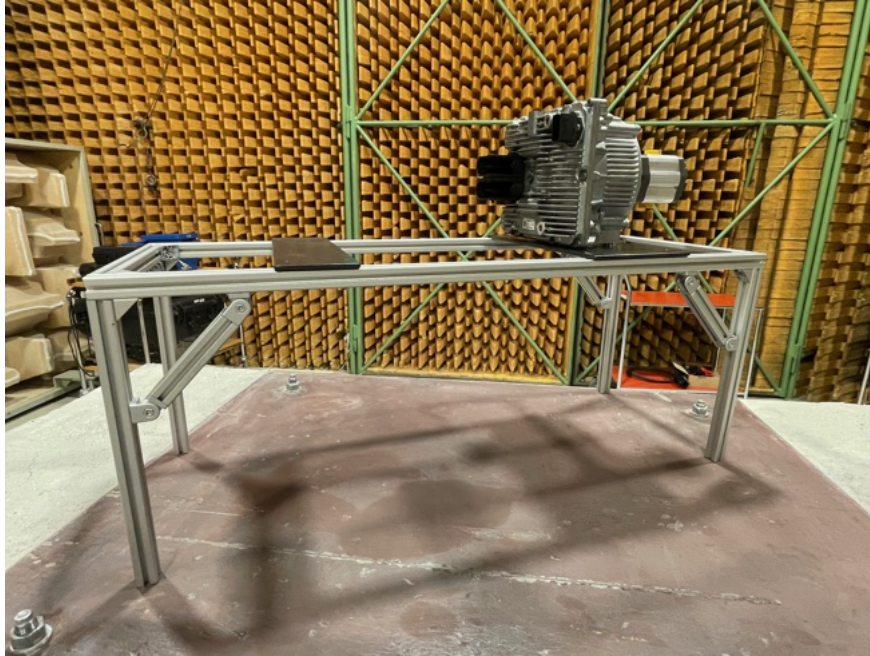




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Structure-borne noise assessment of automotive components using the blocked force approach

Nicholas Man

DEPARTMENT OF ARCHITECTURE AND CIVIL ENGINEERING

CHALMERS UNIVERSITY OF TECHNOLOGY
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Nicholas Man



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Department of Architecture and Civil Engineering
Division of Applied Acoustics
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Gothenburg, Sweden 2026

Method validation for obtaining the blocked force of a source component via in-situ methods

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Cover: Coupled structure representation used for blocked force analysis.

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Abstract

Changes in the vehicle industry have heightened the demand for performing acoustic and vibration analysis earlier in the product development process. This is challenging as NVH issues are often identified during full system integration, when design changes are costly and limited due to frozen vehicle configurations. To address this, virtual prototypes can be developed where components are independently characterized at their interfaces and transferred between assemblies. A major limitation of this approach is that vibration forces from source components are not inherently independent, as interface forces obtained through conventional methods depend on boundary conditions and are therefore not transferable between systems. To overcome this, the in-situ blocked force method has been introduced to characterize source components such that the resulting interface forces are intrinsic to the source and independent of the receiving structure. This thesis presents a case study applying the in-situ blocked force method to an electro-hydraulic steering pump as a source component. The objectives are to validate the in-situ blocked force method, demonstrate that the derived blocked forces are independent of the receiving structure and transferable between systems, and establish a workflow for digital twin-based acoustic and vibration analysis. The results show good consistency of the derived blocked forces across two dynamically different receiving structures. This consistency is confirmed through cross-validation between test configurations. Furthermore, the developed digital twin enables component-based transfer path analysis and the simulation of modification scenarios.

Keywords: Blocked Force, Virtual Prototype, Digital Twin, Component Based Transfer Path Analysis.

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Nicholas Man, Gothenburg, June 2026

Usage of AI tools

In this study, AI tools were used as a language aid to improve grammar and phrasing, as well as to assist in creating text in *L^AT_EX* format. No AI tools were used for data or image processing, or for the analysis or interpretation of the results.

List of Acronyms

Below is the list of acronyms that have been used throughout this thesis listed in alphabetical order:

BF	Blocked Force
CAE	Computer-Aided Engineering
C-TPA	Component-Based Transfer Path Analysis
CoG	Center of Gravity
DAQ	Data Acquisition
DOF	Degree of Freedom
EHPS	Electro-Hydraulic Power Steering
FEA	Finite Element Analysis
FRF	Frequency Response Function
ISO	International Organization for Standardization
MIMO	Multiple Input Multiple Output
NVH	Noise, Vibration, and Harshness
OEM	Original Equipment Manufacturer
RPM	Revolutions Per Minute
SNR	Signal-to-Noise Ratio
TPA	Transfer Path Analysis
VAP	Virtual Acoustic Prototype
VP	Virtual Point
VPT	Virtual Point Transformation

Nomenclature

Below is the nomenclature of symbols and notation used throughout this thesis.

General notation

t	Time [s]
f	Frequency [Hz]
ω	Angular frequency [rad/s]
i, j, k	General indices representing degrees of freedom or measurement locations

System matrices and dynamic quantities

M	Mass matrix [kg]
C	Damping matrix [Ns/m]
K	Stiffness matrix [N/m]
D	Dynamic stiffness matrix [N/m]
H	Frequency response function matrix [-]
Y	Mobility matrix [(m/s)/N]
Z	Mechanical impedance matrix [N/(m/s)]
T	Transformation matrix [-]
λ	Eigenvalue [-]
κ	Condition number [-]

Mechanical quantities

x	Displacement [m]
v	Velocity [m/s]

a	Acceleration [m/s ²]
g	Contact force [N]
f_{bl}	Blocked force [N]

Virtual point and substructuring quantities

R_v	Velocity transformation matrix [-]
R_f	Force transformation matrix [-]
T_v	Velocity transformation matrix [-]
T_f	Force transformation matrix [-]
ρ_v	Response correlation coefficient [-]
ρ_f	Excitation correlation coefficient [-]
γ^2	Coherence [-]

Coordinates and subscripts

x, y, z	Translational coordinate directions
R_x, R_y, R_z	Rotational degrees of freedom about the x-, y-, and z-axes
CG	Center-of-gravity virtual point
C	Coupled structure or coupled system
A	Source component
B	Receiving structure

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1

Introduction

1.1 Problem statement

Characterization of automotive components using the blocked force method enables earlier assessment of NVH systems in the product development process. Recent updates in the literature and data acquisition have provided a comprehensive process to incorporate the blocked force approach into Component-Based Transfer Path Analysis (C-TPA) for both complete-vehicle and component-level testing, enabling dynamic substructuring for NVH development. While industry has been gradually adopting this methodology to reduce product development cycles, additional demonstrations are needed to verify and validate the process so OEMs and suppliers can fully incorporate it into their vehicle and product development programs. By applying the method to a component, a demo case for source characterization can be established to validate the benefits and limitations of the blocked force approach. An adjustable component test bench was designed and built to evaluate the in-situ blocked force method and virtual point transformation on two structures, so that blocked force is shown to be an inherent property of a source component, and future components can be added to a database for assessment in a virtual acoustic prototype for CAE analysis.

2

Theory

The role of a Noise, Vibration, and Harshness (NVH) engineer is to identify noise and vibration issues and develop noise control solutions. Solutions range from selecting acoustic and vibration isolation material to redesigning entire components. The common challenge for NVH engineers across industry is to identify NVH problems earlier in the program development cycle, before designs are frozen, troubleshooting time is limited, and modifications are costly. Therefore, manufacturers and suppliers are shifting away from physical prototypes and moving toward digital solutions called digital twins, whereby using Computer-Aided Engineering (CAE) tools, a virtual acoustic prototype can be developed to identify and solve NVH issues earlier in the development phase.

2.1 Virtual acoustic prototype (VAP)

Virtual acoustic prototypes are digital representations of a physical product or vehicle architecture that capture the vibro-acoustic behavior of a physical system. By virtually dividing a model or system into its individual components, VAPs allow NVH engineers to troubleshoot issues rapidly, perform "what-if" scenarios, and set performance requirements with fewer iterative physical testing loops. VAPs do not replace testing, but assist engineering analyses such as component-based transfer path analysis [1].

2.2 Component-based transfer path analysis

Component-Based Transfer Path Analysis (C-TPA) is a diagnostic method to assess the vibro-acoustic energy propagation in a complex structure by decomposing the assembly into its individual components [2]. Components are defined as structures that can be separated from the assembly and can range from individual elements, such as a bracket, to complex parts like sub-frame assemblies or rotating machinery. By observing the excitation and responses at the interface between components, system-level vibro-acoustic analysis can be performed without knowledge of each component's internal mechanisms. By analyzing the decomposition of an assembly into its components, the dominant airborne and structure-borne paths can be identified, enabling efficient noise mitigation [3]. Components are categorized by their properties: as source components that generate excitations or as receiver components that transmit vibro-acoustic energy.

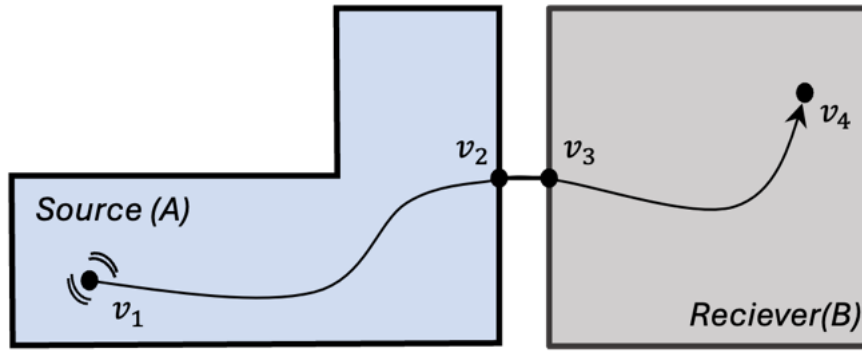


Figure 2.1: Diagram of components in Component-Based TPA shows excitation being generated from the source at point 1 and the vibration energy transferring to the receiver between points 2 and 3 to target location 4

2.3 Receiver component characterization



(a) A simple bracket can be described as a receiver component (b) A truck chassis can be described as a receiver component

Figure 2.2: Receiver structures can vary in size and design complexity

Receiver components do not independently generate excitation; instead, they respond to external excitation and transmit energy to surrounding components. Receiver components are characterized by the frequency response function (FRF), the linear time-invariant kinematic relationship between input and output degrees of freedom (DOF) at interface points. DOFs are physical quantities at the interface and are often force, velocity, acceleration, or sound pressure level. Frequency response functions describe the response and modal characteristics of a component. [4].



Figure 2.3: Frequency response functions have a magnitude and phase component

An example of an FRF used to describe components is dynamic stiffness $D(\omega)$, which represents the relationship between a force $f(\omega)$ and known displacement $x(\omega)$ [4]:

$$D(\omega) = \frac{f(\omega)}{x(\omega)} = -\omega^2 M + i\omega C + K \quad (2.1)$$

where

- M is the mass of the component
- C is the damping of the component
- K is the stiffness of the component

These physical properties are often provided by suppliers, determined from Finite Element Analysis (FEA), or measured if the component is available for testing. Measurements are performed by applying a known force at a defined location using an impact hammer or mechanical shaker and measuring the response at another location with a transducer, such as a microphone or accelerometer. The test procedures are defined in ISO 7626-5 for impact hammer [5] and ISO 7626-2 for mechanical shaker testing [6]. The physical quantities commonly observed at a DOF on a receiver component are force relative to velocity or acceleration. Therefore, the FRFs of mechanical impedance ($Z(\omega)$) or dynamic mass ($M(\omega)$) are commonly used to describe a receiver component. Converting between the FRFs in the frequency domain involves taking the time-based integral or derivative by multiplying or dividing by $i\omega$.

$$Z(\omega) = \frac{f(\omega)}{v(\omega)} = i\omega M + C - \frac{i}{\omega} K \quad (2.2)$$

$$M(\omega) = \frac{f(\omega)}{a(\omega)} = M - \frac{i}{\omega} C - \frac{1}{\omega^2} K \quad (2.3)$$

Additionally, the inverse FRF can be used to describe a system if the output DOF is known and the input is unknown, so long as the FRF meets the requirement of reciprocity.

$$Y(\omega) = \frac{v(\omega)}{f(\omega)} \quad (2.4)$$

2.3.1 Reciprocity

If the system is stable and linear, the DOF relationship for an FRF can be reversed by reciprocity, which states that the same FRF is obtained when the excitation and response locations are interchanged [3].

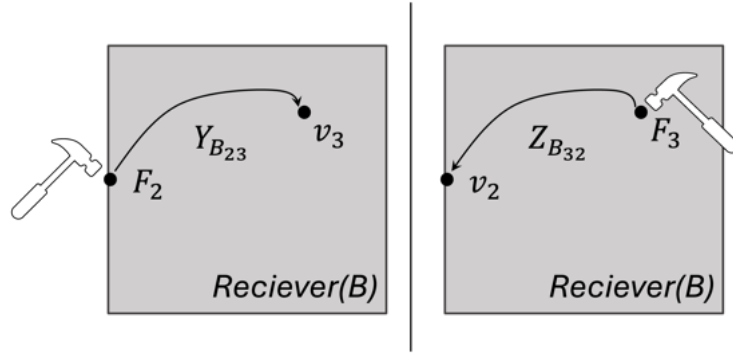


Figure 2.4: If a system shows reciprocity, the FRF of the excitation and the response can be the same

This is advantageous as an inverse FRF can be easier to measure for DOFs such as moments and angular velocity, or for response DOFs with a low signal-to-noise ratio in the frequency range of interest [3]. For example, the velocity at a point can be known if the input force and mechanical impedance are known. However, by the definition of mechanical impedance, there exist cases where the velocity is zero, and the FRF is undefined. Thus, since it is easier to excite the force for a given frequency range, mobility is preferred for TPA. FRFs show stability by comparing repeated measurements between the same DOFs with an average FRF and demonstrate linearity between the two DOF signals with the frequency-based quality indicator of coherence. Coherence greater than 0.9 is considered to be acceptable and is calculated with Equation 2.5

$$\gamma_{xy}^2(\omega) = \frac{|S_{xy}^2(\omega)|}{S_{xx}(\omega)S_{yy}(\omega)} \quad (2.5)$$

where

- $S_{xy}(\omega)$ is the cross spectrum of the input and output signal
- $S_{xx}(\omega)$ is the auto spectrum of the input signal
- $S_{yy}(\omega)$ is the auto spectrum of the output signal

2.3.2 Simulation characterization methods

Characterization of receiver structures can be completed via simulation analysis. Using FEA and SEA tools, the FRF at any point can be calculated and used as input to a VAP. Additionally, once a blocked force is derived, a coupled structure model can be created, allowing the blocked force to serve as an input load to the simulation. To build confidence in the model before adding a simulated receiver to the component database, the model can be correlated by comparison to a modal measurement.

2.4 Source component characterization

Source components, such as pumps and motors, generate vibration due to moving mechanisms. The movement of the mechanisms causes airborne, structure-borne, or flow-borne energy to surrounding components and environments [7]. The internal force mechanisms of source components are often in harsh environments or are impractical to measure directly. However, like receiving components, a source component can be characterized by the DOFs at the interface points. For the source component to be used for C-TPA analysis, it should be represented in such a way that it [3]:

- is equivalent to the internal forces for the same operating conditions
- is defined at its interfaces
- can be practically determined
- is independent and transferable

2.4.1 Contact forces

Contact forces (g) are forces at the interface between components and represent the vibro-acoustic excitation from the source component on the receiver component.

$$v_3 = Y_{B_{32}} g_2 \quad (2.6)$$

Equation 2.6 shows the general relationship between velocity and contact force where

- v_3 is the velocity at receiver location 3
- $Y_{B_{32}}$ is the mobility of subsystem B from interface location 2 to location 3
- g_2 is the contact force at the interface location 2

By representing the source component as a contact force, the response at any location on the receiving structure can be determined, provided the FRF between the interface and the target location is known.

$$\begin{pmatrix} v_2 \\ v_3 \end{pmatrix} = \begin{bmatrix} Y_{B_{22}} \\ Y_{B_{32}} \end{bmatrix} g_2 \quad (2.7)$$

where

- v_2 is the vibration at interface location 2
- v_3 is the vibration at receiver location 3
- $Y_{B_{22}}$ is the driving point of subsystem B at interface location 2

- $Y_{B_{32}}$ is the driving point of subsystem B from interface location 2 to location 3
- g_2 is the interface force at location 2 of subsystem B

Depending on the system, there are three common methods to determine the contact force:

- Direct force method
- Mount stiffness method
- Matrix inversion method

2.4.1.1 Direct force method

The direct force method determines the contact force by direct measurement between the source and receiver components during operational measurements [8][9]. ISO 18312 defines the procedure for obtaining the contact force via the direct force method [10].

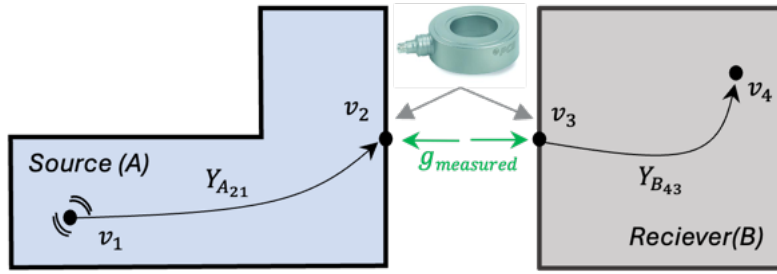


Figure 2.5: A force transducer can be placed directly between components to obtain the contact force

By using a transducer to measure the force directly between a rigid interface, the measured force can be applied to both sides of the interface.

$$g_2 = g_3 = g_{measured} \quad (2.8)$$

While conceptually straightforward, the direct force method is difficult to implement due to mounting constraints, as the transducer mass should not alter the dynamics at the interface at the frequency of interest. Additionally, it is preferred that all 6 DOFs be measured simultaneously, which is challenging because force moments require additional setup [3].

2.4.1.2 Mount stiffness method

If the interface between the source and receiver is mounted resiliently, the mount-stiffness method can be used to indirectly measure the contact force. Also defined in ISO 18312 [10], the mount stiffness method is based on Hooke's law, where force is equal to mount stiffness times displacement across a mount [9][11].

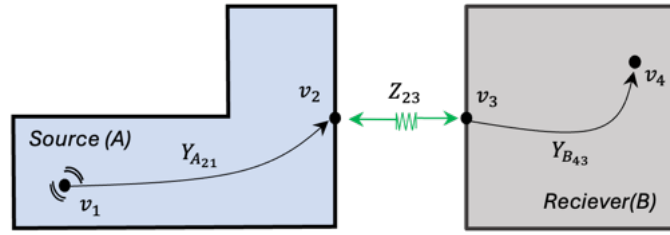


Figure 2.6: The contact force can be obtained with the relative velocity before and after a mount

$$g = Z_{23}(v_2 - v_3) \quad (2.9)$$

The relationship between contact force, velocity and mount stiffness is shown in equation 2.9 where

- Z_{23} is the mount transfer impedance
- v_2 is the velocity on the source side of the mount
- v_3 is the velocity on the receiver side of the mount

A challenge with the mount stiffness method is that it requires mount properties, which are often not provided, because mounts are normally rated for static mount stiffness, not the frequency dependent dynamic mount stiffness [8].

2.4.1.3 Matrix inversion method

In addition to the above methods to directly measure the contact force, an indirect method known as the matrix inversion method can also be used. By measuring the response at additional arbitrary locations on the receiving structure, the blocked force can be determined by addressing the question "What force needs to occur at the interface for the response to occur at these additional locations?"[3]. By rearranging the mobility matrix in equation 2.6, the contact force can be calculated as the product of the response DOF matrix under operating conditions and the inverse of the FRF matrix [3].

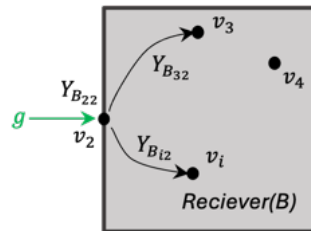


Figure 2.7: The contact force can be derived using the velocity and mobility at a known point on the receiver component

$$g_2 = \begin{pmatrix} v_2 \\ v_3 \\ v_i \end{pmatrix} \begin{bmatrix} Y_{B22} \\ Y_{B32} \\ Y_{Bi2} \end{bmatrix}^{-1} \quad (2.10)$$

Equation 2.10 shows the relationship between contact force, velocity and mobility where,

- (v) is an array of velocities at the interface, as well as an arbitrary indicator located on the receiving structure B from operational measurement
- matrix $[Y_B]$ represents the mobility matrix between the interface and indicator location of the receiving structure

Two measurements are required to determine the blocked force via matrix inversion: an operational measurement of the coupled system to determine operational velocity at the indicator locations, and a mobility measurement of the receiver component only to determine the FRF between the interface and the indicator location [12][3]. The benefits of the matrix inversion method are that no modifications are needed to the components of the system, and that it is possible to derive moment forces [3]. When using the matrix inversion method, caution is warranted when observing local resonances and anti-resonances in the receiving structure as the local minima and maxima result in the dynamic behavior not being captured properly.

