

ON THE FORCED VIBRATION TEST BY VIBRODYNE

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Abstract. *In civil engineering, Experimental Modal Analysis (EMA) dynamic tests are powerful aids to the seismic design of new structures, and useful tools for the structural identification of existing structures. EMA tests require to accurately evaluate the harmonic forcing function that is applied to the structure under testing, in order to correctly apply “model updating” procedures. The present work experimentally investigates on the nature of the forcing function applied by a vibrodyne, and its influence on the results of simulations on the dynamics of a single degree of freedom system. By using wireless accelerometers attached to a vibrodyne, we were able to measure the applied accelerations in the time domain, and the applied forcing function under different frequencies. Such an identification procedure was applied both in presence of 3+3 keyed masses, and in presence of 5+5 keyed masses, considering different angular speeds. In both cases, the forcing function applied by the vibrodyne was accurately determined as a function of time. We found out that the actual forcing function is slightly different from the theoretical sinusoidal profile, featuring marked oscillations. The work is completed by the analysis of the dynamic response a simple degree of freedom system under the action of smooth and oscillating sinusoidal forcing functions. A comparison between the results of the analyzed systems highlights marked mismatches in terms of predicted displacements, velocities, and accelerations. We therefore conclude that an accurate knowledge of the applied forcing function in EMA tests is essential in order to correctly identify the properties of the tested structures.*

1 INTRODUCTION

In civil engineering, dynamic tests can be provided both during the construction phase, as theoretical and technical supports to structural design, and after construction, as useful tools to detect the actual material properties and construction details of existing structures, which have been exposed, e.g., to relevant seismic actions [1-8]. The most frequently employed test procedures involve the application of sinusoidal forcing with variable frequency, providing useful information on a variety of structural properties.

The experimental modal analysis (EMA) consists in exciting a building with harmonic forcing functions at different frequencies, in order to record accelerations, velocities and displacements through applied sensors [6-8]. Numerical simulations are synced to the experimental tests with the aim of simulating the test and identifying target structural properties. Such a procedure requires an accurate knowledge of the real forcing function applied to the tested structure, in order to correctly inform the numerical model of the actual experimental conditions.

The present study deals with an accurate experimental procedure to identify the real forcing function applied by a vibrodyne, and its application to the correct identification of the properties of structural models subject to forced vibrations. Vibration tests were initially introduced in mechanical and aeronautical engineering. Nowadays, such tests are diffusely applied also in civil engineering, since they provide useful results regarding the seismic design of structures.

We tested a real vibrodyne in laboratory, in order to accurately detect the generated force time-history (“experimental forcing function” or EFF). The results of such an identification procedure were compared with the theoretical sinusoidal forcing function (“theoretical forcing function” or TFF), accurately measuring the theory-experiment mismatches. Next, we carried out numerical simulations of the dynamic response of a single degree of freedom oscillator subject to both theoretical and experimental forcing functions applied by the examined vibrodyne. Forcing functions with different frequencies were accounted for, with the aim of investigating the influence of the EFF vs. TFF mismatch on simulation results.

2 VIBRODYNE EXPERIMENTAL SETUP

The vibrodyne is an electro-mechanical excitation machine that is able to generate vibrations, with known frequency and amplitude and is frequently employed in EMA [6-8]. Dynamic identification tests are commonly used to determine natural frequencies, mode shapes and damping ratios of structures [1]. The shaker is able to produce unidirectional harmonic forces, with variable intensity sinusoidal laws, thanks to the movement of lumped masses, which are coupled on disks and are rotated around in opposite directions by an electric motor. Such a system allows to vary with continuity the number of disk laps with moving masses, to modulate the excitation frequency. The counter rotating masses generate centrifugal forces resulting in an unidirectional force. The working principle is based on opposed rotations of the disks. Some cylinder-shaped masses placed on the disks, generate a horizontal force history. The counter-rotating masses generate two centrifugal forces. Their resulting force shows zero axial component, and nonzero component in the normal direction (cf. force R_{tot} in Figure 1).

