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Nonlinear dynamic responses of new electro-vibroimpact system

V.-D. Nguyen[a], K.-C. Woo[b,*]

[a]Thai Nguyen University of Technology, 3-2 Street, Thai Nguyen City, Vietnam
[b]Faculty of Engineering and Computer Science, The University of Nottingham Malaysia Campus, Jalan Broga, 43500 Semenyih, Selangor Darul Ehsan, Malaysia

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Abstract

In this note, we describe dynamics of a new vibro-impact system, which is capable of generating large impact forces and consequently enhanced penetration rates. It consists of a solenoid, connected in series to an RLC circuit. A solid-state relay switches power supplied to the solenoid in accordance to a train of square waves supplied by a function generator. A metal bar, made from laminated sheets, oscillates within the solenoid, and is connected to the main structure by means of a spring. This part of the design facilitates the vibratory motion. The forward progression of the mechanism is impeded by frictional force. The variety of system responses is described, so as to identify the qualitative dynamic behaviour which produces the fastest progression rate.

1. Introduction

While the use of a vibro-impact mechanism in underground tunneling has become a necessity, the means by which it can be realized in practice has proven to be elusive. Back in the 1940s, such a mechanism had been used in pile installation. Such machines [1] were large, and proved that a combination of vibration and impact [2,3] achieved better soil penetration. Further confirmation has been made in theoretical simulation and experimental work of Pavlovskaia et al. [4] and Wiercigroch et al. [5–9], respectively. Pertinent to this work would be the discussion on classification principles of impact systems, as well as the antiphase synchronization of chaos by Blazejczyk-Okolewska et al. [10,11]. Useful groundwork done by Woo [12] can also be explored further. In order to install underground cables and pipes, the diameter of the mechanism should be reduced to 100 mm or even less. The cam system of Lok et al. [13] addressed this issue. However, due to the friction between moving parts and high stresses generated, the reliability of this device posed a problem for further development. A detailed investigation of the vibro-impact mechanism had been performed with an electromagnetic shaker by Franca and Weber [14]. Nevertheless, the diameter of the electromagnetic shaker was still much larger than the 100 mm required.

A novel experimental prototype, using a solenoid-actuated vibrator, possible to deploy in moling machines for its compact dimension has been introduced by Nguyen et al. [15]. The use of a solenoid as a linear

*Corresponding author. Tel.: +60 389248123; fax: +60 389248017.
E-mail address: Woo.Ko-Choong@nottingham.edu.my (K.-C. Woo).

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self-oscillation inductance motor has been studied by Mendrela [16,17]. However, the application of solenoids in vibro-impact mechanisms is relatively new idea.

The new design solves the problem of effective control; it is capable of producing versatile vibro-impact motion controlled by the time history of electric current. It may also be deployed in the field which is far from the energy source. Experimental data pertaining to progression rate and variation in impact force are presented, suggesting the high practicality of actual design. Progression rates and impact forces of the system are the means of identifying the operational modes which elicit the optimum performance of this vibro-impact system.

2. Description of the experimental rig

The principle upon which the experimental rig is based is the combination of resonance in an RLC circuit and a solenoid, to effect impacts of a metal bar on an obstacle block, as shown in Fig. 1. The power is switched on and off in accordance to the frequency of a function generator by means of a solid-state relay. By placing an obstacle block in the path of the bar oscillations, impacts are created. Reversal of the direction of the bar motion is facilitated by the impact force acting on the bar from the obstacle block in addition to the electromagnetic force. The impact force is required for the forward progression of the machine. A spring was also installed to bring the bar back to the other end after impact. A schematic of the new system is shown in Fig. 2.

