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Vibroacoustic Analysis of Trimmed Aircraft through Modal and Principal Field Modelling

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Abstract

Effective interior noise reduction measures require an in-depth understanding of the operational noise and vibration fields, as well as of the intrinsic system characteristics. The former requires detailed mapping of the in-flight sound and vibration responses, whereas the latter requires the proper modelling of the vibro-acoustic system behaviour.

Such experimental modelling 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.

In the case of principal field analysis, a singular value decomposition of the multi-reference FRF matrix can be performed at each frequency. Plotting the singular values as function of frequency gives a more global idea of the dominant frequencies and the number of dominant modes at each frequency.

When mode shapes are more important than exact individual resonance frequencies, multi-frequency analysis techniques can be used which analyse frequency bands 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

When performing a study of aircraft interior acoustics, it is important to relate the observed in-flight vibration and acoustics response to the intrinsic system behaviour of fuselage and cabin cavity.

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. Examples can be found in [1],[2], where also the relation to operational (in-flight) data analysis is discussed. In fully trimmed aircraft however, the damping becomes very large. In [2], this high damping is explained by the friction at the interface between frames and trim panels, including the thermal insulation.

With the very high damping in the trimmed fuselage, the resonance 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 response, the situation is even more complex. It is not straightforward that the same model formulation also holds for acoustic variables. Also, 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 modal analysis of a trimmed aircraft in the acoustically relevant frequency regions is far from trivial. In the sections below, a number of the most critical elements in this procedure are further analysed, and an alternative analysis method based on principal field decomposition is presented.

The discussed principles will be illustrated by means of test results obtained on a twin-engine propeller aircraft.

2. Experimental Acoustic Modal Analysis

An important consideration in the application of experimental modal analysis techniques for acoustic problems is the validity of the modal model formulation and the selection of proper input/output variables. In addition, equipment considerations need to be made to perform the required frequency response function measurements correctly.

2.1 Basic formulation

Consider a three-dimensional closed acoustic system with rigid or finite impedance but non-vibrating boundaries. The governing equation of the system can be written in the form [3]:

$$\nabla^2(\vec{r},t)-\frac{1}{c^2}\ddot{p}(\vec{r},t) = -\rho\dot{q}\delta(\vec{r}-\vec{r}_0) \tag{1}$$

Assuming now that a number of point monopoles of known volume velocity are placed in the cavity and the sound pressure across the volume is sampled at an appropriate number of points, it can be shown that the continuous wave equation can then be substituted by its discrete equivalent:

$$[A]\{\ddot{p}\} + [B]\{\dot{p}\} + [C]\{p\} = \{\dot{q}\} \tag{2}$$

where

p = sound pressure

$\dot{q}$ = volume veloicity

Taking the Laplace-transform and assuming zero initial conditions we get:

$$[s^2[A]+s[B]+[C]]\ \{p(s)\} = s\{q(s)\} \tag{3}$$

As usual in structural dynamics, the inverse of the matrix term can be substituted by the frequency response matrix $H(s)$:

$$p(s) = [H(s)]\ s\{q(s)\} \tag{4}$$

The frequency response matrix can in turn be expressed as a partial fraction expansion of modal parameters:

$$[H(s)] = \sum_{i=1}^{n} \frac{Q_r\{\Psi\}_i\{\Psi\}_i^T}{s-\lambda_r} + \frac{Q_r^*\{\Psi\}_i^*\{\Psi\}_i^{*T}}{s-\lambda_r^*} \tag{5}$$

Now substituting s by $j\omega$ and using Eq. (4) it becomes obvious, that the modal parameters of the system can be gained from the FRF measurements where the sound pressures across the volume are referenced to the volume velocities of the sources. In acoustic terms, the transfer impedances of the field have to be measured:

$$Z_{re}(\omega) = \frac{p(\omega)}{q(\omega)} = j\omega \sum_{i=1}^{n}\left[\frac{(r_{re})_i}{j\omega-\lambda_i} + \frac{(r_{re})_i^*}{j\omega-\lambda_i^*}\right] \tag{6}$$

The expressions (3) to (6) are in complete analogy to the ones being used in structural dynamics. Therefore, the usual structural methods and software packages can in principle be used without modification.

