ACOUSTIC FATIGUE ASSESSMENT IN THE DESIGN OF AEROSPACE VEHICLES(*)

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Abstract

The importance of acoustic fatigue problems is well estabilished in the design of aircraft structures. In the field of aerospace structures the importance of such problems is likely to increase, since the trend in space missions is toward the use of powerful multimission reusable vehicles.

In facing acoustic fatigue problems, both safe-life and damage tolerant design criteria may be followed.

This paper gives an account of the approaches based on both criteria. In particular the role of input data (acoustic field characteristics, damping, fatigue and crack growth data) and the reliability of service life evaluation are discussed.

1. Introduction

Acoustic fatigue evaluation and certification, which is a well established requirement in aircraft design, is now becoming relevant also for the design of space vehicles.

In fact the recent spatial exploits show that we are passing from the stage of conquest to the stage of industrial exploitation of space.

This new stage basically requires the achievement of a better cost-effectiveness, and hence it imposes the choice of reutilizing the vehicles for several missions.

This natural choice, if it doesn't apparently alter the basic requirements of space structures, as a matter of fact it implies a large revision of design criteria. Up to this moment in fact the fatigue problems could be considered only marginally and in a final design phase, but for reutilizable vehicles they are likely to influence the design from the very beginning.

As regards acoustic loading, it may be noted that the high frequency loads coming from the engines in the propelling phase, and from boundary layer turbolence in the re-entry phase, even if they are relatively low, may become responsible of the structure endurance.

The phisical nature of these loads and the fatigue problem itself demand that fatigue evaluations should be based from the beginning on statistical calculations; on the other hand, the high economical investment of a mission and the safety requirements demand an accurate evaluation of the confidence lavel of the estimated life.

The achievement of this purpose appears to be rather difficult if it is kept in mind that the fatigue life estimation of a single component or of the whole structure is affected by several uncertainties, regarding mainly the following subjects:

- the characteristics of the acoustic environment;
- the model adopted for structure dynamics;
- the interactions between the structure and the acoustic medium;
- S-N and da/dN properties of the material;
- the damage law.

Such uncertainties, in some cases, may be reduced within certain limits by performing ad hoc tests on the material employed and on the components, obviously at the expens of product economy.

But in any case the calculation methods are foreseen to become more and more developed and certified in order to supply a design instrument aiming to reduce to the very least experimental tests and their risk of failure.

Moreover this kind of instrument becomes absolutely necessary for huge sized structures for which experimental methods are more or less inapplicable.

All these problems are considered by airworthiness regulations for aircraft and space vehicles; such regulations now require also the safety against fatigue induced by acoustic loads [1,2].

In this paper the requirements of the regulations concerning these problems are reviewed, and a design procedure is proposed for structures or structural components subject to acoustic loading.

Then the reliability of such procedure is briefly examined, and results are given of two researches on this subject.

One research concerns the study of the dynamics and of the fatigue behaviour of stiffened panels, typical of aerospace structures, and apt to evidence some particular problems connected with acoustic excitation.

The other concerns the experimental observation of the extension of through cracks in unstiffened flat panels subject to a bending stress state.

Both researches refer to problems where the phenomenological aspect itself is not yet clarified enough. Actually the fatigue behaviour of stiffened structures subject to wide-band noise, and the extension of defects in combined stress states of bending type are phenomena which deserve further research effort.

(*) This research was supported by the National Researches Council of Italy.

2. The design of aerospace structures subjected to acoustic fatigue.

2.1 The Design Specification Requirements.

The purpose of flight safety against catastrophical crackings in the structure of aircraft and aerospace vehicles is generally pursued by the regulations by imposing the application of damage tolerance criteria in the structure design.

Such requirements appear to be considered by the MIL-STD-1530 and by the MIL-A-8893 also with regard to sonic fatigue.

As far as space vehicles are concerned it is prescribed [2] that "acoustic excitation from aerodynamic sources..... and from sources internal and external to the vehicles shall be accounted for during all phases of the vehicle's sonic life" and it is prescribed in general that the structure must be Fail-Safe or Safe-Life; in this last case "any flows that cannot be detected in regular inspection should not grow enough before the next scheduled ispection to degrade the strength of the structures below that required to sustain loads...." and "the safe life shall be determined by analysis and test to be at least four times the specified sevice life".

On the basis of what has been expressed by the regulations attention may be drawn on the two following design criteria:

I) Crack free life design criterion.

The arising of a defect due to acoustic excitation should be, for economy reasons, a highly improbable event and it must be assured that the stresses around a possible non-detectable defect should be lower than the propagation treshold; this criterion is due to the fact that even small velocities of propagation, in terms of da/dN, may cause a considerable growth of the damage in a specified number of flights because of the vibration high frequency.

II) Safe crack growth design criterion.

Whenever the previous criterion may not be conveniently followed it is necessary to assure that the damage growth doesn't become critical in the interval between two subsequent inspections.

In fact it seems that, when the da/dN propagation rate assumes significant values, it turns out to be difficult to guarantee the durability for sonic cracking of the structure for the whole service life [3].

2.2 Methodology.

The structural design of a component subjected to acoustic fatigue, as briefly illustrated in Fig. 1, develops through the succession of the following fundamental phases:

1) determination of the acoustic field and of the excitation forces induced on the component.
2) the determination of the frequencies and the natural modes, also in terms of stress, and evaluation of the damping.
3) determination of the stress field originated from the acoustic environment.
4) calculation of the fatigue behaviour; crack nucleation phase and crack growth phase.

Obviously for an efficient component design it appears necessary, in general, to proceed to subsequent re-analysis of the component dimensioning.

These design phases up to now do not seem to be sufficiently defined, and, in some cases, did not reach adequate standards.

The acoustic field identification is generally a rather complex problem, in particular as far as the spatial correlation of noise and the estimation of the structure-acoustic field interactions are concerned. In some cases, as for components near jets or directly lapped by the turbulent boundary layer, fairly well grounded methods are available [4,5], where the structure-acoustic field interaction doesn't appear to be very important.

