# VIBRATION REDUCTION OF A HAMMER DRILL WITH A TOP-DOWN DESIGN METHOD

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## ABSTRACT

Designing vibrating systems is challenging due to component interaction. One approach to reduce the resulting complexity is top-down design where requirements on components are formulated such that the overall system achieves the design goal. Previous work showed how to derive quantitative and solution-neutral requirements on components of a vibrating system, expressed as permissible ranges of impedance. This work adapts the methodology to a practical use case and provides a concrete technical solution: A hammer drill that can cause white finger syndromes to users is equipped with an appropriate vibration absorber. The hammer drill is represented by a lumped mass model and validated using experimental data of a reference design. Solution-neutral and quantitative component requirements on the overall dynamics of the vibration absorber expressed by impedance are derived. They provide a clear target for the component design. A vibration absorber in form of a Tuned Mass Damper (TMD) is designed accordingly. The final design is validated experimentally and shown to reduce the vibration by 47%.

**Keywords**: Top-down design method, Computational design methods, Numerical methods, Product modelling / models

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## 1 INTRODUCTION

Hammer drills and other handheld power tools are essential equipment used on construction sites. Hammer drills transmit what are known as "Hand-Arm vibrations", which can be damaging to the user's hands and arms. High levels of vibration exposure can have significant consequences on an operator's performance and dexterity as well as long-term negative health implications like white finger syndrome (Chetter et al., 1998).

To reduce the vibration in hand-held power tools such as hammer drills, research works, and tool manufacturers began to implement vibration isolation and damping measures. Hwang et al. (2010), for instance, used a vibration isolator to reduce vibration transmissibility for a grass cutter. The module consisted of several leaf springs and was placed between the hand of the worker and the power tool. Additionally, he filled the module with silicone gel to enhance further vibration reduction (Hwang et al., 2010). Cherng et al. (2008) redesigned rivet tools by implementing springs, polymer damper, rubber cushions, and a damping coat to the cylinder of the tools to attenuate vibration. A combination of vibration isolators and passive dynamic absorbers was used by Golysheva et al. (2004) to mitigate the vibration of hand-held percussion machines. He placed isolators between the handle to reduce vibration at high frequencies and attached dynamic absorbers to the handle for suppression of low frequency vibration (Golysheva et al., 2004). Tool manufacturers, such as Hilti AG (2022b), and Makita Ltd. (2022) incorporated decoupled handles and a vibration absorber within the casing to their more expensive hammer drill variant to isolate vibration.

In order to reduce vibration, springs and dampers are used in each of these designs either inside the casing or at the point where the operator and the vibrating tool interact. A disadvantage of implementing vibration reduction components inside the casing is space. To accomplish this, the vibrating tool typically needs to be completely redesigned. In the classical design approach, the mass is predetermined by the designer. Based on the mass, the optimum stiffness and damping parameters are obtained by a bottom-up approach. Numerical methods like Evolutionary Algorithms or Den Hartog's design method are common to determine the optimum parameters (Yang et al., 2021). The use of these methods is time-consuming and an iterative process to meet the design goal.

The V-Model is a common methodology which is applied in product development to simplify the design process of such a system (see Figure 1). The starting point of the V-Model is the definition of requirements on the overall system. Once the specifications have been determined, the product development of the system for vibration reduction follows a Top-Down manner and is broken down into the development of each system component (Bhise, 2017). Thus, each component can be designed separately. The design of these components are significantly impacted by the validation of the component design, followed by the system verification. During the development process at least 30 % of the project's time and effort can be spent on validation and verification (Yadav, 2012). Based on a Top-Down design approach Giannini and Carcaterra (2008) proposed a numerical method to define a frequency distribution based on a desired damping function. The component requirement is defined qualitatively and hence, does not ensure that the system requirement is likewise met, even if the component requirement is satisfied. Determining component requirements is crucial to ensuring that the system accomplishes its design goal and to avoiding system-level iterative testing and redesigns.