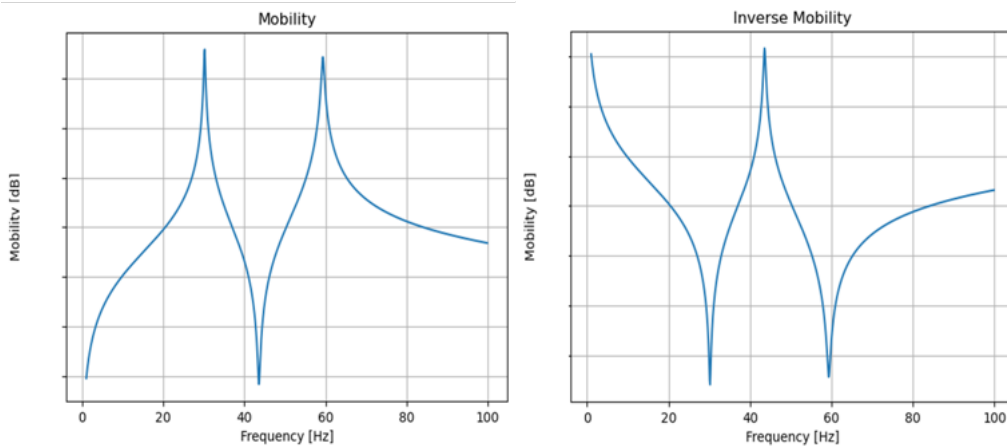


Figure 2.8: When a FRF is inverted, the anti-resonances become an asymptote. This can be alleviated with the addition of damping to the receiving structure

2.4.2 Limitations of the contact force

A significant limitation for the contact force is that all methods to derive the interface forces depend on the mobility of the receiver structure. The effect of this limitation is that the contact force applies only to the system from which it was derived and is no longer valid if the receiver system is changed. Therefore, it does not meet the requirement of being independent and transferable, and it is not possible to answer the question "if we modify or change the receiving structure, will we have noise and vibration issues" with the contact force.

2.4.3 Blocked forces and free velocity

The blocked force (f_{bl}) describes the interface force when the source component is fixed to an infinitely stiff constraint so that the motion at the interface is zero in

all directions. By fixing the motion at the interface, the influence of the dynamic behavior of a receiving structure is removed [3]. The force from the source component is fully counteracted and applied back onto the source component, as seen in equation 2.11

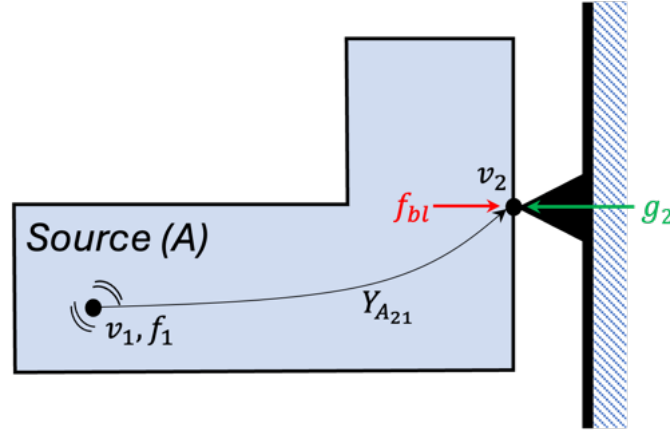


Figure 2.9: Diagram depicting direct measurement of blocked force.

$$f_{bl} = g_2|_{v_2=0} \quad (2.11)$$

Since internal forces generate the vibrational energy of the source component, the blocked force can be mathematically related to internal forces using the driving-point and transfer-point mobilities as seen in equation 2.12.

$$f_{bl} = Y_{A22}^{-1} Y_{A12} f_1 \quad (2.12)$$

The direct method describes an ideal case to measure the blocked force under a fixed condition where an infinitely rigid receiver is difficult to observe in practice unless the source component is very small [2] or the impedance of the receiver structure is significantly larger than the source component [3]. Alternatively, the blocked force can be determined using the free velocity $\tilde{v}$, which observes the velocity at the interface of a source component under free-free boundary conditions. As described in ISO 9611 [13], the source is constrained under free-free boundary conditions, and the operational velocity at the interface is measured [3].

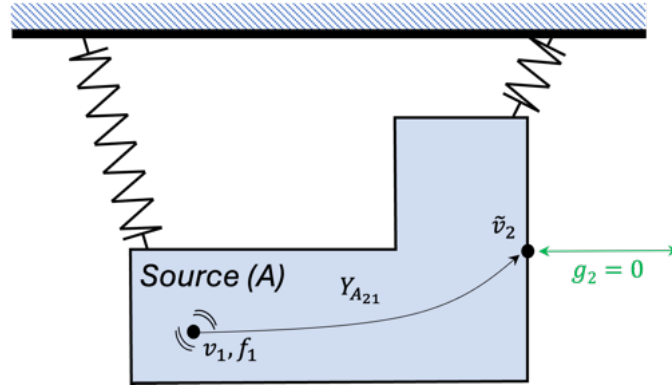


Figure 2.10: Source component suspended under free-free conditions depicting free-velocity.

The free velocity($\tilde{v}_2$) can be measured directly, and the blocked force can be back-calculated using the driving-point mobility at the interface.

$$f_{bl} = Y_{A22}^{-1} \tilde{v}_2 \quad (2.13)$$

Additionally, the free velocity inherent to the source component can be related to the internal forces.

$$\tilde{v}_2 = Y_{A21} f_1 \quad (2.14)$$

The free velocity method has advantages over direct blocked force measurement, as it allows a free-free boundary condition to be achieved using bungee, springs, and other suspension methods. However, the method is challenging to use for removing all flanking paths as source components, as operating components typically have structural connectors, such as cables or pipes, that can induce unwanted rigidity. Additionally, achieving free-field conditions for large, massive sources, such as an engine or powertrain unit, can be difficult[3].

2.4.4 In-situ blocked force method

The direct blocked force and free velocity methods are idealized methods for obtaining an interface force intrinsic to the source component. Due to this, they are limited in their application for many source components given their specific boundary conditions. A method was published by Moorehouse Elliott, and Evans that demonstrated measuring the source in situ and obtaining the blocked force via matrix inversion [14]. By removing the need for a special test setup, the blocked force can be determined with the source mounted under operating conditions, rather than under extreme boundary conditions. Similar to the matrix inversion method described in Section 2.4.1.3, the in situ method relies on defining an FRF, in which the interface force and the mobility between the interface and the target location can determine the response at a target location.

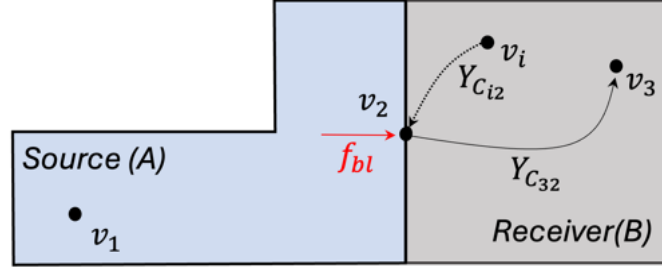


Figure 2.11: Diagram depicting in-situ blocked force

$$\begin{pmatrix} v_i \\ v_2 \\ v_3 \end{pmatrix} = \begin{bmatrix} Y_{C_{i2}} \\ Y_{C_{22}} \\ Y_{C_{32}} \end{bmatrix} f_{bl} \quad (2.15)$$

Equation 2.15 shows the velocity at any location on the coupled system can be determined from the blocked force where,

- (v) is the velocity at the interface and arbitrary indicators obtained from operational measurement
- $[Y_C]$ represents the mobility matrix between the interface and indicator location if the coupled system, is obtained by passive measurement

The measurement process for the matrix inversion is similar to the matrix inverse method in section 2.4.1.3, where two separate measurements are needed to identify the velocity at indicator locations during operation, as well as the system's mobility. The difference can be noticed when comparing equations 2.16 and 2.10 where equation 2.16 requires the coupled system mobility whereas equation 2.10 considered only the receiver mobility [3]. Therefore, Equation 2.16 can be used to determine the blocked force from the in situ method with the operational velocity and the pseudoinverse of the mobility matrix.

$$f_{bl} = \begin{bmatrix} Y_{C_{22}} \\ Y_{C_{32}} \end{bmatrix}^+ \begin{pmatrix} v_2 \\ v_3 \end{pmatrix} \quad (2.16)$$

This distinction implies that, unlike free velocity and direct measurements, the in situ blocked force can be determined without separating the source and receiver components, and that the components' boundary conditions are not critical to determining the blocked force. However, the more profound consequence is that the blocked force, an independent characteristic of the source component, can be determined with the measurement of the coupled mobility and operational velocity at arbitrary points [14].

2.4.4.1 Validity of in-situ blocked force

The in-situ blocked force can be derived by equivalence [14], or by comparing admittance [9]. An impedance-based derivation [3] can also be determined by considering a linear system of equations of a rigidly coupled system with an unknown force internal to the source being excited at location 1, similar to one shown in Figure 2.1

$$\begin{bmatrix} f_1 \\ 0 \\ 0 \end{bmatrix} = \begin{bmatrix} Z_{A11} & Z_{A12} & 0 \\ Z_{A21} & Z_{C22} & Z_{B23} \\ 0 & Z_{B32} & Z_{B33} \end{bmatrix} \begin{bmatrix} v_1 \\ v_2 \\ v_3 \end{bmatrix} \quad (2.17)$$

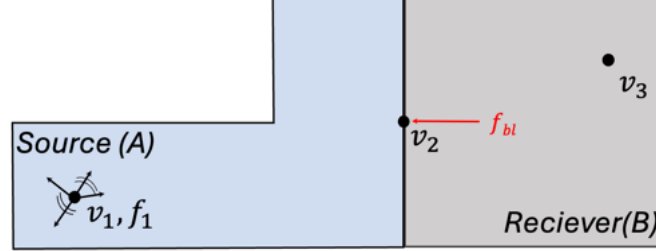


Figure 2.12: Reactionary blocked force due to internal forces

Equation 2.17 can be rearranged so that the equation still describes the system with the internal response DOF v_1 removed, and the new condensed matrix becomes

$$\begin{bmatrix} -Z_{A21}Z_{A11}^{-1}f_1 \\ 0 \end{bmatrix} = \begin{bmatrix} -Z_{A21}Z_{A11}^{-1}Z_{A12} + Z_{C22} & Z_{B23} \\ Z_{B32} & Z_{B33} \end{bmatrix} \begin{bmatrix} v_2 \\ v_3 \end{bmatrix} \quad (2.18)$$

Equation 2.18 demonstrates an alternate force term at the interface is $-Z_{A21}Z_{A11}^{-1}f_1$. This equivalent force has the same dynamic behavior as the internal force f_1 and is independent of the source component.

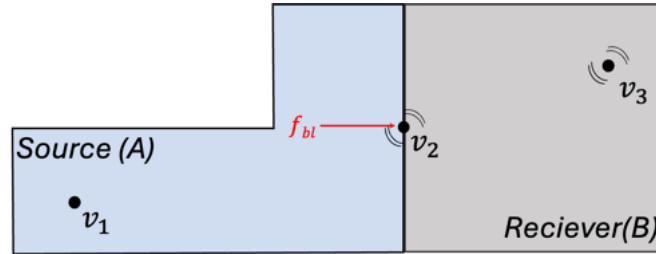


Figure 2.13: Equivalent blocked force with opposite direction

Considering a system based on the ideal blocked-force constraint, where the source component is fixed at the interface, as shown in Figure 2.5, a relationship can be established between the blocked force and the internal force.

$$\begin{bmatrix} f_1 \\ f_{bl} \end{bmatrix} = \begin{bmatrix} Z_{A11} & Z_{A12} \\ Z_{A21} & Z_{A22} \end{bmatrix} \begin{bmatrix} v_1 \\ 0 \end{bmatrix} \quad (2.19)$$

$$f_{bl} = Z_{A21}Z_{A11}^{-1}f_1 \quad (2.20)$$

When compared to equation 2.18, it can be seen that the ideal fixed boundary condition blocked force in equation 2.20 is the same as the reactionary blocked force $-f_{bl}$ from in-situ boundary condition 2.18. Finally, when considering the case where the contact force is applied at the interface

$$\begin{bmatrix} 0 \\ -f_{bl} \\ 0 \end{bmatrix} = \begin{bmatrix} Z_{A11} & Z_{A12} & 0 \\ Z_{A21} & Z_{C22} & Z_{B23} \\ 0 & Z_{B32} & Z_{B33} \end{bmatrix} \begin{bmatrix} v_1 \\ v_2 \\ v_3 \end{bmatrix} \quad (2.21)$$

By rearranging equation 2.21 so that DOF v_1 is eliminated, it can be seen that the reduced matrix is similar to equation 2.18.

$$\begin{bmatrix} -f_{bl} \\ 0 \end{bmatrix} = \begin{bmatrix} -Z_{A21}Z_{A11}^{-1}Z_{A12} + Z_{C22} & Z_{B23} \\ Z_{B32} & Z_{B33} \end{bmatrix} \begin{bmatrix} v_2 \\ v_3 \end{bmatrix} \quad (2.22)$$

This demonstrates that a negative blocked force acting at the interface, when the source is not running, produces the same response as if the internal force were operating. If the coupled mobility matrix is used, the inverse of equation 2.21 can be considered so that the following relationship can be made.

$$\begin{bmatrix} v_1 \\ v_2 \\ v_3 \end{bmatrix} = \begin{bmatrix} Y_{C11} & Y_{C12} & Y_{C13} \\ Y_{C21} & Y_{C22} & Y_{C23} \\ Y_{C31} & Y_{C32} & Y_{C33} \end{bmatrix} \begin{bmatrix} 0 \\ -f_{bl} \\ 0 \end{bmatrix} \quad (2.23)$$

By only considering the response at external DOFs, the matrix can be reduced to the relationship shown in equation 2.24.

$$\begin{bmatrix} v_2 \\ v_3 \end{bmatrix} = \begin{bmatrix} Y_{C22} & Y_{C23} \\ Y_{C32} & Y_{C33} \end{bmatrix} [-f_{bl}] \quad (2.24)$$

2.4.5 Expanded in-situ blocked force equation

Section 2.4 extensively demonstrates how a response at a desired location can be calculated using contact force and blocked force. An expanded equation can be derived so that mount stiffness can be included, and all terms are described by information from individual components.

$$v_4 = [Y_{B34}]([Y_{A22}] + [K_{23}]^{-1} + [Y_{B22}])^+(f_{bl}) \quad (2.25)$$

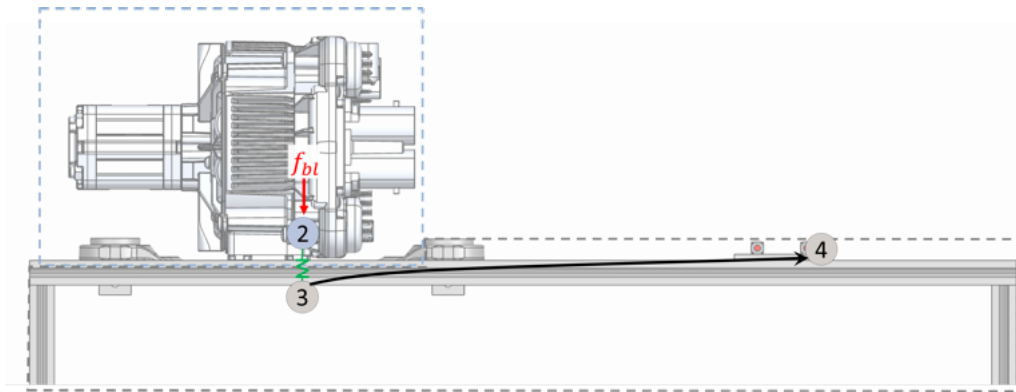


Figure 2.14: Vibration flow path of an example system using blocked force

The expanded equation 2.25 meets all the requirements of source characterization described in the introduction to section 2.4. During the development of a new system, it is possible that the source component changes, where the interface location moves due to modifications to boundary conditions, changed attachment locations, or different definitions of what is included in the source component. If the interface locations were to change, then an additional measurement campaign would be required to obtain the blocked force at the new interfaces. To prevent the need for retesting, the blocked force can be expanded using the definition of free velocity in equation 2.14 so that the interface force can be related to internal force f_1 with the mobility $Y_{A_{21}}$ known. Because the internal mechanism and geometry are not known, the location of the internal force in the source component can be generically described at the center of gravity (CoG) of the component.

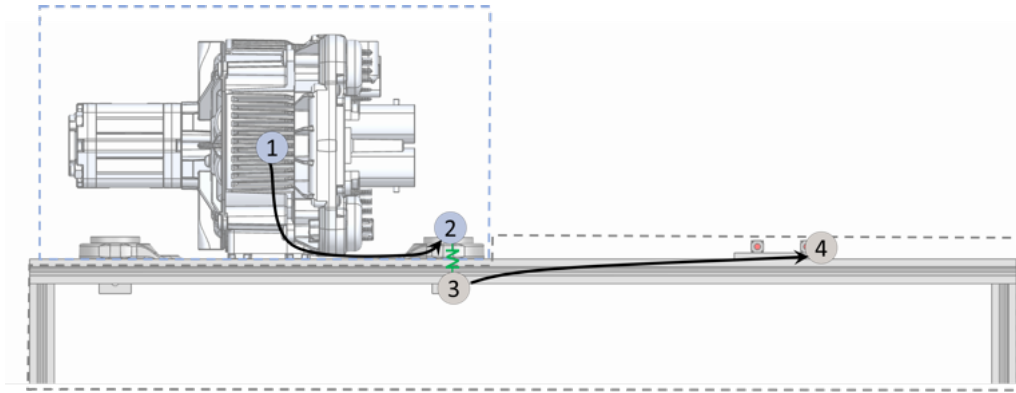


Figure 2.15: Vibration flow path of an example system using CoG force

$$v_4 = [Y_{B_{34}}]([Y_{A_{22}}] + [K_{23}]^{-1} + [Y_{B_{22}}])^+[Y_{A_{21}}](f_1) \quad (2.26)$$

With equation 2.26, the blocked force at the interface location can be determined by the internal force at the CoG and the mobility between the CoG and the new interface. Equation 2.26 is only practical when the mobility of the source component Y_A can be determined. If the source mobility cannot be determined, then equation 2.25 should be used.

2.5 Interface characterization

When describing the connections between components in C-TPA, the interface node must be well defined. When describing the interface between components, it is important that the displacements and the equilibrium of interface forces are satisfied [15]. In numerical tools like FEA, interfaces can be defined as a single-point interface between two nodes connected by a beam element. In physical assemblies, this is not the case, as components can be connected by bolts, welds, and adhesives, which represent different fixed, hinged, and sliding constraints that are modeled differently than fixed connections at a single point. Additionally, interfaces can have damping elements such as mounts, bushings, and dampers. Although there are other meth-

ods for characterizing the dynamics at interfaces, the virtual point transformation method is used in this project.

2.5.1 Virtual point transformation

For C-TPA to be performed, the dynamics at the interface between components must be captured. However, direct measurement of a connector DOF is often infeasible due to limitations such as physical access constraints and operational conditions.



Figure 2.16: Measurement of interface locations is limited due to the accessibility of transducers.

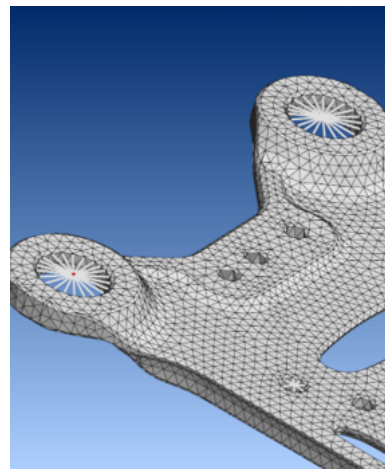


Figure 2.17: FEA can obtain acceleration responses or excite a structure at a non-physical location using a rigid body element.

A direct method to measure the translational DOF at an interface is to place a transducer as close as possible to the interface to represent a single-point connection. This method is allowed under the assumption that the translation DOFs between the transducer and interface point are identical due to rigidity. This leads to challenges such as being able to determine translation DOFs only, not rotations, and correlating the exact DOFs of nodes between an FE model and a physical test object.

To resolve this, the virtual point transformation (VPT) can be used, which transforms the response from a set of arbitrarily positioned response and excitation DOFs located around a non-physical node called a virtual point. The advantage of the virtual point is that all DOFs at the precise location can be estimated based on a set of arbitrary response transducers and forces [3][16].

The measured DOFs from a tri-axial accelerometer can be transformed to a virtual point using equation 2.27.