Further considerations regarding the calculation of forced acoustic fields and the equivalence between structural and acoustical values can be found in [4].

The main consideration for establishing the required experimental database is that the volume velocity is used as input variable and the acoustic pressure as output variable.

In the application section, it will be shown that the actual identification of the modal parameters involved is however far from trivial.

The issue is further complicated by the fact that both subsystems - structural (fuselage) and acoustical (cavity) - are not behaving independently but, on the contrary, are rather tightly coupled. To analyse this coupling, and to assess which acoustic and structural modes are coupled to each other, frequency response functions need to be measured and analysed between structural excitation and acoustic response, or vice versa.

2.2 Equipment requirements

In principle, no correct experimental acoustic modal analysis can be conducted without using a well-controlled volume velocity source. Unfortunately, such actuators are commercially not available. A few experimental systems have been reported on in literature, out of which the converted acoustic driver method seems and actually has been found to be the most practicable.

Imagine an electrodynamic loudspeaker which is provided with a closed, sealed housing behind the diaphragm. The most obvious realisation could be to use a horn driver. Unfortunately, these loudspeakers are generally designed for high frequency sound reproduction and sometimes cannot radiate sufficient acoustic power in the frequency range relevant for acoustic modal analysis applications. A good quality medium-range loudspeaker with closed housing or, in case of even lower frequencies, a closed box loudspeaker unit may be helpful.

An important element in this approach is the measurement of an appropriate reference signal.

If the back cavity's dimensions are considerably smaller than the wavelength, one can assume that the pressure is constant everywhere in the cavity. Then, we have an acoustic capacity excited by the backward radiation of the diaphragm, causing a pressure in the cavity which can be calculated by means of:

$$p = q_{back}\frac{\rho c^2}{j\omega V} \tag{7}$$

By measuring this pressure a good reference signal can be derived.

This method has one single practical drawback, namely that the pressure in the back cavity is very often too high, amenable to measurement. Another alternative is to measure the displacement of the diaphragm, implicitly assuming of course that the whole diaphragm moves with the same amplitude and phase. Substituting $q = Av$ in Eq. (7), it is easy to show that the pressure in the back cavity is proportional to the displacement. In any case, a proper calibration of the source strength and reference signals is essential.

If the analyst is interested in the modal frequencies and the mode shapes of the system only, and a correct modal model is of no importance, the volume velocity source can be substituted by a simple loudspeaker. Then the reference signal can be taken directly from the input clamps of the speaker. It should be remarked that the reference signal cannot be derived using a microphone in the close vicinity of the source. The sound pressure measured in any point of the volume is a response rather than an excitation signal. If one aims at detecting the modal frequencies only, it can happen that even strong normal modes will be missed if the microphone reaches a local maximum. One has to be aware of the fact that in this case, the loudspeaker itself becomes an element of the system to be investigated, and any possible resonances of the exciter appear in the analysis as supplementary modes. These false modes are not easy to distinguish from the actual modes of the system in case of strong damping. The same holds for the microphones. The solution is that before the actual test, the frequency response functions of the actuators and sensors have to be checked carefully.

As far as the sensors for the measurement of the responses are concerned, the difficulties are much smaller, but care should be taken here too. In principle the acoustic field has to be sampled using microphone positions which are closer than $\lambda/6$ (a general rule of thumb used for discrete acoustic methods). In case of larger dimensions, the number of microphone positions can run high. If one does not use an appropriate large number of parallel channels, the total measurement time can be too long, enabling the loudspeaker to heat up the air in the volume. Thereby the sound speed in air can change which in turn can cause the variation of the modal frequencies. Even a slight frequency shift can cause serious problems during post-processing of the data. The problem can be overcome using microphone arrays. Additional equipment considerations can be found in [4].

3. Principal Field Analysis

Since, as already mentioned, the identification of a proper modal model is in many cases very difficult or even impossible, and 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 [5].

$$[H(f)] \quad = \quad [U(f)] \lceil \Sigma(f) \rfloor [V(f)]^H \tag{8}$$

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 of the original (right singular matrix $[V(f)]$) 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).

$$[[H(f_1)] \cdots [H(f_N)]] \quad = \quad [U] \lceil \Sigma \rfloor [V]^H \tag{9}$$

If the matrix of Eq. (9) 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:

$$[H(f)] \quad = \quad [U] \, [H'(f)] \tag{10}$$

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.