In order to reduce vibration, a Tuned Mass Damper (TMD) for a hammer drill is created in this study using the Top-Down Design approach. The dynamics of the hammer drill are analytically evaluated in order to use the Top-Down Design approach. In this research, a testbench is devised and constructed to validate the analytical model. The testbench is used to measure the hammer drill's vibration and perform an experimental model validation. The solution neutral and quantitative component requirements on the dynamics of the vibration absorber are determined from the mathematical model. The aim for reducing vibration across the entire machine is ensured to be met as long as the component requirements are satisfied by the component design. These component requirements include specific targets for component design and requirements for component validation. As a result, only the component level of the design is iterated upon. With the design framework system-level design revisions are avoided, which significantly boosts the efficiency of the entire development process compared to existing design methods. To fulfill the component requirements, a vibration absorber in form of a modular, yet passive TMD is designed. The modular TMD is mounted on the housing of the hammer drill and thus needs no additional space inside the casing. The Top-Down development process is presented in Section 2. In Section 3 the final design is



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validated experimentally to show the vibration reduction. The whole study is outlined and future work is discussed in Section 4.

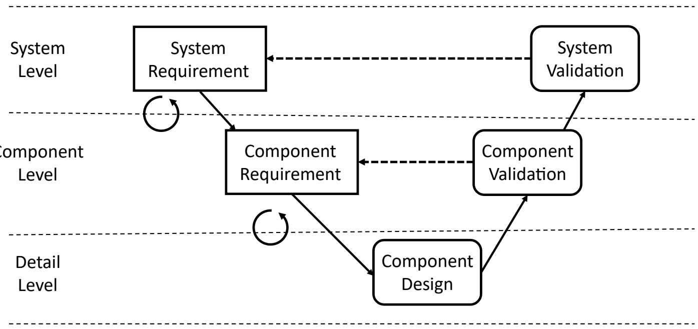


*Figure 1. V-Model*

## 2 TOP-DOWN DEVELOPMENT OF A TMD

### 2.1 Design problem

An operator using a hammer drill to drill into a concrete wall is the scenario under consideration (see Figure 2). This dynamic system consists of three components: the operator (1), the hammer drill (2), and the workpiece (3).The goal is to reduce the vibration of the hammer drill. The quantity of interest is the vibration in acceleration $a(\omega)$ and is frequency dependent. The system requirement is defined as an inequality where $\hat{a}(\omega)$ is the upper limit of the hammer drill's vibration and equals $25\ \frac{m}{s^2}$.

$$|a(\omega)| \leq \hat{a}(\omega) \qquad for\ 80\ Hz \leq \omega \leq 81\ Hz \tag{1}$$

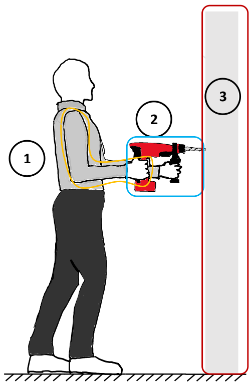


*Figure 2. An operator using a hammer drill to drill into a concrete wall where an operator (1), hammer drill (2), workpiece (3) are depicted*

### 2.2 Dynamic modelling of the hammer drill

The interaction between the hammer drill, operator and their surroundings is studied and described below in order to build an appropriate dynamic model.

#### *2.2.1 Modelling*

The interaction between operator and hammer drill is simulated as a one degree of freedom (DoF) model based on (Golysheva et al., 2004). The one DoF model consists of a mass $m_H$, spring $k_A$, and damper $c_A$ where the mass $m_H$ is the mass of *the hand and the forearm* and the spring $k_A$ represents the stiffness of the arm. Figure 3 depicts the working principle of a hammer drill. The driving shaft receives a rotary motion from the electric motor through a bevel gear. The shaft has a swivel joint attached to it and is connected to the cylinder by a wobble bearing that is mounted to the swivel joint. A Fly-wheel transducer which consists of the wobble bearing and swivel joint then converts the rotary motion from the electric motor into a translational motion by moving the cylinder back and forth. As a result, the drill bit moves translationally (Schäfer, 2018). The displacement $\Delta u$ which symbolizes the translational movement of the drill bit is assumed to be periodic and is expressed as:




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$$\Delta u(t) = A \sin(\omega t) \tag{2}$$

$A$ is the displacement of the drill bit due to the hammering function and $\omega$ is the angular frequency.