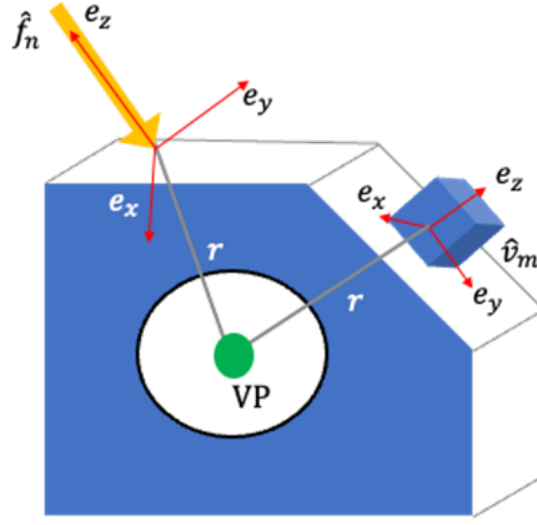


Figure 2.18: Example of coordinates for VPT

$$\underbrace{\begin{pmatrix} \hat{v}_{x,m} \\ \hat{v}_{y,m} \\ \hat{v}_{z,m} \end{pmatrix}}_{\hat{v}_m} = \underbrace{\begin{bmatrix} e_{x,x} & e_{x,y} & e_{x,z} \\ e_{y,x} & e_{y,y} & e_{y,z} \\ e_{z,x} & e_{z,y} & e_{z,z} \end{bmatrix}}_{E_m} \underbrace{\begin{bmatrix} 1 & 0 & 0 & 0 & r_z & -r_y \\ 0 & 1 & 0 & -r_z & 0 & r_x \\ 0 & 0 & 1 & r_y & -r_x & 0 \end{bmatrix}}_{r_m} \underbrace{\begin{pmatrix} v_x \\ v_y \\ v_z \\ v_\alpha \\ v_\beta \\ v_\gamma \end{pmatrix}}_v + \underbrace{\begin{pmatrix} \varepsilon_{x,m} \\ \varepsilon_{y,m} \\ \varepsilon_{z,m} \end{pmatrix}}_{\varepsilon_m} \quad (2.27)$$

where

- vector $\hat{v}_m$ represents the measured velocity in DOF of the triaxial accelerometer
- matrix E_m represents the orientation of the triaxial accelerometer (Euler Angles)
- matrix r_m represents the position/distance vector
- vector v is the translational and rotational DOF at the virtual point
- vector ε_m is the residual flexibility

Although a minimum of 2 accelerometers can be used to determine the 6 unknown DOFs at the virtual point, it is recommended that the system be overdetermined, and at least 3 accelerometers be used [3]. To calculate the virtual point due to multiple sensors, the following equation is used:

$$\begin{pmatrix} \hat{v}_1 \\ \hat{v}_2 \\ \hat{v}_3 \\ \dots \\ \hat{v}_m \end{pmatrix} = \underbrace{\begin{bmatrix} E_1 & & & \\ & E_2 & & \\ & & E_3 & \\ & & & \dots \\ & & & & E_m \end{bmatrix}}_{R_v} \underbrace{\begin{bmatrix} r_1 \\ r_2 \\ r_3 \\ \dots \\ r_m \end{bmatrix}}_{r_m} \begin{pmatrix} v_x \\ v_y \\ v_z \\ v_\alpha \\ v_\beta \\ v_\gamma \end{pmatrix} + \begin{pmatrix} \varepsilon_1 \\ \varepsilon_2 \\ \varepsilon_3 \\ \dots \\ \varepsilon_m \end{pmatrix} \quad (2.28)$$

where R_v represents the linear transform that simplifies equation 2.28 to

$$\hat{v}_m = R_v v_m + \varepsilon \quad (2.29)$$

Due to the assumptions of rigidity, which are further explained in Section 2.5.1.1, the velocity at the virtual point can be determined by multiplying by the Penrose pseudoinverse to transform

$$v = (R_v)^+ \hat{v} = T_v \hat{v} \quad (2.30)$$

where T_v is the transformation matrix from measured velocity to virtual point. Forces can be translated to a virtual point similar to velocity with the following equation for a single force.

$$\begin{pmatrix} f_x \\ f_y \\ f_z \\ f_\alpha \\ f_\beta \\ f_\gamma \end{pmatrix} = \underbrace{\begin{bmatrix} 1 & 0 & 0 \\ 0 & 1 & 0 \\ 0 & 0 & 1 \\ 0 & -r_z & r_y \\ r_z & 0 & -r_x \\ -r_y & r_x & 0 \end{bmatrix}}_{r_n} \underbrace{\begin{bmatrix} e_x \\ e_y \\ e_z \end{bmatrix}}_{e_n} \hat{f}_n \quad (2.31)$$

where

- vector $\hat{f}_n$ represents the measured force
- matrix r_n is the distance between the Virtual point and the measured force
- matrix e_n is the angle of the applied force relative to the virtual point

The virtual point force due to multiple forces can be calculated similarly to equation 2.28

$$\begin{pmatrix} f_1 \\ f_2 \\ f_3 \\ \dots \\ f_n \end{pmatrix} = \underbrace{\begin{bmatrix} r_1^T & r_2^T & \dots & r_n^T \end{bmatrix} \begin{bmatrix} e_1^T & & & \\ & e_2^T & & \\ & & \dots & \\ & & & e_n^T \end{bmatrix}}_{R_f^T} \begin{pmatrix} \hat{f}_1 \\ \hat{f}_2 \\ \dots \\ \hat{f}_n \end{pmatrix} \quad (2.32)$$

The matrix can also be simplified, similar to equation 2.30

$$f = (R_f^T)^+ \hat{f} = T_f^T \hat{f} \quad (2.33)$$

In equations 2.28 and 2.32 the virtual point velocity v and force f_n can be determined by least-square pseudoinverse. If both the velocity and force transformation matrices are determined, the FRF at the virtual point Y can be calculated from measured mobilities.

$$Y = T_v \hat{Y} T_f^T \quad (2.34)$$

In equation 2.33, $\hat{Y}$ represents the FRF of a measured arbitrary response due to a measured arbitrary excitation, it can be seen in equation 2.34 for a virtual point to be valid, it is critical that the dynamic response is correctly captured in the transform matrices T_v and T_f^T . The virtual point transformation is considered valid if two criteria are satisfied: the rank of the measured FRF is acceptable, and the interface is rigid.

2.5.1.1 Rank and rigidity

The rank of a matrix describes the significance of each DOF to the total contribution by evaluating the linear independence of its FRFs by comparing the rows and columns of the mobility matrix. For the case of the mobility matrix at the virtual point, there must be no more than 6 linearly independent FRFs measured to capture the 3 translation and 3 rotation DOFs [3]. Rigidity is defined as the condition in which the structural wavelength is much larger than the interface size. Therefore, rigid conditions are usually acceptable in the low- to mid-frequency ranges[3]. If the rank and rigidity condition is not sufficiently met, the dynamic response of the interface is not captured correctly. To verify that the dynamic response is correctly captured, an indicator is to comparing the measured FRF matrix $\hat{Y}$ with a reconstructed FRF matrix $\hat{Y}'$ which is calculated by multiplying the virtual point mobility $\hat{Y}$ with the pseudoinverse of the transform matrices from equations 2.33 and 2.30. By performing this reconstruction operation, the interface's flexibility is filtered out. Therefore, if the reconstructed FRF is equal to the measured FRF $\hat{Y}' = \hat{Y}$, the rank is ≤ 6 and is sufficiently low. If $\hat{Y}' \approx \hat{Y}$, it is possible that there is flexibility (ϵ_m) in the system, and rigid dynamic conditions are not captured.

$$\hat{Y}' = T_v^+ Y T_f^{T+} = \hat{Y} \quad (2.35)$$

If the system is overdetermined with additional DOFs, the sensor and force consistency criterion can also be calculated to evaluate the rank and rigidity of each response DOF and excitation DOF [15].

2.5.1.2 Response correlation coefficient

Similar to equation 2.35 the product of the recalculated mobility $\hat{Y}'$ a unit force f_h of 1 is applied at arbitrary position h will result in a reconstructed velocity $\hat{v}'$. When correlated with the measured mobility and a unit force, the velocity correlation coefficient at position h ρ_{vh} can be calculated, where ρ_{vh} is a frequency-dependent value between 0 and 1 that quantifies the sensor's rigidity. [3]

$$\rho_{v,h} = \text{Corr}(\hat{v}_h, \hat{v}'_h) \quad (2.36)$$

$$\hat{v} = \hat{Y}_h * f_h \quad (2.37)$$

$$\hat{v}' = \hat{Y}'_h * f_h \quad (2.38)$$

The flexibility of the recalculated response can be filtered out by multiplying by the transform matrices

$$\hat{v}' = T_v T^+ \hat{Y}'_h * f_h \quad (2.39)$$

where $\hat{Y}_h$ represents the FRF between force at position h and the interface response. Because both equations 2.37 and 2.38 are multiplied by unit force (f_h), the relationship between the measured and recalculated velocity can also be described by the measured and recalculated mobility, and equation 2.36 can be rewritten as

$$\rho_{v,h} = \text{Corr}(\hat{Y}_h, \hat{Y}'_h) \quad (2.40)$$

If multiple force locations are considered, each force can determine its own independent $\rho_{v,h}$ and vectorized $\hat{\underline{Y}}$ to determine an overall velocity correlation coefficient (ρ_v).

$$\rho_v = \text{Corr}(\hat{\underline{Y}}, \hat{\underline{Y}}') \quad (2.41)$$

2.5.1.3 Excitation correlation coefficient

Similar to the response correlation coefficient, a coefficient can be determined to evaluate the quality of the excitation.

$$\hat{v}_i = \hat{Y}_i \hat{f} \quad (2.42)$$

where $\hat{v}_i$ is the response at sensor location i due to applied forces on the system.

2.5.2 Limitations and indicators

A limitation of the virtual point method is that the angles and positions during measurement must be highly accurate for the model to be valid. Commercial NVH tools are available for entering detailed CAD data and predefining transducer locations and excitation. The component of the response that cannot be transferred to the virtual point due to the rigidity is considered to be the residual flexibility described by the term ε . Engineering judgment should be considered when selecting the response transducer locations. If the transducer is placed close to the virtual point, the effects of rigidity are less significant; however, there is more error in the absolute placement of the transducer [15]. Similarly, all excitation impacts should be placed to sufficiently excite all six DOFs around the virtual point [15].

3

Method

3.1 Overview of measurement campaign

A measurement campaign was completed following the procedure defined in ISO 20270 [17] to validate the in-situ blocked force source characterization method and to develop a virtual acoustic prototype of the system.

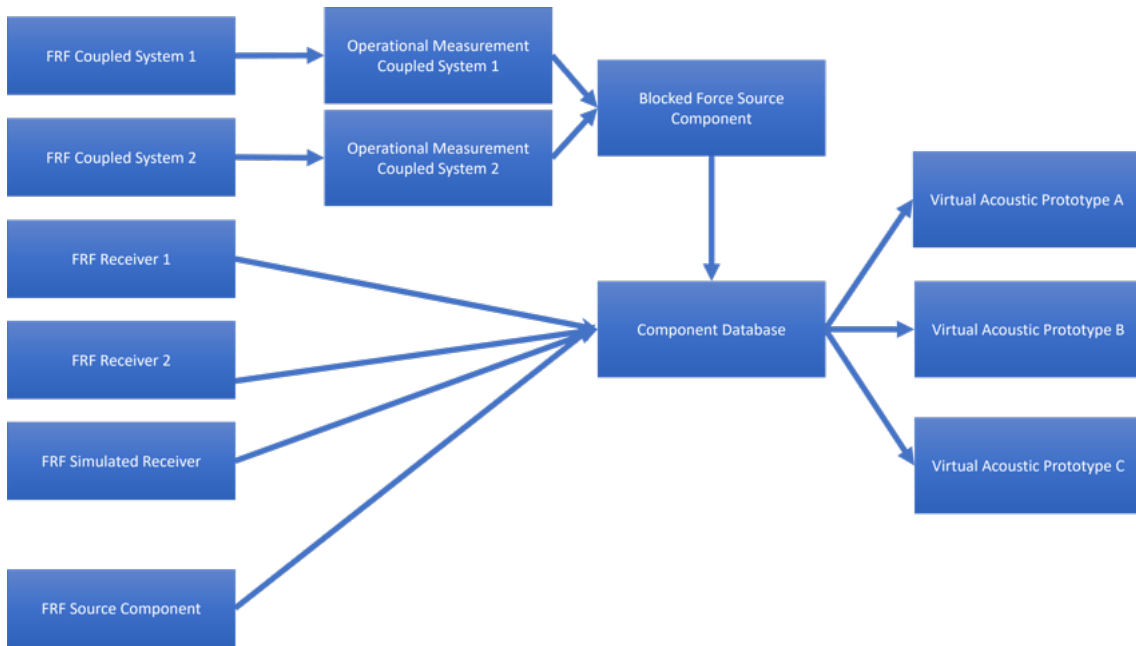


Figure 3.1: Block diagram of the measurement campaign

To demonstrate that the blocked force is independent, transferable, and representative of the internal force of the source component, characterization was performed on two systems that share the same source but have different receiver components. The results showed that the blocked forces were consistent across corresponding DOFs for both systems. The blocked force that represents the source component and the FRFs of the receiver components were used to build a VAP and modifications to the system were introduced to show the applicability of the VAP for predictive engineering. The source and receiver components were added to a component database, and the acceleration at the target location was predicted using a VAP.

3.1.1 Lab setup

Measurements were conducted at the Volvo Group Lundby facility. To isolate the test articles and prevent external influence on the measured mobility, the test fixture was mounted to a large concrete block.

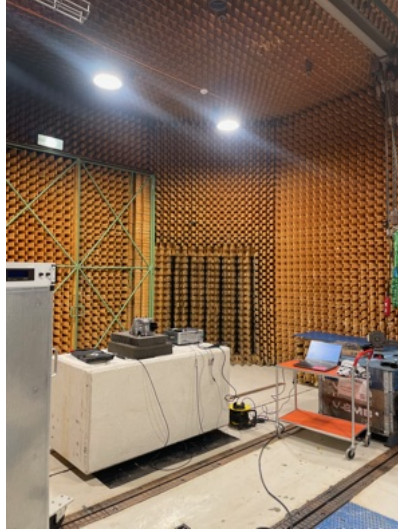


Figure 3.2: Lab space with DAQ and concrete block

3.1.2 Data acquisition

Data acquisition was controlled using the commercial software Siemens Testlab NEO and a Siemens SCADAS mobile DAQ hardware.



Figure 3.3: Siemens SCADAS frontend

For both FRF and operational measurements, PCB triaxial accelerometers were attached to the coupled structure using adhesive (Loctite) and PCB accelerometer clips.

3.2 Components

3.2.1 Electro-hydraulic power-steering (EHPS) pump



Figure 3.4: EHPS suspended via bungee in Free-free condition

The source component characterized was an electro-hydraulic steering (EHPS) pump that assists a vehicle operator's steering input by supplying hydraulic pressure during turning maneuvers. The component can be used on a variety of large vehicles and is a commercial product. A test article was provided by the supplier to the Volvo Trucks group for testing.

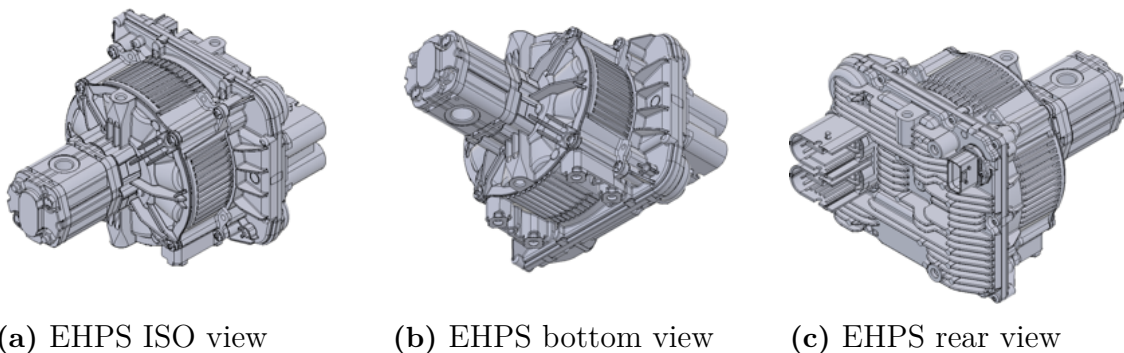


Figure 3.5: Electro-hydraulic power-steering pump (EHPS) used as the test article. Inlet and outlet hoses and electrical connections are not shown.

Detailed design of the internals of the component was not known; operational information was provided:

- the EHPS has an operational RPM range of 900-3200 RPM
- operational oil temperature range is 20-80 Celsius
- operational oil pressure range is 20-100 Bar

3.2.1.1 Measurement position on source component

To obtain the FRF matrix Y_A of the EHPS for the component database, a mobility measurement was done on the test article, which was suspended in a free-free condition with the inlet and outlet hoses and electrical connectors installed.

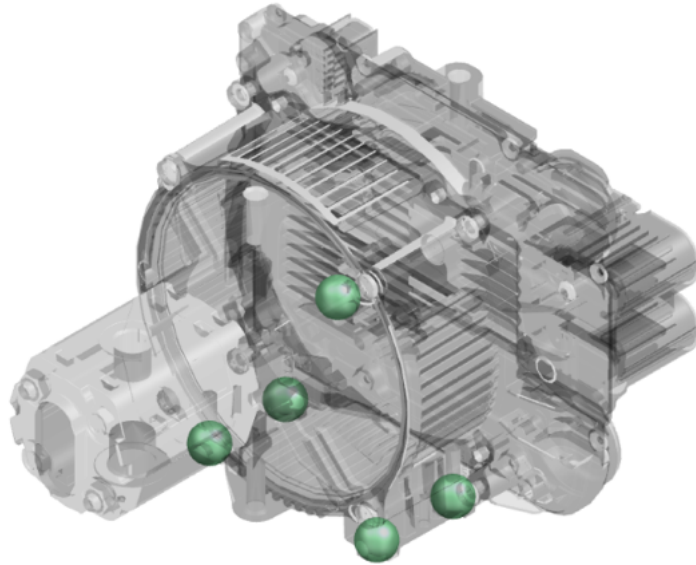
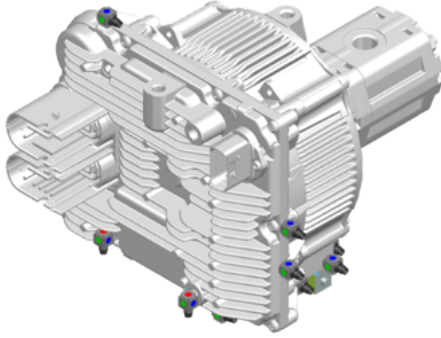
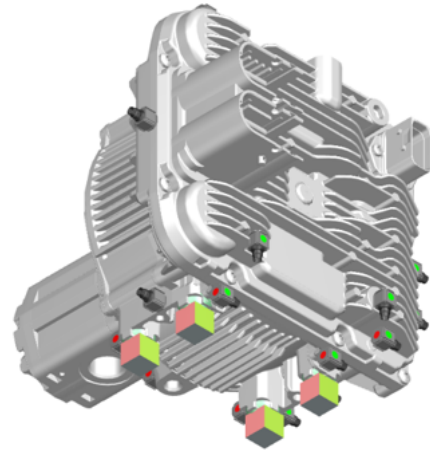


Figure 3.6: ISO view of VPTs on source component

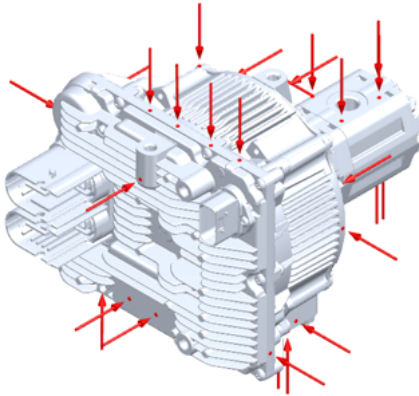
Five Virtual points were calculated to quantify the DOFs, the CoG, and the interface locations. For all calculated translational and rotational DOFs of the VPTs, 4 accelerometers and 12 excitations per VPT were located on the EHPS to generate satisfactory response and excitation correlation coefficients.



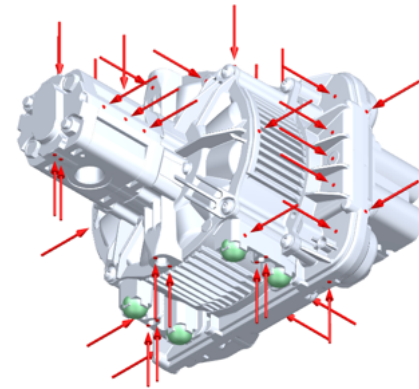
(a) Top view of accelerometers on EHPS



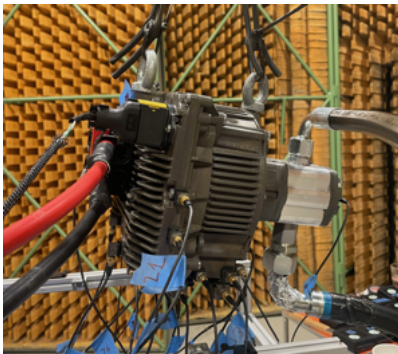
(b) Bottom view of accelerometers on EHPS



(c) ISO of hammer excitation on EHPS



(d) Bottom view of hammer excitation on EHPS



(e) Left side view of EHPS in free-free



(f) Right side view of EHPS in free-free

Figure 3.7: Response and excitation DOFs for source characterization

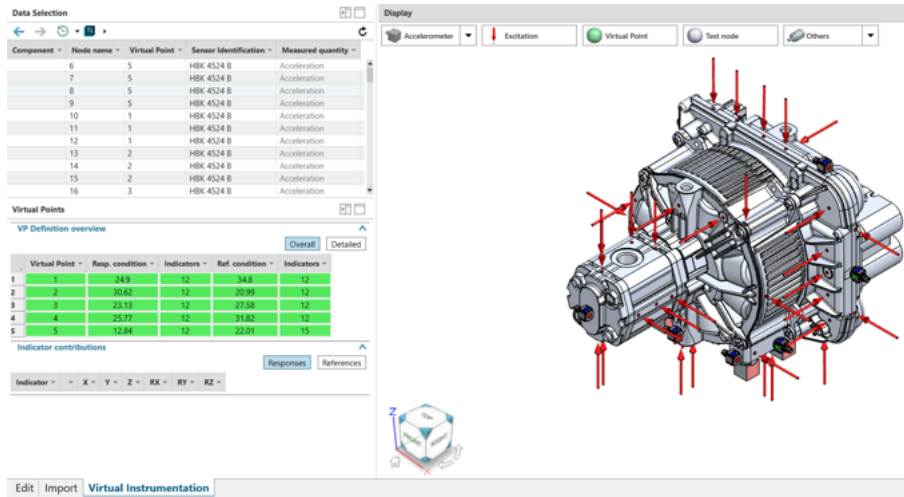


Figure 3.8: Siemens Testlab NEO user interface showing sufficient geometrical positioning for correlation coefficients

3.2.2 Receiving structure 1



Figure 3.9: ISO CAD view of receiver 1

Receiving structure 1 was constructed from commercial 5 mm T-slotted aluminum extrusions. The structure's extrusions were connected via 90-degree brackets assembled with T-slot fasteners. Two 5 mm steel plates were connected to the structure via fasteners and brackets. The source plate (300 mm by 146 mm) had clearance holes for four M10 bolts for attaching the source component, and the target plate (300 mm by 80 mm) had provisions for accelerometers. To minimize surface contact, the steel plates were elevated from the aluminum extrusions with washers.

3.2.2.1 Measurement positions for receiver structure 1

To obtain the FRF matrix Y_B of Receiver 1 for the component database, a mobility measurement was performed on the test frame mounted on the large concrete mass.

