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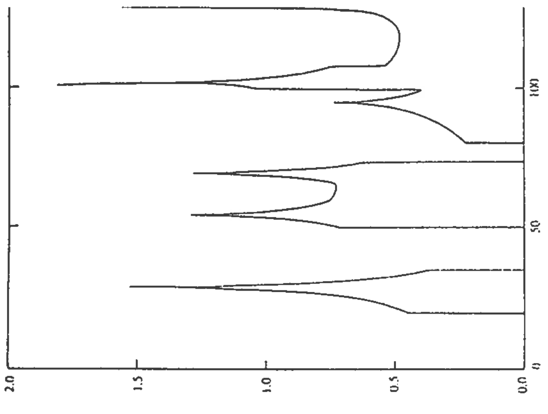

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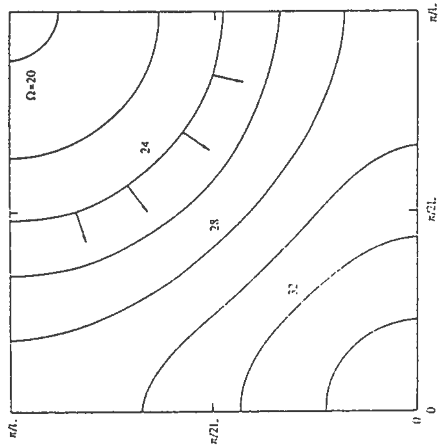

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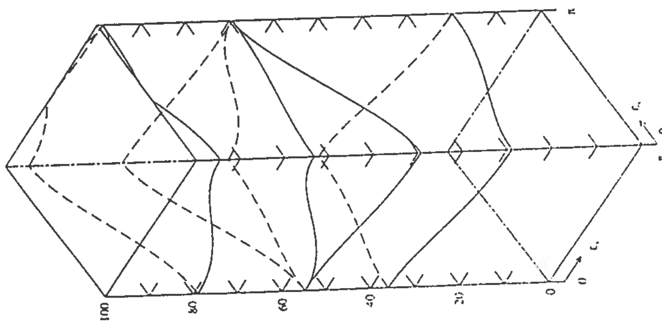

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APPLICATION OF PRINCIPAL FIELD DECOMPOSITION TO AIRCRAFT INTERIOR NOISE ANALYSIS

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ABSTRACT

Experimental modelling of trimmed aircraft cabins in the acoustic frequency ranges is not straightforward due to the high modal density and the relatively high damping of many of the system modes. Hence traditional modal models often fail to yield meaningful results.

An alternative modelling approach is based on principal field analysis, where a singular value decomposition of the multi-reference FRF matrix is performed at each frequency. This approach has been extended to multi-frequency analysis where frequency bands are analysed in a global sense. These techniques have been applied to the analysis of a twin-propeller aircraft in the context of the Brite/Euram project "ASANCA".

1. INTRODUCTION

The classical approach to experimentally model the vibroacoustic system behaviour of a mechanical structure, consists of the identification of the modal system model parameters. The system behaviour is divided into a set of individual resonance phenomena, each characterised by a resonance frequency damping ratio and mode shape. The experimental data set to derive this model from consists of a set of frequency response functions between a limited set of reference degrees of freedom and all response degrees of freedom.

For structural (vibration) responses, this technique is widely spread. Its application in aircraft dynamics is however mainly limited to the low frequency range of the global wing/fuselage/tail modes, which are of importance for flutter studies and structural integrity. In the higher frequency range, which is of relevance for the acoustic behaviour, only a limited amount of results is documented. They nearly all deal with mock-ups or green aircraft.

With the very high damping in the trimmed fuselage, the resonant modes become closely coupled, making it very difficult to obtain a good modal model, and hence to determine the relative contributions of the various modes to the dynamic operational response.

For acoustic responses, the situation is even more complex. It has been shown that a similar modal model formulation can be used to describe the sound pressure in a discretised cavity, where the sources are point monopoles of known volume velocity [1,2]. However, for the case of systems like trimmed aircraft cabins, the damping of the modes is high, resulting in highly overlapping modes with complex mode shapes. Furthermore, measured frequency response functions usually show even at resonance, a propagating acoustic field instead of a standing wave pattern. Again, the trim, and the resulting non-uniform damping properties of the cavity walls, are the probable cause to these phenomena.