4. Application

4.1 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"[6], in which a demonstrator active control system for the reduction of periodic interior aircraft noise was developed.

To realize this goal, an extensive flight and ground test program was set up and executed on four selected aircraft: the Dornier 228, the Alenia ATR 42, the Saab-Scania 340 and the Fokker 100. The results presented here are the ones for the Dornier 228.

The objectives of the tests were the following:

- to understand the spectral and spatial distribution of the in-flight sound field, as well as of selected vibration responses in important locations.
- to obtain primary field data to be used in the design of the optimal control system configuration and in the simulation of the control system performance.
- to obtain system information in the form of field responses caused by secondary sources placed in possible control system actuator locations.
- to obtain system information in the form of dominant system characteristics (mode shapes or principal field shapes).

The latter two objectives are important for the derivation of an optimal control system in terms of minimal numbers of optimally located actuators.

4.2 Flight tests

The in-flight tests consisted of a survey of the acoustic field of the cabin, as well as of a large number of selected vibration responses, for the first few engine tones in particular. The acoustic field was mapped throughout the complete cabin in 17 sections, with 25 microphones per section.

The vibration responses were measured at typically 80 important locations, e.g. on the fuselage, on a number of frames near the propeller plane, as well as on the trim panels. More details on the flight test procedure are discussed in [7].

The resulting operational vibration shapes, divided into left and right propeller contributions, are shown for some frames near the propeller plane in Fig. 1 (first blade-pass frequency). The corresponding acoustical field is shown in Fig. 2 (wire frame model).

It can be clearly noticed that the two propellers excite the cabin cavity in a different way (top-down behaviour versus side-side behaviour).

4.3 System identification tests

In addition to the prediction of secondary sound fields, which is further discussed in [7], specific ground tests have also been executed for selected aircraft, to derive intrinsic system information which should render it possible to explain the observed in-flight behaviour. In this case, simultaneous excitation was applied at multiple loudspeaker and structural locations in order to excite the complete cabin properly. For the Dornier 228, discussed here, responses were measured at the full set of the in-flight response locations.

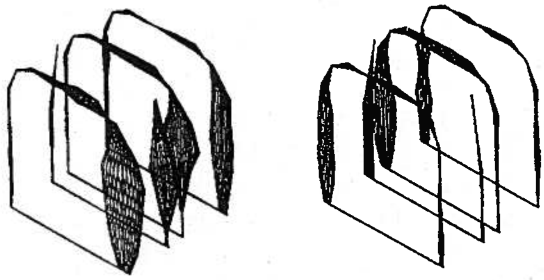

Fig. 1. In-flight vibration shapes

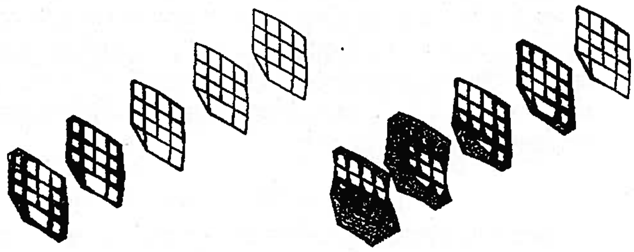

Fig. 2. In-flight acoustic fields

In this case, the excitation was performed simultaneously by four loudspeakers, two longitudinal and two tilted (lateral) ones. The summed FRFs for the structural and acoustical responses are shown in Fig. 3a and 3b.

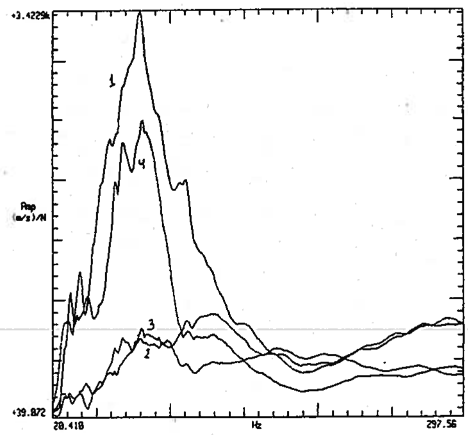

Fig. 3a. Summed FRFs (ac.)