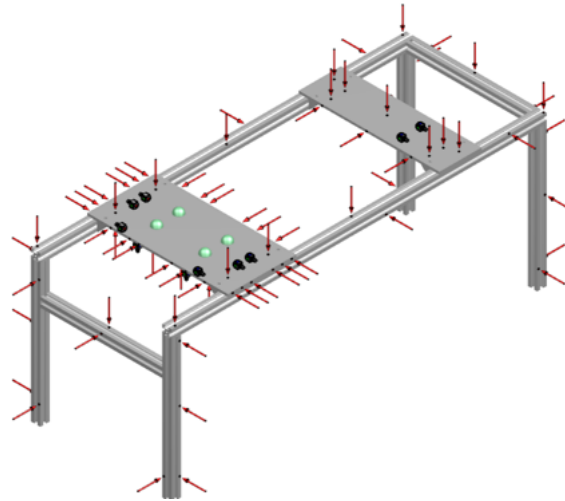


Figure 3.10: ISO view of VPT and DOFs on receiver structure 1

4 virtual points were placed at the clearance holes on the source component to represent the interface between the source and receiver components. 4 accelerometers and 10 excitations per VPT were placed on the source plate, and two accelerometers were placed at a designated location on the receiver plate to represent a target location. Additional excitation was applied to the beams of the Receiver component for modal correlation with the simulation.

3.2.3 Receiver structure 2

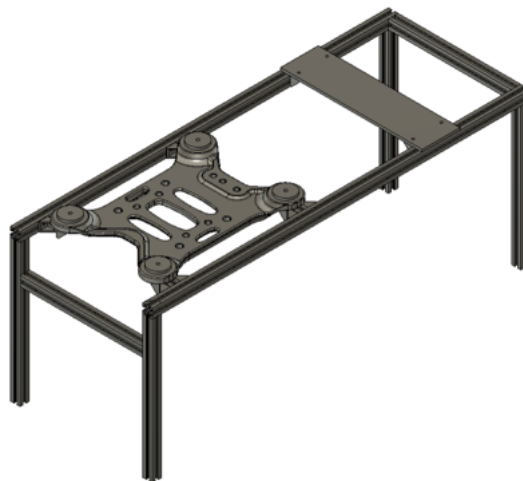


Figure 3.11: ISO CAD view of receiver 2

Receiver structure 2 was designed to represent a "small" change to a receiver structure 1. The modification illustrates the frequently asked question "Will changing Part A with Part B be worse for the NVH performance?" The modification replaced the source plate with a production part called the EHPS sliding plate. The EHPS

sliding plate is an aluminum bracket that connects the EHPS to the vehicle's primary structure. The sliding plate is adapted to the Receiving structure 2, with the addition of aluminum extrusions and thermoplastic bushings, to allow direct mounting of the EHPS sliding plate to the aluminum extrusion structure.

3.2.3.1 Measurement position on receiver structure 2

To obtain the FRF matrix Y_B of Receiver 2 for the component database, a mobility measurement was performed on the test frame mounted on the large concrete mass. 4 virtual points were placed at the clearance holes on the EHPS Sliding plate to represent the interface between the source and receiver components. 4 accelerometers and 10 excitations per VPT were placed on the source plate, and two accelerometers were placed at the same pre-designated location on the receiver plate to represent a target location. Additional excitation was applied to the receiver component's beams to enable modal correlation with the simulation.

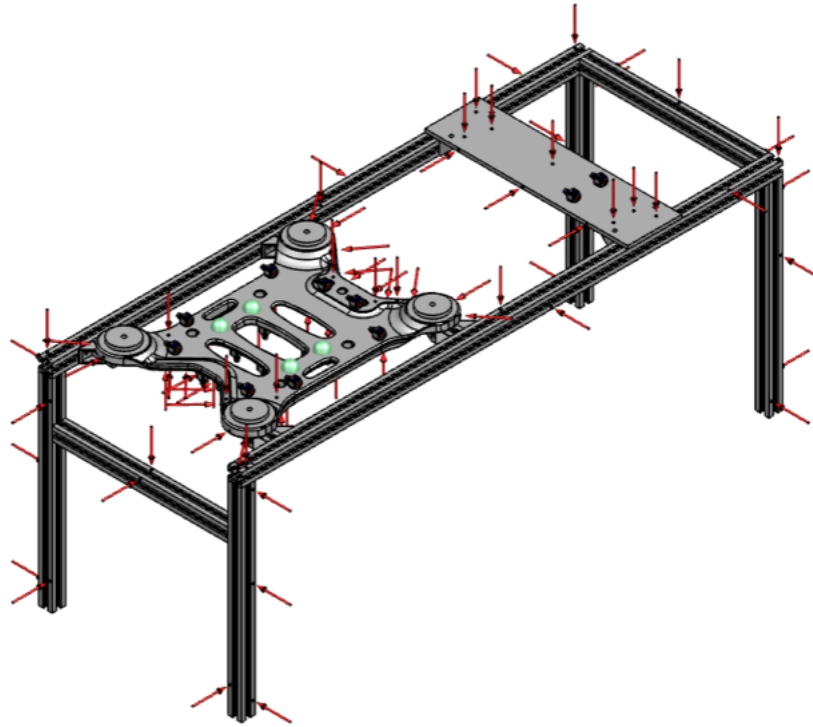
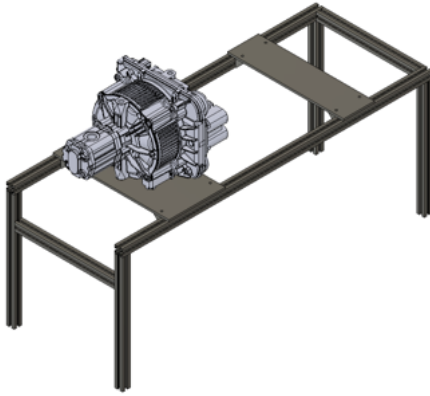


Figure 3.12: ISO view of measurement DOFs on receiver structure 2

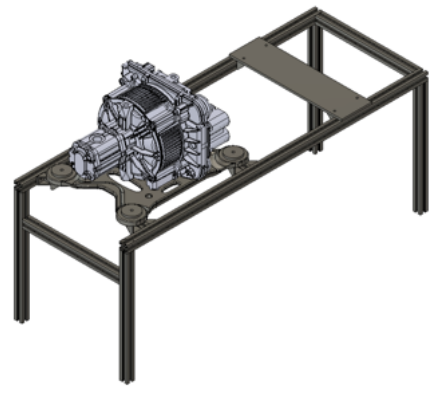
With the new part and interface adaptations, Receiver structure 2 exhibits a different dynamic response from Receiver structure 1 in Figure 3.10.

3.2.4 Coupled Structure 1 and 2

When the EHPS source component is coupled to receivers 1 and 2, the coupled assembly systems are configured for in situ blocked-force measurements.



(a) ISO View of coupled structure 1

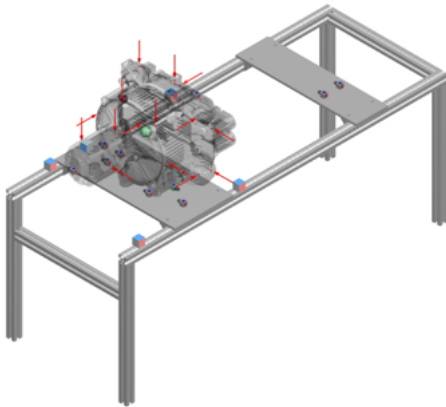


(b) ISO View of coupled structure 2

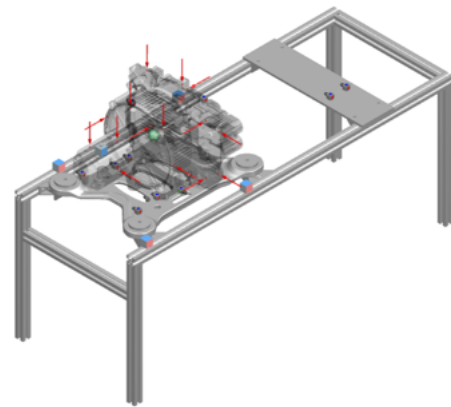
Figure 3.13: ISO View of coupled structures

To obtain the FRF matrices $[Y_C]$ and operational velocities (v_C) of coupled systems 1 and 2 for the component database, mobility and operational measurements were performed on the assembly.

3.2.4.1 Measurement position for coupled system 1 and 2



(a) ISO View of coupled structure 1



(b) ISO view of coupled structure 2

For both coupled systems 1 and 2, a virtual point was placed at the CoG of the source component to represent the source's DOFs. 4 accelerometers and 15 excitation locations were placed on the source structure to measure the required DOFs for the virtual point. Two accelerometers were placed at designated locations on the receiver plate to represent a target location. Additional accelerometers were placed on the receiver structure component, but were not used to calculate the virtual point. All transducer locations were the same for both mobility and operational measurements.

3.2.4.2 Global coordinate system

For all measurements, the orientations of the response and excitation DOFs must be accurately captured. Therefore, all DOFs were placed so that each measurement could be related to and oriented within the same global coordinate system.

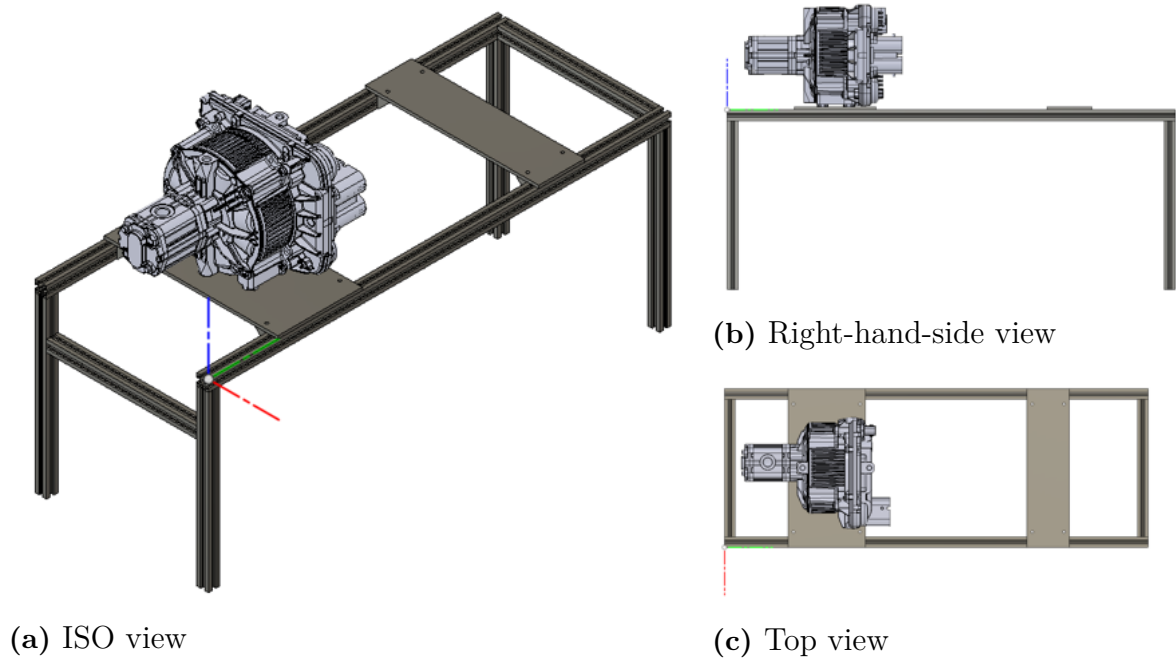


Figure 3.15: Views of the global coordinate system: (a) ISO view, (b) right-hand-side view, and (c) top view.

In Figure 3.15, the origin of the global coordinate system is in the "Back right" corner of the receiving structure. The x-axis is the horizontal red axis line along the shorter edge of the receiver structure, where X+ is in the "Right-hand side" direction. The y-axis is the horizontal green axis line along the longer edge of the Receiver structure, with Y+ in the "Forward" direction. The z-axis is the vertical blue line, with Z+ in the "up" direction. All transducers were placed so that each transducer's local coordinate system matched the global system. However, due to physical constraints, primarily due to cable routing, some transducers were placed with separate Euler angles. These angles were recorded in the software and translated so that all results are displayed in the global coordinate system.

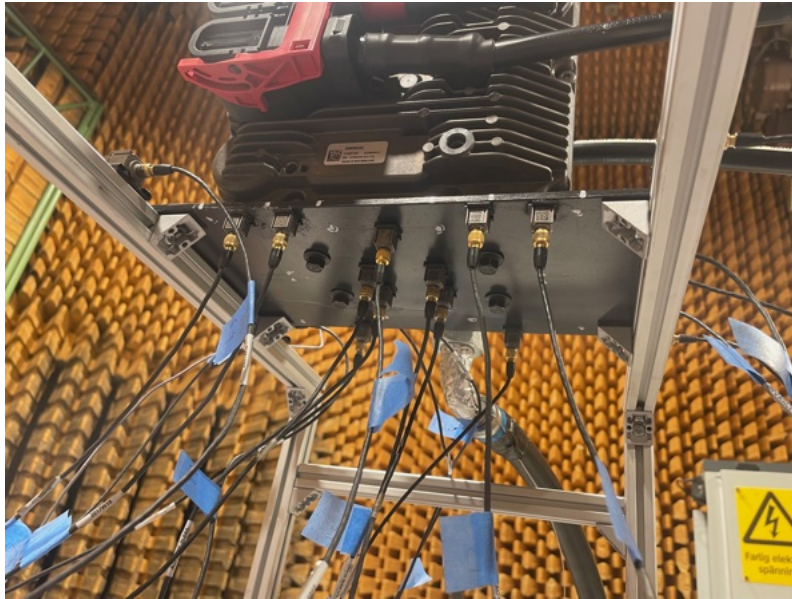


Figure 3.16: Accelerometers placed on the bottom surface of the source plate that do not follow the global coordinate system

3.2.5 Mobility measurement setup

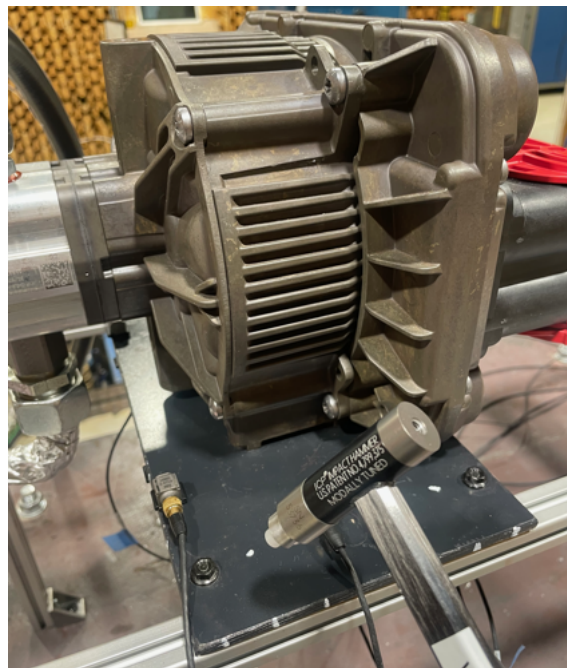


Figure 3.17: Modal hammer with Delrin tip

To obtain the mobility $[Y]$ of the components, measurements were conducted following the guidelines in ISO 20270 [17], in which a modal hammer was used to excite the structure at various locations to capture the FRF for each impact location, thereby capturing the global modal behavior.

3. Method

For all mobility tests, the following settings were used:

- Hammer tip: Delrin
- Sample rate: 12800 Hz
- Resolution: 1 Hz
- Minimum impacts per measurement: 5 impacts
- Upper frequency limit: 3200 Hz

During hammer setup, careful consideration was given to ensure that each measurement had a sufficient signal-to-noise ratio across the frequency range of interest, that excitations were normal to the surface, and that double hits were not used for averaging. Hammer excitation locations were selected so that the test object was energized in each translational DOF and that the excitation would provide rotational DOFs of the test object.

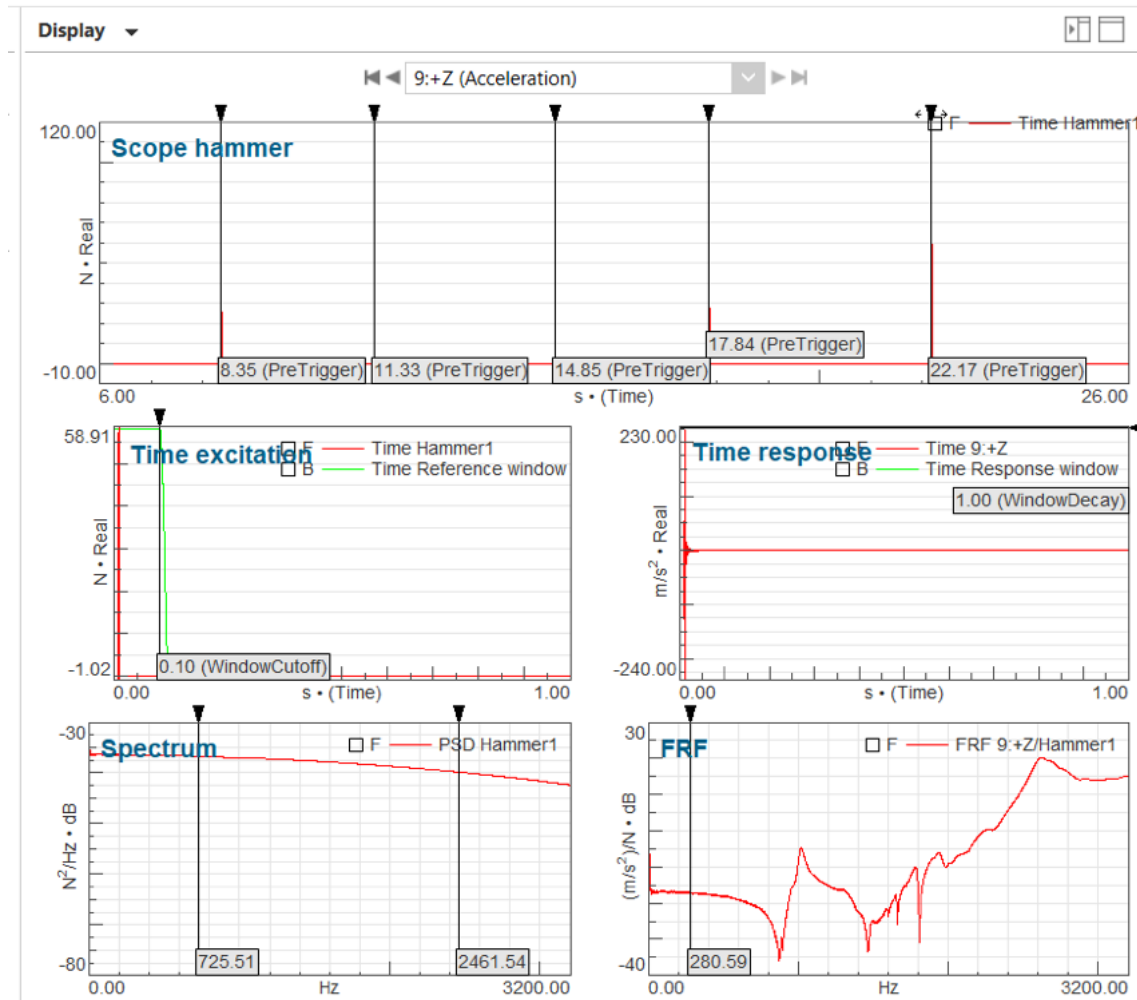


Figure 3.18: Test setup during mobility measurements to confirm time response is sufficient and input excitation spectrum over frequency range of interest does not drop more than 10 dB

Response transducer locations on the components are described for each component in Section 3.2.

The expected output of the mobility measurement is a $(M \times N)$ matrix of mobilities, where the relation between excitation and response for each frequency is shown.

3.2.6 Operational measurement setup

To obtain operational velocity (v), the motor was tested under operating conditions based on the motor specifications in section 3.2.1. A total of 48 test configurations were performed on each coupled structure based on the parameter matrix in Table 3.1

Parameter	Values
Oil Temperature	30, 50, 65 °C
Oil Pressure	20, 50, 90, 100 bar
Steady Pump Speed	1000, 2000, 3200 RPM
Ramp Test	900–3200 RPM @ 15 RPM/s

Table 3.1: Test parameters of the EHPS pump

To control the pump and oil temperatures, the Volvo Group acoustic test team provided a control rig that was operated using PID controllers to monitor cooling, temperature, and pump RPM. Pump operational data were recorded from the SCADAS CAN signal, and pump voltage, current, torque, and oil flow were also measured. A single-axis accelerometer was placed near the inlet oil pipe to enable phase reference tracking.

4

Results

This section highlights the results from the measurement campaign defined in Section 3.1. Section 4.1 shows the results from the mobility test as well as the derived FRFs at the virtual point. Section 4.2 shows the results from the operational measurements. The resultant blocked force is shown in Section 4.3 where the blocked forces for both coupled structures are shown and compared for steady-state and RPM sweep conditions. Validation of the blocked force via onboard and cross-validation is shown in Sections 4.4.1 and 4.4.2. To demonstrate the use of blocked forces for substructuring, virtual acoustic prototypes were built and shown in Section 4.5.

4.1 FRF measurements

The results in this section present selected measured driving-point and transfer mobilities from FRF measurement campaigns on components, calculated driving-point mobilities at all VPTs, and transfer mobilities between the VPTs and the target location.

4.1.1 Measured FRFs

To identify the dynamic behavior of a component, the measured mobility matrix was recorded for each component following the guidelines in section 3.2.5. Figure 4.1 shows an example of the 12 x 15 array of measured FRFs for coupled system 1.