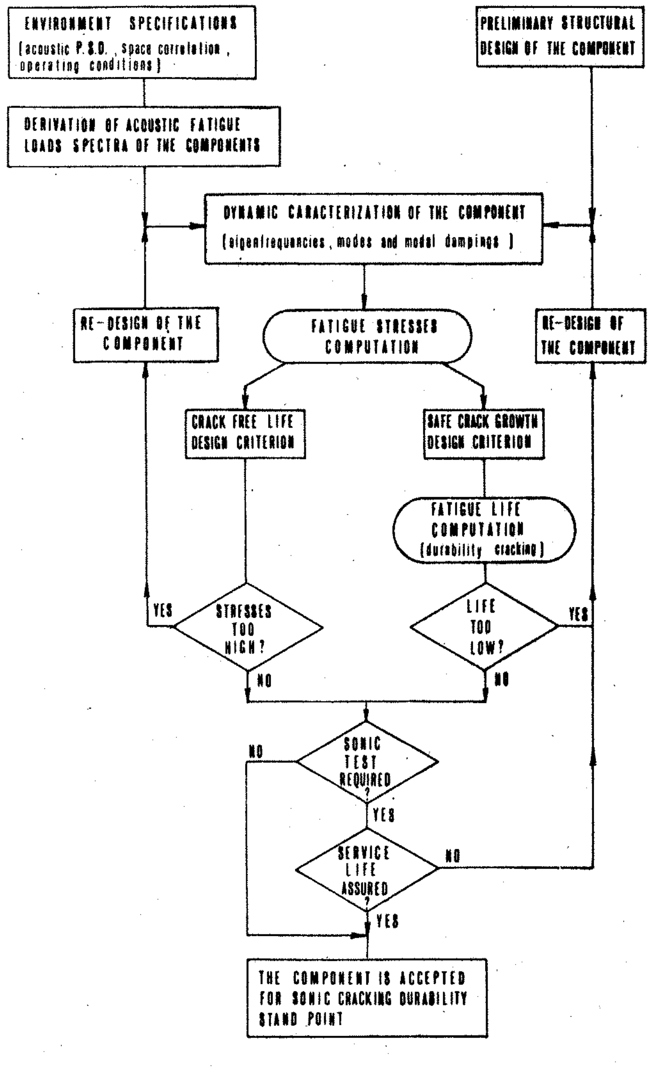

Fig. 1 - Flow diagram of design process of a structure subject to acoustic load.

In other cases such interaction may be more important, as for the noise that builds in the cavity between Space-Lab and Space-Shuttle,and equally efficient methods don't exist.

To this purpose methods based on Statistical Energy analysis are considered convenient [6 7].

The determination of eigenfrequencies and eigenmodes may use simplified methods, in a preliminary design stage. On the other hand a structure representation that may allow the evaluation of detail behaviour in terms of stresses, as required by fatigue life evaluations, demands definitely more refined models, [8,9,10,11,12,13], as, for instance, the ones using finite elements idealization.

Finite elements methods, even if they are more expensive, have the advantage that the accuracy obtainable can be increased just increasing the details of the mesh.

In general, this part of the analysis may be costly, but it is already well defined [14], and not likely to introduce further uncertainties in the evaluation of the stress field due to acoustic excitation. Therefore the major uncertainties about such stress field are coming from the lack of informations about damping. In fact it is well known that damping is a simplified model that is used to take into account several complex dissipation mechanisms, as material damping, friction joints damping and acoustic radiation damping.

Moreover damping plays a basic role in the definition of stress levels, because, due the large frequency band of acoustic excitation,many resonances are likely to occur. For relatively simple components the damping due to joints and to acoustic radiations has been evaluated by Mead and Eaton [15,16].

On the other hand, for complex structures the problem presents formidable difficulties; to make some evaluation acusto-elasticity computations are needed, or it is necessary to make use of experimental values of similar structures, when these are available.

From the above discussion about regulation requirements and design criteria, it comes that, in general, three different evaluation are needed.

The first is the evaluation of the nucleation phase of a crack. The second is a Fracture Mechanics evaluation, used to ascertain if the propagation threshold is ecceeded, for a possible initial defect. The third is the evaluation of the growth time of a defect, pre-existing or caused by fatigue.

The latter evaluation must guarantee that, in any case, a defect non-detectable in one inspection, should not reach critical dimensions before the next inspection.

The determination of the time, or the number of cycles, to the nucleation of a crack proceeds along well known lines, referring to experimental data usually presented in S-N-P curves [17]. The main difficulty of this evaluation is the transfer of data obtained with constant amplitude excitation to the case of random excitation;the application of a linear damage law may solve this problem, without relevant errors. In some cases also fatigue data obtained with random loading are directly available [18].

The check concerning the defect propagation thereshold is made by means of the stress intensity factor K, due to the stress field and to the hypotetical geometry of the defect. Such stress intensity factor K is compared with the experimental threshold value K_{th}. At the present time the data about K_{th} are still rather poor, and with random loads it is not clear how to define the value of K that must be compared with K_{th}.

One possibility, which is certainly conservative, is to use the value of K corresponding to the highest stress of the spectrum; for a narrow band spectrum, typical of single-mode structure response, such maximum stress may be evaluated around 4 times the R.M.S. value [3].

A more accurate procedure is the one to choose that value of K, which, on the basis of a statistical spectrum analysis, is likely to cause the most significant crack growth, in the whole structure life. In any case there is a definite need to go deep into this problem, and to have experimental values of K_{th} obtained directly with random loading.

With the safe crack growth criterion it is necessary to evaluate the extension of a defect having the minimum detectable dimensions, in a specified interval. The evaluation of such extension is made by integrating the propagation velocities corresponding to the distribution of K cycles obtained from statistical analysis of the response spectrum of the structure. The propagation law valid for constant amplitude loading is generally used [3].

This procedure, which is equivalent to the application of a linear damage law, gives acceptable results for narrow band loading (irregularity factor = .9), because in this case the interactions between subsequent cycles of the load history are not significant.

But when the response results from the superimposition of several eigen-modes, the irregularity factor ranging from .3 to .6, the validity of this approach is rather doubtful, even if it is likely to give conservative evaluations.

An alternative approach is the one of using, in the propagation law for constant amplitude loading, a value K_{RMS} of the stress intensity factor obtained from the root meam square value of the stress [19]. The results obtained with this method are so far promising, and they could contribute to improve the reliability of the procedures for foreseing the growth of defects under acoustic loads. Nevertheless it is necessary that such interesting prospects could be validated by a large number of experimental results; in particular, it would be very useful to investigate on how much the irregularity factor of the load spectrum may be significant in affecting propagation velocity.

2.3 Reliability of the design of structures subject to acoustic fatigue.

The attempt to design a structure subject to acoustic fatigue fulfilling the requirements of the regulations, avoiding ad hoc experiments, or, at least, avoiding a sensible probability of failure for the required experiments, demands the reliability analysis of each design step, and then that of the final result.

The present analysis, having a rather general character, does not pretend to define precisely, nor in quantitative terms, the reliability of the design.

Nevertheless it can show the probabilities of success, and emphasize such elements, that, owing to lack in knowledge, have the strongest influence on the accuracy of the design methodology.

For such reason this analysis refers to the simple single-mode response, as, in many cases, it is possible to single out a vibration mode prevailing and determining the response.

The first element that must be established is the accuracy of the evaluation of the stress condition.

The latter comes from the knowledge of the exciting force field (mainly intensity and space distribution), from the application of a computational procedure, and from the specification of the damping factor.

As far as the first point is concerned, the available experience is not sufficient to establish the error possibility in probabilistic terms. In fact the situations that must be faced in specifying the acoustic field are usually so varied that they don't allow such statistical evaluations, as the ones usual for maneuvre and gust loads.

