System analysis and synthesis for the dimensioning of variant lightweight cabin interior

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Nomenclature

A	=	amplitude of oscillation
b	=	damping coefficient
e	=	error (i.e. noise)
F	=	Force
k	=	stiffness coefficient
m	=	centralized mass of a 1 degree of freedom system
N	=	Non-linear restoring force
Q	=	Quality factor
V	=	amplification factor
x	=	deflection, position; input for SISO models
y	=	output (i.e. deflection)
ζ	=	damping ratio
ω	=	(eigen/resonance)frequency in radiant

$A(q), B(q), C(q), D(q), F(q)$ = polynomial functions

I. Introduction and application background

THE dimensioning of aircraft cabin interior has to account for different dynamic loading conditions resulting from varying causes. Beside the loading condition of flight maneuvers and turbulences there are also conditions of emergency landings to account for. In both cases a transient dynamic loading occurs, leading to inertial forces at attachment interfaces. This article, however, focuses on stationary and periodic loads of energy-rich mechanical vibrations below the human acoustics threshold.

These can occur in the sustained engine imbalance conditions as defined in the FAA FAR 25. In this case a turbine blade has failed leading to a rotational imbalance in the turbine. The engine will be switched of, but a windmilling condition keeps up a stationary dynamic excitation being transmitted through wing and fuselage to the cabin interior. Beside this important safety issue to be substantiated, there is also a comfort issue involved when it comes to vibration of interior monuments. Monitors fastened on a partition wall should not vibrate too much during a take-off or landing with slightly uneven runways leading to a random or sweep-like excitation. This issue is clearly comfort driven but may add a good selling point for cabin interior manufacturers.

A. Lightweight design through a more detailed mechanical analysis with better models

As engineering design in the aerospace industry is always trying to minimize weight, structural dynamics become an interesting challenge. If a vibrational problem exists, the first attempt is often to make a design stiffer so that resonance frequencies increase and leave the critical frequency range of a given excitation. Stiffening, however, usually adds mass. With an eigenfrequency for an undamped 1dof system equaling the square root of stiffness over mass, this means that designers have to add a lot more stiffness to counter the inherent mass increase that far that the

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resonance stays out of critical excitation frequencies. Besides a lot of extra mass ruining all weight saving effort, this may add an acoustic problem.
If, however, the necessary parameters for the dynamic behaviors are known, a better prediction of the behavior through simulation offers the opportunity of a proper vibration analysis with corresponding design changes. This way it can be analyzed whether the resonance behavior is critical at all. Maybe damping is high enough so that resonance amplifications stay small. Or the dimensioning could simply account for higher resonance amplifications.

About one year ago, the Hamburg University of Technology's new hexapod testing rig for dynamic testing, funded by the German research foundation (DFG), was taken into service. The servo-hydraulic test rig can be pressured up to 300 bar with a 750l/min flow of constantly cooled hydraulic fluid enabling forces of 500kN vertical and 200kN horizontal. The dynamic excitation range for exact sinusoidal signals lies roughly between 1 and 30Hz. The hexapod design makes it possible to run tests in all independent 6 degrees of freedom using one axes at a time or multiple axes at the same time. A self-tuning iterative control system minimizes crosstalking and provides excitation signals within less than 10% deviation even under heavy nonlinear behavior of test specimens. Cabin interior like a galley can be tested upon the hexapod ring using a stiff sandwich base plate and aluminum backframe as can be seen in Figure 1. Both backframe and baseplate are specifically designed for dynamic analysis tests combining high stiffness and low weight. The galley tests further described in Chapter III-C were undertaken on the hexapod test rig.

Figure 1: Aircraft Galley on the Hexapod test rig.

II. Experimental System Identification for the determination of damping properties

If design optimization regarding dynamic behavior is targeted in dimensioning and substantiation tasks a behavior prediction based on valid simulation models is necessary. When looking at a simple linear 1dof vibrational system and the modeling task in cabin interior dimensioning, different parameters are needed for an adequate model.

$$m\ddot{x} + b\dot{x} + kx = F + N(\dot{x}, x) \qquad (1)$$

F is an external excitation force; N is a non-linear restoring force to be neglected for the further short overview of parameter estimation. While the mass m can often be reasonably well described with densities given for the materials used and stiffness k through parameters from manufacturer data sheets, the damping behavior is often not known. In this example the damping behavior is assumed to be viscous with the coefficient b. Own tests of simple sandwich panels as described under section III-A indicate that this seems to be an adequate analogous model for the damping of the sandwich panels in use with glass fiber layers and aramid paper honeycomb core.
Assuming a low viscous damping in a simple 1dof vibrational system, the damping ratio ζ can be simplified to the following with quality factor Q or amplification V_{max} in the resonance peak.

$$\zeta = \frac{1}{2Q} = \frac{1}{2V_{max}} \qquad (2)$$

The maximum amplification is defined by the maximum response amplitude $A_{resp,max}$ over the excitation amplitude in resonance $A_{exc.,reson.}$.

$$V_{max} = \frac{A_{resp.,max}}{A_{exc.,reson.}} \tag{3}$$

The examples so far were given for conventional linear 1dof. When looking a base excitation in absolute dimensions, the damping ratio estimation from the maximum amplification is calculated slightly differently. Given small damping ratios are present, a rough estimation can be simplified to equation 2 nevertheless.

A standard method for calculating a damping ratio from experimental data is the bandwidth method. Here, a linear 1dof system behavior is assumed. Further vibration modes must be at a sufficiently higher or lower frequency. Additionally this method works only for small damping ratios[3]. The peak amplification at resonance is assumed to be the resonance frequency and the bandwidth is calculated by picking the two half-power points. In these points the amplification is equal to the maximum amplification divided by $\sqrt{2}$ as indicated by the name “half-power”. This corresponds to a lowering by 3dB. Further reading and practical examples can be found in Ref. 1,2,3. With the frequencies ω_1 and ω_2 at the two half-power-points the damping ratio can be calculated.

$$\zeta = \frac{1}{2Q} = \frac{\omega_2 - \omega_1}{\omega_0} = \frac{\omega_2 - \omega_1}{\omega_2 + \omega_1} \tag{4}$$

Parameter estimation from curve fitting with parametric models

Measurement data is often distorted with noise and different modes cannot be separated from each other as necessary for the approaches shown before. To counter these problems, curve fitting algorithms can also be used for parameter estimation. A curve fitting will use an underlying model with parameters it will tune to the best possible fit on a frequency response function in the frequency domain. For this step the measurement data in time domain will have to be transferred into the frequency domain by using a Fourier Transformation[1,2]. If polynomial models are used the needed modal parameters like frequency and modal damping can be derived from the polynomial coefficients. The basis is a simple linear dynamic system with an error input *e* besides the regular input *x* and the regular output *y*.