In Fig. 2, the solenoid (2) causes a laminated-sheet bar (3) to oscillate inside. The bar is connected to a spring support (4). A force sensor is placed at the stop position to measure the impact force. The linear variable displacement transducer (LVDT) (5) measures the displacement of the bar relative to the base board (7), and the second LVDT (6) measures the absolute displacement of the base board, which mimics a moling machine. A set of aluminium rails (8) guides the mole to move in a straight line. The spring support (4) is produced to combine two springs of the same stiffness into one which can act on both sides. To change the frictional force between the moling rig and the rails, a clamp set has been used. When the clamp set is taken out from the moling rig, the frictional force is induced only by the weight of the moling rig and is smallest. By tightening the clamp bolt, the coil spring compression is produced which induces an additional normal force in the system and thus increasing the level of the frictional force. The mass of the metal bar oscillating inside the solenoid was measured as 0.297 kg, whereas the total mass of the board with all components placed on it excluding the metal bar is 2.94 kg. The gap between the centre of the solenoid and the stop was maintained at −2.5 mm. This distance was experimentally found to be suitable for generation of the bar oscillations which caused substantial impacts. It was set to avoid the metal bar experiencing the effect of a magnetic force opposite to its direction of travel. In this way, the bar oscillates mainly in one half of the length of the solenoid where the electromagnetic force acts in the same direction as its travel. The impact stiffness was estimated and

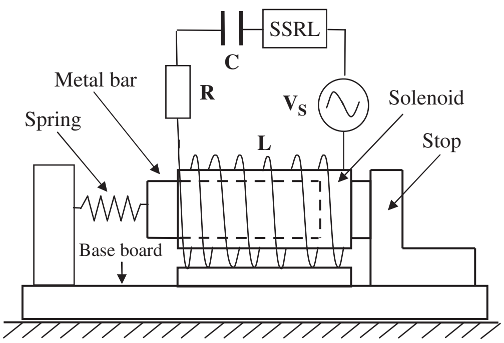

Fig. 1. Schematic diagram of the vibro-impact device mimicking the moling machine. When the bar is at either end of the solenoid, there is a large current flowing through the coils connected in series with the RLC circuit, thus creating a large electromagnetic force pulling the bar towards the centre of the solenoid.