In this situation it is necessary to assume the intensity and the distribution of the exciting forces on the basis of conservative considerations, from the results of similar cases. Then such data must be considered as a deterministic element of the problem.

The application of a computational procedure to determine the stress response of the structure, except for a scale factor depending on damping coefficient, generally does not introduce sensible errors, at least in comparison with the remaining sources of uncertainties.

A suitable choice of the available methods may lead to a good accuracy in the stress field even with the details required in fatigue analyses.

As it is well known, the third step, i.e. the evaluation of the damping coefficient δ, is the one introducing the biggest uncertainties.

An estimate of the uncertainty affecting the evaluation of δ can be drawn from the experimental data reported in [4]. An examination of such data shows that, in a design stage, the value of the damping coefficient can be considered as a statistical variable.

For such variable the ratio between the actual value δ and the established value $\bar{\delta}$ (e.g. the value recommended by Thomson [4]) has a log-normal distribution with a standard deviation $\sigma(\log \delta - \log \bar{\delta}) \simeq .28$.

The hypothesis that the scatter in damping factor is the main responsible of the scatter between actual stresses and computed stresses is strengthened by the comparison between computed values $\bar{s}_{RMS}$ and experimental values s_{RMS}. In fact also for the scatter of the ratio between s_{RMS} and $\bar{s}_{RMS}$ a log-normal distribution can be assumed [4], with $\sigma(\log s_{RMS} - \log \bar{s}_{RMS}) \simeq .28$.

In conclusion in the design stage the stress s_{RMS} can be considered a statistical variable having a log-normal distribution, distinguished by $\sigma(\log s_{RMS}) = .28$.

Obviously different methods for stresses evaluation could be used but the conclusions so far drawn can be considered to have a general character.

In the application of CRACK FREE LIFE methodology the stress acting on the structure must be compared with the fatigue strength, specified by S_{RMS}-N_r curves. It must be assured, with a specified probability, that no crack may arise during the whole life of the structure.

With reference to the S_{RMS}-N_r data reported [4] for 2024-T3 riveted skin specimens in bending stress, it appears that S_{RMS} has a log-normal distribution with $\sigma(\log S_{RMS}) \simeq$ $\simeq .12$, in all the range of N_r of practical interest.

To guarantee the absence of cracks, with a reliability of 99.9%, i.e. with the probability of a crack $P_f = 10^{-3}$, as generally specified by the regulations, it must be [20]

$$P_f = P(S_{RMS} \leq s_{RMS}) = P(\log S_{RMS} - \log s_{RMS} \leq 0) = 10^{-3} \qquad (1)$$

Since log S_{RMS} and log s_{RMS} have normal distributions

$$Z[p_f = 10^{-3}] = \log(\bar{S}_{RMS}/\bar{s}_{RMS}) / \sqrt{\sigma^2(\log S_{RMS}) + \sigma^2(\log s_{RMS})} =$$
$$= 3.09 \qquad (2)$$

whence :

$$\bar{S}_{RMS}/\bar{s}_{RMS} = 8.73 \qquad (3)$$

Owing to the high frequency of acoustic loads the required crack free life of the structure may consist of no less than $N_r = 10^9$ cycles[3]. From the S_{RMS}-N_r data quoted above, the mean value (50% probability) for $N_r = 10^9$ comes out to be $\bar{S}_{RMS} \simeq 16$ MN/m^2; then the computed value must be $\bar{s}_{RMS} \simeq 1.85$ MN/m^2.

It may be observed that the ratio between the ultimate or strength value of the stress and the value computed in the design stage, is higher, by an order of magnitude, than the value 1,4 ÷ 1,5 that is normally assumed in INFINITE LIFE DESIGN[21].

But in the latter case the stress is a deterministic variable, while, in the case so far considered, a rather high error probability must be assigned to the computed stresses, owing to the scatter expected for damping.

The situation may change sensibly only when it is possible, e.g. by means of ad hoc tests, to remove the uncertainties about the actual stress state; in this case, as $\sigma(\log s_{RMS}) \simeq 0$, it is:

$$\bar{s}_{RMS}/\bar{\bar{s}}_{RMS} = 2.35 \quad .$$

Anyhow it must be pointed out that the scatter of the S_{RMS}-N_r relation, for typical specimens with narrow-band random loading, is rather high, being 4 to 5 times higher than the one for sinusoidal loading.

Also for this problem, even if it is not so bad as the one of stress evaluation, it would be much convenient to have a larger number of experimental data.

Similar conclusions are drawn when the design has the further requirement that a non-detectable defect may not grow because of acoustic loads. This can be fulfilled if the threshold value K_{th} of stress intensity factor is not exceeded, when the specified defect is present.

The value of K_{th} for the material must be considered a statistical variable; unfortunately the data available so far are not enough to define with a good accuracy the scatter that may be expected [22].

From a brief analysis of the informations reported [23,24,25], $\sigma(\log K_{th})$ can be valued to have a normal distribution around its mean value, with $\sigma(\log K_{th}) = 0.13$.

On the other hand, the value of K to be compared with K_{th} depends on the size of detectable defect, then on NDE capability, and on the critical region of the structure where the defect has to be located.

In an accurate analysis also the initial defect size had to be considered a statistical variable, whose features depend on the NDI methods employed. Therefore, for the sake of generality, in the present analysis the size of the defect is considered as a deterministic quantity; owing to this assumption, log K coming from the loads applied and from the specified defect must have the same distribution as the computed stress log s_{RMS}, and then $\sigma(\log K) = 0.28$.

When the condition $P_f(K_{th} \leq K) = 10^{-3}$ is imposed, being $\sigma(\log K_{th}) = 0.13$, similarly to what obtained with (1), (2) and (3), once again it comes that:

$$K_{th}/K = 8.74 \quad . \tag{4}$$

To make clear the meaning of (4), let us consider an initial through crack, as specified by the regulation[1], having a size $2a_0 = 0.25$ in, located far from stiffeners.

In a bending stress field the resulting value of stress intensity factor is:

$$K = ((1 + \nu)/(3 + \nu)) \; \bar{s} \sqrt{\pi a_0} \tag{5}$$

where ν is Poisson's ratio.

Since the load spectrum is not a constant amplitude one, in order to evaluate K, it is worth to take into account the maximum stress value $\bar{s} = 4 \; \bar{\bar{s}}_{RMS}$, even if, as outlined below, this may be exceedingly conservative.

With this hypothesis, for 2024-T3 material having $K_{th} = 2.2$ MN/m$^{3/2}$, it comes that:

$$\bar{s}_{RMS} = \left(\frac{1}{8.74}\right)(2.2)\left(\frac{3+\nu}{1+\nu}\right)\frac{1}{4\sqrt{\pi \times 3.175 \times 10^{-3}}} = 1.58 \text{ MN/m}^2 \tag{6}$$

It is worth noting that the limit value $\bar{s}_{RMS}$ of the computed stress needed to avoid the exceeding of the threshold stress intensity factor for the specified defect size, is substantially the same as the one needed to avoid defect nucleation.

