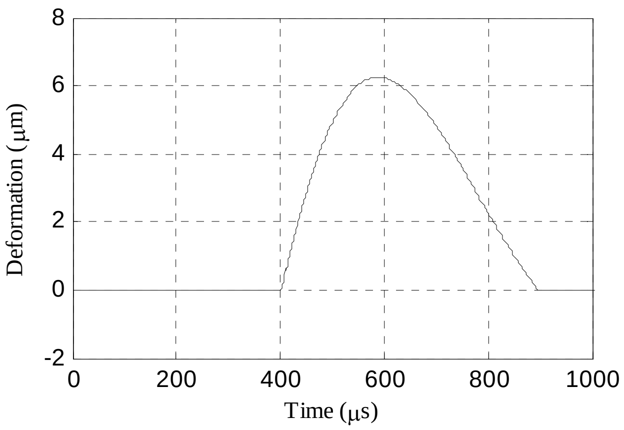

Fig. 7. Calculated deformation of the test object

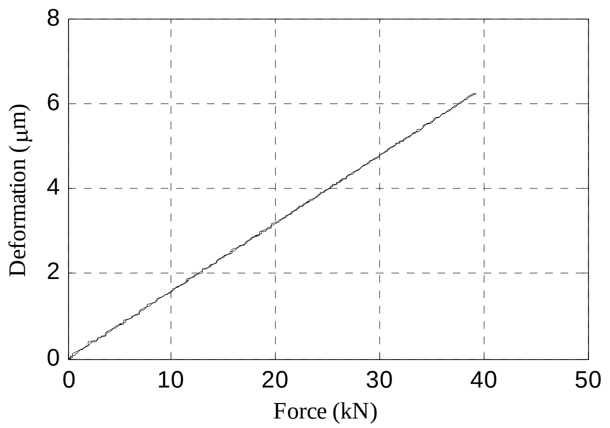

Fig. 8. Force-deformation curve for the simulated system

4. CONCLUSIONS

Split Hopkinson pressure bar (SHPB)-technique is a promising technique for dynamic calibration of force transducers. With this technique it is possible to combine high forces with short impulses. A SHPB-system can be simulated in advance to investigate and optimise the design of the system. It has been shown that appropriate pulses for dynamic calibration of force transducers can be achieved by such a system. The technique also allows for measurement of transducer deformation making it possible to study the dynamic characteristics of the transducer.

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