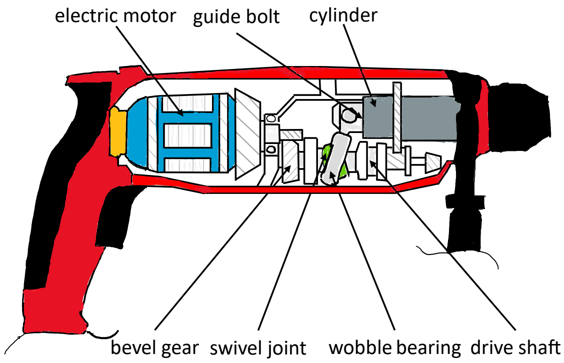


*Figure 3. Working principle of a pneumatic hammer drill based on (Hilti AG 2022a)*

Stated in (Golycheva et al., 2003) the hammering function of a hammer drill is what generates the bulk of its vibration. Thus, the dynamics of the vibrating machine are described using a 1D lumped mass model. The translational movement of the drill bit and cylinder was incorporated into the model to account for vibration in the drilling direction. The mass $m_T$ represents the hammer drill whereas the mass $m_D$ is the mass of the drill bit and the cylinder. Both masses are coupled to another and allow for a displacement $\Delta u$. In addition to rotating and removing concrete powder from the borehole as it drills, the drill bit also transmits pulses of impact energy in the form of hammer blows to the concrete block. There is a compliance between the workpiece and the hammer drill during that action. A spring stiffness $k_W$ is used to describe the compliance between the hammer drill and the workpiece same as in (Schäfer, 2018). The initial lumped mass model is displayed in Figure 4 (a). Since the mass of the hand and forearm $m_H$ and the mass $m_T$ are rigidly connected, they can be represented as one mass $m_{HT}$ (see Figure 4 (b)).

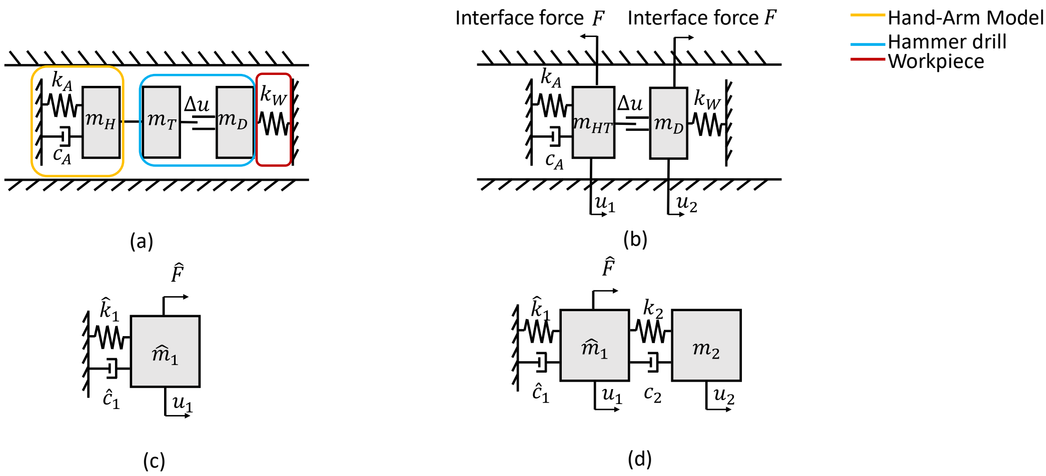


*Figure 4. (a) initial lumped mass model of the system, (b) simplified model, (c) 1 DoF model, (d) model withTMD (left to right)*

To solve a system of equations with three unknowns $u_1$, $u_2$ and the interface force $F$ three equations are needed. The equation of motion for the masses $m_{HT}$ and $m_D$ as well as the relation of $u_1$ and $u_2$ can be used to solve the problem.

$$\begin{cases} m_{HT}\ddot{u}_1 + c_A\dot{u}_1 + k_A u_1 = F\ (I) \\ m_D\ddot{u}_2 + k_W u_2 = -F\ (II) \\ u_2 = u_1 + \Delta u = u_1 + A\sin(\omega t)\ (III) \end{cases} \tag{3}$$

To express the Equation (II) of the equation system (3) as a function of $u_1$ the Equation (III) is inserted in Equation (II).