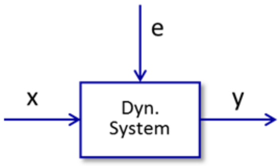


Figure 2: Linear dynamic system with error input

In order to derive a transfer function from the fitted model later a basic equation with the polynomials A, B, C, D and F is used[8].

$$A(q)y(t) = \frac{B(q)}{F(q)}x(t) + \frac{C(q)}{D(q)}e(t) \tag{5}$$

Due to time constraints normally not all polynomials will be used in a curve fitting process. Often it is sufficient to use only some of the A, B, C, D and F, depending on the noise and system behavior. For the work presented in this paper three different polynomial models from the MATLAB System Identification Toolbox™ have been used for parameter estimation: Output Error, ARX and ARMAX.

From the description in Ref. 8 the following block diagram overview of the models used and their use of the polynomial functions can be derived.

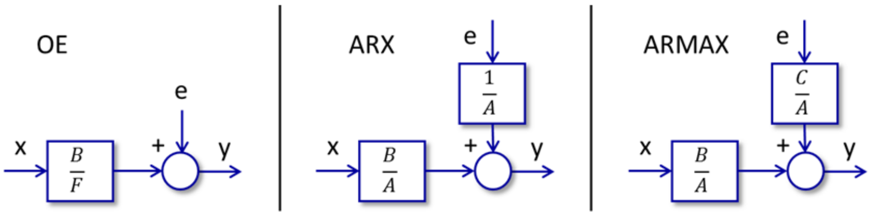


Figure 3: Different models used for the polynomial curve fitting

The output Error (OE) model is the simplest with taking into account an error not coupled to the actual system behavior, but not using a polynomial error model also. The ARX uses a common denominator for error and system description which means they share the same poles. An ARMAX model offers an even more flexible error model with an extra polynomial.

Besides choosing an appropriate model – as good as necessary, but as fast as possible in calculation – it is of high importance to find an appropriate model order. The higher the order is, the more pole pairs and hence estimated modes will be found within the frequency range of interest. However, if too many pole pairs have been used, the curve fitter will fit very local effects on the curve. As the polynomial coefficients are used for parameter estimation, the local curve changes of normal measurement data will result in implausible mode predictions in these points on the curve. Therefore the right model and the right model order have to be chosen for a good curve fit. The parameters derived this way in this paper have been taken from curve fits which were deemed sufficiently close to the measurement data. The evaluation has been based on the aspects *reproduction of the peak*, *the slopes* and *a small least square error of the curve fitting algorithm*.

III. Parameter Estimation for System analysis

In order to model vibrational behavior of cabin interior monuments certain parameters have to be identified. These should account for the material behavior on a macroscopic level as well as for the behavior of combined structures. As stiffness and mass parameters are at least known to some extent, mainly damping parameters are to be identified. A short overview to parameter estimation can be found in Ref. 4 while methods of identifying the dynamical behavior of dynamic systems in general are best described in Ref. 5, 6 and 7. For the described sandwich structures under stationary-periodical forced excitation reliable values of the damping behavior are often missing for dimensioning. This means that excitation for the possibly non-linear damping properties should resemble near-reality conditions. Therefore excitation loads should be stationary vibrational and applied at the actual attachments. Impact excitation often does not bring enough energy in the system to bridge small gaps in composed structures like a galley with inserts so that all masses are coupled into the motion to determine realistic damping properties. The system boundaries are set at these attachments with an ideally stiff surrounding in order to determine the monument or test specimen behavior only and be able to compare the results to computational models.

A. Example: Parameter Estimation for a single sandwich panel

The analysis started with tests of simple sandwich panels from which the combined structures are composed. Here several test sequences with parameter variations have been conducted. The excitation included hammer impact tests as well as sine sweeps through a frequency range on a hydraulic shaker with a slide. In both cases the panel was fastened at the bottom. The hammer impacted at the top end if used. The vibrational excitation was realized through a slide on which the panel was mounted vertically leading to a base point excitation as seen in Figure 4.

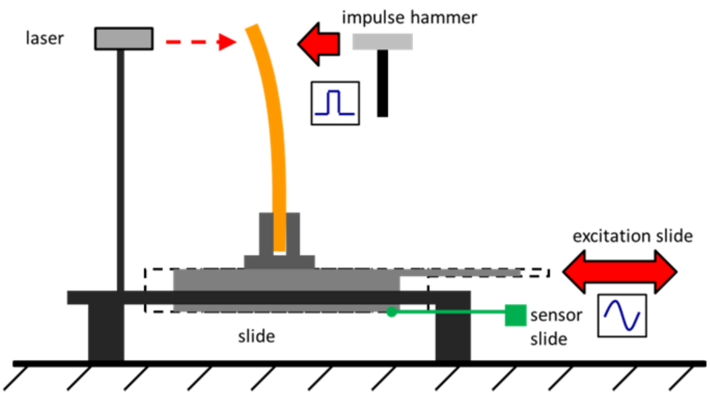


Figure 4: Test slide for panel Parameter Estimation

In test sequence 1 a single sandwich panel with variations of the layers and core orientation was analyzed. The sandwich buildup and the abbreviations and conventions used are in the following.

