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DESIGN AND EXPERIMENTAL VERIFICATION OF THE USB-FLAP STRUCTURE FOR NAL STOL AIRCRAFT

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Abstract

Since the USB-flap of the NAL STOL experimental aircraft is one of the primary design modification of the original C-1 transport aircraft, an acoustic fatigue test of substructural models at elevated temperature was conducted to verify the safety of the flap structure during the planned flight evaluation program.

The five structural models were provied and following items tests were carried out: (1) Detection of the thermal buckling temperature. (2) Vibration tests. (3) Acoustic fatigue test at elevated and room temperature.

The conclusions reached by the present experiment are summarized as follows: (1) Thermal buckling occured on test panels. (2) Resonant frequencies of the flat test panels fall in the range between those of clamped and those of simply supported plates. (3) Through the prescribed fatigue test period, all structural models have been proved themselves to be strong enough to resist both acoustic and thermal loading.

Besides the laboratory verification, analytical treatment on the response has been analyzed and the fatigue life has been estimated the assumption that the flat test panels were uniformly loaded by heat and noise.

I Introduction

NAL is presently developing an experimental STOL aircraft which adopts Upper-Surface-Blowing (USB) flap configuration as a high lift device. Since the upper surfaces of the wing and the USB flaps are exposed to thermal as well as acoustic load due to high temperature exhaust gas of FJR-710 engines, it is required to guarantee that the USB flap structures have enough acoustic fatigue life to stand both acoustic and thermal loads during prescheduled flight test hours. Many design procedure for skin stringer type structures at room temperature can be found in a commercial data bank. [1] However, there are very few informations available on acoustic fatigue at elevated temperatures. Therefore only the report on the structural features of YC-14 [2] and vibration test results [3] of Al or Ti alloy panels could be refered in this study.

Prior to manufacturing, it was programmed to verify the safety of USB flap structure, and simulation calculas and laboratory test of panel structure models have been carried out.

This paper covers results of acoustic fatigue test, fatigue life estimation based on computer simulation [4] or measured real time strain history, and preliminary tests such as thermal loading, free vibration and dynamic response.

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II USB Flap Structure

The cross-section of the USB flap structure is illustrated in Figure 1. It consists of two parts; the fore flap and main flap. The upper skin of both fore and main flaps are fabricated by Ti-6Al-4V sheet panel of 1.8 mm thickness because of high temperature flowing on their surfaces. And lower skins are fabricated by 2.5 mm thick 2024C-T3 panels. The trailing edges of both flap are tailored smoothly by wedge-shaped polyimide honeycomb sandwich structures. This flap configuration has been selected among the possible aerodynamic options. Due to simplicity and feasibility it is decided that the stiffened panel type structure is to be manufactured.

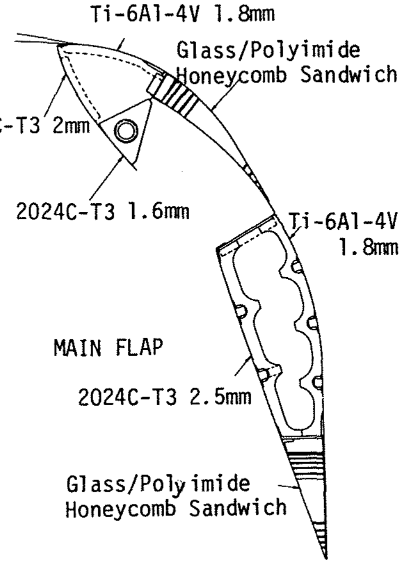

Figure 1. General Layout of USB Flap

III Acoustic Fatigue Design

Our primary concern in the initial design phase is to determine the bay dimensions of panel surrounded by ribs and stringers. Considering over all chord and span lengthes of the flap system there remain few choices for the dimension of the optimum stiffener spacing. Nevertheless, the panel thickness can not be determined until the adequate fatigue life estimation is carried out.

The analytical approach to the fatigue design will be briefly explained in the followings.

Firstly, the acoustic excitation force to the panel should be clarified. To NAL STOL aircraft four FJR-710 engines are to be installed. From the full scale ground test simulating the USB system with the actual engine, it has been shown that the maximum Overall Sound Pressure Level (O.A.SPL) reaches as high as 166dB on the flap surface.

Secondly, the information of the temperature on the flap surface is required. The test data show that the temperature stays in the range from 90 °C to 170 °C and that it has spatial distribution.

Since there remaines ambiguity on the thermal environment, the acoustic fatugue life for the flat plate is estimated by numerical simulation with thickness and temperature being varied as parameters.