The theoretical force amplitude is directly proportional to the distance r between the rotation axis and the centers of the masses; the square of the angular speed (ω) and the summation (m) of the counter-rotating masses. The theoretical forcing function is sinusoidal in time (t) and it can be written as follows

$$F(t) = \omega^2 \cdot r \cdot m \cdot \sin(\omega t) \quad (1)$$

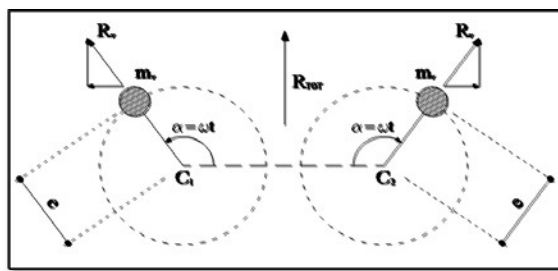


Figure 1: Illustration of the working principle of a vibrodyne.

The maximum force amplitude (in module) is achieved when it results

$$\sin \alpha = \sin(\omega t) = 1 \quad (2)$$

and can be computed as follows

$$F_{\max} = \omega_{\max}^2 \cdot m \cdot r \quad (3)$$

The generated force is instead zero when the masses are next to the plane passing through the rotation axis, since the centrifugal actions produced by each disk are opposite in verse in such a configuration. The sinusoidal force function (1) is applied to the tested structure through the connections placed between the vibrodyne support system and the structure.

2.1 Vibrodyne Design

The electric drive motor movement, with programmable and user-adjustable rotational speed, is transferred, by a timing belt kit, to a rotating part; the rotary motion, in turn, is transmitted to the counter-rotating part, through straight cut gears.

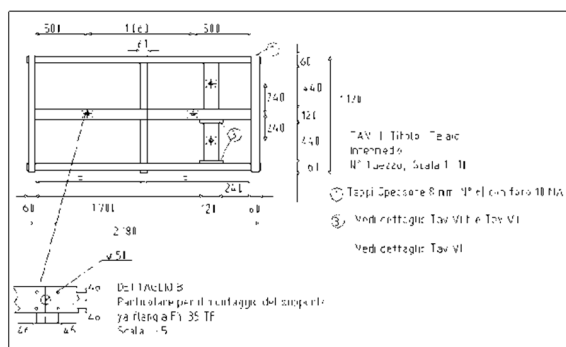


Figure 2: Vibrodyne layout: plan.

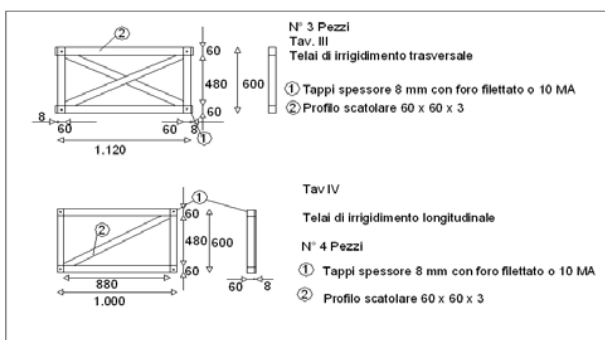


Figure 3: Vibrodyne layout: cross-section.

During the design and the construction of the vibrodyne employed in the present work, a transmission system with timing belt kit on the two belt sides has been adopted, in order to allow the machine to easily work up to a frequency of 50 Hz.

2.2 Test setup

A scrolling system was built to evaluate the forcing function at different frequencies. The scrolling system is composed by rails and rollers put on ball bearings with a very low friction coefficient. Such system permits the vibrodyne to freely scroll according to the force direction. By using wireless accelerometer attached to the vibrodyne, we were able to measure the vibrodyne acceleration in the time domain under different testing frequencies (Figs. 4-6).