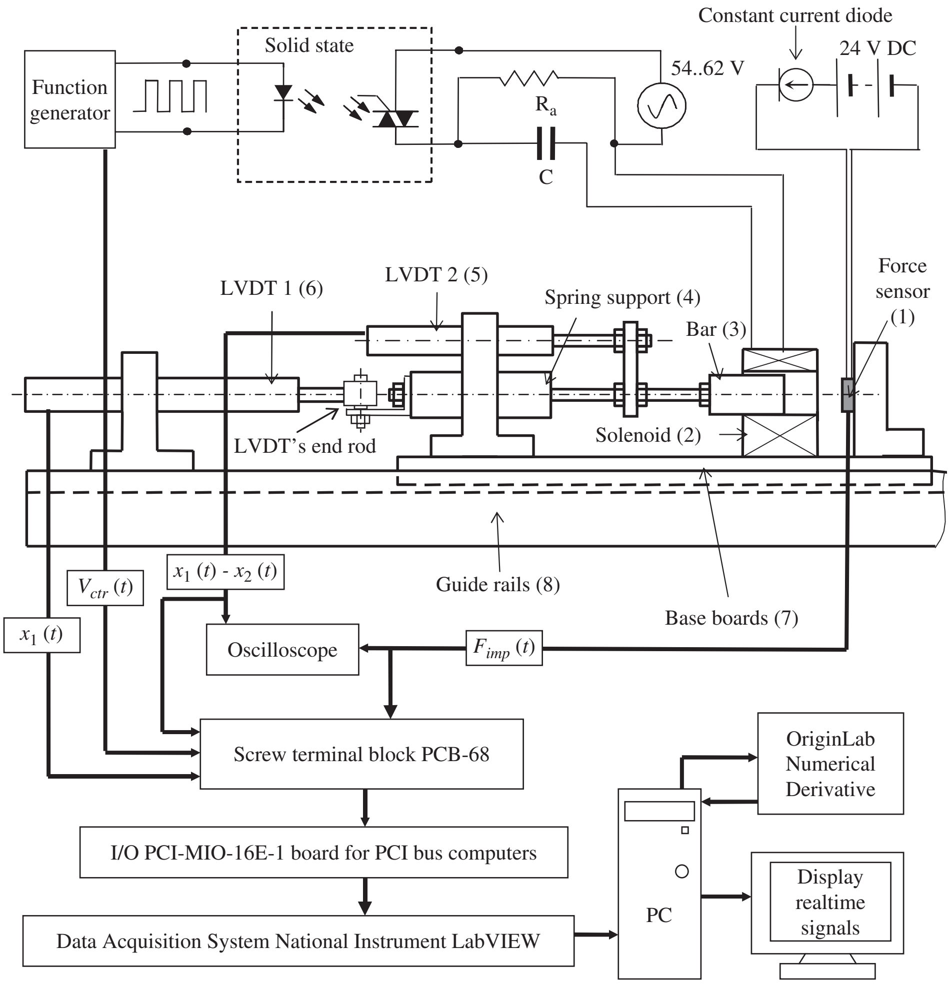

Fig. 2. The comprehensive schematic of the experimental rig. During the experimental test, signals from the function generators, the transducers and the force sensor were connected to the analogue input channels of the data acquisition system. The sampling rate was set to be 10 kHz to be able to capture impulses of the impact force. The experimental data was then saved and analysed off-line by OriginLab and Matlab software.

further refined to match experimental observations of bar motion, resulting in a final value of 11.8 kN/m. The frictional coefficients between the board and the rails was found approximately as 0.235. They were determined using a force balance on the board when it was moving slowly at a constant speed in dynamical equilibrium. To achieve such a slow motion, one end of a piece of string was attached to the board, while the other end was tied to an inertia such that this mass was hanging over the experimental table. In this way, the board and components rigidly attached to it moved forward at a constant speed. The frictional force acting on the board by the rails was balanced by the tension in the string, which was in turn a consequence of the suspended inertia. The stiffness of the spring was measured and found to be 200 N/m. The balance position of this spring was set to be -22 mm, again experimentally determined. When the system does not generate impacts and exhibits bar oscillations only, it is a single degree-of-freedom oscillator. Its undamped natural frequency may be calculated from the spring stiffness and mass. However, the external forcing may assume a different frequency and bar oscillations have been observed to be synchronous to it. This is the frequency of the control signal from the function generator. By measuring the current during pure oscillations of the metal bar, the limits were recorded as 1 and 2.5 A. On conversion to root-mean-square values, the upper and lower limits were 0.7 and 1.75 A. For a typical voltage supply having a root-mean-square value of 54 V, the effective supplied electrical power ranges from 37.8 to 94.5 W, dependent on the position of the metal bar within the solenoid.

3. Results and discussion

To gain insight into the influence of the control signal frequency on the dynamic behaviour of the system, a set of experiments with different voltages and capacitance values was carried out. The solenoid operates effectively from 54 V to values of voltage greater than 100 V. However, when the voltage is higher than 62 V, the impact force generated will exceed the measurement limitation of the force sensor. To obtain values of the impact force, the voltage supply was varied from 54 to 62 V with increments of 2 V; the capacitance was set from 30 to 34 μF with 2 μF steps.

The control signal frequency in a rectangular waveform was varied from 2 to 9 Hz by a function generator. Displacement of the base board and relative motion of the bar for different sets of the control frequency, 58 V and 32 μF are shown in Fig. 3.

From the results obtained, the following can be observed:

1. At values of the control frequency smaller than 5 Hz, as shown in Fig. 3(a) and (b), the motion of the bar relative to the motion of the base plate is irregular (such as at 2.5 Hz) or an oscillation twice the frequency of the control frequency (3.4 Hz). In this range of frequencies of the control signal, the moling machine does not move forward or moves at a very low speed compared to higher frequencies (such as shown in Fig. 3(d) and (e)).
2. At 5 Hz (Fig. 3(c)), the oscillation of the bar is synchronous with the driving frequency of the control signal. However, the bar does not hit the stop at every cycle.
3. At higher frequencies of the control signal (7 and 8 Hz as in Fig. 3(d) and (e)), the oscillation of the bar almost has the same frequency as the frequency of the control signal. High progression rates of the system have been observed for these sets.
4. At the frequency of 9 Hz (Fig. 3(f)), the relative motion of the bar became unstable and the displacement of the moling rig is much smaller than that at 7 Hz (Fig. 3(d)) or 8 Hz (Fig. 3(e)).

Since the bar only oscillates in a half of the solenoid, its magnitudes are always negative. To facilitate the analysis of the process, absolute values of these data will be used in this study. The data have been collected for 67 cycles of the bar oscillation. It was not possible to acquire data pertaining to the 68th cycle onwards because, after this number of periods at the fastest progression rate, the moling rig already reached the limitation of 300 mm of the LVDT used in the rig. Both data of the magnitude of the bar and the displacement of the moling rig for 5 s with respect to frequency of the control signal are shown in Fig. 4.