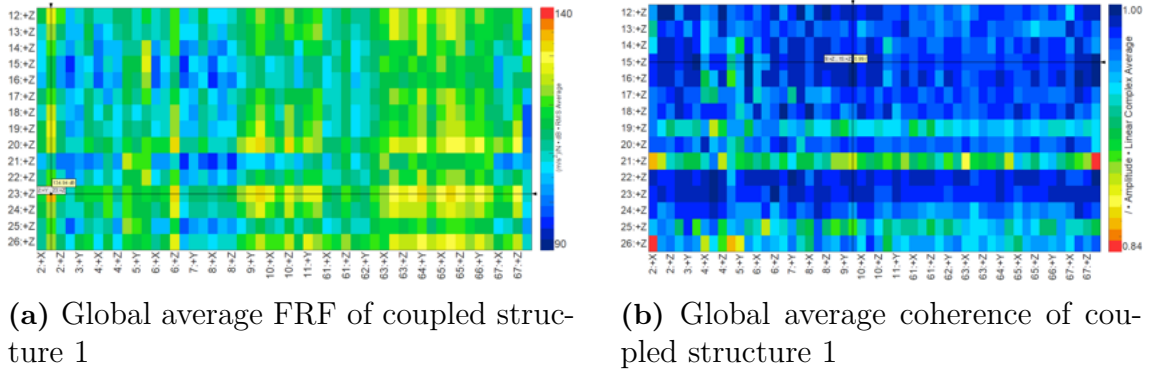


Figure 4.1: Heatmap of 780 average mobility datasets for coupled structure 1 between 0-3200 Hz that represents $[Y_C]$, where the y-axis is excitation DOFs and the x-axis is response DOFs

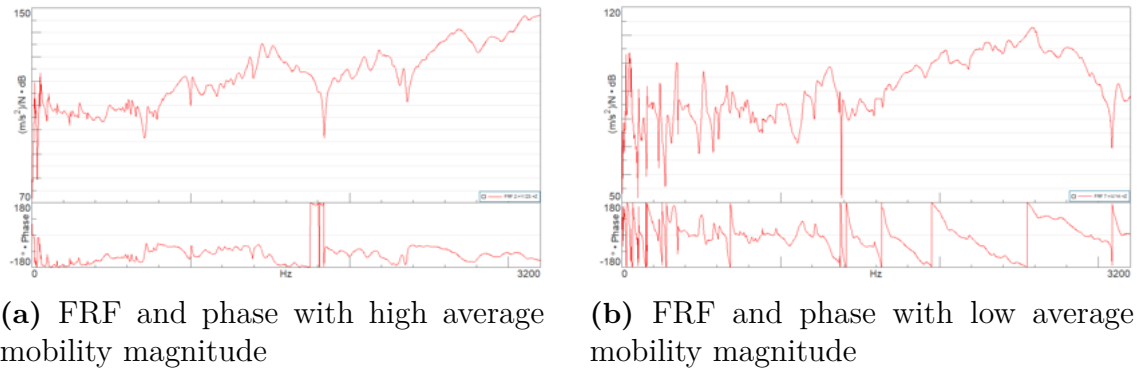


Figure 4.2: FRF of select average mobility in coupled structure 1 to demonstrate that the amplitude of the average FRF does not indicate the FRF response

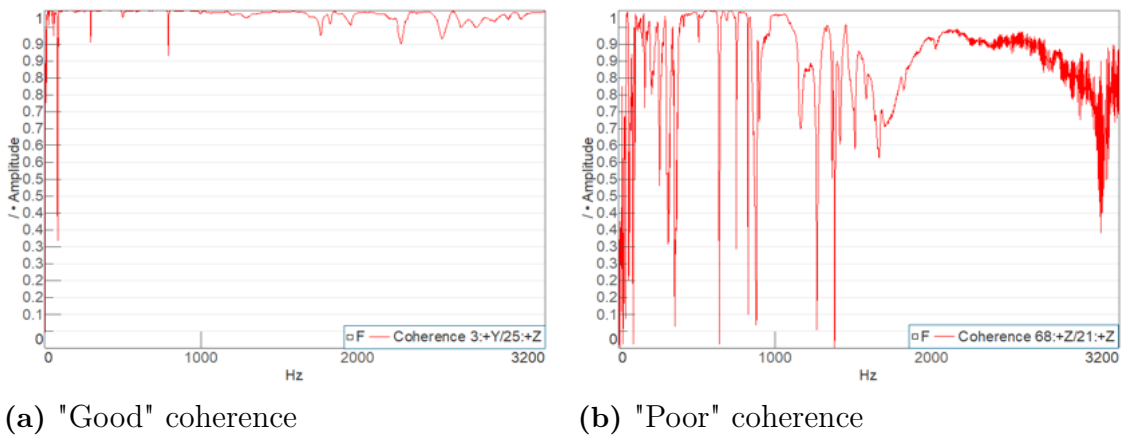


Figure 4.3: Select examples of "good" and "poor" coherence in coupled structure 1

4.1.2 FRFs at virtual point

The dynamic response at all virtual points was calculated with the Virtual Point Add-in in Siemens Testlab, following the calculation methods described in section 2.5.1. Figure 4.4 shows the driving mobility for all rotational and translational DOFs for the virtual point at the center of gravity of coupled structure 1, as shown in Figure 3.14a.

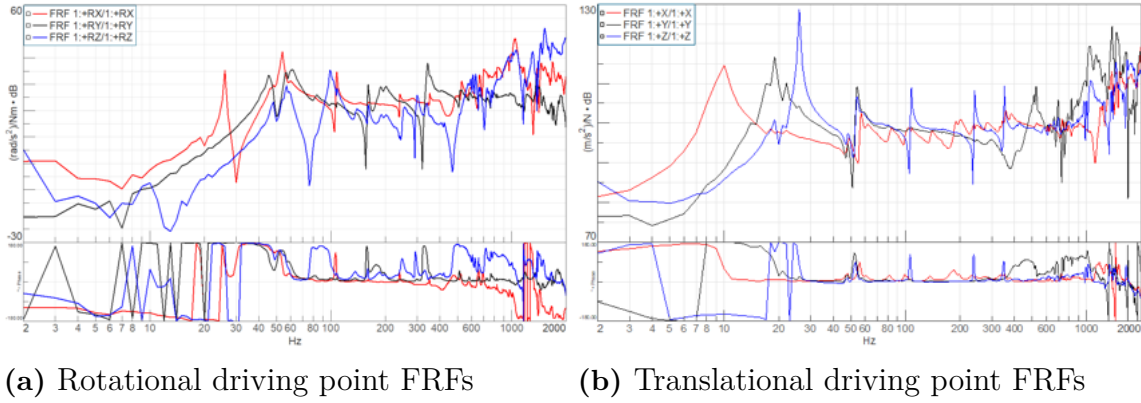


Figure 4.4: Driving point mobilities of 6 DOFs at the center of gravity virtual point of coupled structure 1

Figure 4.5 shows the comparison of transfer mobility from the DOFs at the center of gravity virtual point for both coupled structure 1 (Figure 4.5a and coupled structure 2 (Figure 4.5b).

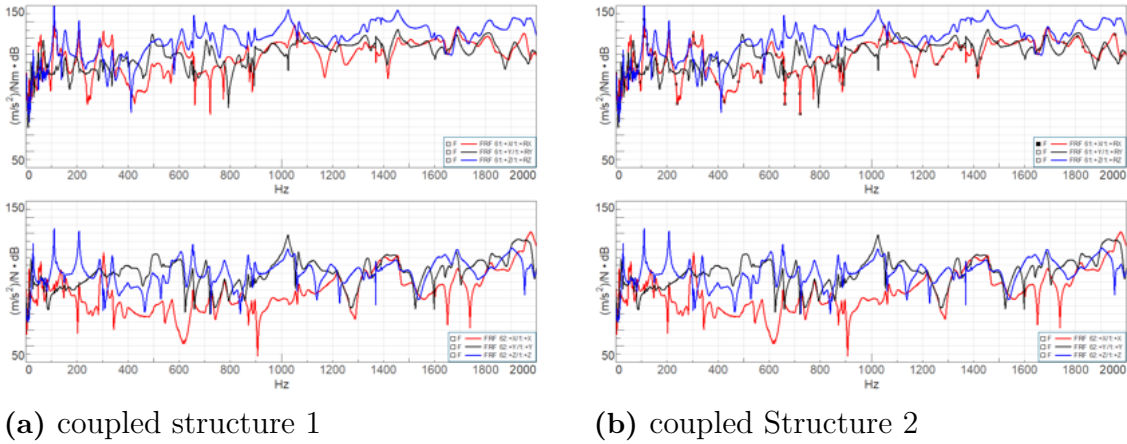


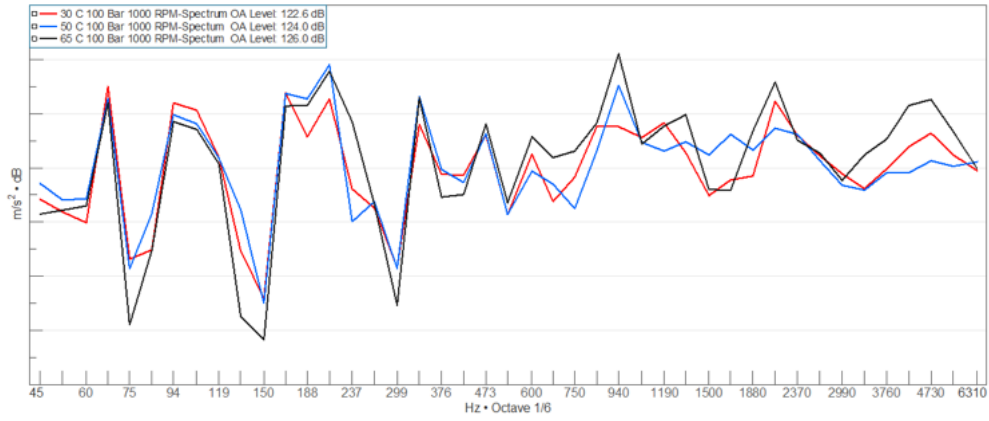
Figure 4.5: Transfer point FRFs between CoG virtual point and target location 61

Figures 4.4 and 4.5 show measured mobility between the CoG virtual point and a target location of the coupled structure $[Y_{C34}]$ described in equation 2.24.

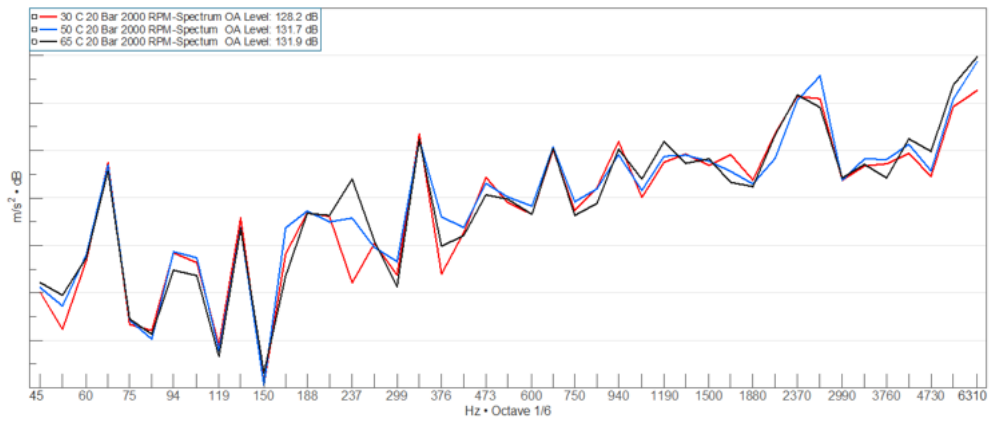
4.2 Operational measurements

Operational measurements were performed on coupled structures 1 and 2 following the steps in section 3.2.6 and added to the component database. Figures 4.6 through 4.8 show the acceleration response on coupled structure 1 operating under steady-state conditions. It can be observed that changes in oil temperature and pressure have a minimal influence on the frequency content and overall acceleration level of coupled structure 1 during steady-state operation. Figure 4.8 shows that changes in pump RPM have a significant effect on the frequency content and a positive linear relationship with overall acceleration level for coupled structure 1.

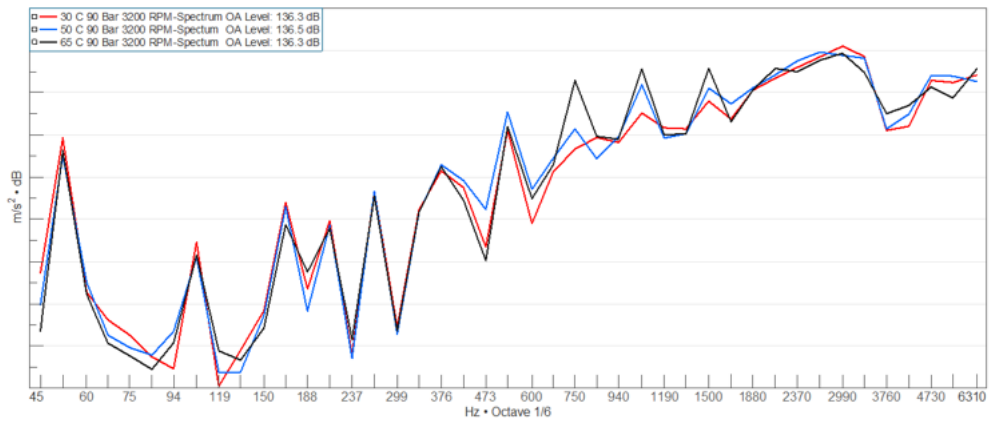
4. Results



(a) Accelerometer 2 x-axis acceleration at 100 Bar and 1000 RPM

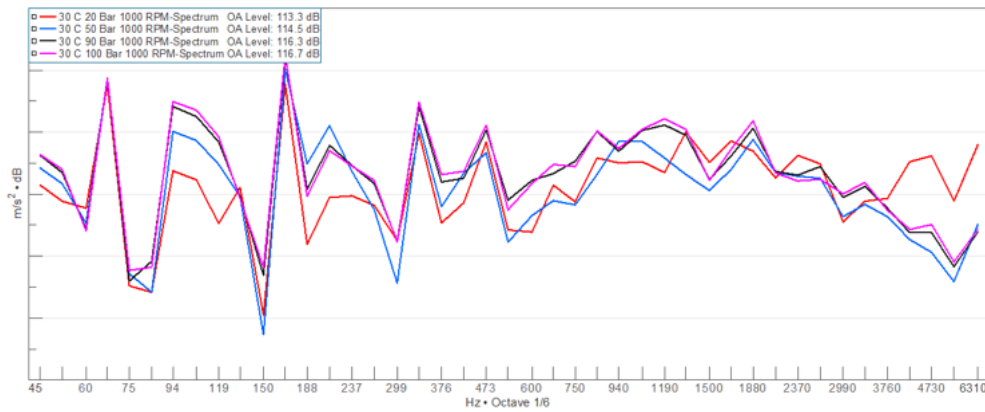


(b) Accelerometer 3 z-axis acceleration at 20 Bar and 2000 RPM

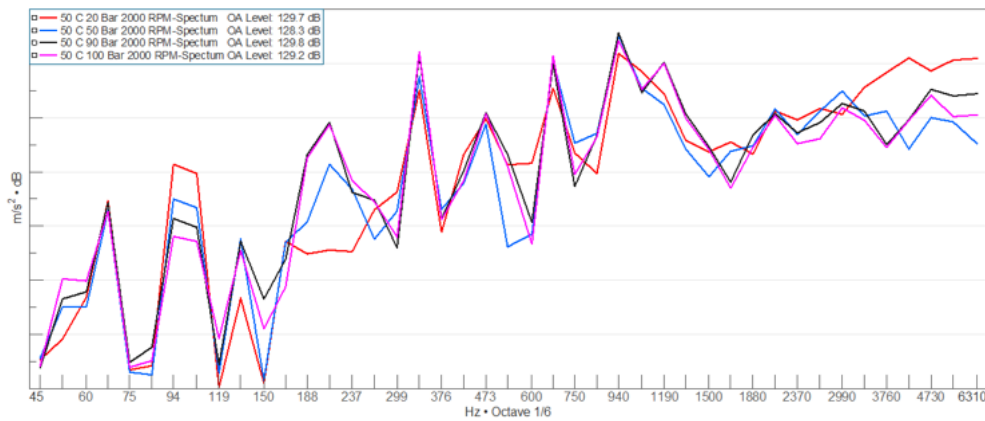


(c) Accelerometer 5 y-axis acceleration at 90 Bar and 3200 RPM

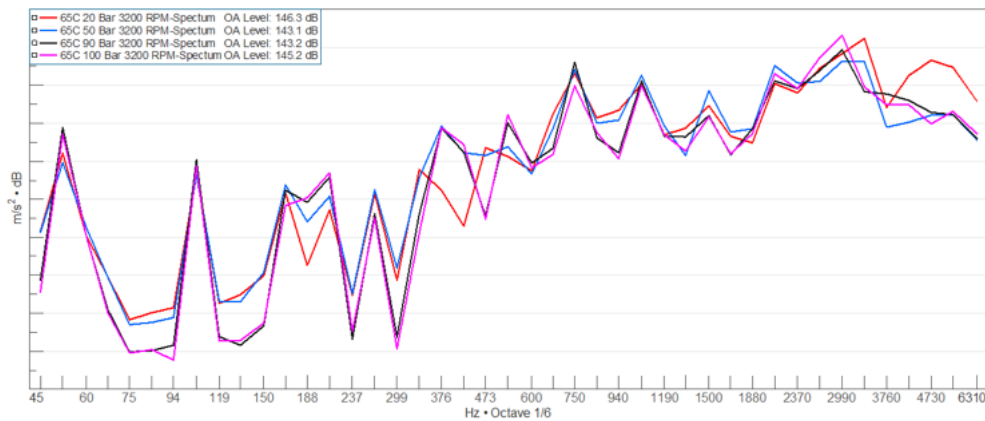
Figure 4.6: Effects of oil temperature on acceleration at various locations for coupled structure 1



(a) Accelerometer 3 x-axis acceleration at 30 C and 1000 RPM



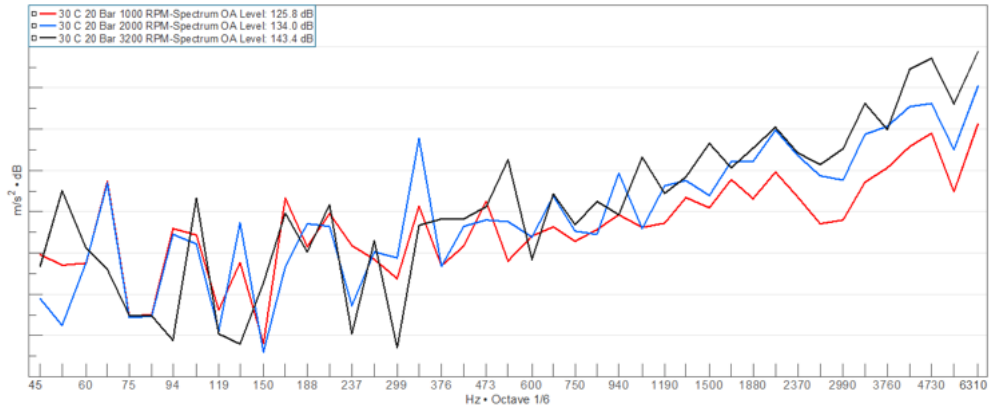
(b) Accelerometer 4 y-axis acceleration at 50 C and 2000 RPM



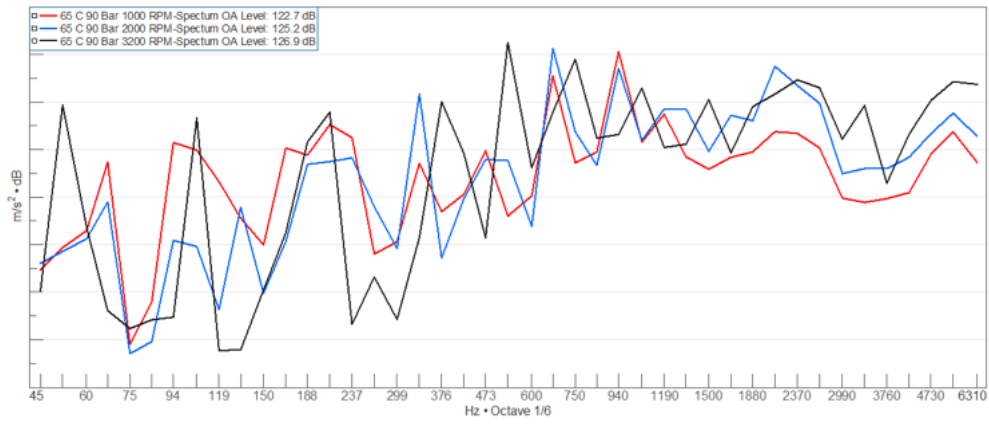
(c) Accelerometer 5 z-axis acceleration at 65 C and 3200 RPM

Figure 4.7: Effects of oil pressure on acceleration at various locations for coupled structure 1. Grid lines denote 10 dB.