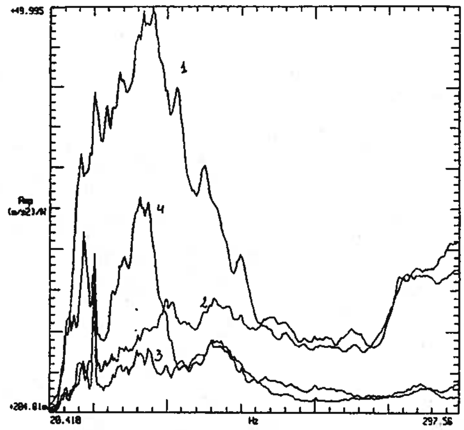

Fig. 3b. Summed FRFs (struct.)

The corresponding singular value analysis plots are shown in Fig. 3c and 3d.

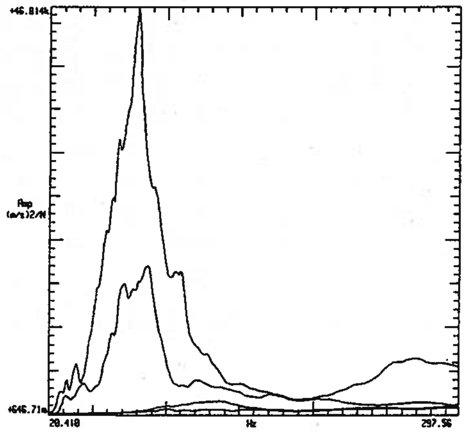

Fig. 3c. Singular values (ac.)

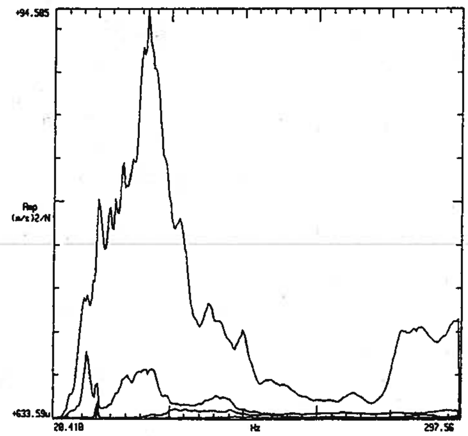

Fig. 3d. Singular values (struct.)

From these figures, it is concluded that:

- the modal density is very high.
- the damping ratios of the various modes are rather high.
- consequently, the modal coupling is high.
- the longitudinally placed loudspeakers (1 and 4) excite the system much better than the tilted/lateral ones (2 and 3).
- one main dominant resonance peak is present.
- the structural locations are only properly excited by the loudspeaker near to them, implying a propagating nature of the sound field rather than a modal one.

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

5. Modal Analysis

5.1 Method

Modal analysis was performed both by peak-picking and by applying a least squares complex exponential curve fitting procedure. The results discussed below are presented in different formats, each allowing a maximum understanding of the nature of the mode.

The analysis can either be performed directly, using the FRFs with respect to each of the loudspeakers, or using the FRFs with respect to the "virtual" references obtained after singular value decomposition. The latter approach corresponds to the "CMIF" or Complex Mode Indicator Function approach [8].

These virtual references are to be seen as a linear combination of the original references, the first one causing the maximal energy response, and hence the most relevant one to be analysed.

A typical stabilisation diagram, obtained when applying the Least Squares Complex Exponential technique (LSCE) is shown in Fig. 4.

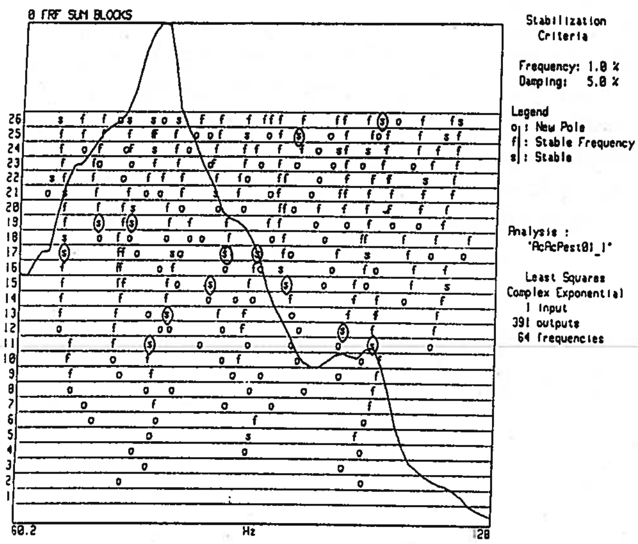

Fig. 4. Stabilisation diagram

that there is much acoustic absorption present in the cabin, distributed non-uniformally, resulting in high damping of the modes and complex mode shapes.

