

3.2 Mechanical parameters identification

Figure 4 and Figure 5 show the plotted response curves as a least square estimation of the measurements for both blocked table and free table configurations.

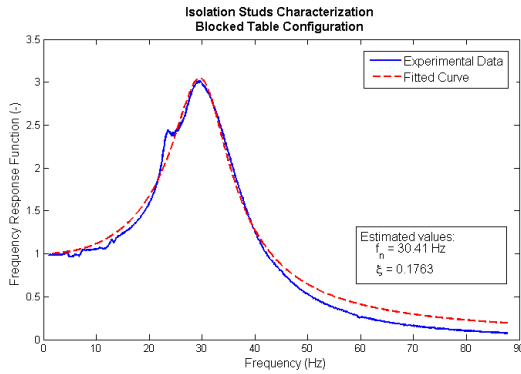


Fig. 4. Response curve of the isolation studs and the one DOF estimation in the blocked table configuration. With f_n the natural frequency and ξ the damping ratio.

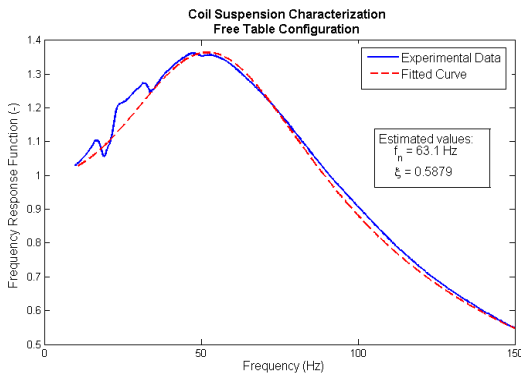


Fig. 5. Response curve of the vibrating table and the one DOF estimation in the free table configuration. With f_n the natural frequency and ξ the damping ratio.

The responses of the system correspond to 1DOF oscillators and using a least square estimation we obtained the values of the mechanical parameters. Those values are given in Table 1 with respect to the parameters given by the manufacturer.

3.3 Modal parameters

The developed method requires an external possibility of making the shaker vibrate and is quite difficult to apply to larger shakers. In this case, modal analysis is an alternative since the mechanical parameters and the modal ones are equivalent to characterize the system.

In this section, a short review of the modal theory will be presented to sufficiently master this equivalence as the relationship between movement, global coordinates, modal coordinates, and modal parameters.

Table 1. Mechanical parameters

Parameter	Symbol	Values	
		Given by the manufacturer[6]	Determined experimentally
Mass of the vibrating table	m_T	60 g	76 g ^a
Mass of the body	M_B	8.3 kg	8.375 kg
Damping of the suspension of the coil	c_T	—	35.43 Ns/m
Damping of the isolation studs	c_B	—	564.3 Ns/m
Stiffness of the suspension of the coil	k_T	$12 \cdot 10^3$ N/m	$11.95 \cdot 10^3$ N/m
Stiffness of the isolation studs	k_B	—	$3.06 \cdot 10^5$ N/m

^aFor testing purposes, the original vibrating table is equipped with an extra 16 g fastening bolt.

3.3.1 Theoretical aspects

If we only consider the purely mechanical system as depicted in Figure 2(a), the global equations (1) can be generalized by:

$$M\ddot{z} + C\dot{z} + Kz = F \quad (2)$$

Where M , C , and K are respectively the mass, damping, and stiffness matrices of the system.

Solving the eigenvalues problem for the free undamped system yields the transformation matrix P between the modal coordinates q and the global coordinates z according [7]:

$$z = Pq \quad (3)$$

The premultiplication by P^T allows to diagonalize M and K since both are symmetrical. Despite that C is also symmetrical, the damping matrix cannot be diagonalized and we can consider a linear combination of M and K [8]. Considering this, we now can express (2) as:

$$m_i \ddot{q}_i + c_i \dot{q}_i + k_i q_i = Q_i \quad (i = 1, 2) \quad (4)$$

The modal parameters m_i , c_i , and k_i are respectively the modal mass, damping and stiffness of the i^{th} mode. Q_i is the modal force and q_i is the modal coordinate of the i^{th} mode.

Whereas the system described by (2) is coupled, the system described by (4) is uncoupled since all the matrices have been diagonalized.

The solutions q_i to the decoupled equations are given by:

$$q_i = \frac{Q_i}{(k_i - m_i \omega^2 + j c_i \omega)} = \frac{Q_i}{P k_i} \quad (i = 1, 2) \quad (5)$$

$P k_i$ is the characteristic polynomial of mode i and j is the complex constant.