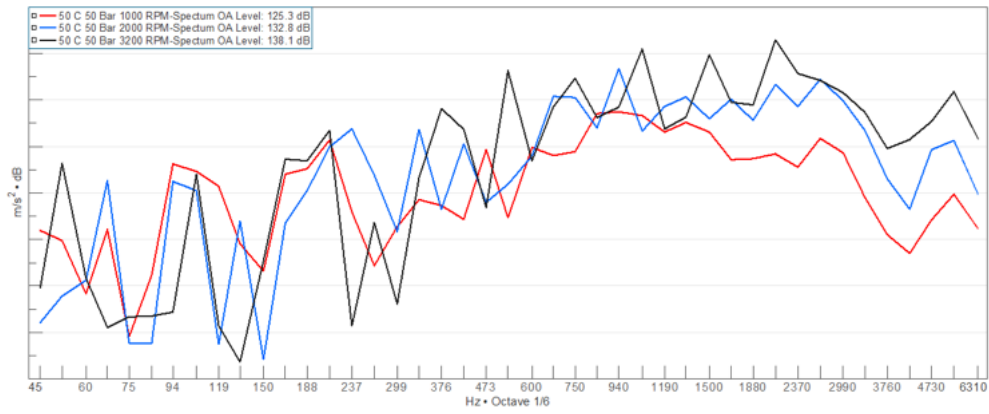
4. Results



(a) Accelerometer 2 z-axis acceleration at 30 C and 20 Bar



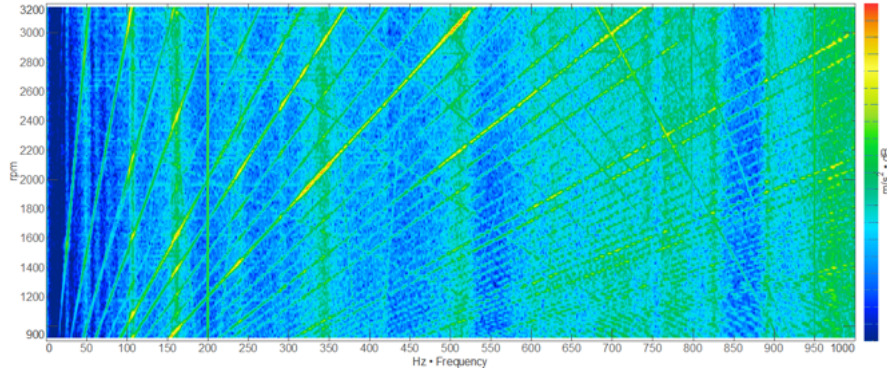
(b) Accelerometer 3 y-axis acceleration at 65 C and 90 Bar



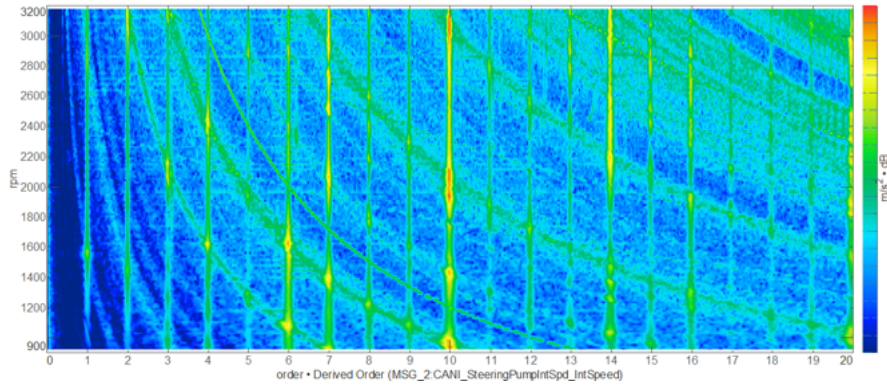
(c) Accelerometer 5 x-axis acceleration at 50 C and 50 Bar

Figure 4.8: Effects of pump RPM on acceleration for coupled structure 1

Figure 4.9 shows the acceleration response at a target location of the run-up measurement on coupled structure 1. Because the steady-state measurement indicated that changes in oil temperature and pressure were not significant for the coupled structure, only the run-up at 30 °C oil temperature and 100 Bar oil pressure will be shown. Figure 4.9b shows a strong dominance from order 7, 10 and 14 on the response at target accelerometer 61. This was observed in both receiving structures.



(a) Acceleration spectrum waterfall



(b) Acceleration order waterfall

Figure 4.9: Waterfall plot of target accelerometer 61 in the z-axis at 30 C and 100 Bar

4.3 Blocked force

The results in this section show the calculated in situ blocked forces at the center of gravity of the source component for both coupled structures 1 and 2. The operating conditions evaluated are the EHPS pump under steady-state and run-up conditions, with the EHPS pump operating at 30 °C oil temperature and 100 Bar oil pressure. The results for both steady-state and run-up conditions showed strong agreement between the blocked forces obtained by the two structures, however some deviations occurred likely due to resonances in the receiving structure being excited during the run-up measurements.

4.3.1 Blocked force at steady-state

Figure 4.10 shows the calculated blocked forces with the system operating at a steady-state speed of 3200 RPM. The blocked forces show strong alignment, with a maximum overall RMS magnitude difference of 4 dB in the moment about the y-axis, and all other DOFs have differences no greater than 3 dB.

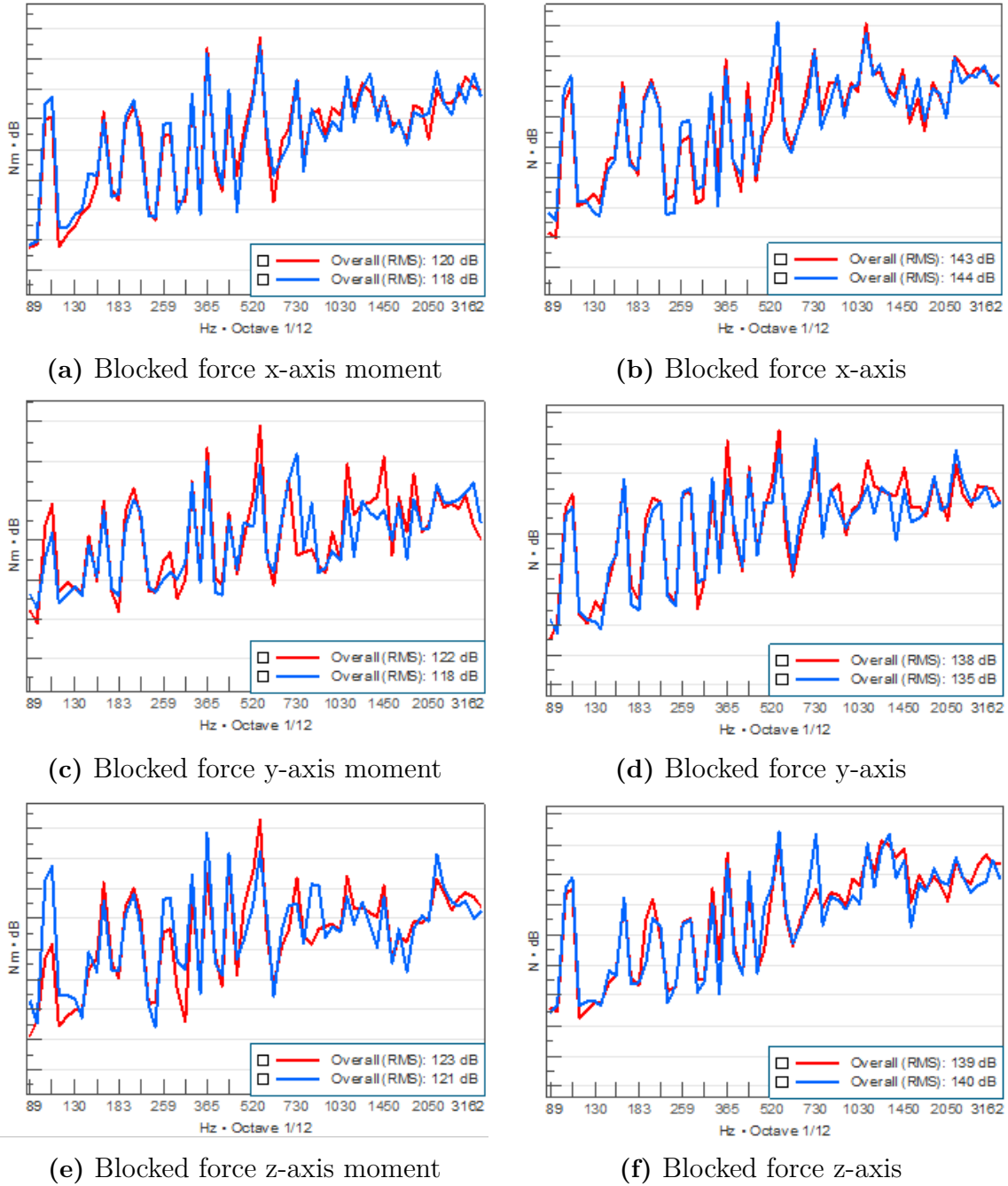
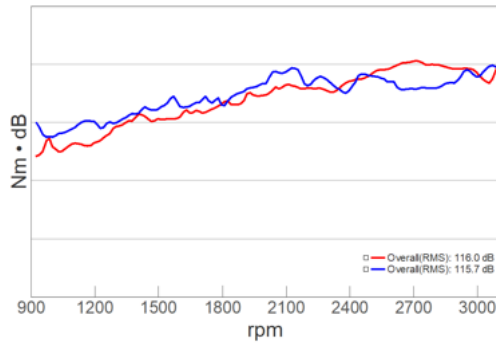


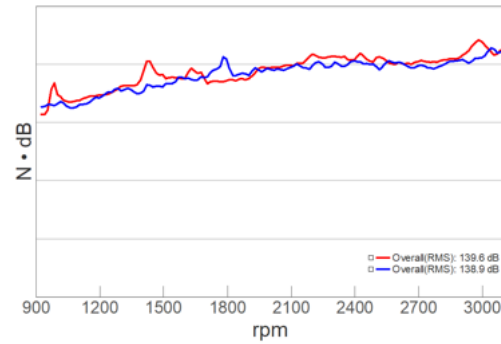
Figure 4.10: Total blocked force at the center of gravity virtual point for coupled structure 1 (red) and coupled structure 2 (blue) at 3200 RPM. Gridlines denote 10 dB.

4.3.2 Blocked Force vs RPM

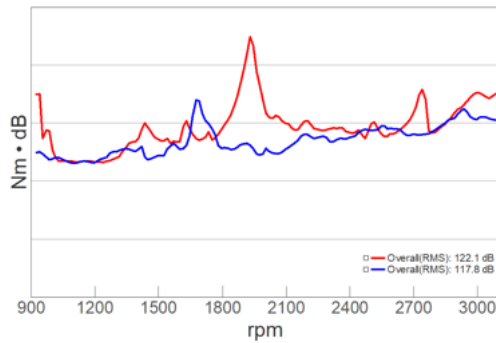
Figures 3.14a and 4.11 show the calculated blocked force at the CoG of the pump during the RPM sweep operation. The blocked force shows good correlation throughout the RPM range, with deviations of less than 3 dB. Figures 4.10c and 4.10e show a deviation in the rotational blocked forces in the Y- and Z-axes at 1931 RPM and 1678 RPM. The peaks are due to a resonance of the receiving structure, as seen in Figure 4.12.



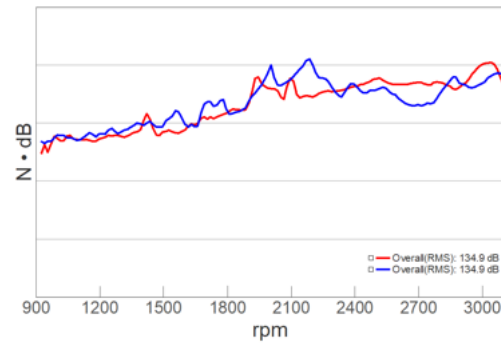
(a) Blocked force x-axis moment



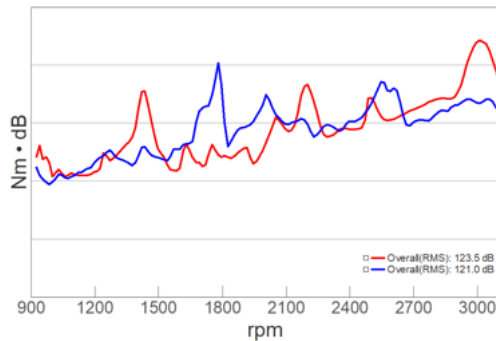
(b) Blocked force x-axis



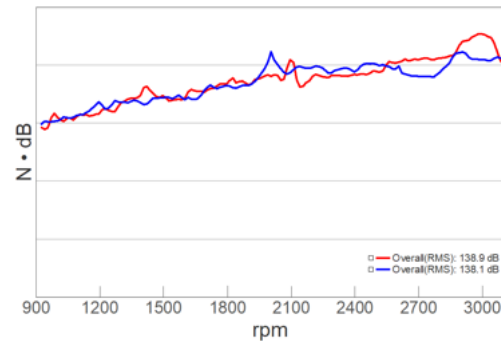
(c) Blocked force y-axis moment



(d) Blocked force y-axis

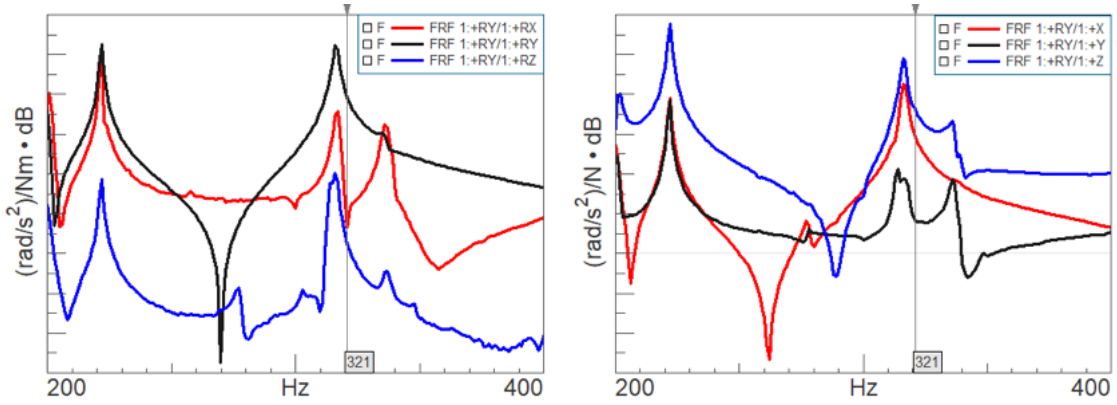


(e) Blocked force z-axis moment

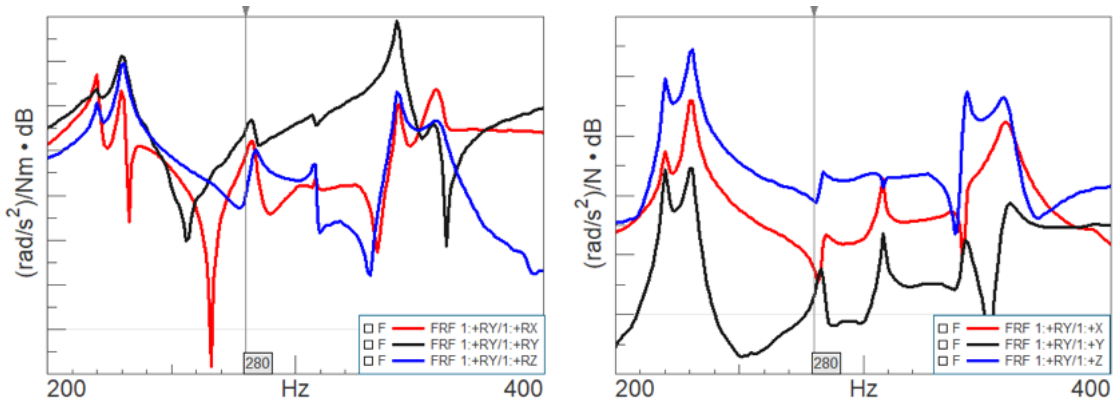


(f) Blocked force z-axis

Figure 4.11: Total blocked force vs. RPM at the center of gravity virtual point for coupled structure 1 (red) and coupled structure 2 (blue). Gridlines denote 10 dB.



(a) Rotational FRF about y-axis of receiver structure 1 due to moments and translation forces



(b) Rotational FRF about y-axis of receiver structure 2 due to moments and translation forces

Figure 4.12: Modal response of receiver structures 1 and 2 shows resonances that coincide with the operational frequency of the source during RPM sweep measurements

4.4 Validation of blocked force

Section 4.3 demonstrated that using the in-situ method on two different receiver structures with virtual point transformation can result in an accurate characterization of the source component. Following completion of the measurement campaign for the source and receiver components, the component database was developed to construct a virtual acoustic prototype. This section shows the results from applying the blocked force to the coupled structure.

4.4.1 Onboard validation

Onboard validation was performed by comparing the measured response at target locations 61 and 62 with the simulated response obtained by applying the blocked force to the coupled structure. To determine the simulated responses, the calculated blocked force was re-applied to the coupled system from which it was calculated. Figure 4.13 shows the comparison between coupled structures under the steady-state

condition of 3200 RPM. It can be observed that the simulated acceleration reproduces the frequency content of the measured acceleration very well. Additionally, the measured overall acceleration in the out-of-plane DOF (z-axis) did not accurately predict the overall level of the in-plane DOFs (x and y axes). This can be caused by including the moment-path contributions for the in-plane axes.

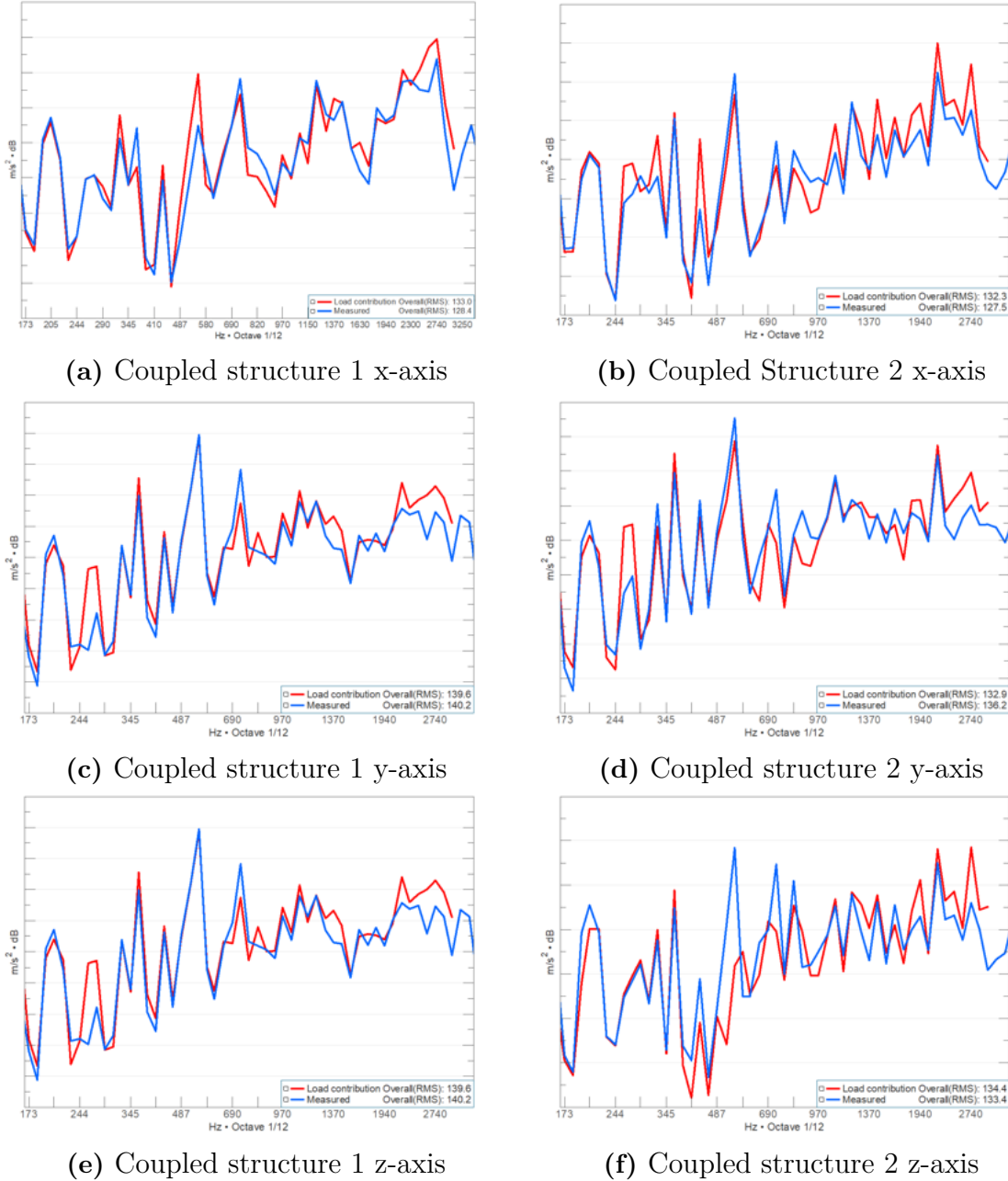
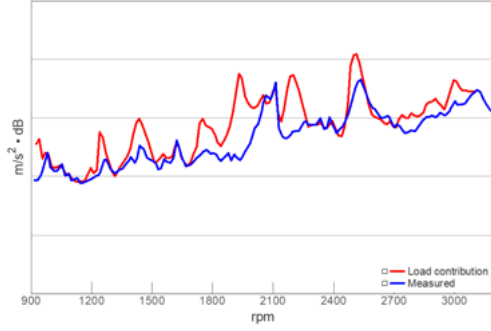
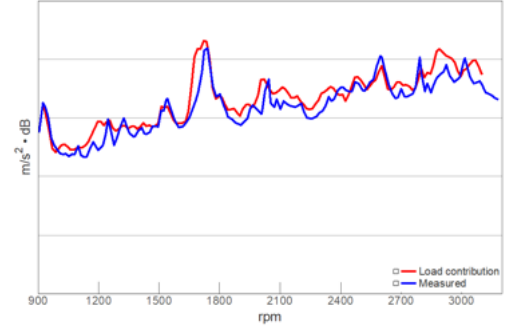


Figure 4.13: Onboard validation of measured(red) vs. simulated (blue) acceleration at the target location on coupled structures 1 and 2 during constant 3200 RPM. Gridlines denote 10 dB

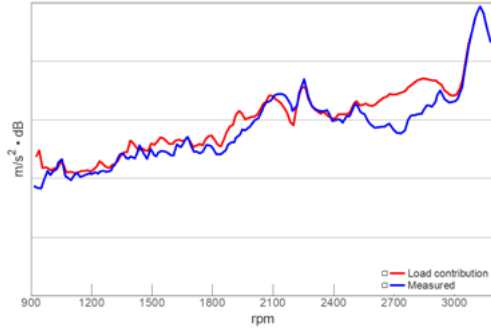
Figure 4.14 shows the comparison between measured and simulated overall acceleration during the RPM run-up operating condition. In all cases, the simulated overall acceleration accurately matched the measured overall acceleration in all DOFs for both coupled structures.