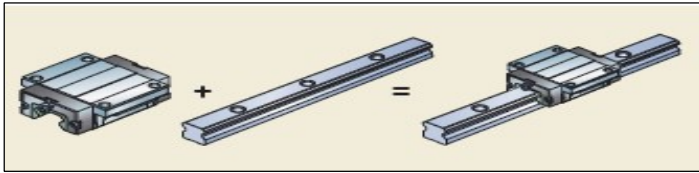


Figure 4: Roller-rail system.

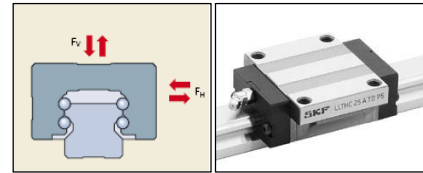


Figure 5: Roller-rail mounting layout.



Figure 6: Vibrodyne on the guides.

2.3 Vibrodyne tests

The system composed by the vibrodyne, the rollers and the guides is secured to two IPE 200 beams through fillet welds (Fig. 7). Such beams are fixed to a steel plate placed on a reinforced concrete slab supported by a pile foundation.



Figure 7: Vibrodyne handling phase.

After fixing the vibrodyne, the eccentric masses were set on the disks. Such operation required an accurate registration of the impellers transmission belt. Tests with 3+3 masses and 5+5 masses were performed. The theoretical forcing parameters generated by the vibrodyne with 5 masses on each disk are shown in Table 1.

2.4 Data acquisition system

A piezoelectric unidirectional accelerometer WiTilt V3 was employed to measure the acceleration produced from rotary motion of the masses. It was attached to the vibrodyne and set in bluetooth connection to a computer for data acquisition and processing. The mass of the overall system was measured by a dynamometer. Upon multiplying the system mass by the measured acceleration, we obtained an accurate recording of the applied forcing function.

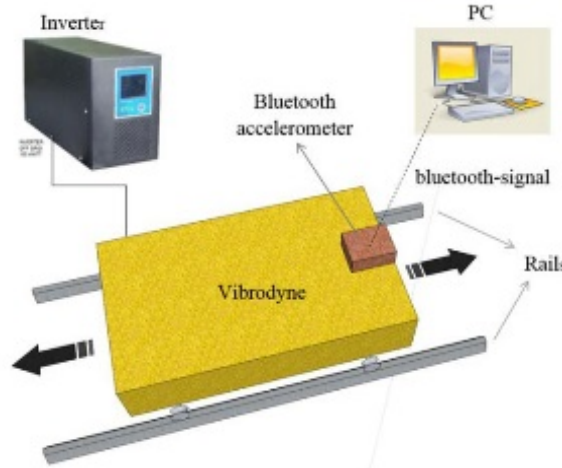


Figure 8: Overall test setup.

rpm	Angular velocity	Angular velocity Ω_{for}	Frequencies f	Period T	Weights on the vibrodyne	Mass	Radius	Forcing F_{max1}	Force generated $F_{max tot}$
60	376,99	6,28	1,00	1,00	153,60	15,66	0,40	247,87	495,74
80	502,65	8,38	1,33	0,75	153,60	15,66	0,40	440,66	881,32
100	628,32	10,47	1,67	0,60	153,60	15,66	0,40	688,53	1377,06
120	753,98	12,57	2,00	0,50	153,60	15,66	0,40	626,95	1982,97
140	879,65	14,66	2,33	0,43	153,60	15,66	0,40	1349,52	2699,04
160	1005,31	16,76	2,67	0,38	153,60	15,66	0,40	1762,64	3525,28
180	1130,97	18,85	3,00	0,33	153,60	15,66	0,40	2230,84	4461,68

Table 1: Maximum force values as a function of frequency.

2.5 Main system parameters

The vibrodyne has a weight of 8,60 kN, net of the masses that can be keyed to the discs in rotation. It is 2120 mm long and 1120 mm large. The machine is equipped with two rotors able to guest up to 11 eccentric masses, 6 of which with a weight of 0,33kN, and additional 5 masses with weight of 0,27 kN. The maximum force value that the machine can apply is equal to 220 kN in presence of eleven masses rotating at 275 rpm. We employed a maximum of 5+5 masses for safety reasons and to avoid an excessive pressure on the test apparatus.

