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A METHODICAL APPROACH FOR DYNAMIC SYSTEM ANALYSIS & SYNTHESIS IN THE DIMENSIONING OF VARIANT LIGHTWEIGHT CABIN INTERIOR

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Abstract

To obtain valid simulation parameters for simulation based lightweight optimization of cabin interior, various system identification tests have been performed with specimens ranging from sandwich panels to fully loaded galleys. Furthermore, a methodical approach using dynamic substructuring algorithms was developed to support simulation based vibration behaviour prediction of a modular product structure and improve the dimensioning for highly variant lightweight structures.

1 Motivation and background

The dimensioning of aircraft cabin interior monuments is traditionally based on basic simulations using the Finite-Element-Method (FE). Because of sometimes very coarse simplifications these are normally combined with worst-case validation using quasi-static testing.

1.1 Dynamic excitation loads in the cabin

Over the last years however, dynamic loading has been an ever growing topic in the dimensioning of the aircraft cabin. Especially in new designs the dimensioning is not focusing solemnly on transient loads from the emergency landing conditions but also takes low frequency stationary vibrations into account. This has comfort driven reasons as wells as safety driven requirements. This contribution will focus on stationary/periodic dynamic excitations below the hearing threshold with a lot of energy being transmitted for the dimensioning of cabin interior.

A **comfort-driven** issue of low frequency vibrations is the shaking of wall or ceiling mounted entertainment monitors during flight, which may cause discomfort. Another aspect of discomfort is the rattling of larger cabin monuments like galleys, which can occur during taxi, ground acceleration, take-off, turbulences and landing.

A clearly **safety-driven** aspect of dimensioning is the sustained engine imbalance (SEI) or blade loss windmilling condition. In this case, a fan with a blade lost and still rotating in the airstream causes a high rotary imbalance and hence vibrations that are transmitted into the cabin, where heavy monuments like galley have to stay in place. The trend of larger fan diameter engines to be used on single-aisle aircraft in order to save fuel burn makes this an ever more pressing topic because smaller planes get higher excitations than before. [1]

1.2 The challenges with highly variant product structures in dimensioning

Simulations with valid and accurate model parameters provide reliable predictions and support a weight saving design optimization. But the high variety of cabin interior monuments makes optimization with detailed simulation based mechanical analysis very tedious. As airlines follow the trend towards an ever more individual cabin look and layout, the cabin interior gets highly customized as depicted for various partition variants in Fig. 1.

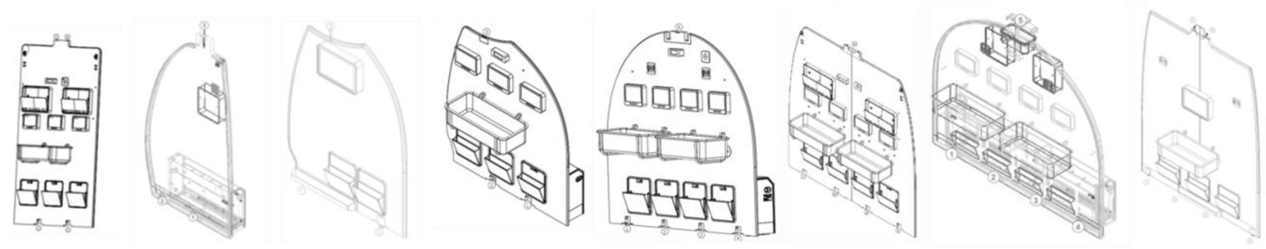

Fig. 1. Overview of partition variants from cooperation partner Diehl Service Modules (source: Diehl Service Modules)

With a small given time slot for the dimensioning, there is often hardly any time for optimization loops of each and every variant. Therefore overdimensioned designs are much favoured in the hope that these will to cause the least problems in later substantiation under safety aspects. Inferior design solutions and high weight are the outcome. Furthermore, important simulation parameters of the monuments for a valid model are often unknown. Therefore, very rough guesses have to be used with safety demanding a huge overdimensioning through very conservative assumptions.

The integrated PKT-Approach for Developing Modular Product Families pursues the objective of reducing internal variety for the company and offering optimized external variety for the customer by combination of a few modules to form many different product variants. [2] This idea of a modular product structure is now transferred to vibration dimensioning using dynamic substructuring algorithms.