In Fig. 4, when the frequency of the control signal is less than 5 Hz, the magnitude of the bar displacement has approximately two typical magnitudes: suggesting period two motion as shown in Fig. 3(a) and (b). When the frequency increases from 5 to 8.3 Hz, the magnitude of the bar converges to a sampled value of 35 mm and the progression rate of the moling rig increases. However, when the frequency approaches 9 Hz, the magnitude of the bar displacement is reduced and there is a decrease in the progression rate of the moling machine.

For the set of parameters investigated, the moling machine will move faster if the oscillation of the bar is stable and exhibits large average magnitudes. This condition is satisfied when the frequency of the control signal is in a range of 5–8.3 Hz. In this range, the higher the frequency, the higher the progression rate of the moling machine. The dynamics of the bar oscillation in this range are important for the optimization of the system. When the frequency is greater than 8.3 Hz, the impact force decreases and the progression rate of the moling rig was reduced. Moreover, the magnitude of the bar displacement decreased correspondingly. The undamped natural frequency of the oscillator without impacts and forward motion is 25.82 rad/s. The optimal frequencies of the control frequency are an increase from the undamped natural frequency of the single degree-of-freedom oscillator (4.13 Hz). The actual range of 5–8.3 Hz are shifts in the increasing direction, similar to the shift in natural frequency of impact oscillators.