3 COMPARISON BETWEEN THEORETICAL AND EXPERIMENTAL FORCING

We compared theoretical and experimental forcing parameters in presence of 3+3 and 5+5 keyed masses at different angular speeds. The comparison between theoretical and experimental (or real) forcing functions was carried out by plotting the corresponding force-time laws for angular frequencies varying in the range 60 ÷ 180 rpm (cf. Figs. 9-12).

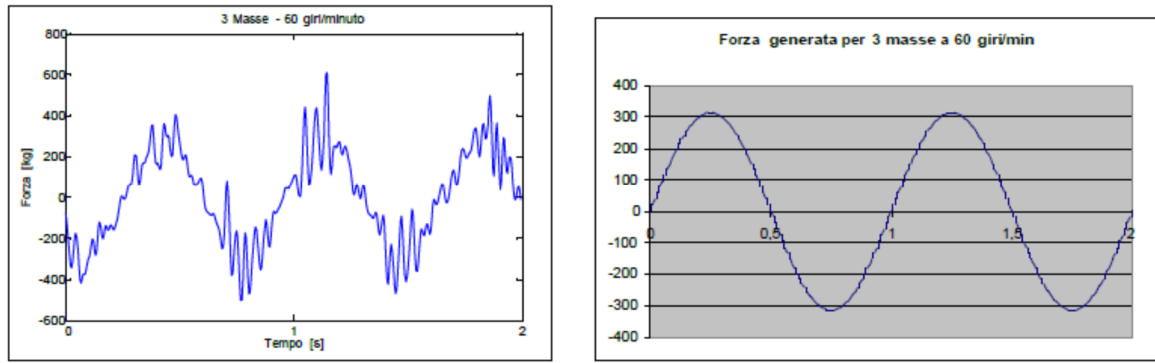


Figure 9: Experimental (left) vs theoretical (right) forcing function with 3+3 masses rotating at 60 rpm.

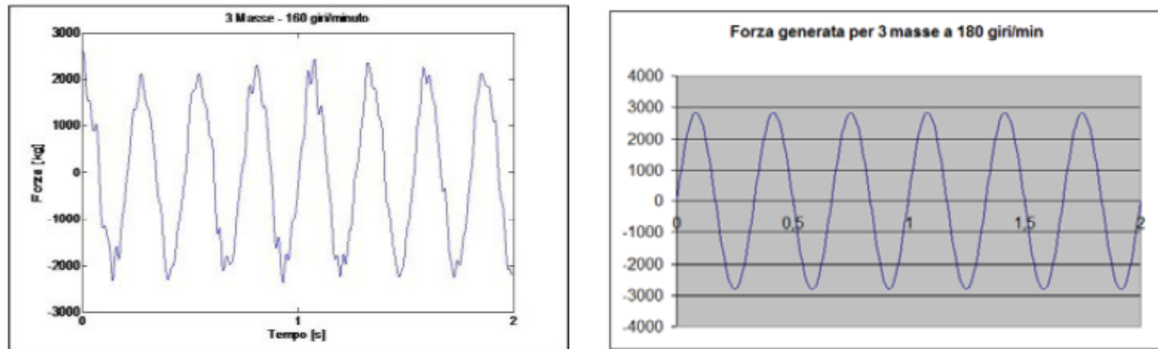


Figure 10: Experimental (left) vs theoretical (right) forcing function with 3+3 masses rotating at 180 rpm.