1.3 Means to be utilized for dynamics dimensioning of highly variant structures

In the light of the high number of variants to be dimensioned, the following approaches for redesign were found in literature or industry for a vibrational problem with resonance in excitation frequency band:

- stiffening for frequency push
- interface dampers
- revision of the variants offered
- geometric redesign and reallocation
- damping increase of the structure

If the resonance occurs only at the upper end of the excitation range, a frequency push may be used by stiffening the structure. This approach does not need a well-defined simulation model of a detailed mechanical analysis and can be achieved by engineering judgement of experienced engineers in the absence of a detailed mechanical analysis as coined by [3]. However, as the resonance frequency of a linear one-degree-of-freedom system depends on a stiffness/mass ratio, the inherent mass increase with every increase in stiffness limits the frequency push significantly. With sandwich materials used commonly for cabin interior and a limited design space, the stiffness increase is usually realized through stiffer face sheets of the sandwich. Therefore even a small frequency push induces a high mass increase, contradicting the critical lightweight ambitions in general aircraft design. The difference of the stiffening approach compared to others is explained in [4] by distinguishing between narrow-band (a slight frequency push may be used) and broad-band applications (high frequency push factor is needed).

One approach is the use of interface dampers at the attachments of the cabin monument to the aircraft structure. This approach introduces significant further weight and relative deflection as the dampers need to support for high emergency landing interface forces as well.

Another approach could make use of the information on the specific dynamic behaviour of individual variants by not offering selected variants with extreme resonance behaviour due to low damping. This however, may not be an option in fierce economic competition.

As a less harsh alternative to the former approach the critically vibration variants could be redesigned for a better vibrational behaviour, for example by placing components with a high damping contribution further towards places with high deflection rates in resonance. This however, may also conflict with the customers wishes, but probably less significantly than the former approach.

The approach that targets the root of the elevated resonance behaviour is the increase of the damping of the damping structure. As shown in [5], the sandwich panels used commonly only offer a critical damping ratio of 1% (referring to a structural damping of 2%) in the first global mode. The increase of sandwich structure damping is still in fundamental research stage.

All but the very weight-costly stiffening approach require a good understanding of the vibrational behaviour, with at least a prediction of the relevant global modes regarding their frequency, damping and resulting interface forces or local amplifications.

In order to support any of the approach, the Institute PKT has conducted several small and large scale tests of cabin monuments and components to derive the needed simulation parameters (shown in Fig. 2) as well as sought for ways to support faster and better dynamic dimensioning over high numbers of variants.

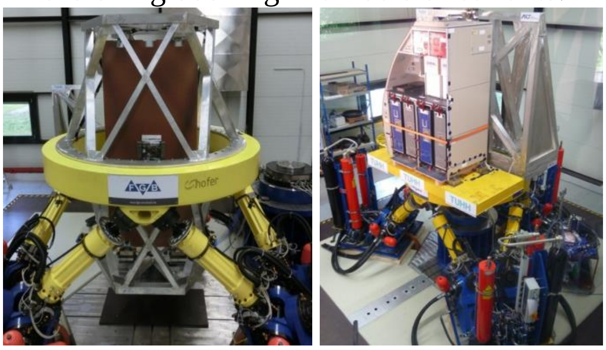

Fig. 2. Partition panel with literature pocket & magazine (left), fully loaded G2 galley (right) on 6dof shaker

2 Dynamic substructuring as an approach

In literature, the term "dynamic substructuring" refers to the coupling of models in structural dynamic analysis. By using coupling and decoupling algorithms, an implementation of the modular substructure in dimensioning is possible. Furthermore, the interchangeable use of substructure models from simulation or real test data is possible by coupling Frequency Response Functions (FRF) from simulation or test origin. This can be very helpful if relevant simulation parameters are not available with the needed accuracy or if the dynamic behaviour of a substructure is very difficult to model, for example due to non-linearities. Because of the possibility to use test data without the reduction to simplified one-degree-of-freedom-models in the modal domain, the Frequency Based Assembly (FBA) is chosen over the Component-Mode-Synthesis (CMS, i.e. see [6]) in this context. For a description of the technical implementation side of the approach presented, see also (Plaumann et al. 2013).