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$$m_D(\ddot{u}_1 - A\sin(\omega t)\omega^2) + k_W(u_1 + A\sin(\omega t)) \;=\; -F \tag{4}$$

Now Equation (I) of the equation system is inserted in Equation (4) to eliminate the interface force $F$ on the right-hand side of the equation. By rearranging the equation, the force $F$ is represented in terms of $A\sin(\omega t)$. Furthermore, the masses $m_D$ and $m_{HT}$ and stiffness coefficients $k_A$ and $k_W$ are summarized as $\hat{m}_1$ and $\hat{k}_1$, respectively. The Equation is now only dependent on the variable $u_1$.

$$m_D(\ddot{u}_1 - A\sin(\omega t)\omega^2) + k_W(u_1 + A\sin(\omega t)) = -m_{HT}\ddot{u}_1 - c_A\dot{u}_1 - k_A u_1$$

$$\Big(\underbrace{m_D + m_{HT}}_{\hat{m}_1}\Big)\ddot{u}_1 + \underbrace{c_A}_{\hat{c}_1}\dot{u}_1 + (\underbrace{k_A + k_W}_{\hat{k}_1})u_1 = \underbrace{A\sin(\omega t)\,(m_D\omega^2 - k_W)}_{\hat{F}}$$

As a result, the hammer drill can be regarded as a one DoF model (see Figure 4 (c)).The equation of motion becomes an equivalent one DoF system.

$$\hat{m}_1\ddot{u}_1 + \hat{c}_1\dot{u}_1 + \hat{k}_1 u_1 \;=\; \hat{F} \tag{5}$$

or represented in the frequency domain

$$(\hat{k}_1 + i\hat{c}_1\omega - \hat{m}_1\omega^2)u_1 \;=\; \hat{F} \tag{6}$$

with

$$\hat{m}_1 \;=\; m_D + m_{HT}$$

$$\hat{k}_1 \;=\; k_A + k_W$$

$$\hat{F} \;=\; A\sin(\omega t)\,(m_D\omega^2 - k_W)$$

By attaching a TMD to the hammer drill the 1 DoF model is extended to a 2 DoF model ( see Figure 4 (d)).

### *2.2.2 System parameter determination*

The arm stiffness $k_A$ is set as 7.4 $\frac{N}{\text{mm}}$ as in (Cronjäger et al., 1984). The mass $m_H$ is 1.946 kg and is obtained from Plagenhoef et al. (2013) study on the weight of the hand and forearm. Parameter $c_A$ is estimated to be 330 $\frac{Ns}{\text{m}}$. In this paper, a hammer drill with a weight of 2.7 kg is used. The hammer drill's center of gravity is located near the handle. Thus, the mass $m_T$ is set as 2.5 kg. The mass $m_{HT}$ which is the sum of the mass $m_T$ and the mass $m_H$ is consequently 4.646 kg. The mass of the drill bit is measured and is 0.1 kg and the cylinder mass is assumed to be 0.2 kg, hence $m_D$ is 0.3 kg. The stiffness $k_W$ and the parameter $A$ are obtained by matching the model with the experimental measurement. The excitation frequency $f$ is obtained experimentally. The model parameters are summarized in Table 1.

*Table 1. model parameters*

| $m_{HT}$ | $m_D$ | $c_A$ | $k_A$ | $k_W$ | $A$ | $f$ |
|---|---|---|---|---|---|---|
| 4.446 kg | 0.3 kg | 330 $\frac{Ns}{\text{m}}$ | 7400 $\frac{N}{\text{m}}$ | 72608 $\frac{N}{m}$ | 0.0232 m | 81 Hz |

### *2.2.3 Experimental validation*

To validate the dynamic model, a testbench is developed (see Figure 5 (a)). Based on the modeling, linear guides and carriages are used to limit other translational drilling movement to a single horizontal movement. Similar to the actual vibrating system seen in Figure 2, the test rig is made up of three parts. In order to get reproducible test results, the human (1) in Figure 2 is represented by a physical Hand-Arm model as in (Cronjäger et al., 1984). The second component is the hammer drill (2). The workpiece (3) represents the last component and is marked in red in Figure 5 (b). For the three components a base out of aluminium profiles is set up and screwed to the ground which is the rigid boundary condition of the testbench. On top of the profiles, linear guides with carriages are placed. A platform made out of plastic is used to connect the linear carriages. The hammer drill (2) is fixed with a support structure to a metal plate. To represent the stiffness of the arm, a compression


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spring with a stiffness of 7.4 $\frac{N}{\text{mm}}$ is chosen based on parameter $k_A$. The spring is placed between a thick metal block and the metal plate. The right section of the test rig is an apparatus in which the workpiece (3) is moved with a stepper motor towards the hammer drill. To measure vibration during operation triaxial accelerometers are placed on the handle and the housing of the hammer drill.