In addition to the longitudinal wave propagation, some transversal modes could also be found using the tilted loudspeakers as one single input, assuming that the poles were estimated appropriately. 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 modes were found insufficient to separate them completely.

6. Principal Field Analysis

6.1 Method

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.

6.2 Results

6.2.1 Frequency band 60-100 Hz

Fig. 9 shows the singular values for the frequency range 60-100 Hz. It is clearly difficult to give an exact estimate of the number of important singular values. The principal response FRFs for the first singular value indicate a resonance at 80 Hz, dominantly excited by loudspeaker 1 and 4 (Fig. 10). The corresponding real field shape is shown in Fig. 11a (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. 11b). Generally, this field shows a longitudinal shape, complementary to the first one (different node lines).

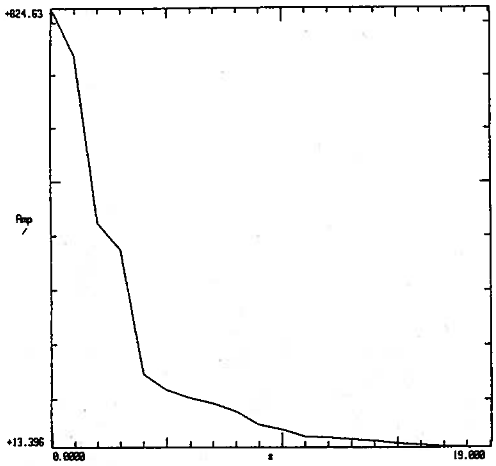

Fig. 9. Singular values (60-100 Hz)

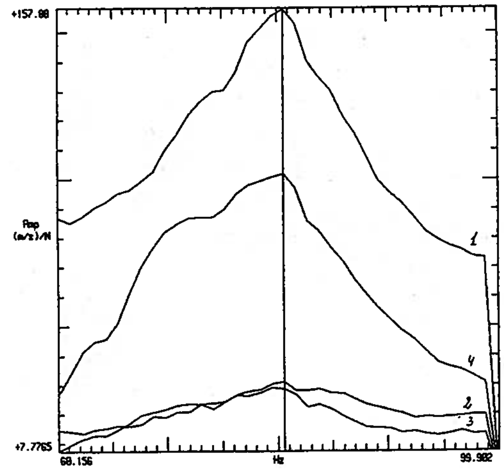

Fig. 10. Principal FRFs

Also, the third singular value corresponds to a field shape with a rather longitudinal nature (Fig. 11c). Again, a top-down mode is superposed at the first frame field shape.

From the fourth field shape (Fig. 11d to 11f), 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.

5.2 Results

The most dominant FRF response is found for the loudspeakers 1 and 4, around 80 Hz. The peak-picking and LSCE application yields similar results.

An example is given in Fig. 5. In the upper part the field shape obtained from an analysis of loudspeaker 1 is shown, in the lower part the one for loudspeaker 4. The pressure value is represented by an axial line segment; the view is a top view along the cabin.

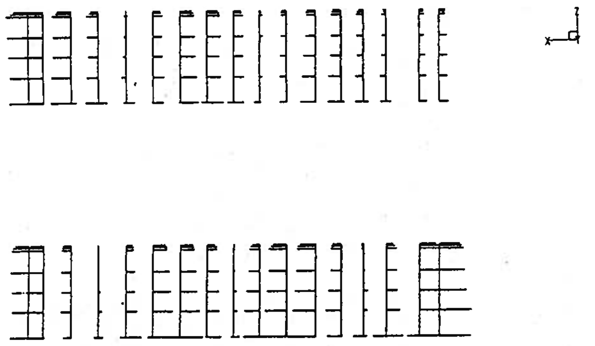

Fig. 5. 80 Hz mode (LS 1, 4)

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 analysing 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. 6, 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.

