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The estimation of r.m.s. stress in stiffened skin panels subjected to random acoustic loading

Associated software: ESDUpac A7205

Endorsed by
The Royal Aeronautical Society

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THE ESTIMATION OF RMS STRESS IN STIFFENED SKIN PANELS SUBJECTED TO RANDOM ACOUSTIC LOADING

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THE ESTIMATION OF RMS STRESS IN STIFFENED SKIN PANELS SUBJECTED TO RANDOM ACOUSTIC LOADING

1. NOTATION

a	width of skin plate or frame pitch	m	in
b	flat panel width between stringers or curved panel arc length between stringers	m	in
E	Young's modulus of plate material	N/m^2	lbf/in^2
f	fundamental natural frequency of skin plate assuming all edges fixed	Hz	Hz
$G_p(f)$	spectral density of acoustic pressures at frequency f	$(N/m^2)^2/Hz$	$(lbf/in^2)^2/Hz$
K_δ	damping ratio correction factor		
$L_{ps}(f)$	spectrum level of acoustic pressure at frequency f	dB*	dB*
p	uniform static pressure on plate	N/m^2	lbf/in^2
p_{rms}	root mean square fluctuating pressure	N/m^2	lbf/in^2
R	radius of curvature of panel	m	in
r	ratio $(1-\nu^2)/0.91$ (when $\nu = 0.3$, $r = 1$)		
S_0	ratio of stress at rivet line in assumed mode shape to applied uniform static pressure on plate		
S_{rms}	rms stress at rivet line due to acoustic loading	N/m^2	lbf/in^2
t	thickness of panel	m	in
V	velocity parameter for plate material†		
δ	damping ratio of mode of vibration of skin panel		
ν	Poisson's ratio of skin material		
ρ	density of skin material	kg/m^3	‡

* The reference pressure is 20 μPa (*i.e.* 2×10^{-5} N/m^2) or 2.90×10^{-9} lbf/in^2.

† The velocity parameter is defined in SI units by $V = (E/\rho)^{1/2}/5080$ or in British units by $V = (E/\rho)^{1/2}/200\,000$. V is approximately unity for all common structural metallic materials.

‡ A density value expressed in British units as pounds per cubic inch has to be divided by 386.4 before it can be used in the formula for V given here or in the computer program in Appendix A. (A force of 1 lbf acting on a mass of 1 lb produces an acceleration of 386.4 in/s^2.) For example, if the material density is 0.097 lb/in^3, the value used in the calculation of V should be 2.51×10^{-4} $lb\ s^2/in^4$.

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2. NOTES

2.1 General

This Data Item gives a method of estimating the rms stress in rectangular skin panels subjected to random acoustic loading. The reference stress is the rivet line stress*; stresses at other positions on the plate may be found by applying a suitable factor to the reference stress.

The rms stress for a stiffened panel subjected to random acoustic loading on one side is given approximately by the expression

$$S_{rms} = \left[\frac{\pi}{4\delta} f\, G_p(f)\right]^{½} K_\delta S_0 \ . \tag{2.1}$$

In the case of a control surface or similar structure where two skins are coupled by ribs and both vibrate in response to a loading largely on one side, the stress level is about $1/3$ of the value given by Equation (2.1). However, where the acoustic loading is of similar intensity on both sides of a box structure the stress level is about ½ of the value given by Equation (2.1).

In deriving Equation (2.1) it has been assumed that the predominant form of skin vibration is one in which individual plates within the stiffened panel vibrate independently in the assumed mode which, for this calculation, has been taken to be the fundamental fixed-edge mode. This restricts the use of this Data Item to skin-stringer panels where the bending stiffness of the stringers is sufficiently high to approximate to a fixed-edge condition for individual plates. It is not applicable to stiffened panels where individual plates are much stiffer and the supporting stiffeners relatively more flexible than for conventional skin-stringer panels.

The pressure is assumed to be uniform and in phase over the whole of each individual plate and the spectrum level of the acoustic pressure is taken to be constant over the range of frequencies close to the fundamental natural frequency of the panel.

The rms stresses given are the nominal unfactored rivet line values and in using these stresses to calculate a fatigue life care must be taken to ensure that due account is taken of any geometric stress concentrations in the region of the rivet line.

This Item may be used for etched skins, or skins with bonded reinforcing, by first obtaining the nominal rivet line rms stress for the panel, assuming it to be of uniform thickness, and then applying a correction with the aid of ESDU 67029 (Reference 15).

For conventional structures without special damping treatment, f may be taken as the undamped natural frequency.

A typical value of damping ratio for the fundamental mode of a conventional aircraft structure without special damping treatment is 0.017. This value has been used in producing the design nomographs although a correction factor is presented which enables rms stresses to be estimated at other values of damping ratio.

In Appendix A, a computer program is described that estimates the rms skin panel stresses using the same analysis method as that used for the graphical calculation procedure. However, the computer program is not limited by all the restrictions that are imposed on the graphical procedure (see Section 2.2).