A first presentation of the frequency based substructuring can be found in [7]. The authors describe the vibrational analysis of a helicopter that is segmented into five dynamic substructures. The dynamic models are defined separately for each substructure by using Frequency Response Functions, which are then coupled together. A more recent source describing the current state of dynamic substructuring is [8]. Here however, only the benefit of having different groups working on the same complex system is mentioned regarding aspects of variety. A literature review did not provide references to improve variance handling among the advantages of the approach in these and other publications like [9,10] except for [11]. This publication however only refers to Component-Mode-Synthesis, remains on a fairly non-technical level and is not compliant to the integrated PKT approach for the development of modular product families.

The main challenge when using dynamic substructures is a consistent interface definition for all possible combinations of modules. Without consistent interface definitions over all variants, no coupled calculation is possible. This can be supported by a methodical support.

2.1 Overview of the proposed approach of modular dynamic analysis and synthesis

A specific dimensioning approach has been developed with the aim to simplify the prediction of the vibrational behaviour of composed structures such as cabin monuments with the given high number of variants. It uses the mechanical formulation of the dynamic substructuring approach but adds the necessary methodical guideline to be able to perform the

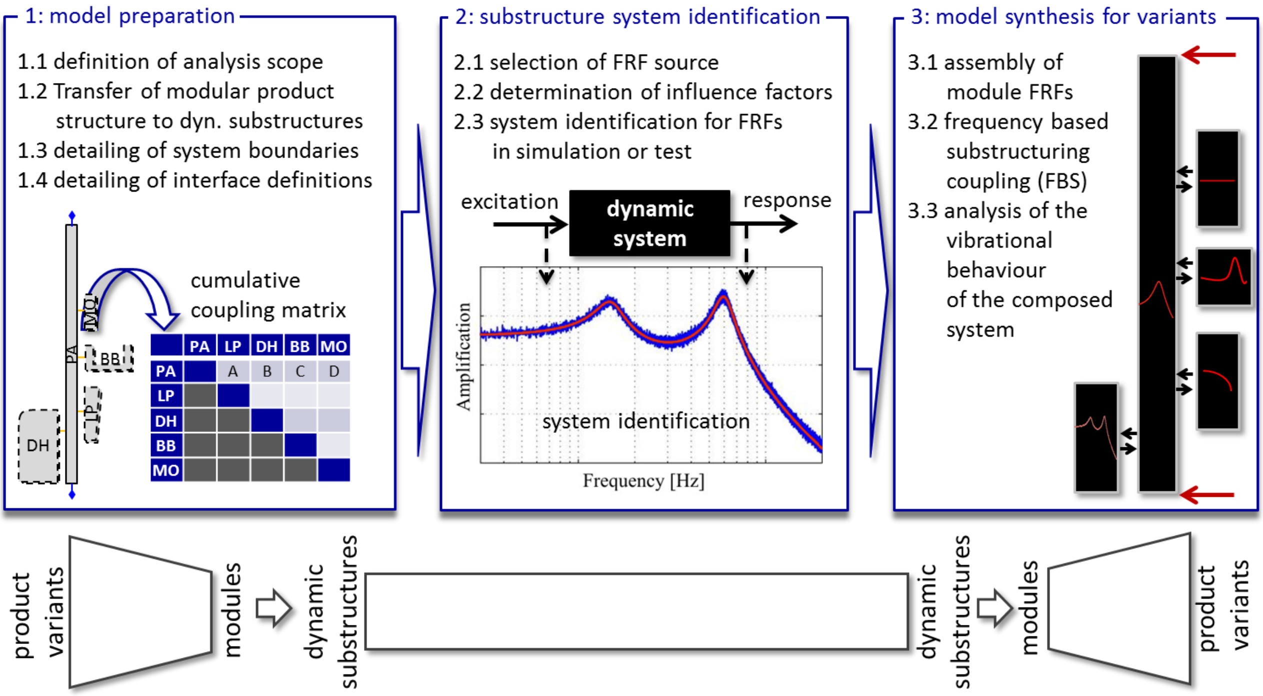

Fig. 3. Proposed approach for the dimensioning of variant cabin interior under dynamic loading

predictions with a large number of variants using only a few modules in a modular product structure.