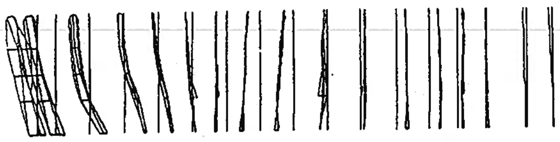

Fig. 6. 110 Hz mode (LS 2)

In Fig. 7, a side view of the mode shapes at 132 and 166 Hz is shown. They can probably be related to the 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. 8 shows the field shape at 96 Hz (comparing to Fig. 1 with regard to the in-flight behaviour, reveals that these modes are well excited in-flight).

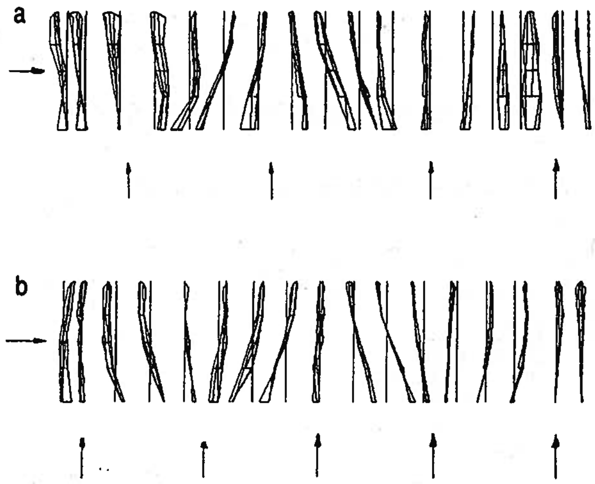

Fig. 7. 132 Hz and 166 Hz modes

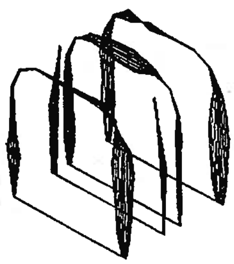

Fig. 8. 96 Hz mode shape

5.3 Summary

The acoustic behaviour of the cabin is characterised mainly by longitudinal modes. Yet, in the majority of cases, these modes do not show the typical features of a standing wave inasmuch as no unique nodal planes can be identified and the places 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 complex mode shapes.

In addition to the longitudinal wave propagation, some transversal modes could also be found using the tilted loudspeakers as one single input, assuming that the poles were estimated appropriately. 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 modes were found insufficient to separate them completely.

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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.

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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. 11b). Generally, this field shows a longitudinal shape, complementary to the first one (different node lines).

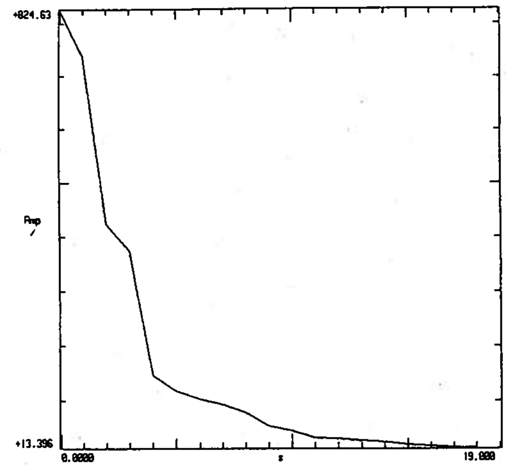

Fig. 9. Singular values (60-100 Hz)

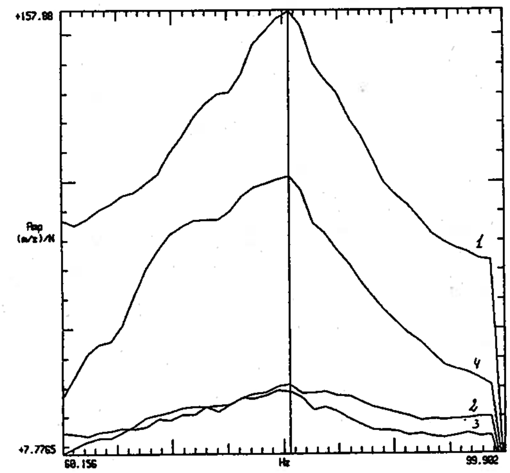

Fig. 10. Principal FRFs

Also, the third singular value corresponds to a field shape with a rather longitudinal nature (Fig. 11c). Again, a top-down mode is superposed at the first frame field shape.