In this simulation analysis, it was assumed that the intensity of acoustic load was 160 dB and the noise spectrum shape could be approximated by two straight lines as drawn in Figure 2. This assumptions resulted from the difference between the flat panel and box structure. [1]

The results shown in Figure 3 clearly described the general trend of the fatigue life. The fatigue life first decreases as the temperature rises but above the certain temperature the life increases.

It is anticipated that the parabolic tendecy of the fatigue life corresponds to the occurence of oil canning type vibration around the thermal buckling temperature of the panel. Numerical simulation approach has played a primary role to determine the skin thickness to the flap system and the conclusion has been reached that the plate of 1.8 mm thickness seems adequate enough to withstand the maximum input noise more than 1000 hours at 200 °C.

Since the noise level (O.A.SPL) varies depending on the flight condition or on the mode of maneuver, the structural damages to the panel from various noise levels should be all adjusted so that their relative contribution are explained in terms of the equivalent duration at the maximum noise level.

Thus it is concluded that 140 hours endurance test with O.A.SPL=160 dB at 150 °C would be good enough to clear the safety requirement of the flap structure.

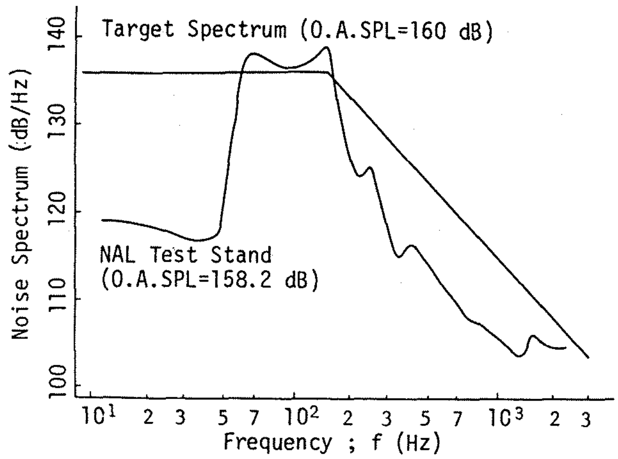

Figure 2. Noise Spectrum

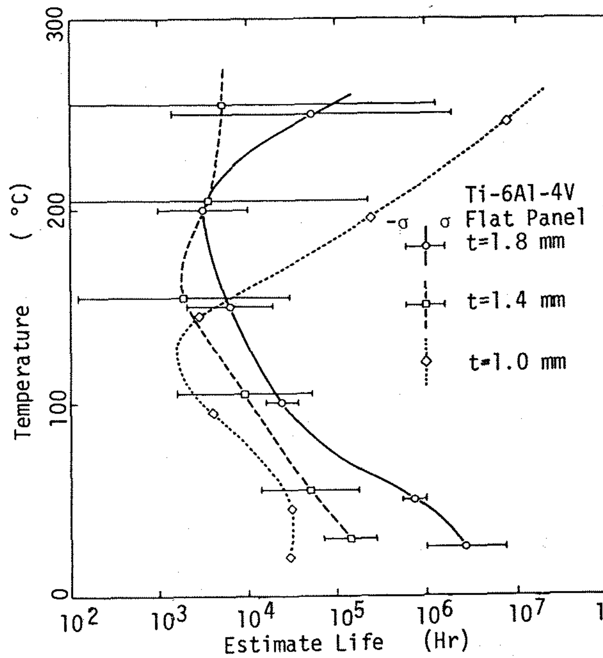

Figure 3. Fatigue Life Estimate by Numerical Sim.

IV Test Panel

Provieded are 3 test panel structure models which typify so-called 9 bay semi-monocoque structures, that is, the face sheet panel stiffened by two stringers and ribs on the back side; The first one for upper surface of the main flap (A), the second one for lower surface of it (B), the third one for the upper surface of fore flap (C).

Although the first two are flat models, the third one has the same curvature as real structure and all three have the real dimensions in terms of panel thickness and stiffener spacing.