The approach, as shown schematically in Fig. 2, enables a system synthesis from the dynamic behaviour models of several substructures, which have been identified from a modular product structure, to predict the dynamic behaviour of the fully composed system. This way, a partition panel substructure model can be coupled with substructure models of monitors, baby bassinets or literature pockets to form different product variants, for example. This improves the current situation with decisions being mostly made by engineering judgement and vastly simplified FE calculations due to missing validated model parameters and the variety multiplication factor mentioned before. Now variants can be easily calculated and their vibration behaviour predicted to support the worst case identification and further optimization. This is further supported by coupling only the FRFs of the substructures in a hybrid model, which can consist of FRFs from modal parameter estimation, detailed FE analysis or black box tests.

The main challenge in the primarily methodical model preparation of step 1 is the consistent modelling of substructures, especially regarding its interface definition. The methodical guideline is further elaborated in [12]. Also in step 1, dynamic substructures are mapped to a modular product structure as defined by the integrated PKT approach, for example. If designed also with the later dynamic substructuring application in mind, a suitable modular product structure will enable the generation of many product variants by the pure combination of a few modules. The modules are then mapped to matching dynamic substructures with more details regarding exact interface definitions and subsystem boundaries on the single part level.

Step 2 with the system identification derives Frequency Response Functions (FRFs) at interface nodes and other relevant internal nodes. This model reduction of describing the overall dynamic behaviour by selected functions at the substructure's interfaces can be based on either a black box test, a detailed and validated Finite Element simulation or a modal estimation model if detailed behaviour is yet unknown in early design stages.

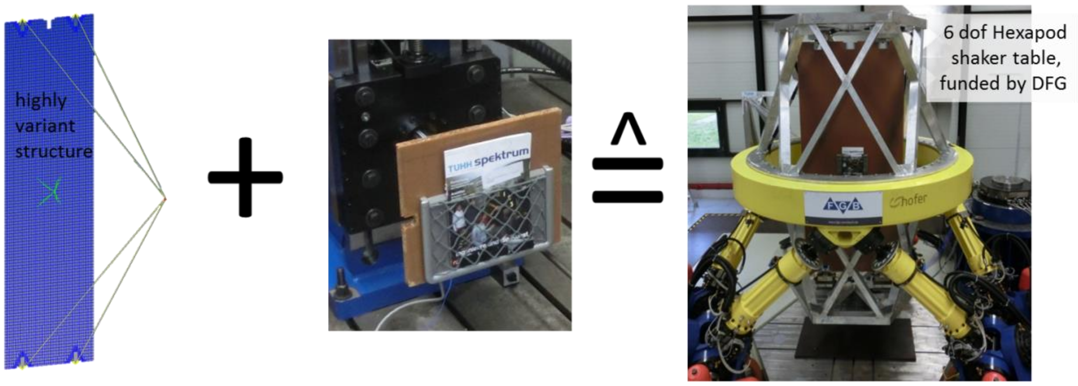

Fig. 4. Demonstrative example of coupling two substructures compared to the benchmark of the assembled system.

The last step 3 then combines the dynamic substructures according to the modular product structure. This model synthesis generates behaviour predictions for each product variant of interest by calculating the behaviour of the coupled Frequency Response Functions representing the dynamic substructures with their behaviour reduced to the relevant behaviour at its interfaces and other points of interest. The variant calculation has to be performed for each variant but takes very little time in comparison to a detailed FE model for every variant, as the calculation is based only on coupling reduced FRFs.

3 The demonstrative example: Partition panel with attached literature pocket and magazine

For the demonstrative example a simplified partition panel of 2340mm height, 1000mm width and 18mm thickness was excited at two bottom attachment points and two upper attachment points. The weight of the partition panel was 7290g including the aluminium attachment beams that were screwed into to sandwich panel inserts.

Attached to the panel middle was a common literature pocket manufactured by the Arthur Krüger GmbH. Different loading variations of the literature pocket with magazines were analysed when subjected to an swept sine of 0.25g constant acceleration level with a frequency ascend of 0.5 octaves/minute between 3 and 25Hz. The resonance frequency of that system is at 10.5Hz for the empty sandwich panel in the first global mode, which is reduced slightly by a literature pocket with one magazine (both together 478.5g extra) attached in the middle to 10.2Hz. The critical damping ratio increases from ca. 1% (equivalent to 2% structural damping) of the empty panel to ca. 4.5% (equivalent to 9% structural damping) in the assembled system. For more information on other test parameter and loading variations, see [5]. The test is used as a benchmark for the substructuring approach and depicted in Fig. 4 on the right.