Once again the cause of such a low value of $\bar{s}_{RMS}$ lies firstly in the high uncertainty affecting stress evaluation, and secondly in the lack of knowledge about the phenomenon of the threshold for bending stress fields and random spectra.

The stress levels needed to fulfill the CRACK FREE LIFE requirement are so low, that it is worth to discuss also the reliability connected with the application of SAFE CRACK GROWTH design requirement.

The growth time may be evaluated from a propagation law employng the stress intensity factor coming from the R.M.S. stress[19]:

$$da/dN = C_1 \Delta K_{RMS}^{m_1} \tag{7}$$

An equivalent approach consists in the use of a linear damage law with the load succession, assuming that the peaks have the Rayleigh distribution. This may be done means of a suitable propagation law valid for constant amplitude loading, as for instance the Paris law:

$$da/dN = C_2 \Delta K^{m_2} \tag{8}$$

With reference to the first procedure, the number of cycles N needed to reach the critical defect size a_{cr} is:

$$N = [1/a_0^{(m_1-2)/2} - 1/a_{cr}^{(m_1-2)/2}] / [(m_1-2) C_1 \; \bar{s}_{RMS}^{m_1} \; \pi^{m_1/2} \; Y^{m_1}] \tag{9}$$

where Y is a suitable function of structure geometry and of the loading mode.

The expression (9) may be semplified, considering that a_{cr} is always larger than a_0 by an order of magnitude, and that $m_1 > 2$, as follows:

$$N \simeq 1/\left\{a_0^{(m_1-2)/2} \; (m_1-2) C \; \bar{s}_{RMS}^{m_1} \pi^{m_1/2} \; Y^{m_1}\right\} \tag{9'}$$

Since a_0 has been considered a deterministic datum, the estimate of N is affected only by the uncertainty of the value $\bar{s}_{RMS}$ and of the propagation law. For the latter generally all the scatter is ascribed only to the constant C_1.

As both s_{RMS} and C_1 have a log-normal distribution around their mean values, also log N has a normal distribution around the mean $\bar{N}$, and a standard deviation equal to:

$$\sigma(\log N) = \sqrt{\sigma^2(\log C_1) + m_1^2 \; \sigma^2(\log s_{RMS})} \tag{10}$$

Here $\log C_1$ and $\log s_{RMS}$ are considered statistically independent.

The value to be ascribed to $\sigma(\log s_{RMS})$

has been already discussed above.

The value of $\sigma(\log C_1)$, as resulting from specific propagation tests in bending stress fields, is of the order of 0.2. In general, depending on different conditions, $\sigma(\log C_1)$ may range from 0.1 to 0.25(*).

The second method leads to identical results, as the linear damage law does not introduce new elements affecting the scatter of N [26], the latter being determined substantially by the scatter of stress evaluation and by that of C_2. Moreover the scatter of C_1 and C_2, as well as m_1 and m_2, have nearly equal values.

From (10) it can be easily observed that, as in general $m_1 \approx 2.5$, almost all the possible error in the evaluation of N comes from the evaluation of the stresses.

If the probability that a defect may attain its critical size in one inspection period must be $P_f = 10^{-3}$, the ratio between the computed value $\bar{N}$ and the value of N corresponding to one inspection period must be as high as 180.

This may cause the safe crack growth life design methodology to be inapplicable, as it requires an exceedingly high inspection frequency.

Nevertheless such methodology may be applied with a different approach, i.e. renouncing to evaluate the inspection interval in the design stage, with a high reliability. But in this case it is necessary to measure subsequently the actual stress in the actual structure in operational conditions, and from such measure to evaluate the inspection period with a high reliability. Then the scatter related to the evaluation of N is mainly due to the scatter of the propagation velocity; the ratio between the computed inspection period and the one to be fixed may range from 2 to 4.

The discussion so far carried on has pointed out the difficulties in designing a structure affected by acoustic fatigue, with a high degree of reliability.

Adopting the CRACK FREE LIFE criterion it is necessary to reduce drastically the allowable stress level;similar difficulties are encountered whith the SAFE CRACK GROWTH criterion.

The basic reason of such difficulties is the high possibility of error given to the damping and stress evaluation, which appears from the comparison between the actual and the computed stresses.

Further difficulties are connected with the knowledge, so far inadequate, both of fatigue nucleation and propagation, in the stress conditions generated by acoustic loads.

(*) The evaluation of the influence of $\sigma(\log C\ )$ on $\sigma(\log N)$ by means of (10) is questionable, as it has been pointed out [26] but anyhow it is a conservative estimate.

3. Research work carried out in the acoustic fatigue area.

In the preceding paragraphs it was shown that it is necessary to increase and deepen the knowledge of several aspects of the phenomenon of acoustic fatigue.

Hereafter the first results of the research on two important features of the phenomenon are reported and analysed.

3.1 Some aspects of the dynamical and fatigue behaviour of a stiffened panel subject to acoustic loading.

It is well known that, for the sake of structural efficiency, very often aerospace structures are built up by stiffened panels. Usually the stiffness obtained for such panels is high enough to keep the stresses induced in the panel itself by noise at a very low level. On the other hand the stringers may suffer acoustic fatigue, as it is aknowledged in literature, because they use to be relatively deformable.

The response of such structures, when acoustically excited, resulting from the superimposition of many eigen-modes, may be rather complex. Even in a single-mode excitation, the stress field can hardly be represented by simple stress situations, so that it may be difficult to foresee the evolution of possible cracks in the structure.

To investigate this behaviour a series of tests has been planned, to be conducted on stiffened panels having the shape shown in Fig. 2; such panels had to be tested in the progressive waves acoustical chamber of Centro Ricerche FIAT (FIAT Research Center)[27] with a longitudinal pre-stress of 98 MN/m^2.

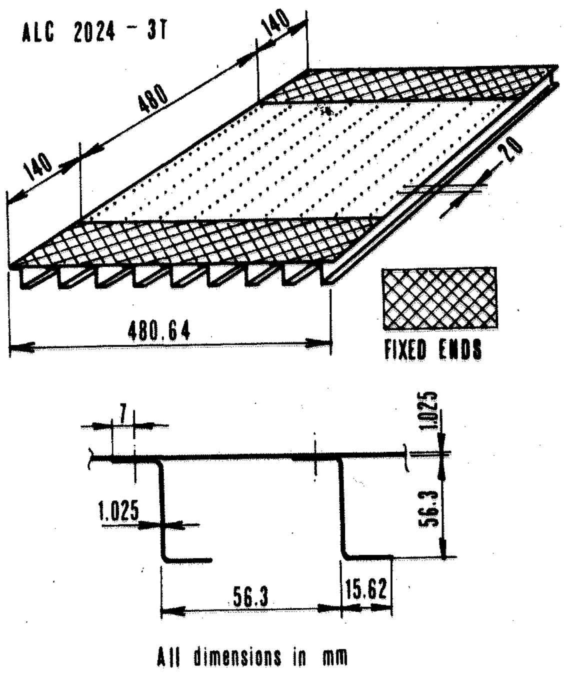

Fig. 2 - Dimensions of tested panels.