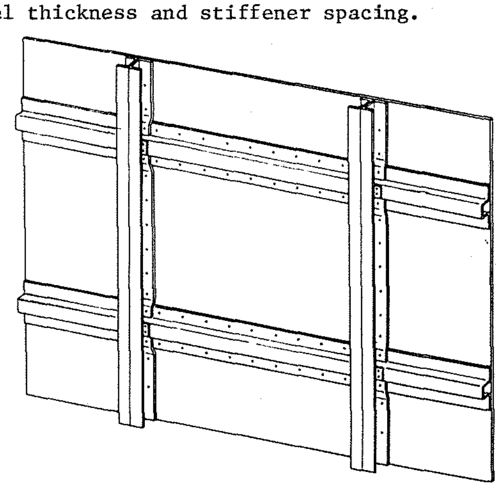

Figure 4. Flat Panel Structure Model

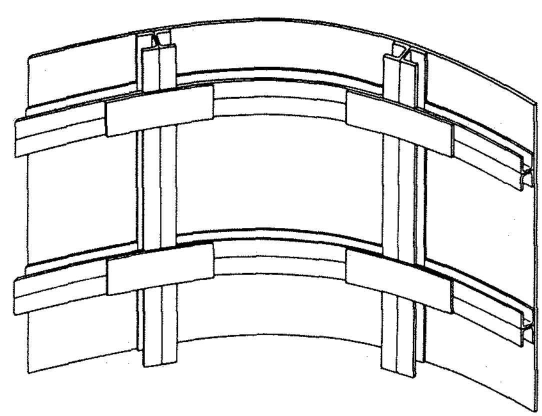

Figure 5. Curved Panel Structure Model

In addition to the previous three test panels, provided are another pair (D,E) for upper and lower skin structures of the flap whose face panels have reduced thickness as compared with A and B, while other dimensions being the same with them.

Table 1 summarizes test panels A-E with their center bay dimensions and thickness of the face sheet. Clearly the center bay of each structure model is of the primary interest in the fatigue strengh point of view.

Figure 4 and 5 sketch the back side of the test panels which show the cross-section of the stiffener on which fastened are the face panels.

V Static & Dynamic Characteristics of the Test Panel

Resonant Frequency, Mode Shape and Damping Coefficient

Two types of the test methods, sinusoidal sweep and impact, are applied to all test panels. The resonant frequencies of the test panel (C) are summarized in Figure 6. As some values were detected by the impact method and the other by the sweep method, the two methods seem to complement each other.

An example of mode shape is illustrated in Figure 7. The allows show the ribs and stringers.

The damping coefficients are plotted in Figure 8 and 9. The values range from 2×10^{3} to 2×10^{2} and they do not have any peculiarity with regard to the structural type (flat or curved in the case) or material difference. The straight line of Figure 8 and 9 recomended in Reference 1 as a design guide does not seem appropriate.

Static Thermal Response

In order to detect the thermal buckling temperature of test panels (A),(C) and (D), the mean thermal strain trends are pursued during the gradual heating on the face side up to 200 °C.

Figure 10 shows the test result of the panel (A). From the mean strain trend the thermal buckling temperature of (A) is considered to be about 160 °C. The relationship between the buckling temperature and face plate thickness was illustrated in Figure 11, which confirms the validity of the factor 6.38 (See Reference 3); the experimental value is well estimated by multiplying this factor

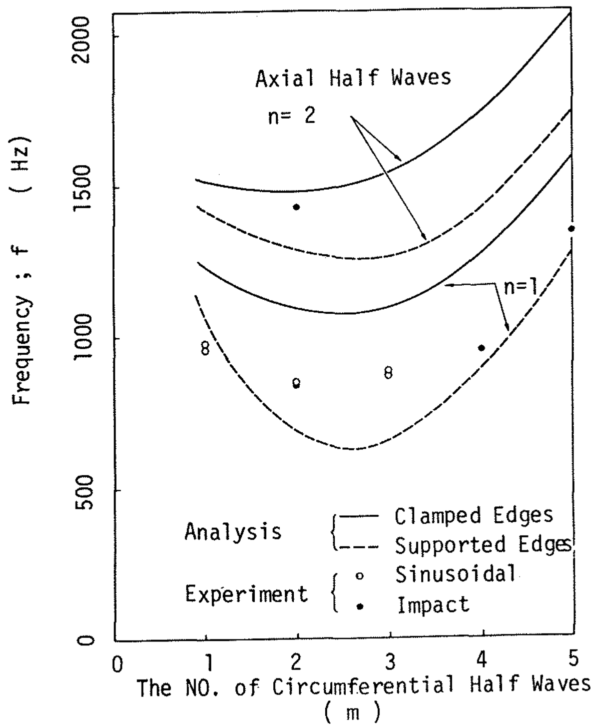

Figure 6. Resonant Frequency of the Curved T. P.