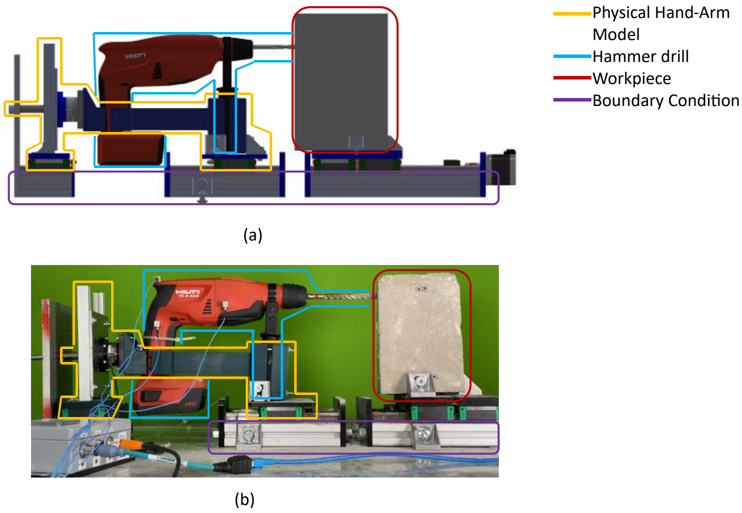


*Figure 5. (a) schematic diagram of the testbench, (b) experimental rig*

The model of the hammer drill is compared with the experimental vibration measurement on the testbench. Figure 6 shows the comparison between them in the time domain. To evaluate and validate both signals, the vibration total value $a_{hv}$ is computed from the experimental measurement. The vibration level is evaluated by the Hand-Arm vibration $a_{hv}$ which is a common quantity for hand-held power tools (VDI-Richtlinie 2057-2). The vibration value is calculated as the root mean square of the frequency weighted acceleration $a_{hw}$ in the drilling direction over a period of time $T_m$.

$$a_{hv} = \sqrt{\frac{1}{T_m} \int_0^{T_m} a_{hw}^2(t)\, dt} \tag{7}$$

The vibration value $a_{hv}$ obtained from the experiment equals 17.4 $\frac{m}{s^2}$. The vibration value $a_{hv}$ from the dynamic model is slightly higher and has a value of 18.6 $\frac{m}{s^2}$.

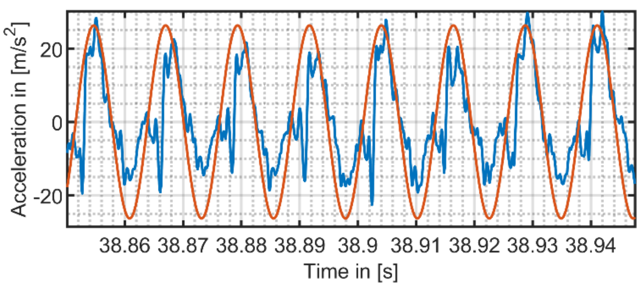


*Figure 6. Comparison between experimental measurement and dynamic model in the time domain; orange line obtained from dynamic model & blue line obtained from measurement*

## 2.3 Top-Down design approach

An overview of the entire system is shown in Figure 7. The vibration absorber and the hammer drill with the hand-arm model, which correspond to the one DoF model in Figure 4 (d), make up the entire system. In the Top Down Design Method, the vibration absorber is regarded as unknown and as a "black box." The TMD component model makes up the black box, which has two degrees of freedom and is marked in yellow in Figure 7 as "C2". The hammer drill with the hand-arm model is marked in red and refers to component 1 "C1".



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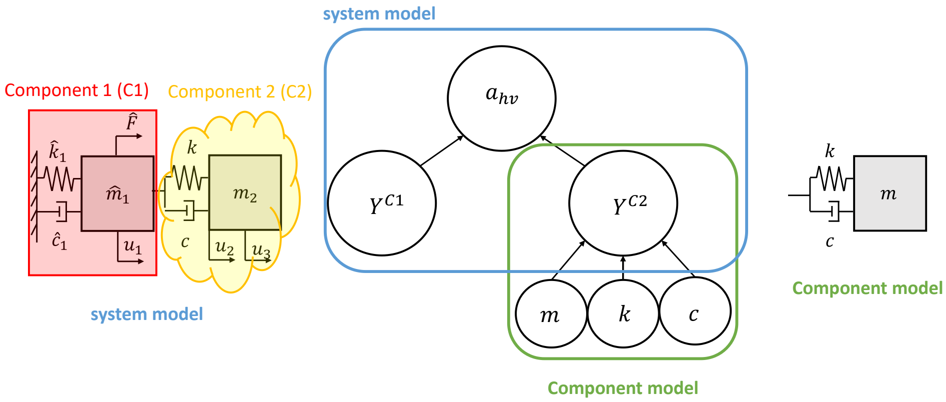