These tests are now in progress; so far tests have been completed for 4 panels with sinusoidal excitation.

Excitation frequencies have been choosen in order to excite stringers deformation modes; such frequencies were previously found by experimental and numerical analyses.[31]

With numerical analyses, carried out for a panel having simply supported ends, with a finite-stripes idealization [13], the first 60 eigen-frequencies and mode shapes were evaluated by means of method described in[28]; some of these results are reported in figure 3a and 3b.

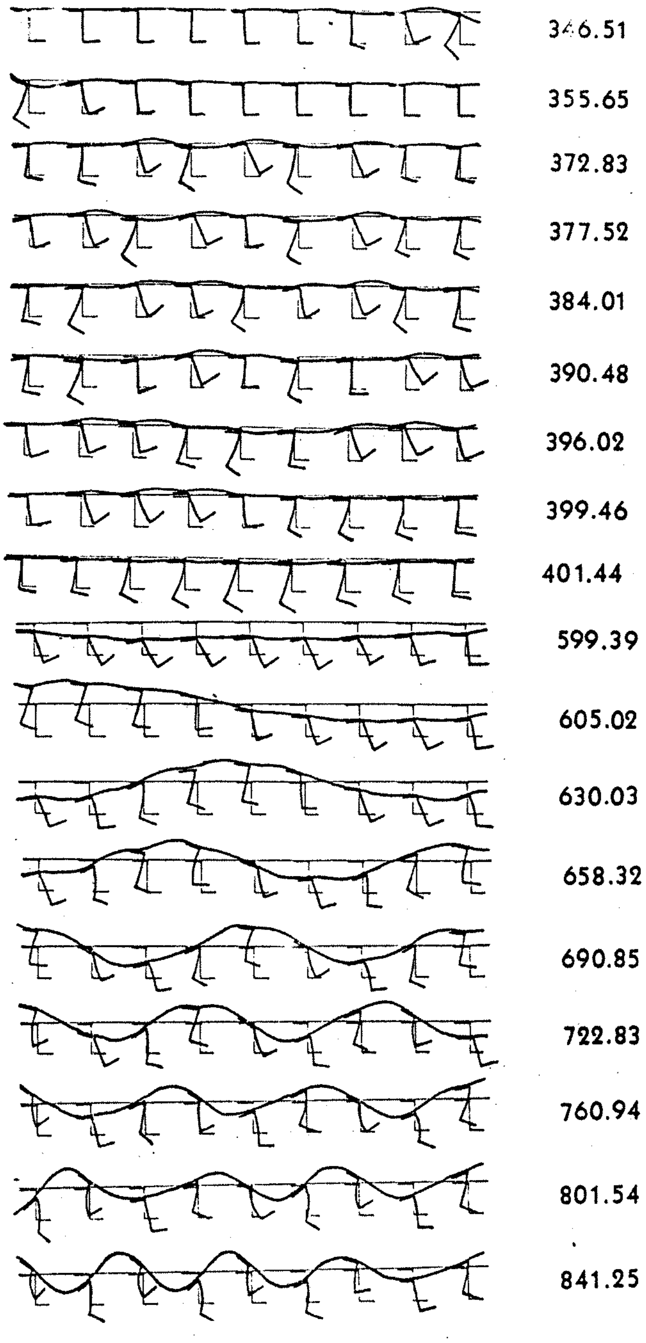

Fig. 3a - Computed natural modes and frequencies of the panel of Fig. 2, simply supported, not prestressed. Equivalent length = 388mm. One longitudinal half-wave.

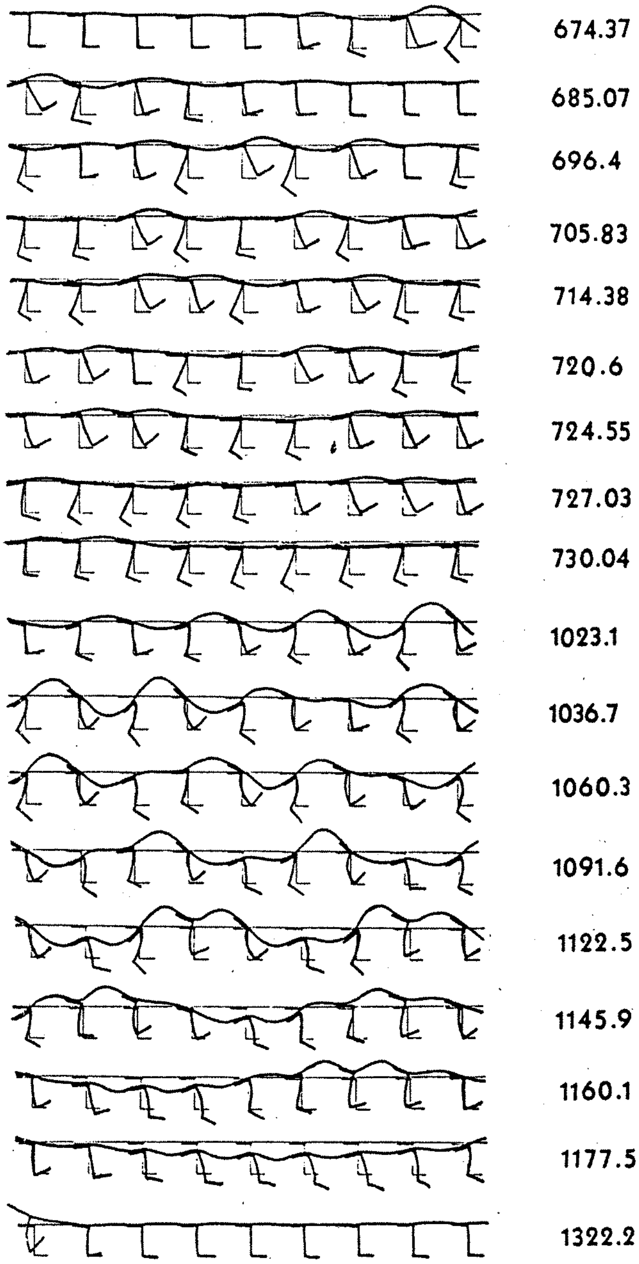

Fig. 3b - Computed natural modes and frequencies of the panel of Fig. 2, simply supported, not prestressed. Full length = 480mm. Two longitudinal half-waves.