The conclusion from all this, is that an experimental vibroacoustic modal analysis of a trimmed aircraft in the acoustically relevant frequency regions is far from trivial.

2. PRINCIPAL FIELD ANALYSIS

Since, notwithstanding above mentioned problems, one still wants to obtain relevant system information in terms of dominant mode shapes, a complementary, non-parametric technique was developed, referred to as principal field analysis.

With this approach, a singular value decomposition of the multi-reference FRF matrix is performed at each frequency. Plotting the singular values as a function of frequency gives a first global idea of the dominant frequencies and the number of dominant modes at each frequency [3].

$$[H(f)] = [U(f)] \lceil \Sigma(f) \rfloor [V(f)]^H \tag{1}$$

The equivalent of the system "eigenmodes" is then found by the corresponding left singular vectors. The importance of each left singular vector $\{U_i\}$ is given by the corresponding singular value. They can be considered as "principal" field shapes, denoting the system's response to excitation in "virtual" or "principal" references. The latter are formed by unit linear combinations (right singular matrix $[V(f)]$) of the original references.

However, the exact value of the individual resonance frequencies is of less importance than the actual modal field shape.

Therefore, a more advanced technique, based on multi-frequency singular value analysis was investigated. In this technique, a band of frequencies is analysed in a global sense, also using all FRFs for all excitations at the same time. For each frequency band, these principal field shapes are then calculated in descending order of importance (corresponding to the singular values).

$$[[H(f_1)] \cdots [H(f_N)]] = [U] \lceil \Sigma \rfloor [V]^H \tag{2}$$

If the matrix of Eq. (2) is expanded by the FRF matrices for the corresponding negative frequencies, the left singular vectors can be proven to be real.

For each frequency in the band of interest, the following relation holds:

$$[H(f)] = [U] [H'(f)] \tag{3}$$

where

$[H(f)]$	: (N_{rsp}, N_{ref})	-	FRF matrix
$[U]$	: (N_{rsp}, N_{prc})	-	matrix of normalised principal field shape vectors
$[H'(f)]$	: (N_{prc}, N_{ref})	-	matrix
N_{prc}	: the number of dominant singular values		

The left singular vectors denote a set of orthogonal, (real) vectors, representing in fact a set of dominant shapes, which may be close to the system modes. Generally, only the shape, corresponding to the first singular value, may show resemblance with the dominant system modes. Mostly, the other calculated "principal" shapes are just unknown linear combinations of the other system modes.

The singular value decomposition reduces the number of responses to a number ($N_{prc} \leq N_{rsp}$) of "principal responses", of which the FRFs related to the given references are given by $[H'(f)]$ (principal FRFs). Evaluating this matrix, it is possible to estimate the relative contribution of each excitation, in function of frequency, to each singular value and its corresponding shape. Principal FRFs that reveal only one dominant peak in the concerned frequency band, may correspond to real system modes. As mentioned above, this mostly occurs only for the first singular value. The reason is that the system modes are generally not geometrically orthogonal, whereas the principal shapes are orthogonal per definition.

Although the interpretation of the other obtained results is clearly to be done with care, especially when modal interpretations are given to principal field shapes, the main advantage lies with the fact that no parametric model needs to be fitted to the data.

3. APPLICATION

3.1 The test case

The discussed procedures have been applied to the analysis of the vibro-acoustic behaviour of a twin-propeller aircraft, the Dornier 228.

The analysis was performed in the context of the Brite/Euram project "ASANCA" [4], in which a demonstrator active control system for the reduction of periodic interior aircraft noise was developed.

In addition to a detailed mapping of the in-flight acoustic field and of the secondary sound fields for a large number of actuator positions, which are both discussed elsewhere [5], specific ground tests have also been executed to derive intrinsic system information which should render it possible to explain the observed in-flight behaviour.