*Figure 7. system and component model*

To describe the dynamics of the overall system, the admittance matrix $Y^{Ass}$ is computed based on the admittance of each component and their connection, using Dynamic Substructuring (Klerk et al., 2006):

$$Y^{Ass} = (I - YB^T(BYB^T)^{-1}B)Y \tag{8}$$

Matrix $Y$ refers to the unassembled state and is constructed by the admittance matrix of each component $Y^{Ci}$. Equation (9) displays admittance matrix of the component in the local coordinate system. The second matrix in Equation (9) represents the admittance matrix in the global coordinate system.

$$Y = \begin{bmatrix} Y_{11}^{C1} & 0 & 0 \\ 0 & Y_{11}^{C2} & Y_{12}^{C2} \\ 0 & Y_{21}^{C2} & Y_{22}^{C2} \end{bmatrix} \rightarrow \begin{bmatrix} Y_{11} & 0 & 0 \\ 0 & Y_{22} & Y_{23} \\ 0 & Y_{32} & Y_{33} \end{bmatrix} \tag{9}$$

$I$ is the identity matrix. $B$ is the signed Boolean matrix and enforces the compatibility condition at the coupling.

$$Bu = 0 \tag{10}$$

Using the fact that $a(\omega) = -\omega^2 u(\omega)$, the quantity of interest $a$ is represented in terms of $u$ for the derivation of the component requirement. $u$ is defined by the assembled admittance matrix $Y_{11}^{Ass}$ and the force $\hat{F}$. By computing $Y_{11}^{Ass}$ the system performance is only dependent on the unassembled admittance matrices of component 1 $Y_{11}$, and of component 2 $Y_{22}$.

$$u = Y_{11}^{Ass} \cdot \hat{F} = \frac{Y_{11}Y_{22}}{Y_{11}+Y_{22}} \cdot \hat{F} \tag{11}$$

The Equation (11) is now inserted in Equation (1).

$$\left| \frac{Y_{11}Y_{22}}{Y_{11}+Y_{22}} \cdot \hat{F} \right| \leq \hat{u} \tag{12}$$

The force $\hat{F}$ is rearranged to the right-hand side.

$$\left| \frac{Y_{11}Y_{22}}{Y_{11}+Y_{22}} \right| \leq \frac{\hat{u}}{\hat{F}} \tag{13}$$

By taking the reciprocal of Equation (13) and simplifying it, the term on the left-hand side is expressed by the reciprocal admittance, the impedance, as:

$$\left| \frac{1}{Y_{11}} + \frac{1}{Y_{22}} \right| \geq \frac{\hat{F}}{\hat{u}} \tag{14}$$

The unassembled admittance matrix $Y_{11}$ refers to the admittance matrix of the hammer drill which is the one DoF model. Hence, $Y_{11}$ can be analytically computed:

$$Y_{11} = \frac{1}{\hat{k}_1 + i\omega\hat{c}_1 - \hat{m}_1\omega^2} \tag{15}$$

The admittance matrix $Y_{22}$ refers to the component of the vibration absorber. Since the vibration absorber is unknown and seen as a black box, $Y_{22}$ is assumed to be complex and thus, can be represented with a real and imaginary part. The real and imaginary part of the impedance are the design variables.