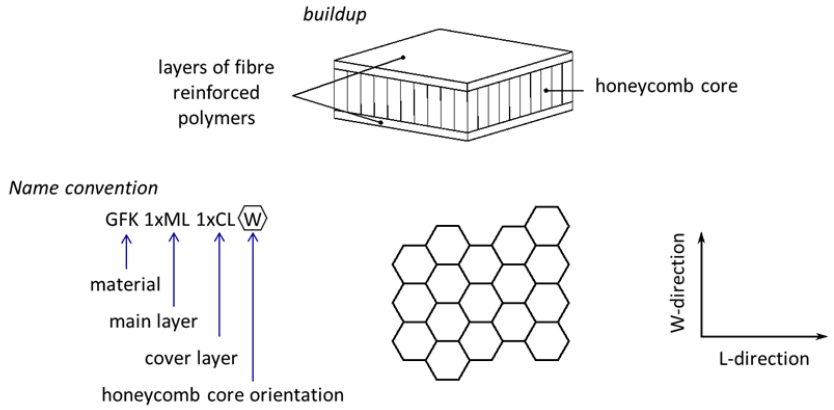


Figure 5: Sandwich buildup and description syntax

The test specimens used for test sequence 1 are displayed in the following Figure 6 together with the results of the damping parameter estimation under impact testing (free vibration with base fixation) and sine sweep excitation (forced vibration through a clamp at the base). Because the excitation range of the shaker was below the resonance frequency of the short and stiff panels, extra masses of 589.8 g (1ML) resp. 858.3 g (2ML) have been added.

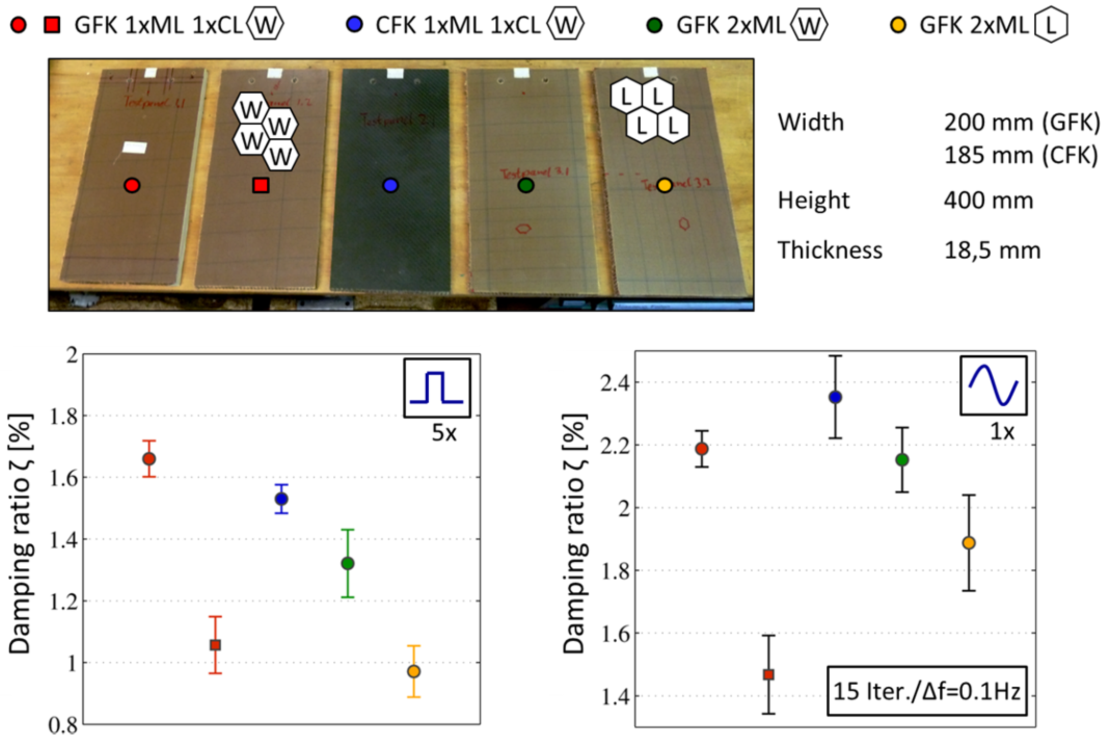


Figure 6: Overview of damping ratios from panel test sequence 1

The impact test was taken 5 times for each panel. The sine sweep ran through a frequency range from 2 to 20 Hz with an incremental step of 0.1Hz and 15 sine iteration cycles per frequency step while exciting with a constant maximum displacement amplitude of 0.2mm.

The results with the standard deviations generally show damping parameters as this can be expected from a rigid sandwich panel in literature[9,10]. Also different results for free and forced vibration were found early as described in Ref. 9. The first test panel to the left (marked with the red dot) was predamaged from overexcitation while adjusting the test rig and test parameters. It is assumed that local damages at the bending line just above the clamp dissipate further energy leading to a slightly higher damping ratio than the exact copy of the panel marked with the red square.

Based on the results of test sequence 1 a second test sequence was run with the aims to optimize the testing procedure and to study further parameter variations, in particular concerning the effect of different attachments of the panels. The length of the panels was increased so that resonance frequencies stayed well within the test rig excitation range without adding extra masses as in test sequence 1.

The variations focus on the base fixation and are briefly visualized in Figure 7. They include

- the machine clamp from test sequence 1,
- a fastening with tubes put into tubular panel inserts like at an upper attachment of a partition
- an aircraft cabin seat track attachment
- two sandwich panels glued together and screwed onto the slide