The excitation was performed simultaneously by four loudspeakers, two longitudinal and two tilted (lateral) ones. Acoustic responses were measured throughout the cabin in 17 sections, each being a grid of 25 microphones. Vibration responses were evaluated at 80 locations on fuselage frames near the propellor plane and on some trim panels. The summed FRFs for the structural and acoustical responses are shown in Fig. 1a and 1b.

From these figures, it can already be concluded that both the modal density and the corresponding damping ratios are high.

These data have then been processed further by modal analysis and by multi-frequency principal field analysis.

3.2 Modal analysis.

Modal analysis was performed by applying a least squares complex exponential curve fitting procedure to the measured FRFs. Since the most dominant FRF response is found around 80 Hz (especially for loudspeakers 1 and 4), the region around this frequency was analysed first.

The modeshape at 80 Hz is shown in Fig. 2. In the upper part, the field shape obtained from the analysis of loudspeaker 1 is given, in the lower part that for loudspeaker 4. The pressure value is here for clarity represented as an axial line segment; the view is a top view along the cabin (cabin front is left).

The corresponding mode shapes clearly indicate a longitudinal modal behaviour. However, the nodal lines are different for the two loudspeakers, which indicates that the corresponding acoustic field is a combination of a standing and of a travelling longitudinal wave.

Further analysis of the FRFs with respect to loudspeakers 1 and 4 reveals "mode" shapes that are in fact mainly longitudinal travelling waves. Transversal modes only show up very locally near the excitation location and decay rapidly along the aircraft cabin.

The transversal modes however show up more clearly when analysisng the FRFs with respect to the tilted loudspeakers 2 and 3. For example, at 110 Hz, a mode is found which is characterised by a strong top-down motion in front of the cabin(Fig. 3, which is a side view along the cabin; the pressure is represented as the motion of a wire frame model). This top-down mode, though, decays quickly until only longitudinal travelling waves remain.

At higher frequencies, pure standing waves with clear nodal planes can be observed. In Fig. 4, a side view of the mode shapes at 132 and 166 Hz is shown. They can probably be related to the analytical modes (4,0,1) and (5,0,1).

Also, the structural responses caused by the same acoustical excitation, are analysed using the modal method. The thus obtained coupled modal field shapes are all similar at the first 4 maxima in the frequency response functions, and related to the fundamental structural bending of the fuselage frames. Hence, it is clear that these modes couple well with the cavity. Fig. 5 shows the mode shape at 96 Hz.

3.3 Principal field analysis

For the multi-frequency singular value analysis, two frequency bands were considered:

- 60-100 Hz
- 100-140 Hz.

In these bands, all data from all loudspeakers and all frequencies are combined, and from this global set, the most dominant field shapes are calculated.

This approach's advantage is that only the really dominant field shapes (which should be very closely related to the actual modes) will "survive" this global analysis. Also, the calculation process is straightforward, and unhampered by curve fitting method characteristics or operational uncertainties. All frequency specific information within each band is lost however. The only clue to a frequency value of the modes, is provided by the maxima in the "principal" FRFs.

3.3.1. Frequency band 60 - 100 Hz.

The principal response FRFs for the first singular value indicated a resonance at 80 Hz, dominantly excited by loudspeakers 1 and 4. The corresponding real field shape is shown in Fig. 6a (wire frame model, limited number of frames) and plainly reveals the existence of nodal lines.

The second principal FRF reveals a less clear dominant frequency. Consequently, the corresponding field shape looks less like a mode. In the first frame, some top-down behaviour is already superimposed on the global longitudinal behaviour (Fig. 6b). Generally, this field shows a longitudinal shape, complementary to the first one (different node lines).

From the fifth field shape (Fig. 6c), other frames also reveal top-down and side-side modes, where top-down seems to dominate the frames in front of the cabin and side-side the frames at the rear. The less important the singular value, the less clear the dominant frequencies become. This means that the obtained "principal" shapes are formed by (unknown) linear combinations of the system modes.

For the structural vibrations, the results are similar. The first two principal fields are shown in Fig. 7a and 7b.

For most of the principal FRFs, the absence of a pronounced peak frequency reveals the bad correlation between the principal shapes and the structure modes. For many principal deflection shapes, the different frames are behaving in a different way. The resemblance with simulated modes is poor in general, except for the first one.