From the fourth field shape (Fig. 11d to 11f), 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 4 principal fields are shown in Fig. 12a to 12d.

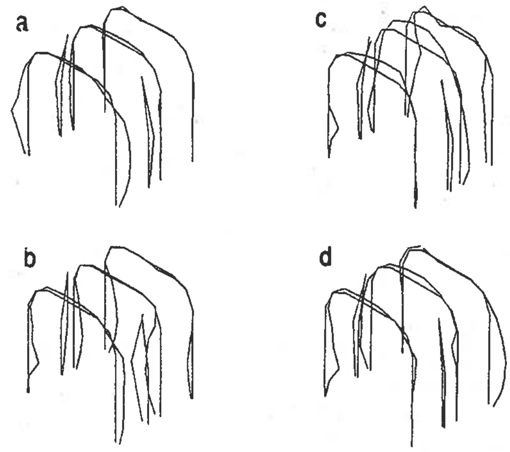

Fig. 12. Principal fields (60-100 Hz)

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.

6.2.2 Frequency band 100-140 Hz

The first four principal field shapes are shown in Fig. 13a to 13d.

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 CMIF plot.

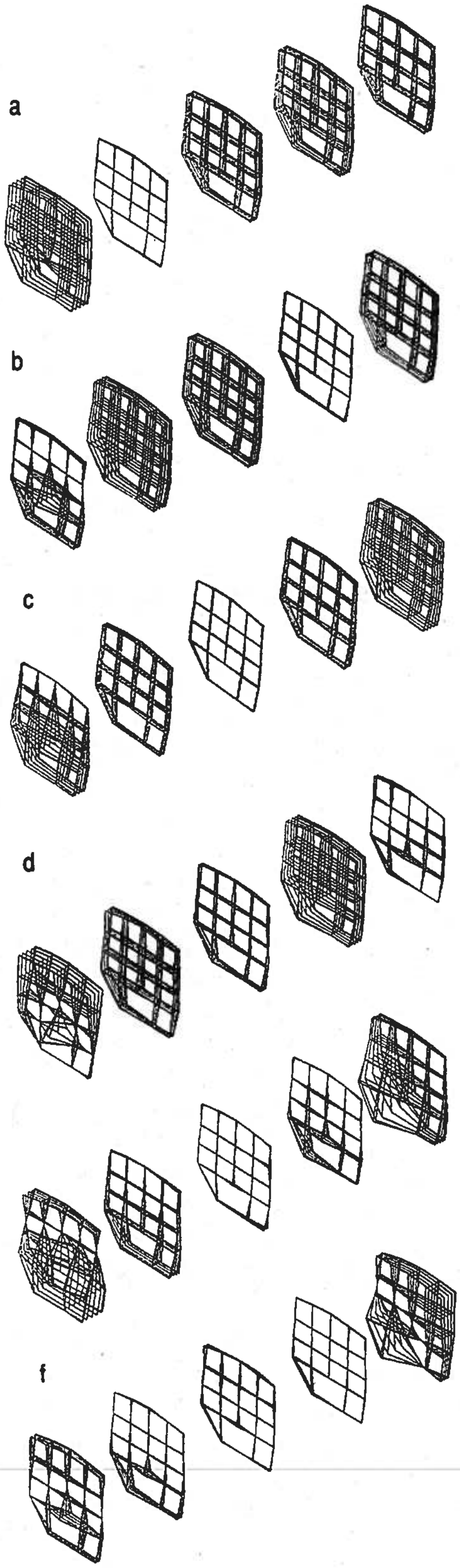

Fig. 11. Principal fields (60-100 Hz)