In this example the partition panel is one module to which different other add-on modules like literature pockets may be added, which generates various product variants. In the example here, the partition panel will be regarded as a corresponding dynamic substructure that will be modelled using a Finite Element model in order to obtain the FRFs describing the dynamic behaviour. The coupling FRFs should be based on a FE model for in-house parts which are subject to in-house variation and optimization. This is, for example, the case for cabin interior manufacturers like Diehl Service Modules GmbH that have a strong selling point on individually designed interior. As with all FE modelling, tests are necessary to produce reliable model parameters and to validate the simulation results.

As the vibrational behaviour of for example literature pockets may be very hard to describe in simplified modal models, the direct use of the frequency response functions from a black box test greatly reduces modelling effort and increases the prediction accuracy as shown in the validation section. Therefore the add-on module of the Literature Pocket (LP) of this simple example is represented by a dynamic substructure with the necessary FRFs derived from a black box test. Black box tests as FRF sources are also recommended for outsourced parts with low variety.

The two constituent substructures will then be coupled together to see how well their predicted behaviour fits the measured benchmark of the assembled structure in a test.

3.1 System identification with cabin interior

Vibration tests of various cabin interior monuments and parts thereof have been performed on different test rigs at the institute PKT ranging from 1-axis hydraulic shakers up to the 6dof Hexapod shaker, funded by the German Research Foundation DFG. The test specimens range from simple sandwich panels with different attachments used in the aircraft over doghouses, partitions up to fully loaded aircraft galleys as seen in Figure 1. The results give valuable simulation parameters for a computational prediction of the vibrational behaviour of cabin interior, as for example interface loads depending on the local acceleration excitation for the further dimensioning. Furthermore, frequency response functions for the relevant substructures were generated which can be used directly in frequency based substructuring simulations without the need to reflect them in a fully detailed simulation model or modal domain models. These black box substructure models [13] were derived from vibration test of monument add-ons like literature pockets, monitors and baby bassinets.

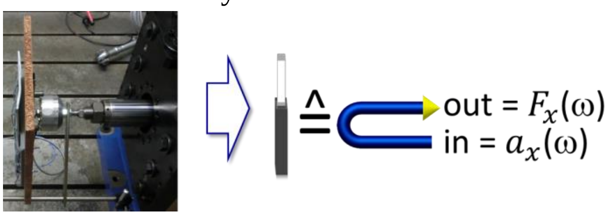

Fig. 5. Dynamic mass FRF from test for literature pocket with one magazine on hydraulic 1d shaker

Fig. 5 depicts the generation a Frequency Response Function of a force feedback depending on an acceleration input for a literature pocket with one magazine from a test.

As these FRFs describe the contribution of the add-ons to the general monument vibration behaviour, they can be used in coupled dynamic substructure modelling to derive interface forces to a global excitation. This will be described in the next section.

The test fixture depicted in Fig. 5 has a significant mass influence on the measured force data. Therefore the following FRFs were subjected to a decoupling using a measured FRF of the empty fixture and an own MATLAB implementation of basic coupling algorithms as shown in [14] and [15]. The frequency domain decoupling results have been compared to simple time-domain decoupling for verification by subtracting the fixture's mass multiplied by the actual acceleration from the actual measured force. This yielded well acceptable results in comparison. The derived FRFs for a literature pocket (LP) with different loading variations (0, 1, 6 magazines) and an equivalent simple mass dummy for LP+1 are shown in Fig. 6 after a linear smoothing over 500 out of 78000 FFT values and a basic frequency domain decoupling of the test fixture.

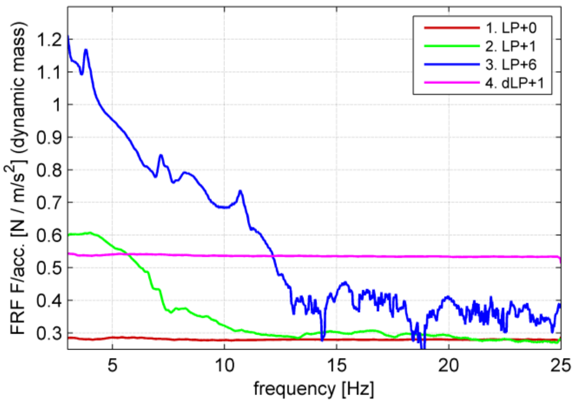

Fig. 6. Dynamic mass Frequency Response Functions of various Literature Pocket (LP) loading variations and weight dummy simplification (dLP)

Due to the non-linearity regarding the constant acceleration excitation level of the FRFs, an anchor point as near as possible to the excitation level in the later coupled system is chosen. In this case, the excitation level was 2g.