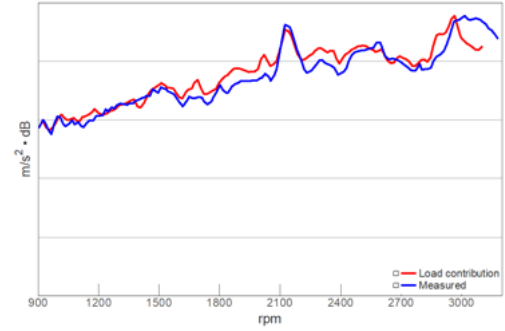
(a) Coupled structure 1 x-axis



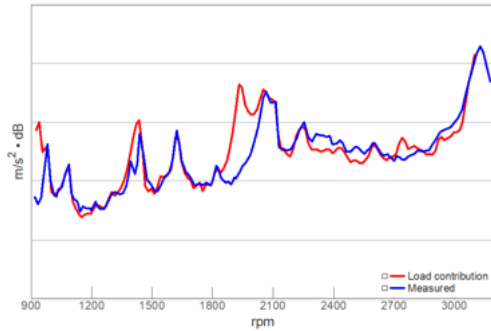
(b) Coupled structure 2 x-axis



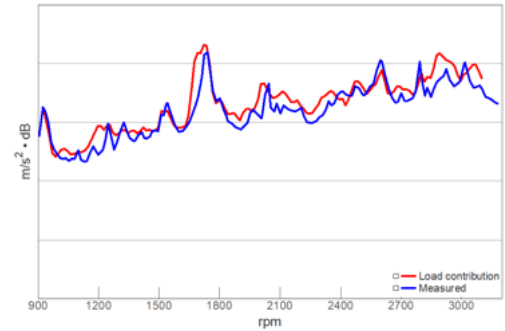
(c) Coupled structure 1 y-axis



(d) Coupled structure 2 y-axis



(e) Coupled structure 1 z-axis



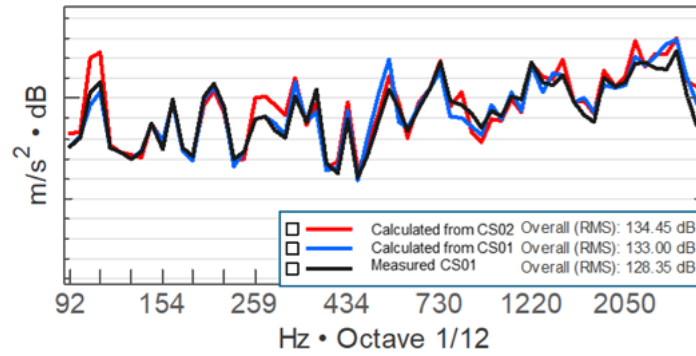
(f) Coupled structure 2 z-axis

Figure 4.14: Onboard validation of measured(red) vs simulated(blue) acceleration at the target location on coupled structure 1 during run-up operation. Gridlines represent 10 dB.

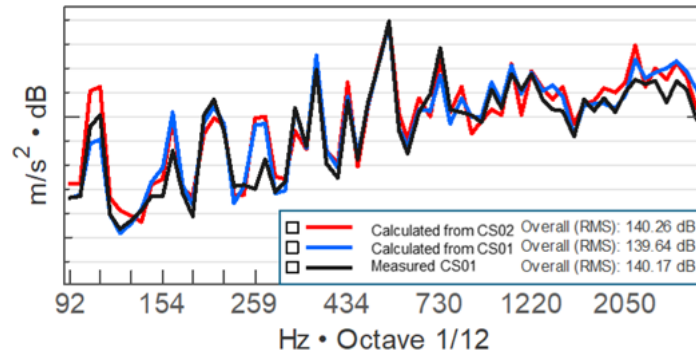
4.4.2 Cross-Validation

Cross-validation was performed where the derived blocked force from one coupled structure was applied to the new structure.

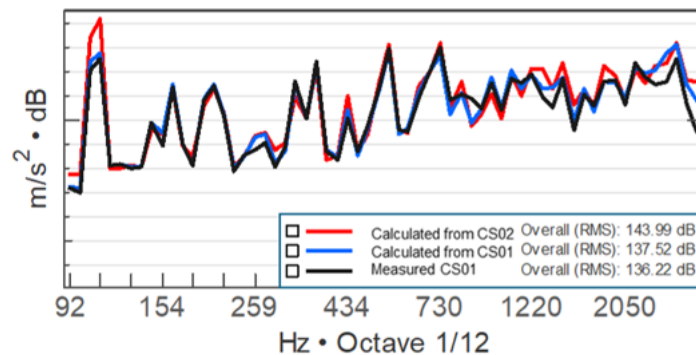
Figure 4.15 shows the acceleration at measurement location 62 for two calculated responses and one measured response with the pump operating at 3200 RPM. The calculated spectra successfully captured the frequency spectrum of the measured value. Variations in magnitude at particular frequencies were due to resonances when the source was operating at 3200 RPM, in both the derivation of the blocked force and in the resonance of the coupled structure itself. However, even with the deviations, the overall correlation of magnitude over the frequency range was acceptable.



(a) Target location 62 x-axis acceleration response



(b) Target location 62 y-axis acceleration response

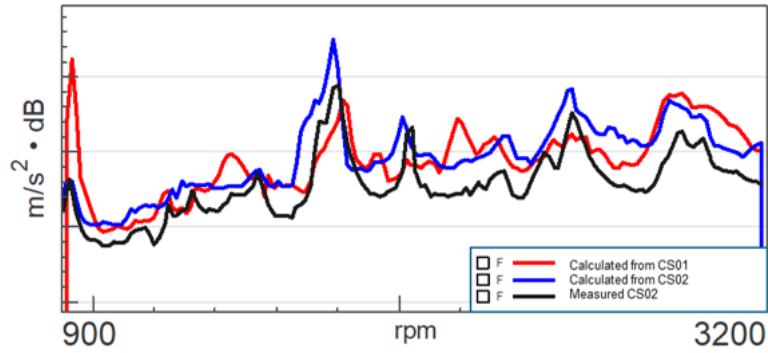


(c) Target location 62 z-axis acceleration response

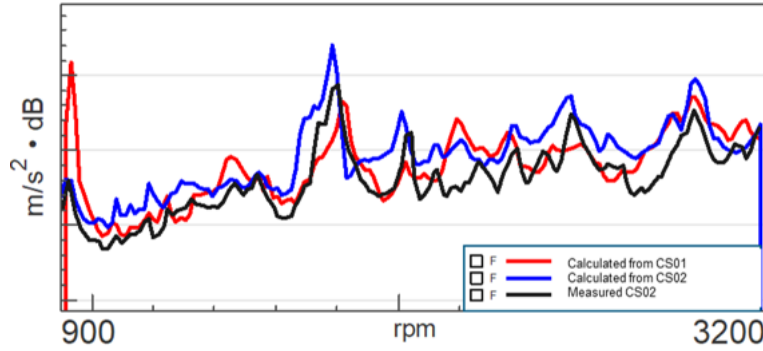
Figure 4.15: Transferability validation of coupled structure 1 during steady-state conditions using blocked forces from coupled structure 2 (red) and coupled structure 1 (blue), compared with measurements on coupled structure 1 (black).

Gridline denotes 10 dB.

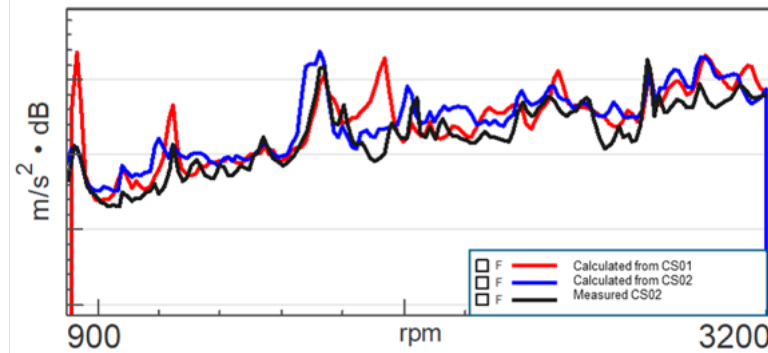
Figure 4.16 shows the acceleration at measurement location 61 for two calculated responses and one measured response during an RPM sweep. Similar to Figure 4.15, the simulated acceleration at the target location successfully captured the frequency trend of the measured values, and the variance in magnitude at particular frequencies is due to resonances. However, even with the variances, the overall correlation of magnitude over the frequency range was acceptable.



(a) Target location 61 x-axis acceleration response



(b) Target location 61 y-axis acceleration response



(c) Target location 61 z-axis acceleration response

Figure 4.16: Transferability validation of coupled structure 2 during RPM sweep using blocked forces from coupled structure 1 (red) and coupled structure 2 (blue), compared with measurements on coupled structure 2 (black). Gridline denotes 10 dB.

4.5 Virtual prototype

Following sections 4.4.1 and 4.4.2 show the validation and transferability of the derived blocked force from two separate coupled structures. This section demonstrates the application of a virtual prototype for NVH analysis.

4.5.1 Path contribution

Virtual prototype models sum the contribution of each input DOF to the target location. The contribution from each path can be evaluated to provide guidance on NVH solutions. Shown in figure 4.17 is the effect of each DOF at the center of gravity virtual point, on the +z-axis at target location 61 under steady-state operation. When comparing the overall calculated vs measured values, the calculated vibration matches measured values between 0 and 3000 Hz. It can be observed that a major source of 0-600 Hz frequency range is due to the rotation about the y-axis.

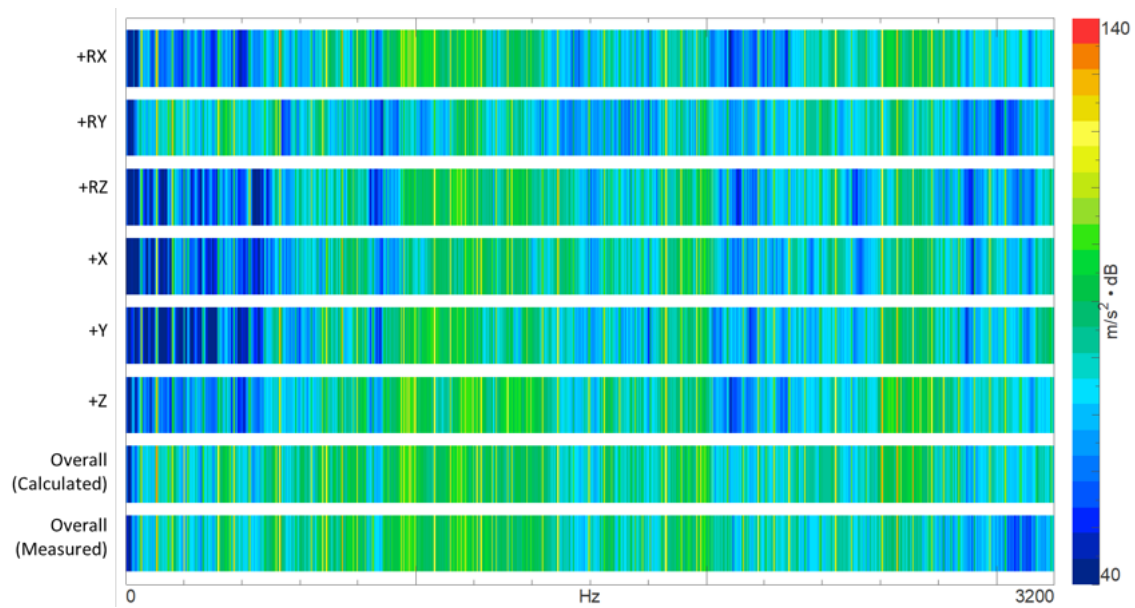
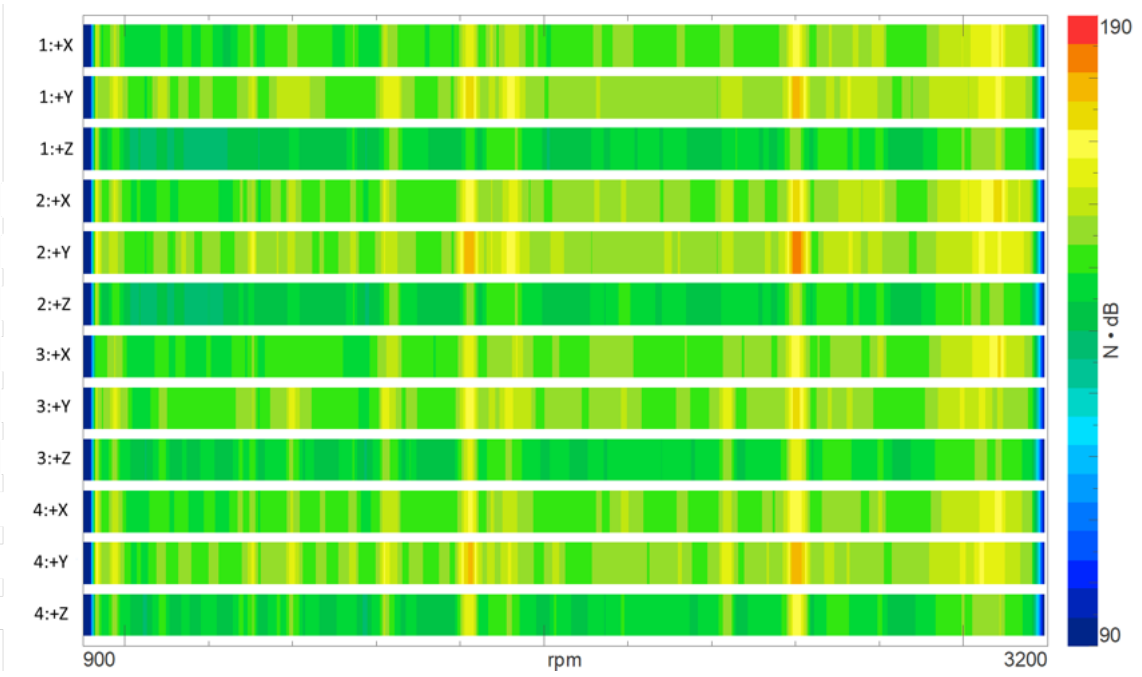
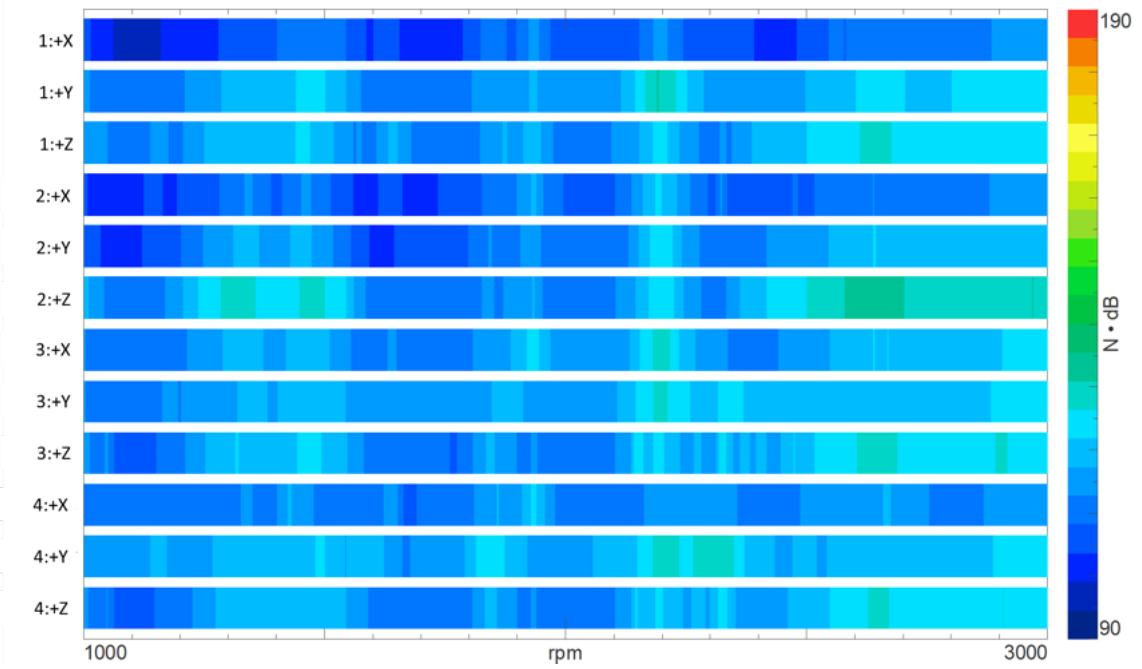


Figure 4.17: Path contribution at center of gravity during steady-state condition

Because virtual prototypes are constructed by building an assembly from its individual components, it is possible to extract information at any interface in the measured path. Figures 4.18 and 4.19 demonstrates calculated translational and rotational loads at the four interface locations between the source and receiver 1. Figures 4.18a and 4.19a show the load case with the source hard mounted to the receiver, whereas 4.18b. For all interfaces the loads were higher in the in-plane axis (x and y). This could be due to the out-of-plane axis (z axis) having the higher compliance. Figure 4.19b show the load cases with mount A from 4.5.2 installed on the system.

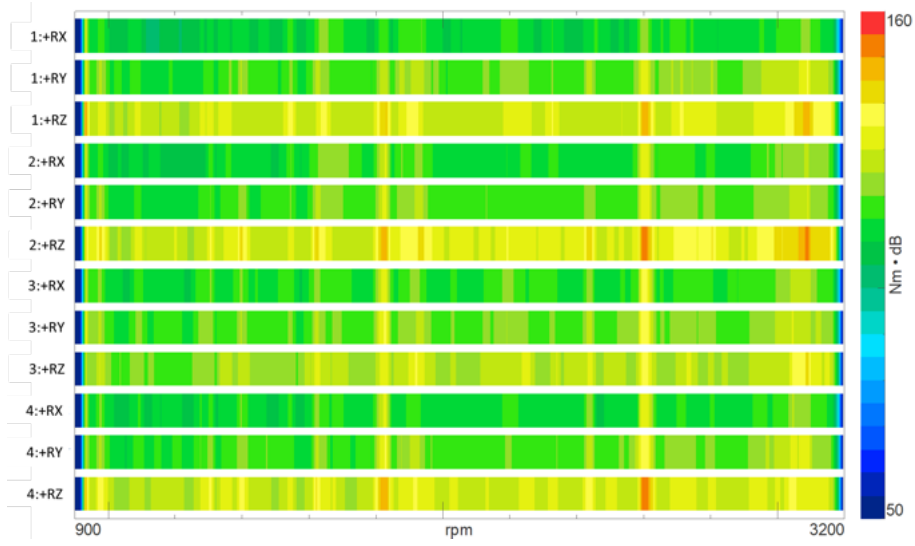


(a) Translation loads at interfaces

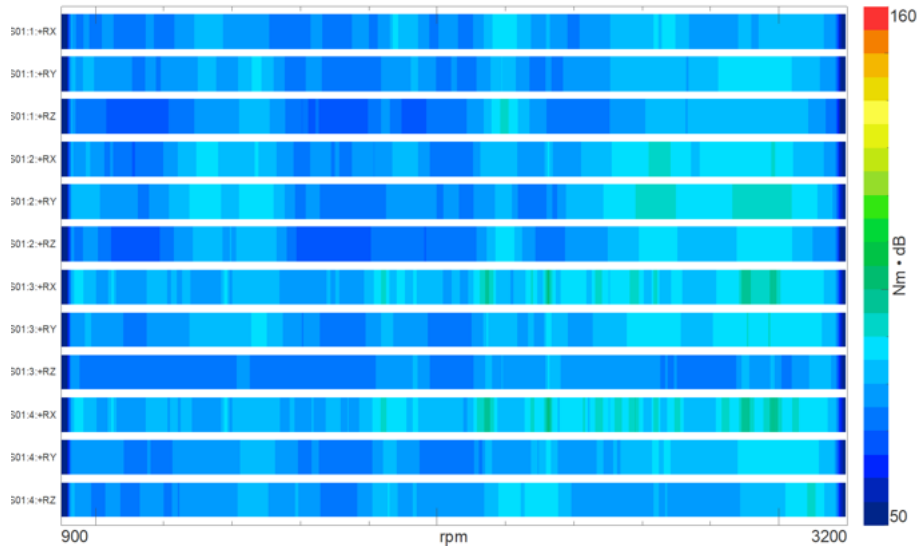


(b) Translation loads at interfaces with Mount A

Figure 4.18: Translation and rotational loads calculated at all interface locations of Receiver 1 during run-up



(a) Moment loads at interfaces



(b) moment loads at interfaces with mount A

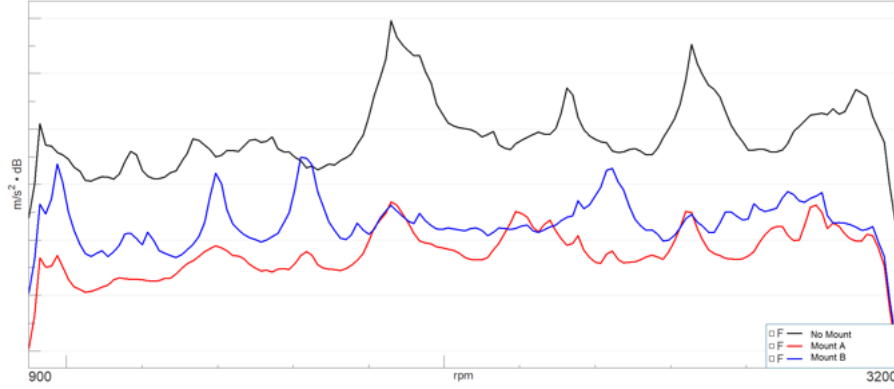
Figure 4.19: Rotational loads calculated at all interface locations of Receiver 1 during run-up

4.5.2 Scenario simulation

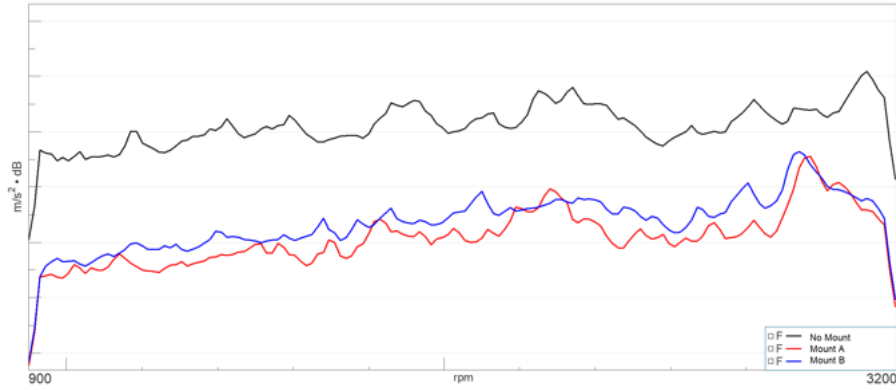
This section describes how to apply the blocked force to evaluate different scenarios, such as mount selection. Since the blocked force is independent and transferable, virtual prototypes can simulate scenarios involving changes to the system. Figure 4.20 shows the simulation of selecting a mount from supplier-provided data. An outside source provided measurement data on the mount stiffness of an isolator. By incorporating the mounts into the virtual prototypes of coupled system 1, the effect of the mounts could be predicted. Three test RPM sweep scenarios were simulated: No mount, Mount A, and Mount B. Using a virtual prototype, the effects of adding mounts were observed. Mount B was observed to introduce additional excitation, particularly at 975, 1394, and 1629 RPM. The spectrum map shows that these

4. Results

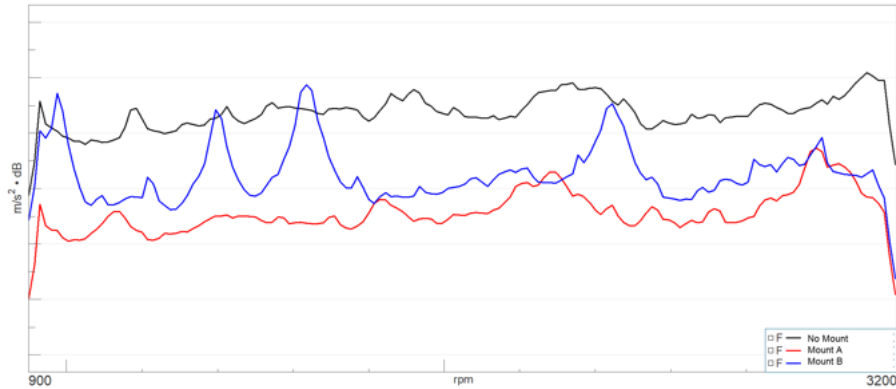
spikes were caused by changes in the system's dynamic behavior, during which the new system's resonances were excited. With this simulation scenario, the effects of changes to the system on dynamic performance can be evaluated.