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$$a = Re\left\{\frac{1}{Y_{22}}\right\}, b = Im\left\{\frac{1}{Y_{22}}\right\} \quad (16)$$

By knowing $Y_{11}$ and expressing $Y_{22}$ in a real and imaginary part, the Equation (14) is formulated as:

$$\left|\hat{k}_1 + i\omega\hat{c}_1 - \hat{m}_1\omega^2 + a + b \cdot i\right| \geq \frac{\hat{F}}{\hat{u}} \quad (17)$$

By squaring the Equation (17) on both sides, the component requirements is now the equation of a circle where the center of the circle is $(-\hat{k}_1 - i\omega\hat{c}_1 + \hat{m}_1\omega^2, 0)$ and the radius is $\frac{\hat{F}}{\hat{u}}$.

$$(a + \hat{k}_1 + i\omega\hat{c}_1 - \hat{m}_1\omega^2)^2 + b^2 \geq \left(\frac{\hat{F}}{\hat{u}}\right)^2 \quad (18)$$

Based on the derivation of the component requirements expressed by impedance, the range of the permissible component dynamics is represented in the form of a tube-shaped area (see Figure 8). All designs whose impedance lie outside the tube for frequencies between 80 and 81 Hz fulfill the component requirement. Thus, the solution spaces represent the area outside the tube. To check whether the design achieves the desired target goal, the component dynamics $Y_{22}$ is derived using Equation (6). For the 2 DoF TMD, the equation system is defined as the following:

$$\left[\underbrace{\begin{bmatrix} k_2 & -k_2 \\ -k_2 & k_2 \end{bmatrix}}_{K} + i\omega \underbrace{\begin{bmatrix} c_2 & -c_2 \\ -c_2 & c_2 \end{bmatrix}}_{C} - \omega^2 \underbrace{\begin{bmatrix} 0 & 0 \\ 0 & m_2 \end{bmatrix}}_{M}\right]\begin{bmatrix} u_1 \\ u_2 \end{bmatrix} = \begin{bmatrix} f_1 \\ f_2 \end{bmatrix} \quad (19)$$

Using the fact that $Z = [K + i\omega C - \omega^2 M]$ and $Y = Z^{-1}$, the component dynamics $Y_{22}$ is computed as:

$$Y_{22} = Y_{11}^{C2} = \frac{1}{k_2 + i\omega c_2} - \frac{1}{m_2\omega^2} \quad (20)$$

In this study, the mass of the component 2 has been iteratively added, starting from 1 % of the hammer drill's mass, until the component requirement is fulfilled. The stiffness $k_2$is computed correspondingly based on the mass in each iteration. For the damping coefficient $c_2$ an approximated value is used based on the material of the TMD. For the vibration reduction of the hammer drill the TMD with the following parameters: $m_2$ = 0.28 kg, $k_2$ = 72.2 $\frac{N}{mm}$ and $c_2$ = 8 $\frac{Ns}{\mathrm{m}}$ is the optimal design that can barely fulfill the component requirement.

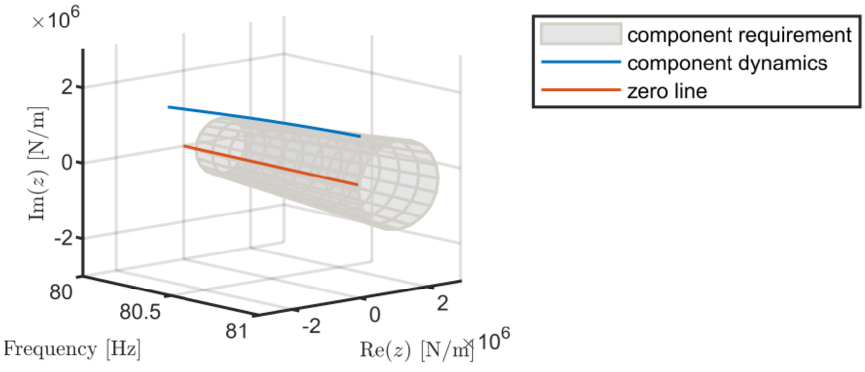


*Figure 8. Solution spaces for vibration reduction and component dynamics of the TMD*

## 2.4 TMD concept design

The TMD is designed to be mounted on the casing of the hammer drill and can furthermore, be applied to a variety of hand-held power tools.The idea of the TMD is to use a cuboid mass made of conventional steel. The cuboid mass has two through holes and is set based on the component requirement as 0.28 kg which is 10% of the hammer drill's mass. Plain-bearing bushes are then inserted with a press fit to the cuboid mass block. Additionally to the plain-bearing bushes, two steel shafts were inserted into the through holes to limit the mass's movement to a horizontal plane. Compression springs are positioned between the mass block and the TMD's case on each of the opposing sides. By placing them in that configuration, the springs