3.3.2. Frequency band 100 - 140 Hz.

The first two principal field shapes are shown in Fig. 8a and 8b.

The first singular value refers to a peak frequency of 110 Hz, mainly caused by loudspeaker 1. The corresponding field shape is less clear and seems still influenced by the 80 Hz mode, as could be anticipated from the singular value plots which are considerably less pronounced.

From the second singular value, top-down and side-side modes, or linear combinations, clearly become more important, again vertical is favoured at front and lateral at the rear of the cabin.

3.4 Discussion

Using modal analysis methods, the dominantly longitudinal modes in the lower frequency range are identified. Yet, in the majority of cases, these modes do not show the typical features of standing waves inasmuch as no unique nodal planes can be identified and the locations of the pressure maxima are in continuous motion

away from the loudspeaker. The explanation of this strange behaviour may be that there is much acoustic absorption present in the cabin, distributed non-uniformally, resulting in high damping of the modes and the complex mode shapes.

In addition to the longitudinal wave propagation, some transversal modes could also be identified using the tilted loudspeakers as one single input. The frequencies of these modes are in satisfactory agreement with simple calculations of the cabin as a rectangular cavity. The various modes are, however, strongly coupled and the available curve fitting methods were found insufficient to separate them completely.

The multifrequency singular value decomposition on the other hand offers a global and straightforward "principal field" decomposition tool, but the resemblance with system modes is generally rather poor.

Under the applied loudspeaker configuration the acoustic behaviour is dominated by longitudinal phenomena; front and rear seats undergo higher amplitudes in general. Vertical and lateral modes are generally superimposed to them, especially for frequencies above 100 Hz. The occurence of top-down shapes is dominant in front of the cabin, whereas side-side shapes are favoured in the back.

4. CONCLUSION

A multi-frequency singular value decomposition technique was developed to analyse frequency response functions of vibro-acoustic systems with high modal overlap. The technique yields a set of orthogonal "principal" field shapes which can be compared to (but which are in general not equal to) the system mode shapes. In those cases where the estimation of an experimental modal system base is not possible due to the system complexity, this approach offers a feasible alternative.

Similar to mode shapes, the resulting orthogonal fields can then be applied to further system synthesis, sensitivity analysis and forced response calculation techniques required to the vibroacoustic behaviour optimisation. They also form a suited vector base for the decomposition of operating (in-flight) fields, allowing a better insight in the system related in-flight behaviour.

The method was applied to the analysis of a fully trimmed aircraft as a complement to a traditional vibrational and acoustical modal analysis.

It was shown that the multi-frequency singular value decomposition offers a global and straightforward "principal shape" decomposition tool, but the resemblance with system modes is generally rather poor.

ACKNOWLEDGMENT

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The discussed test results have been obtained in cooperation with Metravib RDS and Dornier Luftfahrt GmbH.

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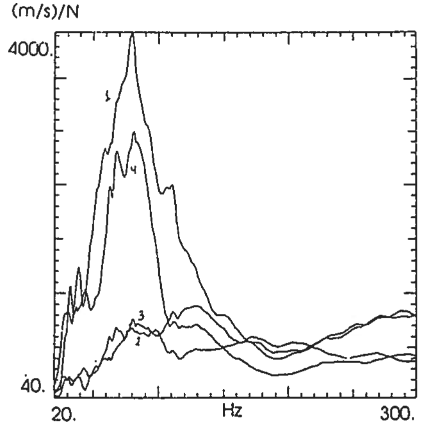


Fig. 1a. Summed FRFs (ac.)

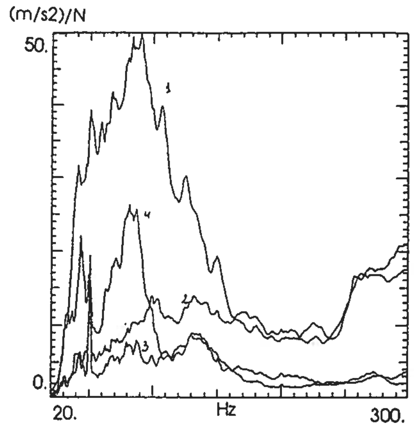


Fig. 1b. Summed FRFs (struct.)