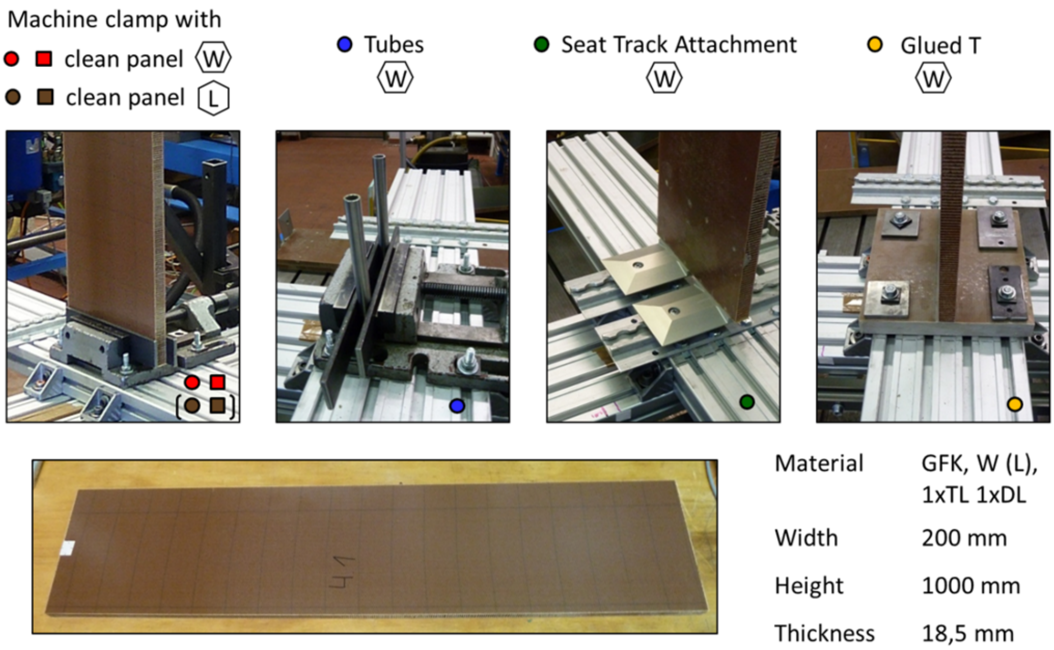


Figure 7: Further test sequence with varying panel fixations

The results in the following figure show that the machine clamp contributes most to the overall damping while the most direct contact with the glued T shows the least damping. The tubular attachment resembling an upper attachment used for cabin partitions showed a highly nonlinear behavior resulting from tilting and gap contacts in the tubular inserts with varying contact surfaces.

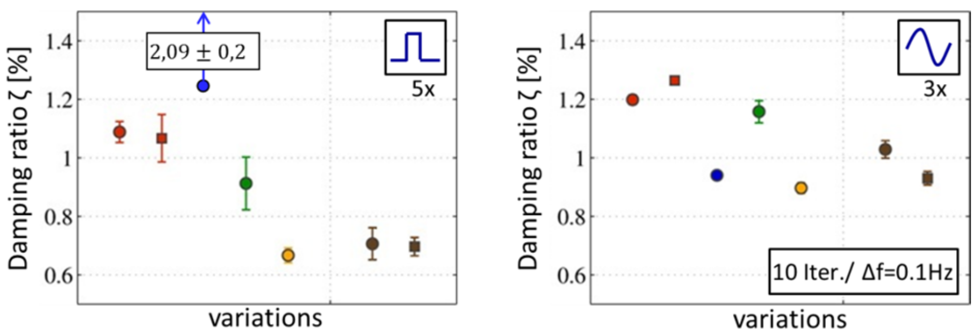


Figure 8: Overview of damping ratios from panel tests with varying fixations

B. Example: A Doghouse in Parameter Estimation

Furthermore the test and analysis of a cabin doghouse – a storage compartment behind the last row of seats in a cabin section – was conducted. This is a more complex structure consisting of more interfaces, leading to a higher damping ratio which was also estimated from the test data. The particular test specimen is attached only through two seat track fastenings at the bottom side.

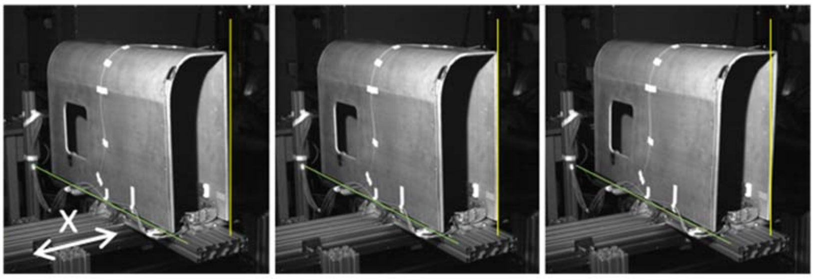


Figure 9: Movement of an empty doghouse without drawer on a 1-axis excitation slide

The doghouse was excited in airplane x-direction through its normal basepoint seat track attachments. The setup was mounted on a test slide similar to Figure 4. The doghouse has only one global vibration mode within the frequency range of up to 30Hz which can be seen in Figure 9. Other vibrational modes have higher frequency values and cannot be excited with the servo-hydraulic test slide. Therefore the analysis focusses on this first global mode in x-direction with the system's output set to the upper right corner and the inputs set to the test slide. Simulations, high speed video analysis and prior impact testing have also shown that this is the only global vibration mode within the test range. The excitation spectrum was a stepped sine function from 2-30Hz with a frequency step of 0.1Hz and 15 sine iteration cycles per frequency step at a constant maximum deflection amplitude of 0.2mm or 0.25mm.

The test specimen was a typical doghouse from aircraft cabin interior supplier Diehl Service Modules, known as Mühlenberg Interiors before. The tests were conducted for the configurations, also shown at the bottom in Figure 10:

- Empty doghouse without drawer
- Doghouse with empty drawer
- Filled drawer in the doghouse with 1kg of rags
- Filled drawer in the doghouse with 2kg of rags
- Filled drawer in the doghouse with 3kg of rags

Filling the drawer with rags was done to achieve a very high damping in order to have a wide range of measurement data regarding damping values. The doghouse drawer was capable to hold up to 12kg of cargo weight. However it was completely filled when 3kg of lose rags had been put into the drawer.

The results in Figure 10 show quite a low damping when simply exciting the empty doghouse but damping increased significantly when the still empty drawer was put in. It increased even more when the rags were put in as it would be expected of such a setup. The lose rags dissipate a lot of the vibrational energy in the system.

Inserting the drawer being fixed to the doghouse frame through sliding bearings with a small but certain amount of free play also introduces energy dissipation. The resonance frequency lay around 18Hz with a slight decrease with higher damping rations and more mass being put into the system.