Experimental analyses, carried out with a panel having clamped ends have shown that the mode shapes in a plane perpendicular to the stringers are very near to the shapes computed, while the corresponding frequencies present some differences, due to the differences in end restraints.

Briefly, the vibratory behaviour of the panel shows that the eigen-modes can be divided into groups of 9 modes, 9 being the number of the stringers. The modes of the same group have the same number of longitudinal half-waves and common characteristics of the transverse shape. In particular, the modes of the first group are mainly concerned with the deflection of the stringers, and the modes of the second group with the deflection of the panel.

These preliminary research allowed to find out two frequencies, i.e. 465 Hz and 620 Hz, as the most significant for sinusoidal load tests in the sonic chamber.

Fatigue tests have been made with 155 db overall sound pressure level; a preliminary test whith wide-band noise has also been performed, recording the output of 17 strain gage bridges and 7 microphones.

Figure 4 show the position of microphones and strain gages.

The measurements were recorded on a ten-channels analog tape, with an AMPEX FR 1300 Recorder, using a tape speed of 30 i.p.s..

Analog recordings were converted into digital recordings, with a sampling rate of 8000 points per second.

Such digital recordings are now processed by means of time series analysis programs.

This work is still in progress at present, but the first results seem to confirm the validity both of the numerical and of the experimental modal analyses previously carried out.

The graphs reported in figures 5a and 5b show the transfer and response functions of strain gage bridges no. 12 and 13, measuring, respectively the bending in the web of a stringer, and the transverse elongation in a point of the panel.

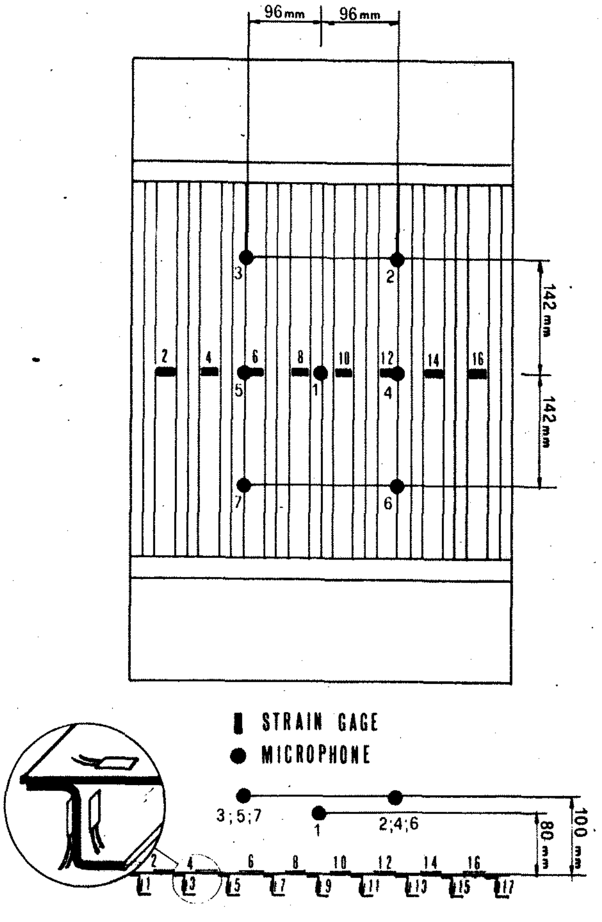

Fig. 4 - Location of strain gages and microphones on panel tested in acoustic fatigue chamber.

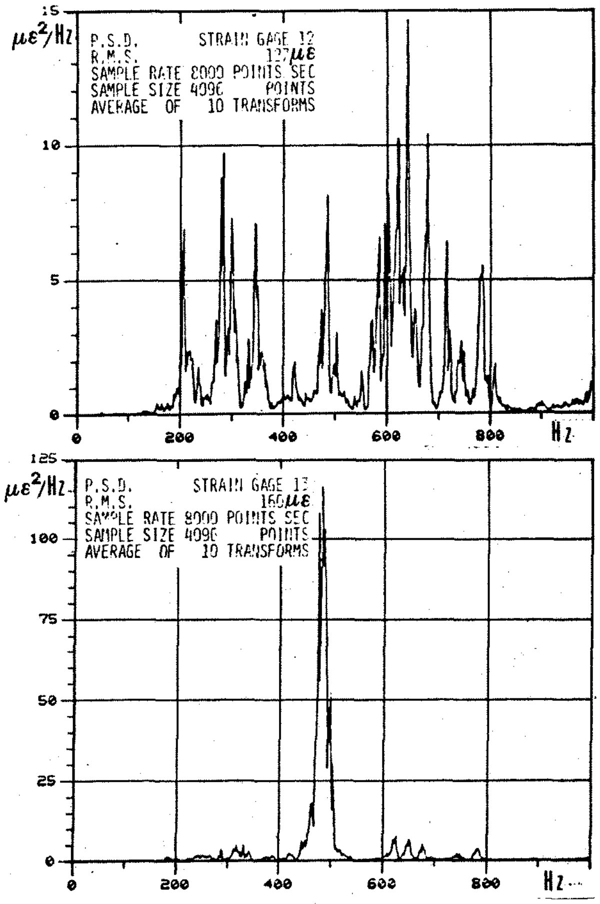

Fig.5a - Autopower spectral densities of the strain gages n. 12 and n. 13.

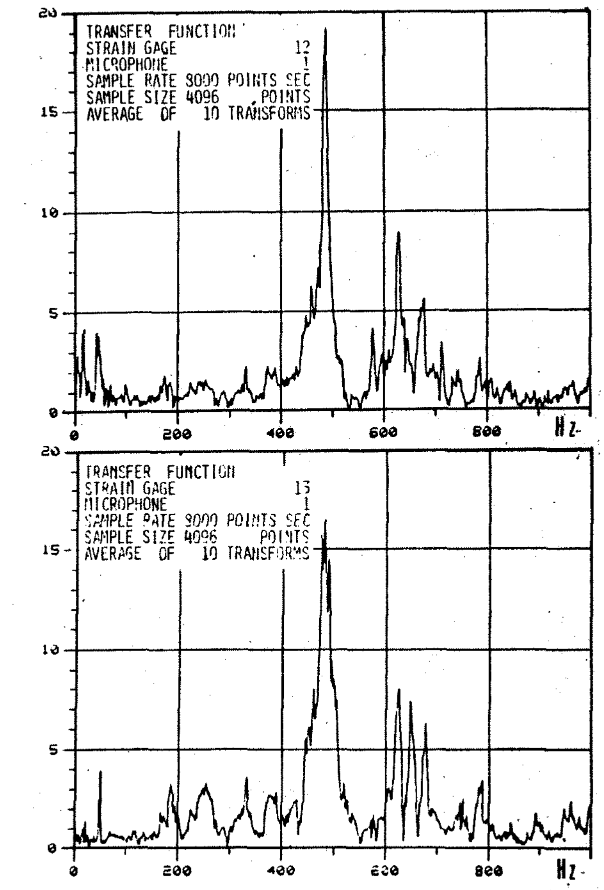

Fig. 5b - Transfer function of the strain gages n.12 and n.13 and microphone n.1.