(a) Target location 61 x-axis acceleration response



(b) Target location 61 y-axis acceleration response



(c) Target location 61 z-axis acceleration response

Figure 4.20: Comparison of simulated results of no mount (black) mount A (red) and mount B (Blue) scenarios on acceleration at target location

5

Discussion

5.1 Virtual point transformation quality checks

During the measurement campaign, quality checks were performed to evaluate the quality of the measurements. Condition numbers were calculated to evaluate the sensitivity of the VPTs, as described in Section 5.1.1, and measured and reconstructed FRFs were used to evaluate the linearity of the VPTs, as described in Section 5.1.3.

5.1.1 VPT condition number

The condition number for a VPT describes the sensitivity of the virtual point to the responses of the measured accelerometers. For VPTs to meet the requirements in Section 2.5.1.2 and 2.5.1.3, the condition numbers for the response and excitation DOFs are below 50. Outputs from Siemens Testlab showed that all VPTs meet the condition number requirements, as seen in Table 5.1. Following the FRF measurements, using 9 DOFs instead of 10 or 12 DOFs to calculate the VPT improved the condition number.

Response Condition Number (12 Indicators)						Excitation Condition Number (10 indicators)					
Virtual Point Component	COG	Mount 1	Mount 2	Mount 3	Mount 4	Virtual Point Component	COG	Mount 1	Mount 2	Mount 3	Mount 4
Receiver 1	-	36.35	36.22	35.75	35.94	Receiver 1	-	48.32	48.32	48.32	48.32
Receiver 2	-	35.57	42.29	36.68	38.24	Receiver 2	-	45.77	45.79	46.49	45.54
Source	24.9	30.62	23.13	25.77	12.84	Source	34.8	20.99	27.58	31.82	22.01
Coupled System 1	11.73	-	-	-	-	Coupled System 1	11.73	-	-	-	-
Coupled System 2	21.85	-	-	-	-	Coupled System 2	21.83	-	-	-	-

Table 5.1: Excitation and response condition number of VPT with all indicators

Response Condition Number (9 Indicators) (0-2000Hz)						Excitation Rigidity Condition Number (9 indicators) (0-2000Hz)					
Virtual Point Component	COG	Mount 1	Mount 2	Mount 3	Mount 4	Virtual Point Component	COG	Mount 1	Mount 2	Mount 3	Mount 4
Receiver 1	-	36.96	36.88	38.89	38.74	Receiver 1	-	42.13	42.13	42.13	42.13
Receiver 2	-	10.35	42.89	37.13	42.37	Receiver 2	-	44.78	43.5	46.49	42.89
Source	18.85	24.92	36.77	29.04	36.67	Source	25.66	33.46	37.14	28.41	39.34
Coupled System 1	23.17	-	-	-	-	Coupled System 1	35.79	-	-	-	-
Coupled System 2	23.17	-	-	-	-	Coupled System 2	36.65	-	-	-	-

Table 5.2: Excitation and response condition number of VPT with reduced indicators and frequency range

5.1.2 VPT rigidity

For the VPT to capture the system's dynamics, the measured responses used to calculate the virtual point must meet the rigidity requirement. However, if the components or systems are inherently nonlinear, such as the use of mounts at the interface, or if the source component operates with internal fluids sloshing, when a force is applied to the system, the response is not expected to maintain a linear relationship. For coupled structures 1 and 2, 15 reference and 12 response indicator FRFs with varying degrees of coherence were used to calculate the virtual point rigidity for the reference and response between 0 and 2000 Hz. The VPT rigidity was calculated using the automatic indicator selection tool in the Siemens Testlab software, which applies an optimization algorithm to determine the 9 optimal indicators for the response and excitation rigidity. It is recommended [16] that the average rigidity across the frequency range be greater than 0.9.

Response Rigidity (9 indicators) (0-2000Hz)						Excitation Rigidity(9 indicators) (0-2000Hz)					
Virtual Point Component	COG	Mount 1	Mount 2	Mount 3	Mount 4	Virtual Point Component	COG	Mount 1	Mount 2	Mount 3	Mount 4
Receiver 1	-	0.932	0.921	0.930	0.949	Receiver 1	-	0.995	0.997	0.992	0.992
Receiver 2	-	0.954	0.940	0.960	0.948	Receiver 2	-	0.944	0.973	0.928	0.957
Source	0.961	0.928	0.912	0.912	0.907	Source	0.952	0.962	0.949	0.950	0.950
Coupled System 1	0.945	-	-	-	-	Coupled System 1	0.942	-	-	-	-
Coupled System 2	0.955	-	-	-	-	Coupled System 2	0.956	-	-	-	-

Table 5.3: Excitation and response condition number of VPT

5.1.3 Recalculated FRFs

In section 2.5.1.1, the requirements of rank and rigidity describe the requirement to compare the measured FRF with a recalculated FRF. Figure 5.1 shows an example of a recalculated FRF for the source component at accelerometer 6 in the y-axis due to excitation 22. Figures 5.2 and 5.3 show the recalculated FRF compared to the measured FRF for accelerometer 9 in the x-axis for excitation 45 and accelerometer 18 in the z-axis for excitation 45.

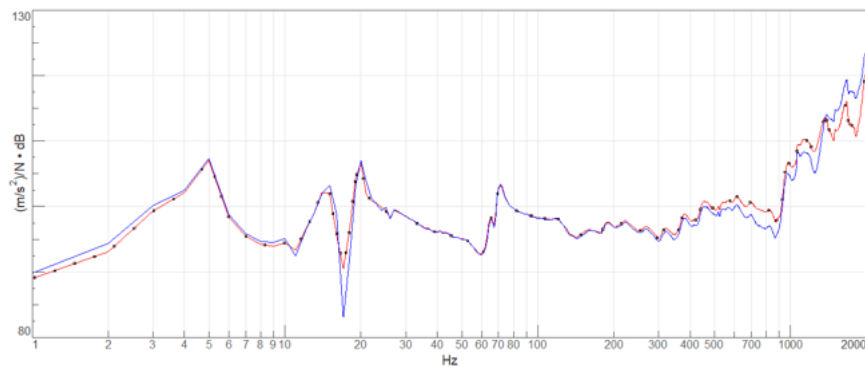


Figure 5.1: Recalculated FRFs from response(Red) and excitation(Blue) DOF 6 y-axis

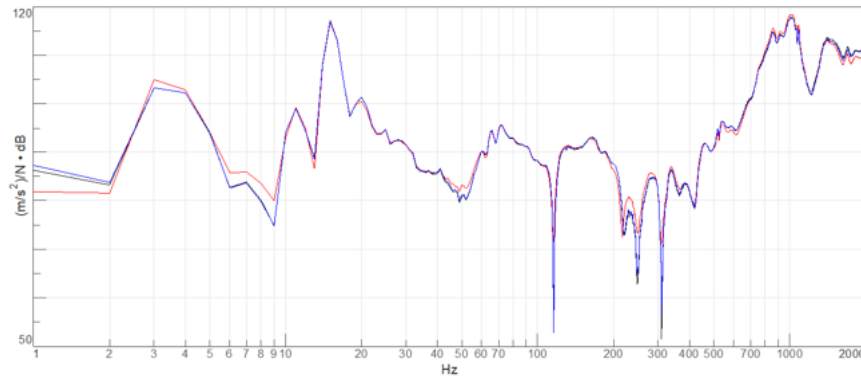


Figure 5.2: Measured (black) FRF vs recalculated FRFs from response(red) and excitation(blue) DOF 9 x-axis

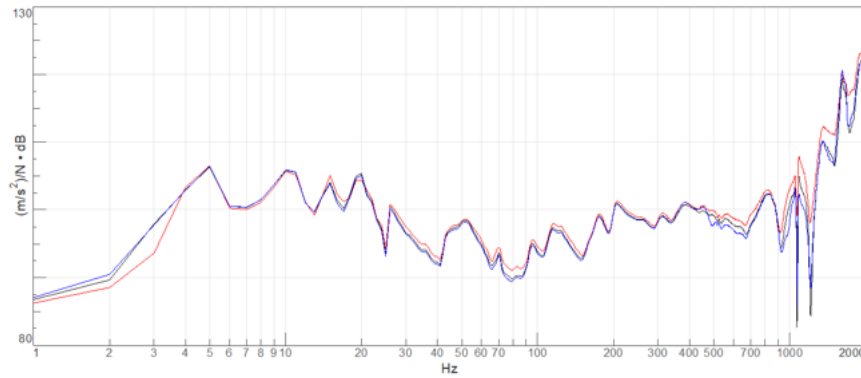


Figure 5.3: Measured (black) FRF vs recalculated FRFs from response(red) and excitation(blue) DOF 18 z-axis

5.1.4 Limitations and uncertainty of VPT

During the setup of the measurement campaign, consideration was given to the placement of the VPT transducers. Because the transducer location in the CAD model can be precisely defined, the physical placement of the transducer had to match the model's geometric location. Due to manual application and the use of mounting clips and adhesives, the exact positioning of the transducers, both translational and rotational, must be performed carefully.

An additional challenge of VPTs is the number of measured DOFs required to calculate multiple VPTs. The initial setup for the measurement campaign had 4 virtual points, one at each interface location. Because the measurements targeted 1.5 times the total number of VPT DOFs, a total of 36 DOFs, or 12 triaxial accelerometers, are needed for the measurement. Due to the geometry of the source component, placing the required number of accelerometers while capture the dynamics of all degrees of freedom was challenging. If the required number of measured DOFs cannot be achieved, a reduction of VPT DOFs may be necessary, such as capturing only translational DOFs.

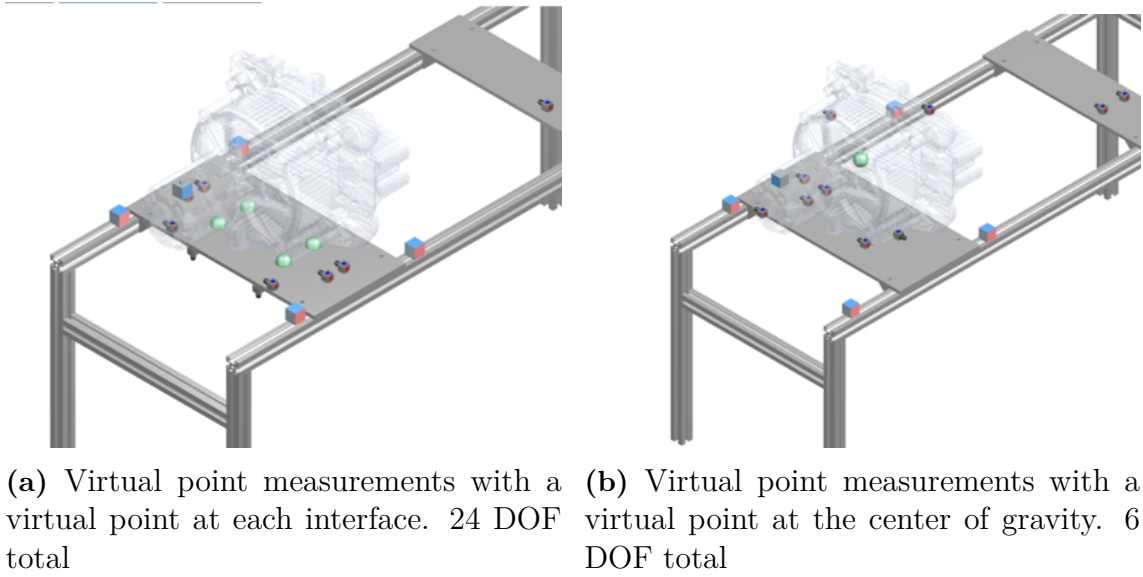


Figure 5.4: Comparison of virtual point at each interface vs center of gravity

5.2 Blocked force quality check

Following the measurement campaign, the blocked force was calculated using the in situ method. Similar to Section 5.1, condition number calculations can be performed to determine the quality of the derived blocked force.

5.2.1 Blocked force condition number

During the calculation of the blocked force, a condition number is computed to assess how well the pseudo inverse captures the dynamic behavior. The condition number is determined by comparing the ratio between the largest and smallest eigen value at each frequency between the DOFs. A high condition number does not inherently indicate an incorrect result; they indicate high uncertainty in the blocked force calculated by the system. While the eigen value cannot be correlated with a specific DOF it describes, using more DOFs to calculate the blocked force requires capturing more unique dynamic behavior. High eigen value amplitudes can be attributed to pseudo-matrix inversion when calculating the blocked force, where, following the inversion, the anti-resonances in the system exhibit high dynamic amplitudes. In the 4 virtual point measurements seen in figure 5.4a, 24 eigenvalues were used to determine the condition number. As seen in figure 5.5a, the condition number exceeded the requirement, as the condition number of the blocked force should not exceed 100. This was thought to be due to the location of the measured DOFs, where the dynamic behaviors of specific DOFs was too similar because of their proximity. To improve the condition number of the blocked force, truncation can be applied to use fewer DOFs in the calculation. While this method improves the condition number as seen in Figure 5.5b, it is important that the truncation does not remove so many DOFs that the dynamic behavior loses accuracy. If eigenvalue truncation does not result in an acceptable condition number, as seen in Figure 5.5,

results that accurately depict the system's dynamic behavior may not be obtained. New measurement points need to be selected to capture its dynamic and geometric behavior. Figure 5.6 shows the effects of reducing the VPTs from 4 locations at the interface to 1 location at the CoG and placing accelerometers in strategic locations to capture the global dynamic behavior, so that each measured DOF describes a different dynamic of the system.

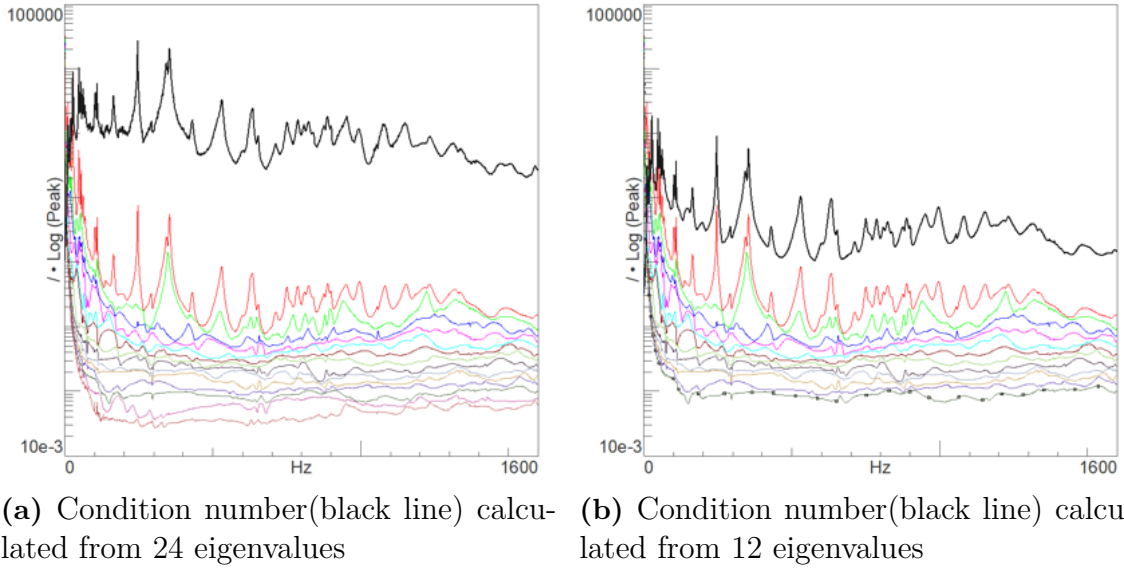


Figure 5.5: Condition number is calculated from the ratio of amplitudes from the highest and lowest eigenvalues

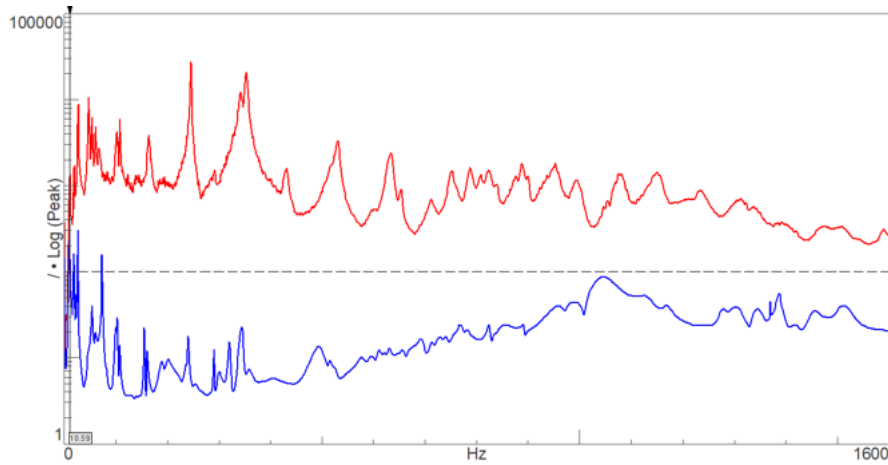


Figure 5.6: Condition number of 4 virtual points(red) and virtual point at center of gravity. An acceptable condition number is below 100 (dotted black line).

5.2.2 Blocked forces at center of gravity

The measurement process to derive the blocked force at the center of gravity does not meet the substructuring requirement of being "at the interface". To obtain the

interface forces, the transfer function between the center of gravity and each interface must be known. To identify this transfer function, the source component was tested in free-free conditions to obtain H_{21} . The advantage of using the blocked force at the CoG is that, if the source component has various mounting configurations, the interface force can be identified as long as the transfer function between the CoG and the interfaces is known. A challenge with this method is that, to obtain the matrix H_{21} , the source component should be tested under free-free boundary conditions. This can be a challenge for large sources with limited modifications to the boundary conditions.

5.3 VAP Challenges

To build a VAP, extensive characterization of components is required to create a component database. For an OEM, developing the database requires significant time and capital resources to conduct the measurement campaign and validate every component. While simulation tools can be used to obtain FRFs, source components require significant in-house development efforts and may not be practical for smaller OEMs with a limited range of products. Blocked forces could be obtained by requiring component suppliers to perform the measurement and provide the data; however, this requires extensive coordination to agree on the interface method, roles in the measurement campaign, and to share sensitive and possibly proprietary data.

6

Conclusion

The objective of this project was to verify and validate the method to characterize a source component using the in situ blocked force method. A comprehensive measurement campaign was conducted on an EHPS pump mounted on two dynamically different receiving structures. The results showed that the derived blocked forces are independent of the receiving structure and therefore represent an inherent property of the source component. Using the VPT interface method, blocked forces were additionally derived for all translational and rotational DOFs at the CoG. The validity of the blocked force approach was confirmed through both on-board and cross-validation between the two test configurations. With the creation of a component database, the VAP process was demonstrated through scenario simulations and path contribution analysis. Further work is recommended in several areas. First, applying the in situ blocked force method on a complete vehicle setup to allow comparison with test bench measurements. Second, the sensitivity of the results to accelerometer placement should be investigated. Finally, the influence of mounting compliance and damping should be studied by including flexible mounts between the source and receiver when deriving blocked forces.

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