The system identification has been performed with an own MATLAB based GUI tool, specifically developed for test data processing, FFT transformation, data analysis and parameter estimation. It has been used in several research and industry projects.

Further substructure identification tests beside the literature pockets included different sandwich panels as used in cabin monuments with various attachments, a doghouse, a baby bassinet with and without a baby dummy, a wall-mounted office monitor and a galley oven dummy as described later in this contribution. Further parameter estimation results of some of the identification tests can be found in [5].

3.2 System synthesis for variants

The shown partition panel with FRFs generated in a validated FE model (depicted in grey in Fig. 7) was coupled with the test-derived FRFs of the literature pocket shown in Fig. 6. The results for the test variation of Literature pocket with 1 magazine (LP+1) is shown in red.

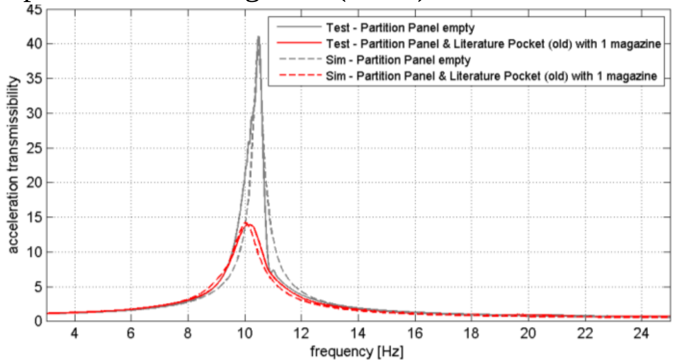

Fig. 7. Global mode acceleration transmissibility for a partition panel with literature pocket and magazine attached

The results represent a good match with the test benchmark of the equivalent assembled system mentioned above. Similar reductions in frequency as well as in acceleration transmissibility from excitation to the maximum of the global mode in the middle of the plate can be obtained. The coupling was performed using the Frequency Based Assembly (FBA) features of MSC Nastran. By using a common FE solver for the frequency based substructuring with coupling algorithms as in [8], a good tool and process integration is achieved from common FE calculations of substructures to frequency based substructuring of the product variants.

The main influence on the result accuracy is seen in the accuracy of the FE model of the supporting panel structure, especially around the frequency of the resonance of the coupled system. Amplifications of small errors in measured FRFs from an inversion of an ill conditioned flexibility matrix with high differences in FRF orders (see [16] and [17]) were not seen as an influence of importance as the constituting FRFs did not show highly different orders and other influences have shown a much larger impact in the given example.

In addition to the system identification support specifically developed at the Institute PKT for the needs presented, a framework support tool for the whole approach is currently under development. This will provide a graphical user interface for consistent substructure definition over the whole product family, support the database handling of substructures and their derived FRFs as well as help with the integration of the needed substructure FRFs for variant generation in Frequency Based Assembly runs.

4 Validation example of a G2 aircraft galley with fully loaded ovens

The demonstrative example of a partition panel has shown the feasibility with a simple setup of FE-derived FRFs for the sandwich panel and test-based FRFs for the literature pocket with one magazine, which is otherwise very hard to capture in a FE model with reasonable modelling effort.

The next example shows excerpts from the validation of the approach using the A320 G2 aircraft galley from Fig. 2. In the full-scale tests of the assembled system, different loading variations have been subjected to swept sine excitation between 3 and 25Hz at various excitation levels between 0.5g and 1.3g. Of particular interest in the galley test is the Y (sideways) direction excitation as this direction shows the first and only global mode with significant vibration behaviour of the galley in the frequency range. For more information see [5] also. The loading configurations tested are “empty”, “fixed items only”, “all loads without trolleys” and “fully loaded”. The “fixed items only” variation is of particular interest, as in this case only the two ovens (filled to max gross weight) and a small beverage maker were mounted in the otherwise empty galley. The relatively low damping of around 10% critical damping ratio in the Y excitation global mode