These graphs show clearly the presence of many resonance peaks, very near to each other, that seem to gather in two main groups, around 465 Hz and 620 hz, respectively.

Furthermore the peaks of the first group are more sensible for the strain gage applied to the stringer, while the ones in the second group are more sensible for the strain gage applied to the panel.

It must be noted here that the latter results where obtained from the average of 10 samples, each sample having 4096 points. This may not be enough for a good accuracy, since, as it is well known, more samples are needed for a higher confidence level, and more points in each sample are needed for a better frequency resolution, even if this may bring to very high computing costs.

Moreover it is important to note that the trends of the autopower spectra of strain gages n. 12 and 13 are quite different.

The situation is typical, namely the power spectra of the stringer deformations resemble that of a narrow band noise with on irregularity factor of the order of .9; on the contrary, the transverse elongations of the panel show an energy content which is spread out over a larger frequency band.

Nevertheless the R.M.S. values of all the strain gage signals are quite similar.

As previously noted this analysis is not finished at present; more results will be available in the near future.

In the first fatigue test, run at 620 Hz, the panel did not show any crack after 10 hours.

In the subsequent 3 tests, run at 465 Hz, longitudinal cracks have been detected after about 1 hour, in the stringers, near the stringer-panel riveting, Fig. 6.

Such cracks were originated in the central part of the panel, and propagated towards both constrained ends. Propagation velocity started with values around 1.10^{-6} m/cycle, then falling more and more as the apex approached the constraints.

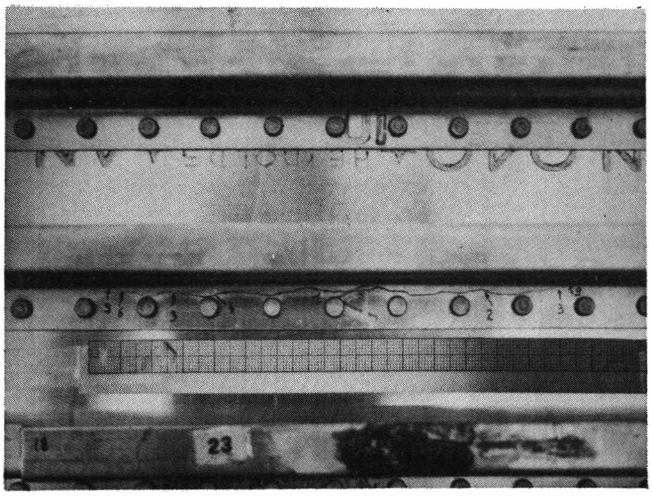

Fig. 6 - Fatigue cracks in the stringer of panel tested in acoustical chamber.

The reason of this decrease in propagation velocity lies mainly in the decrease of stress levels near panel ends, due to the longitudinal mode shape.

These first tests show once again that acoustic fatigue is more likely to damage the stringers than the covering, unless particular care is taken of detail design of the stringers themselves.

Furthermore, the Z shape of the stringers employed in test panels, that may be very convenient for other reasons, does not seem to be the best choice in this respect.

The tests have also shown that accurate computation methods are needed to evaluate stress levels in such details as the stringer panel riveting, and that such simple methods, as the ones indicated by ESDU Data Sheets [29] may not be useful in these cases.

The finite-stripes idealization, used for computing eigen frequencies and mode shapes, seems to be adequate, also for the evaluation of the stresses.

3.2. Crack propagation in panels in bending.

The features of fatigue crack propagation in a bending stress field, typical of panels subject to acoustic excitation, are so far relatively unknown. For instance the non-linear effects connected with partial crack closure have still to be explained, both from the phenomenological point of view, and in respect to their quantitative influence on damage growth.

The problem is rather hard to solve, even in the case of a single-mode response, and then it is much more complex for multi-mode responses, typical of acoustic excitation.

For these reasons an experimental research has been planned, in order to investigate the features of crack extension in bending stress fields.

In a first stage crack propagation is observed in simple test specimens; subsequently tests will be carried out also with random excitation, both on simple specimens and on more complex structural components.

The experimental equipment is so far composed by a 568 Kg shaker, the specimen constraints and a closed circuit TV system.

The camera is stiffly mounted on a device which can run in the direction of the growing crack. Moving the camera the crack tip can be driven on a reference line on the monitor. Trough the accurate measurement of the camera displacement the growth of the crack is obtained. Trough the measurement of the time spent in the growth of the crack, the crack growth rate can be estimated.

The specimen is a flat rectangular panel, having the longer sides clamped to the constraints and free on the shorther sides. The crack have the direction of the longer sides, so that a sufficiently large part of crack extension can take place in a fairly constant stress field. The constraints allow to apply to the specimen a constant membrane pre-stress.

The experimental set-up is shown in Fig. 7.

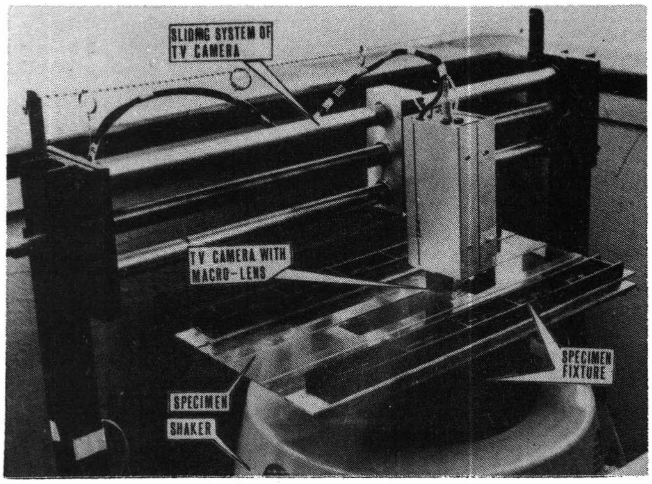

Fig. 7 - Experimental set-up for crack propagation tests of panels in bending.

The test data $\{a_i, N_i\}$ are processed with a computer program[30] through the following steps, namely,

- computation of the crack growth rate $(da/dN)_i$ processing the date $\{a_i, N_i\}$ by the five points diveded difference approach;
- regression analysis of the data $\{(da/dN)_i - \Delta K\}$ to obtain the characteristic constants of the Paris, Forman, Collipriest K-rate laws. The regression analysis is further substantiated by a set of statistical meaningful data (correlation coefficient and normality test);
- statistical test of data coming from different tests to ascertain if they belong to the same population. If the tests are positive the regression analysis is further performed on the complet set of data to obtain the new values of the K-rate laws characteristics constants.

Such a procedure was applied to process the data coming from testing six panels 600 x 200 x 1.23mm. made of 2024-T3.