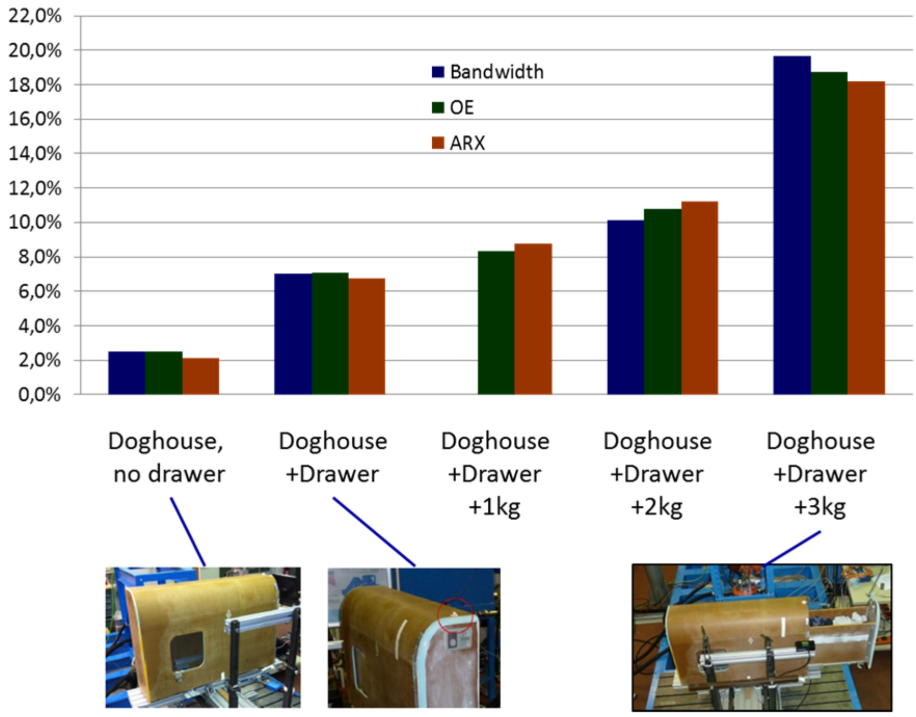


Figure 10: Results for global mode, excitation in x-direction

The curve fitting algorithms used for parameter estimation were based on either an Output-Error or an ARX from the MATLAB System Identification Toolbox™. Because the curve fitting is based on the values in time domain with the later results being transformed into frequency domain the curve fitting was automatically applied for the full frequency range of the measurement data. Therefore higher order models were used so enough complex pole pairs lay within the relevant frequency range of 5 to 25Hz. The parameter estimations were conducted by choosing the best curve fit for every test run. Therefore the models used for estimation were of different order depending on which order showed the best fit as described under section II.

C. Example: Parameter Estimation for the global dynamic behavior of a full aircraft galley

When taking the newly built hexapod into service, a first test sequence using an A320 single aisle aircraft galley (position G2) from our cooperation partner Diehl Service Modules could be run. The galley was attached to the yellow base plate in Figure 11 at 6 hardpoints along with one Flutter point. Additionally 2 tierods were connecting the upper attachments of the galley to the aluminum backframe. For the analysis so far the global damping behavior was evaluated from resonance search sine sweeps between 3 and 25Hz for different loading conditions between an empty and a fully loaded galley as can be seen in Figure 11.

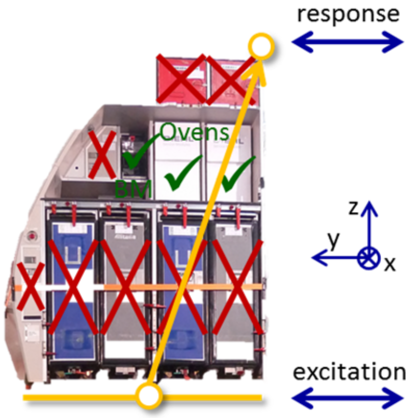
Figure 11: An A320 Aircraft galley G2 on the Hexapod test rig

This highly composed sandwich structure has many contact interfaces in itself, it has loose and fixed compartment fillings and shows differently damped overall behavior. Tests were conducted for different loading conditions:

- Full galley
- Full upper galley without trolleys
- Fixed items only (ovens and build-in compartments)
- Empty galley

The galley was instrumented with 3D force transducers at the base plate, single axis forces transducers in the tierods and 6 3D accelerometers on the structure. Additionally 5 accelerometers were used for measuring the excitation levels at various points on the fixture backframe, the base plate and the hexapod ring to be able to judge that tilting and fixture deflection do not become relevant.

The results were analyzed using several parameter identification methods. In all cases the aim was to determine one global damping parameter for the single dominant global mode. As resonance searches have shown, only one dominant global mode exists in the excitation range of 3 to 25Hz. This mode shows when exciting the galley sideways in airplane y-direction. This sideways excitation is depicted in Figure 12 over a schematic diagram of the galley.
Some aspects of the analysis conducted are exemplarily described in the following for the test configuration "fixed only". This refers to a configuration where only those items remain in the test galley, which are usually permanently fixed in there. These are the two ovens and the beverage maker (BM) in this particular test galley as shown schematically in Figure 12. The ovens were filled to maximum load capacity. Trolleys and standard unit containers were taken out. The filling consisted partly of 0.5l plastic lemonade bottles filled with tap water and partly of A4 office paper in packs of 500 sheets. The load distribution for the test configuration is also given in the table of Figure 12. The weight of the sole galley structure was 135kg with some extra weight from attachments, tubes, filters, etc. The test configuration "fixed-only" weight with the filled ovens and beverage maker amounts to 231kg.



	Oven1	Oven2	BM
gross weight [kg]	44,5	44,5	7,25
empty weight [kg]	23,0	23,0	7,25
water bottles [kg]	9,0	9,0	0,0
paper sheet packs [kg]	12,5	12,5	0,0

Figure 12: Demonstration of the configuration "fixed-only" under excitation in y-direction with the global mode transfer function

An analysis with amplification over frequency plots describing the amplification between a sideways base point excitation and the sideways acceleration at the top describing the first global mode was conducted for different excitation levels. The resulting waterfall diagram of the fixed only configuration (Figure 13) shows a non-linear behavior over different excitation levels. While the resonance frequency stays more or less the same, the amplification sharply declines for higher accelerations indicating a high damping at the frequencies. It is assumed that the load replacement filling of the compartments and equipment in the galley have a higher amount of mass being coupled into the motion when excitation levels get higher. As the filling of the compartments is not fully fixed, it can slide within certain gaps. The amount of coupled mass will therefore increase with more mass in the consequent sliding motion. This will probably contribute to more energy being dissipated.
Similar observations were made for other configurations. Even when looking at the empty galley a non-linear behavior was observed which is probably caused by the split-line interfaces with gap contact areas spread over larger contact areas.