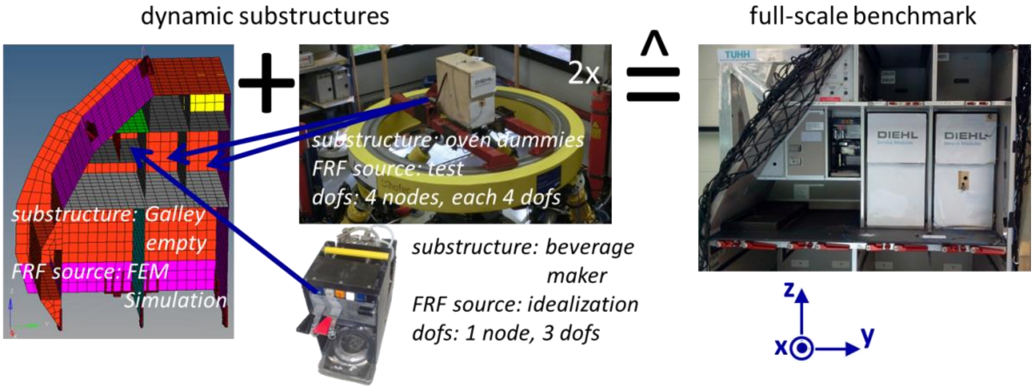

Fig. 8. Validation example of G2 galley in "fixed items only" test variation with benchmark between test of assembled system and coupled hybrid calculation

causes high interface forces in the oven attachments for the “fixed items only” load variation. In the tests conducted, the oven attachments have loosened partially or fully several times. Higher loading variations have shown a significantly less violent vibrational behaviour. The example therefore focusses on the Y excitation of a galley in “fixed items only” loading. It compares the measured benchmark of the assembled system with the coupled calculation of the empty galley structure (FRF from FE simulation), to which two oven dummies with test-based FRFs and a beverage maker with a lumped mass FRF estimation are attached, see Fig. 8.

Following the guidelines presented in [12], a FE model was chosen as the FRF source for the galley structure in order to support further optimization. Since it is difficult to model the non-linear behaviour of the loaded ovens, they were incorporated into the model by using the test derived FRFs directly.

In the benchmark galley test, wooden oven dummies were used instead of the original oven because of fine dust release from the crushed insulation of the original ovens. The oven dummies (empty 23kg each) were filled with 18 0.5l PET water bottles (together 9kg) and 5 packs of 500 sheets of office paper (together 12.5kg) to simulate max gross weight loading.

Using the same system identification procedure as shown in section 3.1, the dynamic mass FRFs in Fig. 9 were derived for the description of the vibration behaviour of the oven dummies. The measured behaviour is non-linear regarding the excitation level which shows in different FRFs depending on the excitation level. Therefore different excitation levels (1g for X and Z; 3g for Y) have been used in the translatory directions to give anchor points for linearization and later linear coupling calculation. These anchor points reflect the actual local excitation in the assembled system in simulation as well as full scale test. The shown X-X, Y-Y and Z-Z as well as the bending contribution Y-Z proved to be a sufficient representation of the relevant contributions, which are used in the coupling simulation of the behaviour of the G2 galley under Y excitation.

The uneven force distribution between the oven attachments may result from a jamming of the rear pin connection on the left side or from a weakened supporting structure on the oven near the rear attachment on the right. However, the measured scenario with higher interface forces on some connections than on the other is the one to be considered for a conservative

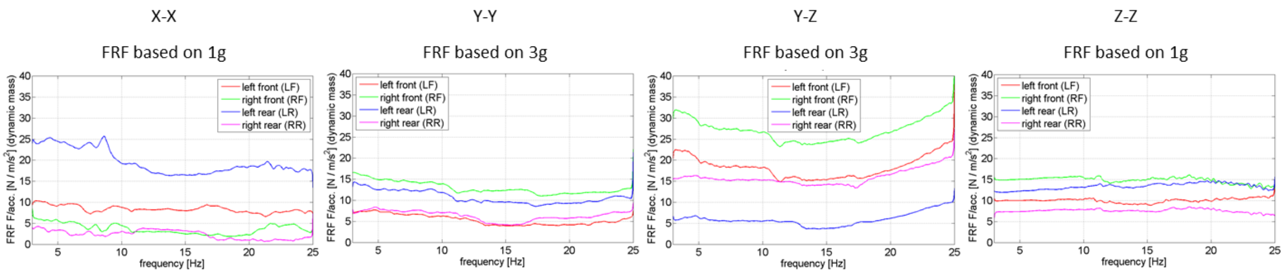

Fig. 9. Relevant dynamic mass FRFs of loaded oven dummy for later coupling

dimensioning instead of an even force distribution over all oven attachments. Furthermore, the substructure identification tests have been performed after the same oven dummies were used in the full galleys tests without making modifications to the ovens, so the measured substructure behaviour is very likely to describe the actual behaviour in the full galley tests.