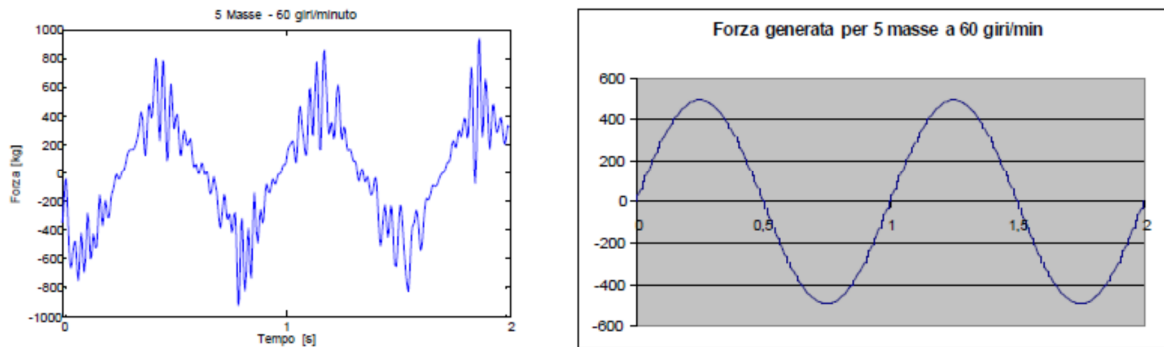


Figure 11: Experimental (left) vs theoretical (right) forcing function with 5+5 masses rotating at 60 rpm.

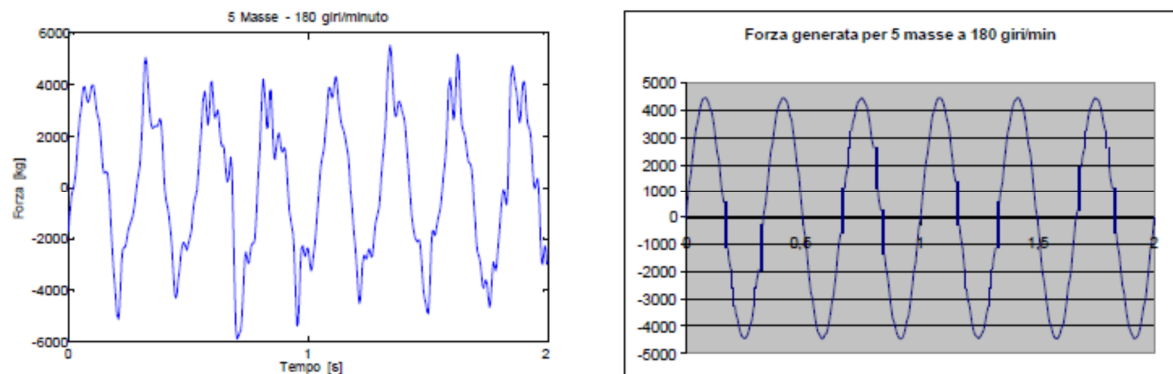


Figure 12: Experimental (left) vs theoretical (right) forcing function with 5+5 masses rotating at 180 rpm.

The results in Figs. 9-12 highlight that the mismatches between theoretical and real forcing functions increase with the angular frequency of the rotating masses, as a consequence of larger friction effects in the high frequency regime. An extensive numerical analysis has shown us that the theory-experiment agreement is satisfactory up to the 160 rpm rotational frequency, and pretty rough for higher rotational frequencies. The results of such an analysis have allowed us to build up a rich numerical database collecting the real forcing functions of the examined vibrodyne over a wide range of test conditions.

4 RESULTS OF DYNAMIC SIMULATIONS UNDER EXPERIMENTAL AND THEORETICAL FORCING FUNCTIONS

The present section aims at investigating the influence of the actual shape of the forcing function applied by a vibrodyne on the results of dynamical simulations of a single degree of freedom (SDOF) system in the elastic regime (Fig. 13). Numerical simulations of the dynamic response of such a system under experimental (real) and theoretical forced functions of the examined vibrodyne were conducted in correspondence with the test setup with 3+3 masses rotating at 180 rpm, on examining a wide range of forcing frequencies. Comparisons between the response of the SDOF system under experimental and theoretical forcing functions are shown in Figs. 14-16 in the frequency domain.