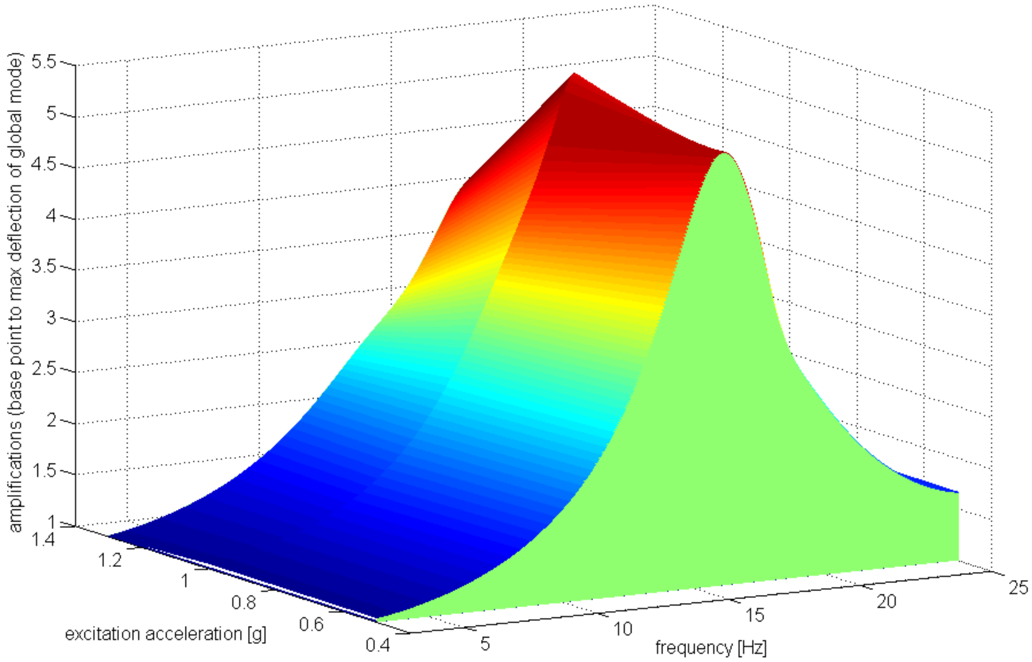


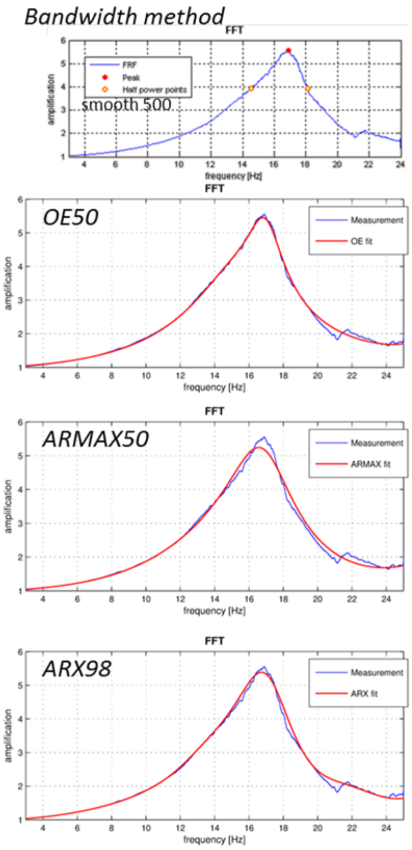
Figure 13: Waterfall diagram of the global mode amplification transfer function from basepoint excitation to maximum deflection point (upper corner) for test configuration fixed-only.

An excerpt from the current research with parameter estimations is shown in Figure 14 where different analysis methods have been used in order to identify parameters for a simplified vibrational model consisting of one global mode for the configuration “fixed-only” under a constant maximum acceleration sine sweep of 1g level in y-axis. With the non-linearity of excitation levels in Figure 13, the process has to be repeated for other excitation levels also unless one is only interested in the 1g level excitation which will be discussed in a little more detail:

The non-symmetric peaks can indicate a slight possible non-linearity or a second mode with higher damping nearby. The polynomial model curve fitters try to take this into account with a second overlapping mode at a lower frequency. However if the two overlapping modes are put to close together they cannot be viewed separately anymore for a simplified 1-mode-replacement model. The results are discussed also in Figure 14.

The identified models (whether they contain just one mode or some more) can be used in Finite-Element-Analysis (FEA) to predict the response of calculated excitation behavior which might for example result from a blade loss windmilling condition. For this a FE-model with mass and stiffness values of datasheets as well as the damping value from the analysis described before is subjected to a stationary periodic excitation. The results can be analyzed for maximum deflections and maximum interface forces. Also interface forces within the structure in the global resonance can be roughly estimated if the model is accurate enough. To make sure of this, however, other transfer paths have to be measured also.

To give practical example: Given that the oven interface forces depending on the different excitation load cases present might be of interest. For this the model should be validated at least with transfer functions from the excitation to the global mode as well as to the connections of the ovens. But with a model validated as accurately enough one could predict the oven interface forces in the galley for various load cases.



results of modal parameter estimation — comments

Bandwidth method

Mode	fn (Hz)	D [%]
1	16,9	10,5

-non-symmetric peak: probably slightly non-linear
-a good first guess

OE50

Mode	fn (Hz)	D [%]
1	15,5	18,9
2	17,0	6,9

-very good fit with two pole pairs close together
-unfortunately too close together for 1 damping parameter of a 1-mass-vibration system
-probably not correct to use just 6,9% as one overall value

ARMAX50

Mode	fn (Hz)	D [%]
1	11,5	92,2
2	14,5	39,2
3	**16,9**	**12,2**

-in spite of the most sophisticated noise model used not a perfect fit of the peak region with a slight overestimation of the damping (fit is below real peak)
-slopes fitted well
-considering the slight overestimation of damping because of the peak fit: damping probably slightly below 12%

ARX98

Mode	fn (Hz)	D [%]
1	1,2	-
2	13,5	15,7
3	**17,2**	**10,9**
4	21,8	17,1

-good fit with one overall damping value for the peak: 11%
-fits very well with estimations from bandwidth method

Figure 14: damping estimation for simplified vibration model consisting of one dominant global mode

Analyses of the further test configuration measurements often show that the damping increases with higher acceleration levels. This could be explained by more sliding masses being coupled into the movements with higher deflections and more gaps being opened and closed at interfaces within the structures.
Analysis has also shown that an empty galley shows higher interface forces at its hardpoint and tierod connections than a fully filled one in its respective resonance. The extra mass added by the inserts can multiply the weight of the single structure by factor 6 or more, but damping increases so much with the extra articles put in the galley that the overall interface forces in resonance become less than for an empty galley.