Interestingly, a decoupling of loading masses over frequency, equivalent to 10kg under a 1g excitation, can be noted, especially when all forces of the X-X plot are summed up. This decoupling does not show so clearly in the Y-Y case.

The MSC Nastran FE model of the empty galley structure, which is described in more detail in [18], is used to derive the FRFs necessary for coupling. The local compliance of the structural oven attachments has been measured on the real galley and incorporated into the galley model using local spring elements.

The beverage maker of 7.25kg mass is incorporated by a lumped mass FRF estimation because its size is significantly smaller than the galley structure with very local influence on the connection area. The irrelevant influence of the small beverage maker connection area has been verified in a small FE study.

As described in section 3.2, the frequency based substructuring calculations are performed with MSC Nastran. In Fig. 10 the interface load sum of the empty galley in test and FE simulation is shown in grey. While the coupled simulation based on FRF represented substructures for the “fixed-only” galley yields well acceptable results (blue) , the typical FE simplification using a lumped mass with stiff connections elements running to the identical local interface elements showed significantly larger deviations (purple).

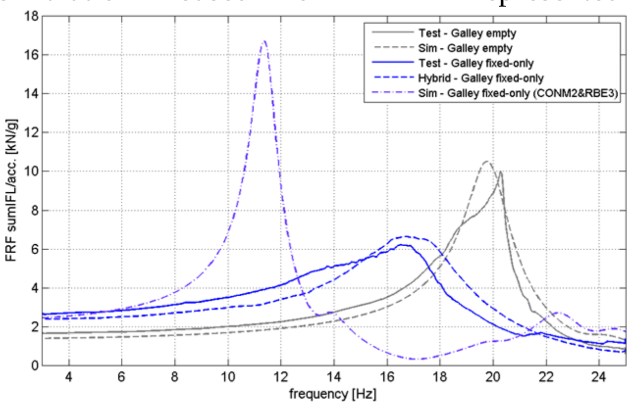

Fig. 10. Results of summed interface loads of the galley from coupled hybrid calculation and classical FE modelling with lumped mass

The coupled system simulations could now be used to predict the vibration behaviour of other variants by combining different modules. A key output of the calculations shown above is the interface force calculation. The interface forces are not only calculated for the galley attachment to aircrafts structure but also for the attachments of the ovens in the galley. These multi-axial forces in the attachments are not measurable in the exact in-vitro situation, but can be predicted using the substructuring approach as presented. For the “fixed items only” configuration shown above, the oven interface forces were calculated to amount up to Y:500N and Z:1000N per attachment under 1g Y (sideways) excitation of the galley in first global mode resonance. This load has to be borne by the local inserts and attachments for as long the excitation remains in the global galley resonance. Also, acceleration transmissibility can easily be calculated based on the FRF models.

5 Conclusions

After an introduction to the challenge of dimensioning variant lightweight structures under stationary dynamic excitation it has been strengthened that a proper lightweight dimensioning needs adequate predictions of the vibrational behaviour of all variants, which is very time-consuming using state-of-the-art detailed FE simulations for each variant with validated model constituents. A methodical approach has been developed that enables the use of a modular product structure with dynamic substructures of Frequency Response Functions (FRFs) representing the modules to support the vibration dimensioning of variant cabin interior. The approach is demonstrated and validated using two examples of a partition panel with a literature pocket plus magazine in it as well as a G2 galley with two ovens and a beverage maker.

With an understanding of variant-specific product development and modularization, an adequately modularized product structure and a consistent use of interface definitions, the approach gives many benefits to the vibration dimension of variant lightweight structures:

It supports the dimensioning by using only a few modules to generate the combinatory variety of many variants. This reduces the modelling effort and thereby helps to save time for more parameter variations in optimization. It further reduces the testing effort by using smaller and simpler test setups of a few modules instead of full scale tests for each variant. By adding the choice between test and simulation derived FRFs in a hybrid approach it helps to increase prediction accuracy and reduces the modelling effort.

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