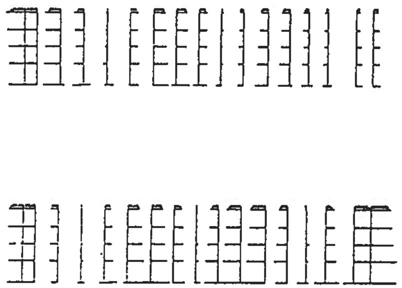


Fig. 2. 80 Hz mode (LS 1, 4)

Fig. 3. 110 Hz mode (LS 2)

Fig. 4a. 132 Hz mode (LS 2)

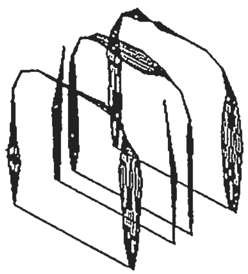
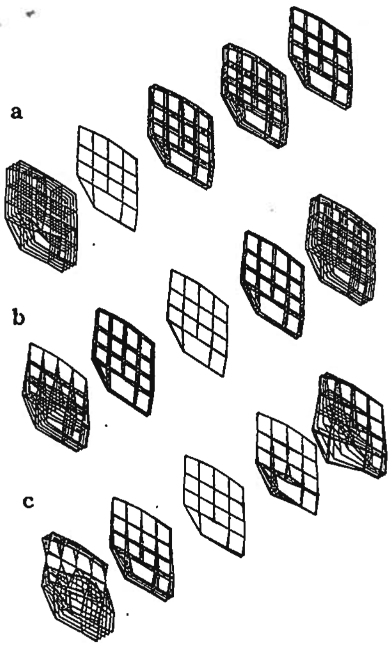


Fig. 5. 96 Hz mode shape

Fig. 4b. 166 Hz mode (LS 2)



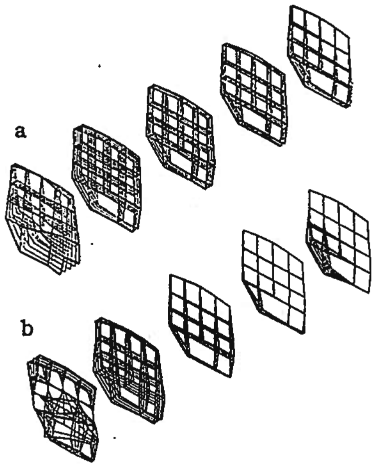


Fig. 8. Principal fields (100-140 Hz)

Fig. 6. Principal fields (60-100 Hz)

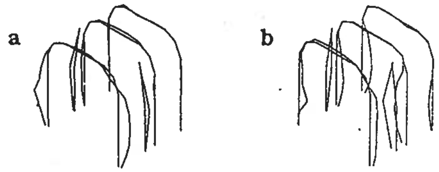


Fig. 7. Principal fields (60-100 Hz)

The Prediction of The Effects of Stiffness & Damping on Noise Reduction of Small Enclosure

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ABSTRACT

This paper reports the application of a theoretical analysis for predicting insertion loss and higher order mode shape relation between source plate and test plate of close-fitting enclosure. Experimental work has been carried out for various plate designs with several common structural materials using special test rig. The theory and the experiments state two important points; 1) test plate is forced to vibrate in the same mode as the source's; 2) the resonance frequencies occurs when combined impedance of those of test plate and source plate equals zero. Furthermore, comparisons of various test plates point out that wood is the best material for low frequency noise reduction due to high stiffness-mass ratio. Air damping also improves noise reduction and can be achieved effectively by extra treatment such as double layer structure and attachment of stiffener.

1. INTRODUCTION

A common constraint imposed on the enclosure design is that its volume is limited for the purpose of saving-space. Enclosure of this kind, described as "close-fitting", is considered as acoustic-structure interrelated system, and includes some properties different from those of large enclosure. Previous works by Fahy [1] and Oldham [2] do not include on the effect of the higher cavity mode on the close-fitting enclosure system. In this paper, theoretical analysis, which include the consideration of the effect, is given to predict the insertion loss of the close-fitting