The measurement data taken from the described test runs allow further detailed analysis in various other aspects also. For example Frequency Response Functions (FRF) from interface forces to interface accelerations also give room for further analysis. The development of accurate dynamic simulation models based on Finite-Element-Analysis (FEA) based on these measurements is another focus in current research at the institute PKT. To make industrial application successful they have to be as simple as possible but as precise as required. Finding a feasible trade-off in the model definition for the application described is an interesting challenge under research.

However, as Chapter IV describes, the high product variety makes industrial application difficult. Questions arise whether the results for the test of this one galley can simply be transferred to a different variant with a different structure and other configurations with ovens, beverages makers, trolleys etc.

IV. The next step: Model synthesis in dynamic substructuring for variant specific dimensioning

Determining damping parameters for dominant global vibration modes of such complex structures as galleys is extremely helpful for the dimensioning of the product. However, the results of the dynamics tests cannot simply be “transferred” to another galley product variant. A different product variant means that parts of the product architecture have been exchanged or modified (for further reading please see Ref. 11). In order to achieve a high

external product variety (required by the customer) with a low internal variety (beneficial for the producing company) a modular product structure can be used. The variant problem is however of high importance in the cabin interior of today's airlines and their strong demands towards individual cabin design in order to distinguish them from each other.
The estimation of worst-case configuration variants may often still be possible when looking at emergency landing conditions which may be tested with quasi-static forces corresponding to the inertial loads. In static load cases the identified worst-cases can be tested and a broad set of less-critical configuration variants can be substantiated with this single test as well. But in structural dynamics the worst-case identification over a product family with different variants is not that easy as the subsystems are dynamically coupled and their behavior intermingled. This is of particular importance if the dimensioning aims for a high performance design with an optimization towards minimum weight. Lightweight issues of modular product families with the cabin interior background have been discussed in Ref. 12.

In order to tackle the variant problem in dimensioning under dynamic loading without performing tests for nearly every variant, a substructuring approach as presented at Ref. 13 and 14 can help. For a modular product architecture the dynamic behavior of single modules can be described in adequate models using frequency response functions. These abstract descriptions of a module's behavior at its interfaces regarding the correlation of forces and deflections (or a derivative like acceleration) can be then coupled with other modules' dynamic behavior models. The full product structure can be described in this way regarding its dynamic behavior as a coupled system. The underlying black box modeling is described in Ref. 5. Further reading on general substructuring methods can be found in Ref. 15. Approaches targeting the presented problems are further described in Ref. 8 and 16.
The parameter identification presented in the previous chapters with the examples from small panels over a doghouse to a complex composed structure is intended to set the basis for the development of a substructuring approach that is focused on the needs of the dimensioning of variant cabin interior monuments under dynamic loads. The approach supports fast modeling and calculation of combinatory variants for dimensioning under dynamic loads.
A first view on the approach is presented in the following Figure 15 using a cabin partition as demonstrational example.

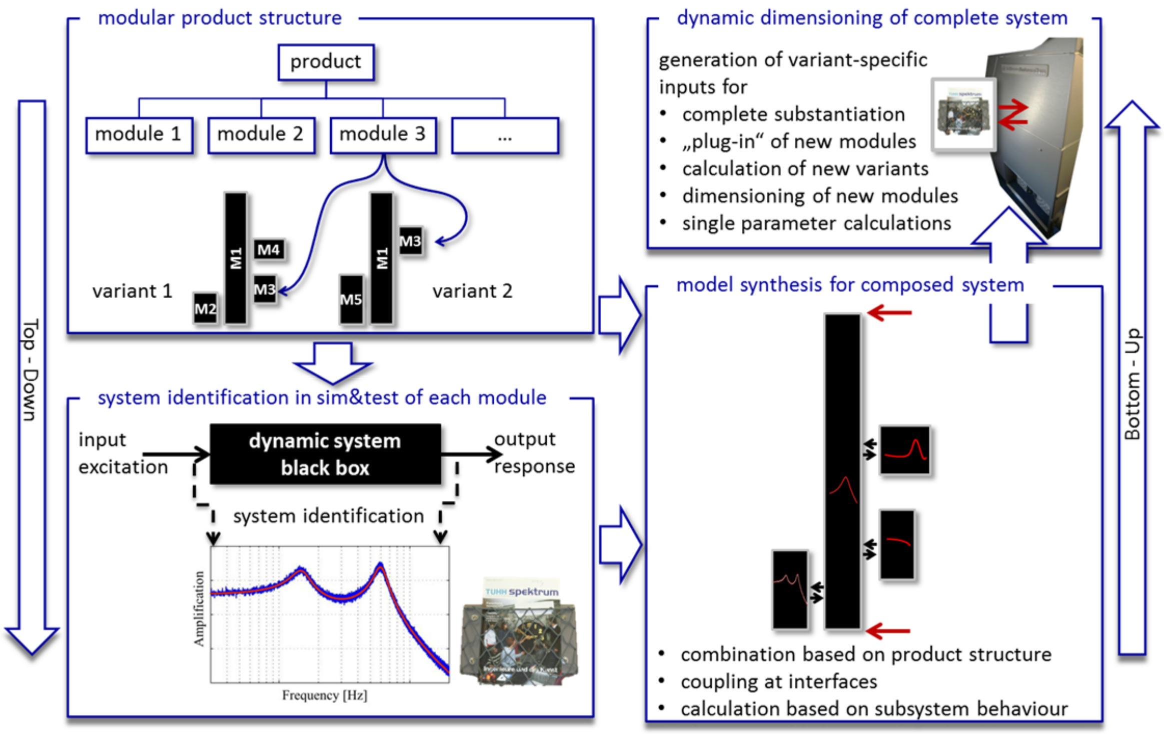


Figure 15: Approach supporting modeling and calculation of combinatory variants for dimensioning under dynamic loads

Based on modular product structures – a general segmentation generated using the integrated PKT approach (see Ref. 11) – the substructuring will be conducted. If necessary, the subsystem/module boundaries have to be further detailed to support later mechanical calculation. To give an example: It has to be decided if an interface exhibiting a relevant dynamic behavior should be considered to be part of one or the other substructure or if it should become another substructure itself.
Following this step the dynamic behavior of each module has to be identified. The approach uses a frequency based coupling on the basis of the frequency response functions. By doing so it can combine frequency response models directly from testing with models from prior Finite-Element-Analysis (FEA). Three different sources for a module's dynamic model are taken into account as shown in Figure 16.