* Throughout this Item the location of the reference stress is referred to as the rivet line. For non-riveted structures, this location should be considered to be the panel edge restraint line.

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2.2 Limitations and Accuracy

The means of estimating the rms skin surface stresses in panels under acoustic loading provided in this Item is constrained by the assumptions, given in the previous Section, made in deriving Equation (2.1). The rms stresses predicted are those across the rivet line at the mid position of the longer side of the plate. This is the position of the highest nominal stress when the individual plates vibrate in their fundamental fixed-edge mode. The assumption that each plate responds only in its fundamental mode is usually a good approximation if the vibration modes are well separated in frequency.

The method of analysis uses thin plate theory and is applicable to panels having shallow curvature. The thin plate criterion is satisfied if b/t and, for curved panels, R/t are less than 20. The condition for shallow curvature is met if b/R is less than 1. Further, it is assumed that the plate bending stress is within the linear region where it is directly proportional to the normal pressure, that is p/E less than about 20 $(t/b)^4$.

The data provided are restricted to panels having a/b not less than 1. Also, for the graphical procedure, a/b should not exceed 3. This upper limit on a/b does not apply to the computer program where the limitations (see Section A.1) are dictated by practical computational considerations. The computer program is not restricted by assumptions of linear variations that had to be applied to the graphical procedure when considering the changes of S_0 with curved plate parameters. Consequently, it is recommended that the graphical procedure be used only for flat plates and those having very shallow curvature.

In the evaluation of the response stress no allowance is made for the stress concentration factor due to rivet holes *etc*.; this is expected to be negligible at distances greater than 2.5 times the rivet diameter from the centre line of the rivet along the rivet line.

The accuracy of the prediction procedure is illustrated by a comparison of estimated and measured stresses in Figures 4 and 5. See Section 5 for information on the data presented in those Figures.

3. NOTES ON THE FIGURES

Figures 1 and 2 give nomographs for S_{rms}. The nomographs are entered at a value of $L_p(f)$, each quadrant being used in turn in the direction indicated through ranges of a/b, b/t, b^2/Rt and t/V. Figure 2 is an extension of the stress range of Figure 1. Figures 1 and 2 are drawn for a value of $\delta = 0.017$ which should be used, in the absence of other information, for structures having no special damping treatment. If a value of δ is known from experimental results, the value of S_{rms} is calculated for the measured value of δ using the correction factor K_δ which is plotted against δ in Figure 3. In the derivation of the nomographs it is assumed that the Poisson's ratio of the skin material is 0.3.

The higher range of Figure 2 is for use in estimating stresses in titanium and high strength steel panels. However, high values of stresses obtained from this Figure should be treated with caution as no extensive measured data are at present available to check the validity of the simple response theory over the higher stress range.

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4. CALCULATION PROCEDURE

4.1 Graphical Calculation Procedure

The graphical procedure for estimating S_{rms} in a general case is as follows.

(i) Estimate the fundamental natural frequency of the panel. For flat plates use ESDU 75030 or ESDU 87002 (References 19 and 20 respectively), or evaluate the highest frequency of the first group of natural frequencies using ESDU 72003 (Reference 16). For curved plates calculate the natural frequency using ESDU 87002.

(ii) Obtain the value of spectrum level of acoustic pressure $L_{ps}(f)$ at the calculated frequency. If the band pressure level only is known, it is first corrected to pressure spectrum level (unit bandwidth) using ESDU 66016 (Reference 13).

(iii) Calculate the parameters a/b, b/t, b^2/Rt and t/V and read the value of S_{rms} from Figure 1 or 2.

(iv) For values of δ other than 0.017, factor the estimated value of S_{rms} by K_δ obtained from Figure 3. The value of $\delta = 0.017$ is typical of aircraft structures without special damping treatment.

4.2 Spectral Density of Acoustic Pressure

Within the nomograph the spectrum sound pressure level is converted into the spectral density of acoustic pressure. The spectrum sound pressure level is converted into the root mean square fluctuating pressure in units of (N/m^2)/Hz (see expression below or ESDU 66018 – Reference 14) and then squared giving a value in units of (N/m^2)^2/Hz^2. Since unit bandwidth is used this is numerically equal to the spectral density of acoustic pressure $G_p(f)$ in units of (N/m^2)^2/Hz. When p_{rms} is in units of N/m^2,

$$L_{ps}(f) = 20(\log_{10} p_{rms} + 4.70) , \tag{4.1}$$

and when p_{rms} is in British units of lbf/in^2

$$L_{ps}(f) = 20(\log_{10} p_{rms} + 8.54) . \tag{4.2}$$

5. COMPARISON WITH MEASURED DATA

Figure 4 shows a comparison of estimated and measured stresses in flat or curved plates. Figure 5 shows a comparison of estimated and measured stresses in control surfaces and other coupled skin box type structures. For these estimated values the factors given in Section 2.1 were applied to the stresses given by Equation (2.1). In estimating rms stresses to compare with measured values, δ has been assumed to be 0.017.

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