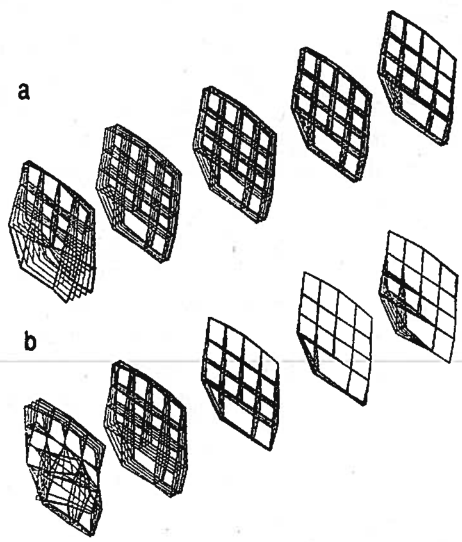

Fig. 13a,b. Principal fields 1-2 (100-140 Hz)

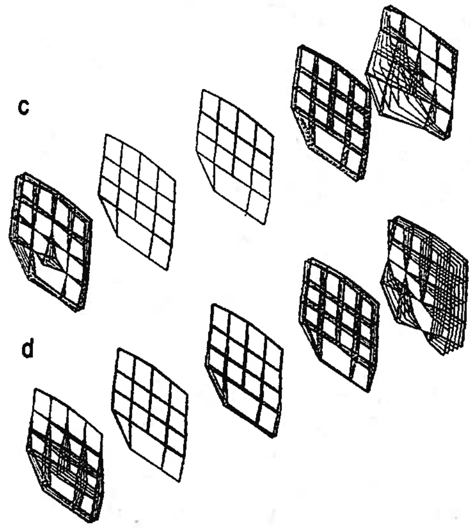

Fig. 13c,d. Principal fields 3-4 (100-140 Hz)

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.

6.2.3 Summary

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.

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

7. Conclusion

An extensive in-flight and ground testing survey of the interior noise and vibration characteristics was performed on four aircraft in order to optimize the design of a multi-channel active control system for interior noise reduction (EC project "ASANCA").

Whereas in-flight tests and secondary source position evolutions can readily be performed with modern multi-channel test equipment, the analysis of the system characteristics of the complex vibro-acoustic fuselage/cavity system is less obvious. Complementary to standard modal analysis, applied to structural as well as acoustical responses, a new approach based on multi-frequency singular value analysis was evaluated.

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References

(1) U. EMBORG, W. HALVORSEN, " Coupled Propeller Contributions to Aircraft Noise and Vibration", Proc. Internoise-90, Götheborg (Sweden), 13-15 Aug. 1990, pp. 821-824.

(2) W. HALVORSEN, U. EMBORG, "Interior Noise Control of the Saab 340 Aircraft", SAE paper 891080, presented at the SAE General Aviation Aircraft Meeting, Wichita-KS (USA), 11-13 Apr. 1989.

(3) CH. H. KUNG, R. SINGH, "Acoustic Modal Analysis Experiment", J. Acoust. Soc. Amer. 77 (2), pp. 731-738, 1985

(4) F. AUGUSZTINOVICZ, "Acoustic Modal Analysis", Proc. 4-th Seminar on Applied Acoustics, Katholieke Universiteit Leuven (B), Dept. of Mech. Eng., 30 Aug. - 1 Sept. 1993.

(5) D. OTTE, H. VAN DER AUWERAER, M. GUSTAVSSON, U. EMBORG, "Vibro-Acoustic Analysis of Propeller Aircraft Integrating Advanced Experimental Modelling with In-Flight Data Analysis", Proc. 14th DGLR/AIAA Aeroacoustics Conference, Aachen (G), 11-14 May 1992, pp. 964-970, paper DGLR/AIAA-92-02-162.

(6) I. BORCHERS et al., "Advanced Study for Active Noise Control in Aircraft (ASANCA)", Proc. 14th DGLR/AIAA Aeroacoustics Conference, Aachen (G), 11-14 May 1992, 12 pp., paper DGLR/AIAA-92-02-192.

(7) H. VAN DER AUWERAER, D. OTTE, G. VENET, J. CATALIFAUD, "Aircraft Interior Sound Field Analysis in View of Active Control: Results from the ASANCA Project", Proc. Noise-Con 1993, Williamsburg-VA (USA), 2-5 May 1993, pp. 219-224.

(8) W. LEURS, F. DEBLAUWE, F. LEMBREGTS, "Modal Parameter Estimation Based on Complex Mode Indicator Functions", Proc. 17th International Modal Analysis Seminar, K.U. Leuven, div. PMA, 23-25 Sept. 1992, pp. 583-598.