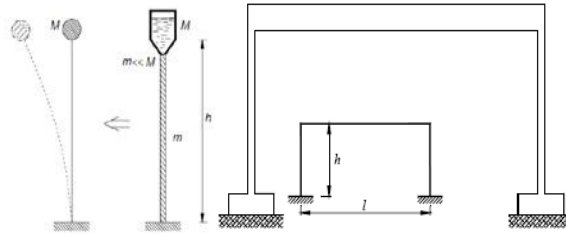


Figure 13: Single degree of freedom system analyzed in dynamical simulations of forced vibrations.

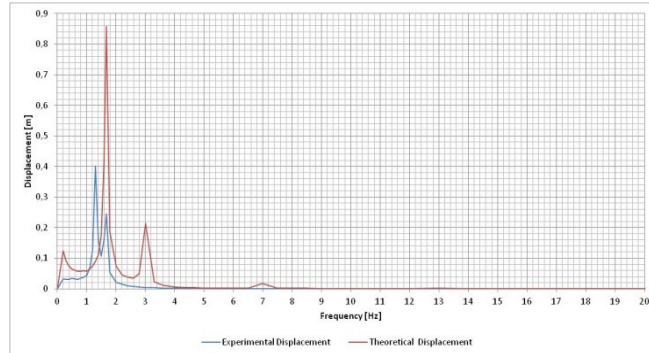


Figure 14: Theoretical (red) vs. experimental (blue) displacement curves in the frequency domain.

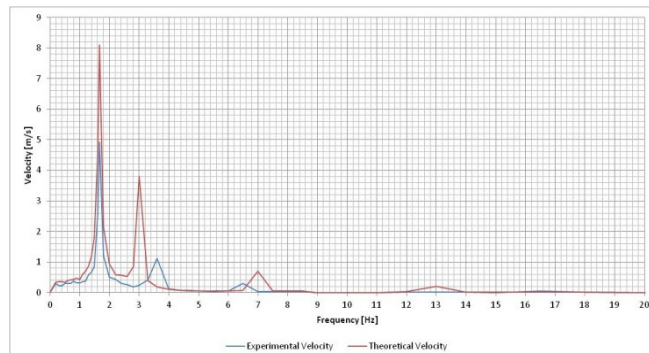


Figure 15: Theoretical (red) vs. experimental (blue) velocity curves in the frequency domain.

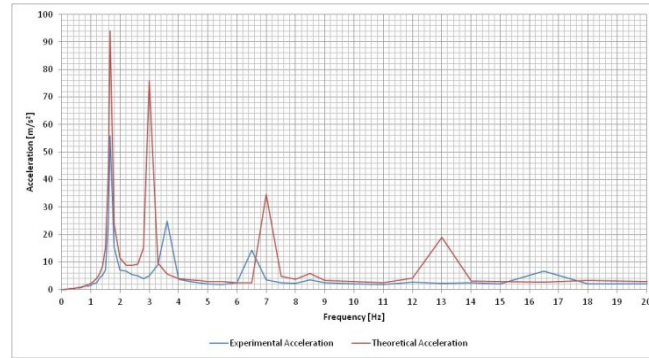


Figure 16: Theoretical (red) vs. experimental (blue) acceleration curves in the frequency domain.

5 CONCLUSIONS

The results shown in Figs. 14-16 of Sect. 4 highlight marked differences in predicted displacements, velocities and accelerations of a SDOF oscillator under real and theoretical forcing functions of a vibrodyne. The study conducted in Sect. 3 showed that such mismatches are particularly significant in correspondence with high angular frequencies of the keyed masses, i.e. for angular frequencies greater than 160 rpm. We are therefore led to conclude that an accurate measurement of the real forcing function applied by a vibrodyne is of fundamental importance when conducting EMA dynamic tests. It appears important to build up a database of the real forcing function applied by such a testing machine over a wide range of operating conditions. Proposed extensions of the present research will deal with accurate vibration tests for diverse structural models, with special focus on the identification of the elastic and damage properties of innovative truss and frame structures [9-15].

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