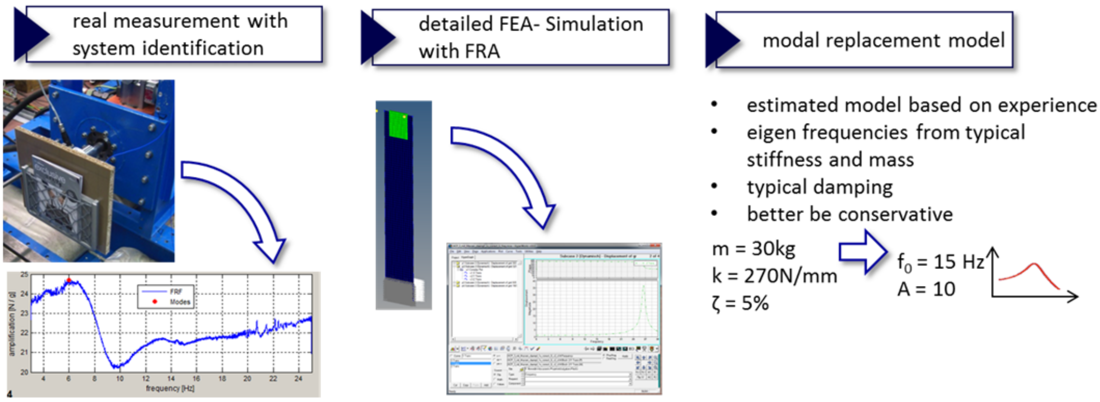


Figure 16: Different sources for a frequency response function model

- The dynamic behavior can be derived from the acquired test data by transferring it from time domain into frequency domain. Frequency response functions like the receptance or the inertance can be used because they describe a substructure's dynamic behavior with a relation between force and deflection (or a derivative of it). A good overview on FRF generation from measurements is given in Ref. 2. The test data may have to be smoothed and fixture influences may have to be removed as described in Ref. 17, 18 and 19.
- A dynamic model of a module may also be derived from a physical model of finite elements in a Frequency Response Analysis (FRA). For this approach detailed information on stiffness, damping and mass parameters have to be present. Especially the damping information makes tests as described in section III inevitable.
- In the lack of better information a simplified model may be generated using simple 1-dof-vibration systems with estimated stiffness, damping and mass parameters. But even in this approach the damping is likely to be determined by a test only.

The next step, following the overview in Figure 15, is the coupling of the modules according to the product structure defined earlier. Using a coupling algorithm in the frequency domain as described in Ref. 15 the overall system behavior can be calculated. The process of calculating the overall system behavior at certain dofs from the substructure's frequency response function is often referred to as Frequency Based Substructuring (FBS) or Frequency Based Assembly (FBA). It works by coupling the interface degrees of freedom of one module with the interface degrees of freedom of another. It differs from the coupling in the modal domain, known as Component Mode Synthesis (CMS), because it couples the frequency response functions from measurement directly without the simplification to a modal system. The process circumvents the problems of making the simplification to a modal model accurately which can prove to be difficult or even impossible for substructures exhibiting a slightly non-linear behavior. By leaving out the simplification step from frequency (non-parametric) to modal model (parametric model) it minimizes possible inaccuracy. Further information on coupling in physical, modal and frequency domain is found in Ref. 15, while an extensive description on the CMS approach is found in Ref. 20. The original description of the FBS or FBA approach is given in Ref. 21, a more recent presentation can be found in Ref. 22 and an extensive description with the example of a wind turbine is given in Ref. 16.

The dimensioning of variant lightweight structures under dynamic loads can be supported by the prediction of the composed system's behavior if

- A new variant is to be dimensioned and the overall dynamic behavior has to be estimated. The extent of application can vary between a dynamic substantiation and the identification of a worst-case-combination for further analysis or substantiation tests.
- A single new module is to be developed. A new design's impact on the behavior of the composed system can be evaluated. Usually this will be done with a white box FEA model of the module. Or the impact of the surrounding of a module to be developed can be derived. This could be for example maximum interface forces which the new module has to withstand.

With the necessary mechanics mostly in place (see especially Ref. 15) a remaining challenge lies in the consistent modeling of the interacting substructures over a full product family. Interface definitions with the coupled degrees of freedom have to be chosen carefully when modeling one module of the product structure so that the respective model of this module is capable of interacting with all the other models of modules it is supposed to interact with following the product structure of a whole product family.
As there is hardly any literature to be found which addresses methodically the substructure model definition in a variant product structure that has to be dimensioned under dynamic loads, further research will be conducted in this field. A modeling guideline which improves model consistency of FRF substructure definitions over a combinatorial variant product family is currently under development.

V. Summary

After a general introduction on dynamic loads, in particular stationary-periodic loads, for the dimensioning of aircraft interior, the basic principles of parameter estimation methods in structural dynamics were explained.
Based on the shortly described estimation methods, the main part of the contribution focusses on the system identification for different cabin interior monuments, including detailed analysis of the contribution of the sandwich panel material and some of the attachments used on the structures. The tests and some aspects of the analysis were presented for sandwich panels, a doghouse stowage and a single aisle galley.
As a short-term outlook, an approach was presented that will help to speed up dimensioning of different monuments based on a modular product structure using dynamic substructuring methods for coupling the separately generated models for each module. The dynamic behavior of each module can be derived directly from test data (without modal synthesis), detailed finite element-models or modal replacement models from scratch. Either way will resort to the system identification testing performed as described before. Further research will focus on the consistent modeling of combinatorial variant product families to further develop the presented approach.

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