The panels were tested with the experimental equipment previously described in pure sinusoidal bending at 175 c.p.s..

In all the tests the panels were loaded in such a way to **maintain** 40 MN/m^2 as maximum stress in the points far from the crack zones.

Some results of this approach are summarized in Fig. 8. there the best-fit straight line of the data Log da/dN - Log ΔK is shown together with the scatter band defined by the 10% and 90% confidence intervals. The figure shows also the results drawn with the same procedure from crack propagation data relevant to the case of a stress constant in the thickness. Such data were obtained with panels loaded with R = -1 at a frequency of 6 c.p.s..

From such results some conclusions can be drawn, namely,

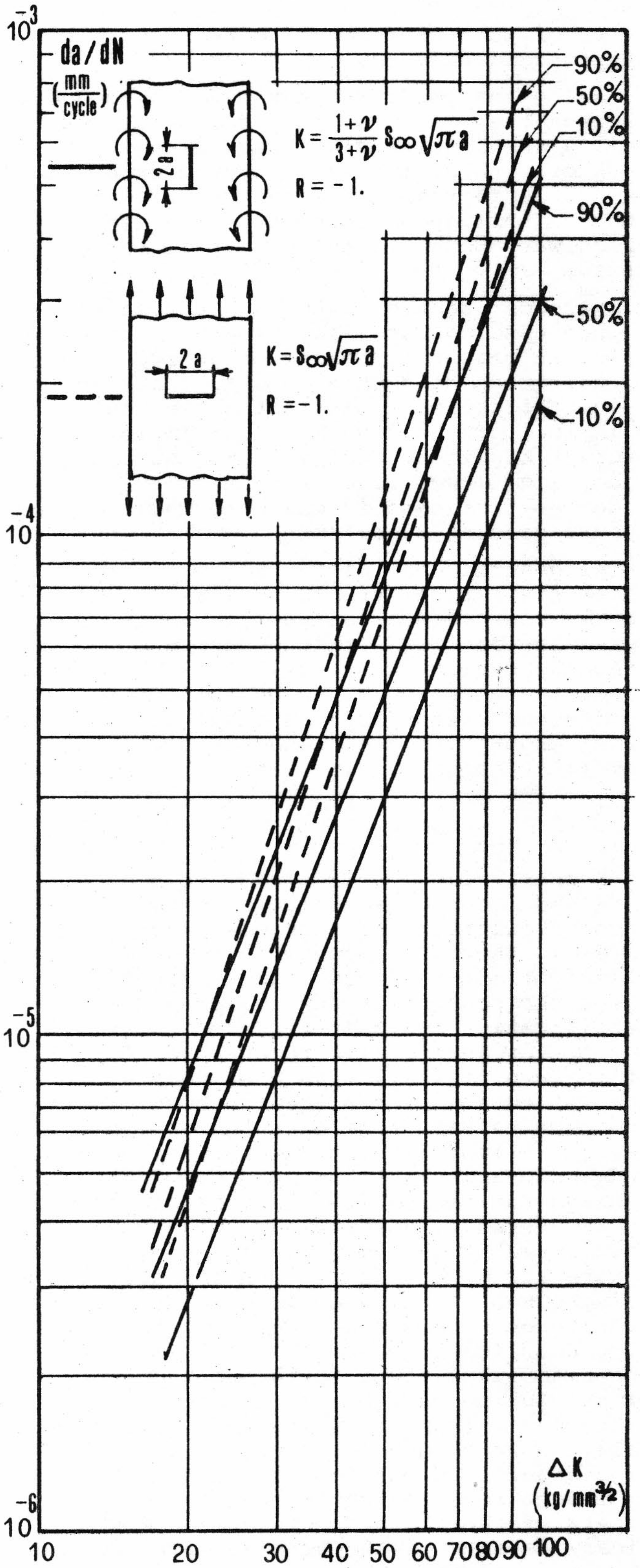

Fig. 8 - Crack propagation data in bending and in tension. Paris law.

- the ΔK - da/dN relationship in the Paris formulation is a suitable tool to takle the problem of crack growth under pure sinusoidal bending. The same conclusion further holds for the Forman and Collipriest laws;
- a noteworthy difference exists between the ΔK - da/dN best-fit curves in the two ca-

ses of pure bending and stress constant with thickness.

Among the possible reasons of such a difference the following ones can be quoted, namely the different accuracy with which K is known in the two cases (the pure bending case is affected by greater uncertainties) and the influence of the test frequency which was much higher in the pure bending tests,

- a broader scatter band is found in the pure bending case. The errors in the crack length measurements can be quoted among the causes of such a result. The crack tip identification is in fact a difficult task in the pure bending mode. An improved crack tip identification procedure ought to be envisaged to ascertain if the broader scatter band is characteristic of the pure bending mode.

As general conclusion the test data and their statistical treatment show the need to rely on ad hoc fatigue and crack growth data when dealing with the design of acoustically excited structure.

Further data ought to be collected both on the nucleation and the propagation of a crack under both broad and narrow band random fluctuating bending load.

4. Conclusions

Acoustic fatigue problems are becoming more and more relevant for structural design of aircraft and reusable spacecraft.

It is therefore necessary to develope efficient and reliable design methods; this gives the possibility of avoiding expensive certification tests in the acoustical chamber or, at least, of a remarkable reduction of failure risks. It has been done a very semplified analysis of design possible methodologies with regard at the obtaining result reliability with the present knowledge situation. It can be observed that to achieve the high reliability degree normally requested by aircraft and spacecraft structures it is necessary to undergo very severe structural design conditions. The main reason is the uncertainty with which phenomenon data are known: first the damping coefficient value and therefore all the data inherent to fatigue phenomenon either during nucleation or defect propagation phase. In fact fatigue data are generally known enough for constant amplitude sinusoidal load but not for random load.

To get some insight into the different facets of the acoustic fatigue some significant aspects of the problem were tackled through experimental and theoretical investigations.

Tests were conducted in an acoustical chamber on stiffened panels to measure their dynamic response and successively their fatigue endurance. To study dynamic behaviour of such structures it was realized the need of developing sophisticated computation methods (finite elements or finite stripes methods) taking into account the bending-torsion as well as the load deformations of the stiffeners which are the elements more liable to cracking by acoustic loading. The results untill now obtained do not allow to draw general conclusion on the problem; nevertheless the stress spectrum irregularity factor has been shown to be an important parameter in phenomenon.

Another investigation significant in respect of acoustic fatigue was the measurement of the growth of crack in a sheet loaded in pure bending, in the frequency range typical of the acoustic excitation. The data were treated on the basis of ΔK crack propagation rate relationship and compared with existing data on low frequency, uniform stress in thickness data. The result of such a comparison shows that existing low frequency crack growth data are inadequate to predict the high frequency, pure bending crack growth rate.

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To be published on: L'Aerotecnica, Missili e Spazio.