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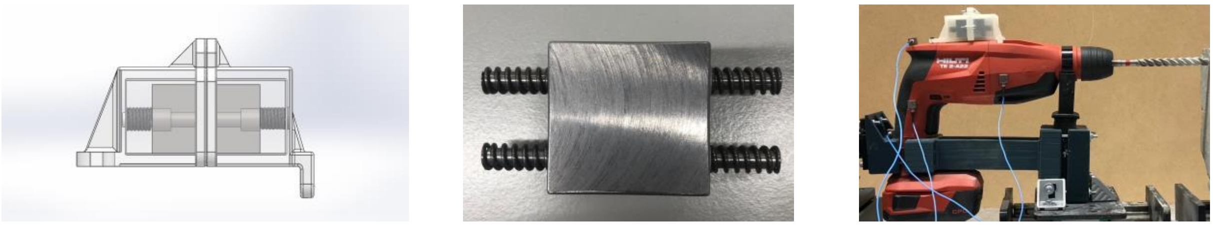
are in parallel and hence, each spring's stiffness is a quarter of the overall absorber's stiffness $k_2$. Figure 9 depicts the components of the TMD.



(a) (b) (c)

*Figure 9. (a) CAD model of the TMD (b) components of the TMD showing the cuboid mass, shafts and compression springs, and (c) deployment of the TMD on the hammer drill*

## 3 VALIDATION

For the validation of the component design, the TMD is mounted on the hammer drill and tested for drilling into concrete.The effect of the vibration absorber on the hammer drill is compared in the time domain with and without the vibration absorber and is displayed in Figure 10 (a) where a significant reduction in vibration due to the TMD can be observed.

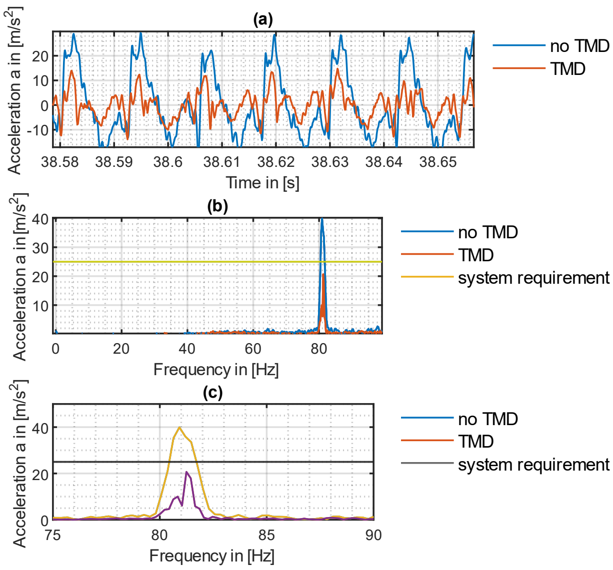



*Figure 10. (a) results from the validation measurement; (b) comparison with the system requirement in the frequency domain; (c) detailed view of (b)*

The vibration value $a_{hv}$ of the hammer drill without TMD is 17.43 $\frac{m}{s^2}$ whereas the hammer drill with TMD is 9.4 $\frac{m}{s^2}$. Hence, a TMD with a mass equal to 10% of the hammer drill's mass reduces the hammer drill's vibration by around 47 %. The acceleration of the hammer drill with and without TMD is displayed in the frequency domain to compare with the system requirement of 25 $\frac{m}{s^2}$ ( see Figure 10 (b)). The acceleration of the hammer drill during operation is 40 $\frac{m}{s^2}$. With the designed TMD the overall vibration of the hammer drill is reduced to 20 $\frac{m}{s^2}$ and hence, achieves the target goal ( see Figure 10 (c)).

## 4 CONCLUSION

In this paper a systematic method is proposed to derive component requirements quantitatively, so that the whole system meets the design objective. The proposed Top-Down Design methodology is utilized



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in the study to reduce hammer drill vibration.. A one DoF model is used to represent the hammer drill, operator, and workpiece. Utilizing a testbench, the one DoF model is then empirically validated. The component requirements are derived to systematically design a TMD based on the system model. The component requirements provide a clear target for component design, avoiding design iteration at the system level and improving efficiency in the development process. The impedance represents the component requirements as a tube-shaped region. The overall dynamics (impedance) of the TMD must lie outside the tube in order to satisfy the component requirements and, consequently, the system requirement to reduce vibration. The mass of the TMD is manually added after being designed in this fashion. The vibration absorber is shown to reduce vibrations by 47 % and hence fulfills the system requirement.

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