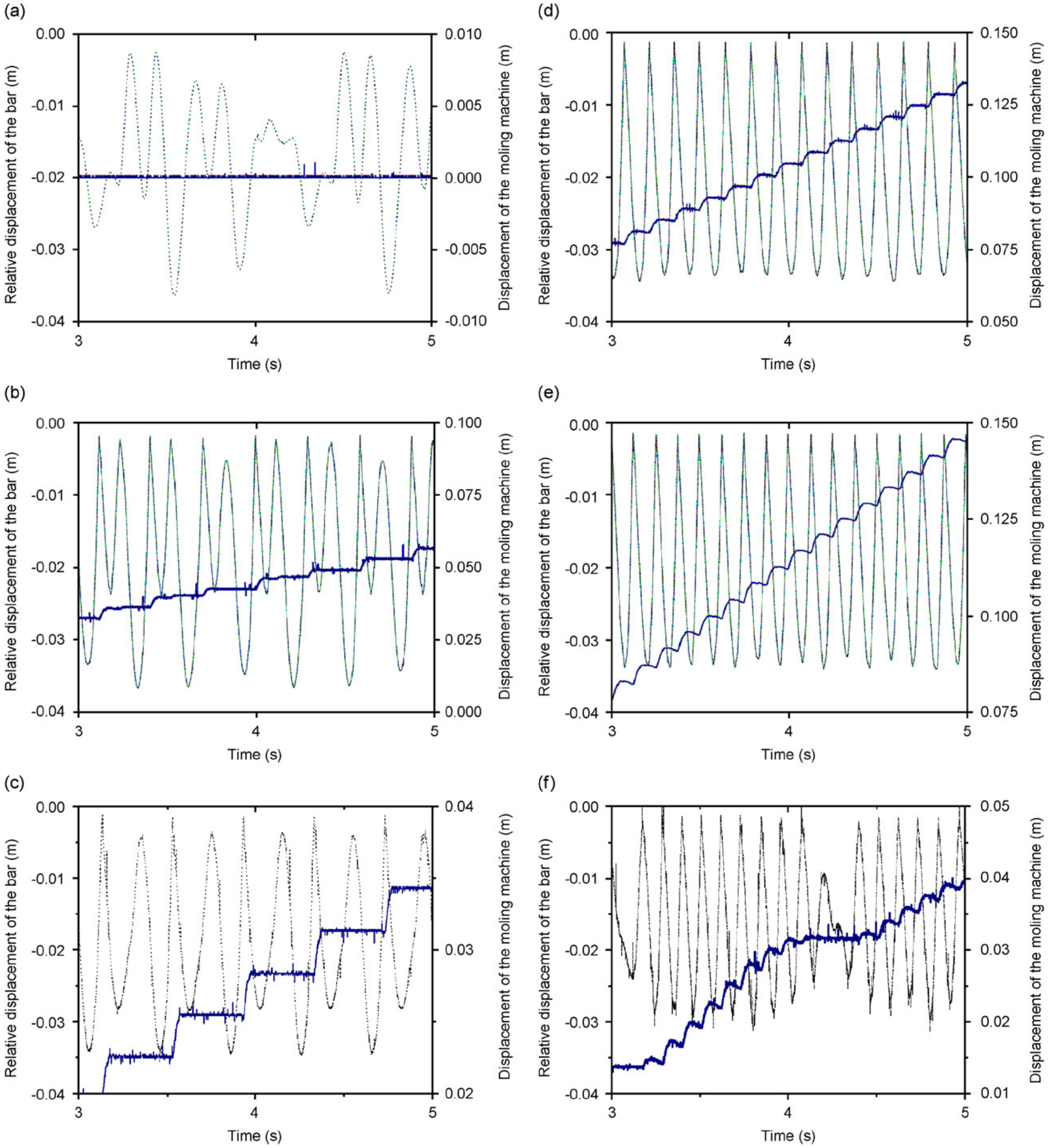

Fig. 3. The absolute displacement of the rig (continuous, blue) and the relative displacement of the bar (dash, black) for different frequency of control signal; (a) 2.5 Hz, (b) 3.4 Hz, (c) 5 Hz, (d) 7 Hz, (e) 8 Hz, and (f) 9 Hz. To scrutinize the relationship between the motion of the bar and the displacement of the moling rig for different frequencies of control signal, two aspects of those motions have been chosen. For the progression rate of the moling rig, the displacement-time history of the base board for 5 s, from the third to the eighth second has been used. For each value of the control frequency, the displacement of the bar was collected as a function of time.

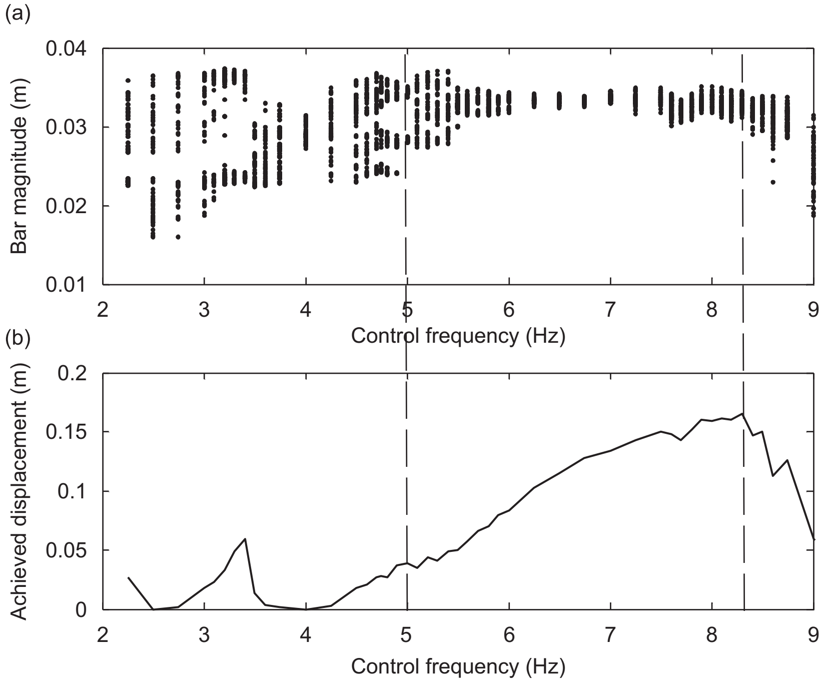

Fig. 4. Bifurcation diagram of the (a) amplitude of the bar and (b) achieved progression of the moling rig for 5 s, respect to the control frequency.

4. Conclusion

A new electromagnetic vibro-impact testing rig has been designed, manufactured and tested in the laboratory. By using a solid-state relay in tandem with a spring, the switching on and off of the power maps the velocity of the metal bar to large velocities at the instant of impact. By varying the input frequency of the function generator, the corresponding frequency of switching is altered. Impact forces and progression rates have a maximum value at a particular frequency. At frequencies higher than this optimum value, there is an abrupt drop in the progression rate. This has been consistently observed for input voltages ranging from 54 to 62 V.

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