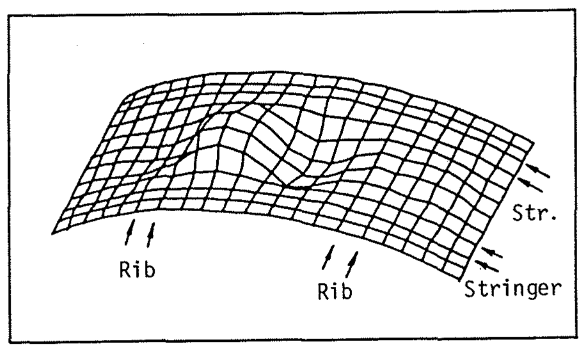

Figure 7. A Mode Shape Example of Curved Panel

Test Panel		A	B	C	D	E
Center Bay Dimension (mm) ; Edge Line		92 × 244	113 × 244	274 × 144	92 × 244	113 × 244
〃 ; Rivet Line		116 × 268	137 × 268	298 × 168	116 × 268	137 × 268
Face Panel Thickness (mm)		1.8	2.5	1.8	1.4	2.0
Face Panel Material		Ti-6Al-4V	2024C-T3	Ti-6Al-4V	Ti-6Al-4V	2024C-T3
Vibration Frequency and Mode at Room Temp.		0	0	0	0	0
Frequency Shift by Temperature		0	×	0	0	×
Static Thermal Loading		0	×	0	0	×
Dynamic Response Test	at Room Temperature	0	0	0	0	0
	at Elevated Temperature	0	×	0	0	×

Table 1. Test Panel Center Bay Dimension and Test Items

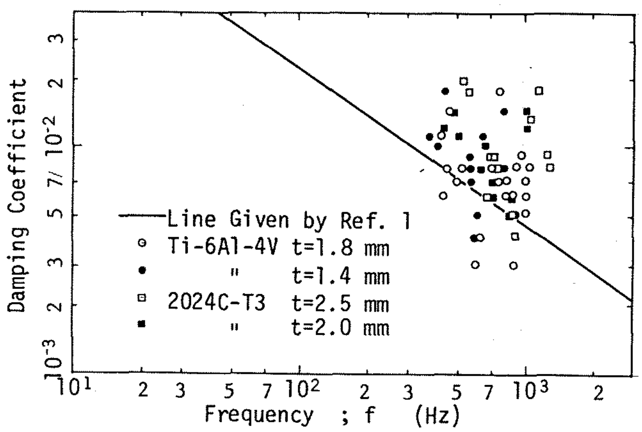

Figure 8. Damping Coefficient of the Flat Panel

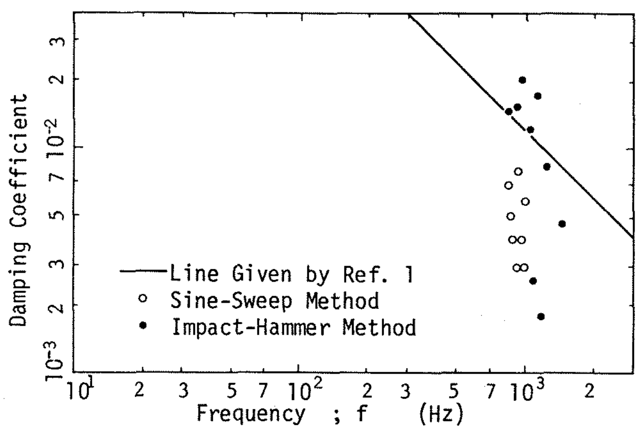

Figure 9. Damping Coefficient of the Curved Panel

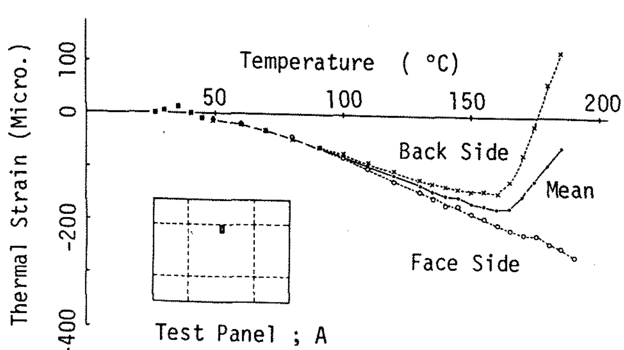

Figure 10. The Strain Response under Thermal Load

to the theoretical buckling temperature (Tc , See Figure 11.) of the simply supported plate under uniform heating. In the calculation of Tc,there are used to following values, a=269.0 mm, b=116.0 mm, ν=0.34 and $\alpha=9.45\times10^{-6}$°C^{-1}.

Dynamic Response

The typical dynamic response characteristics of the test panel (A) is illustrated in Figure 12 and 13; strain response spectra at 150 °C and room temperature (R.T), respectively. Both at 150 °C and R.T, the spectrum levels by the numerical simulation are higher than those by the laboratory test.

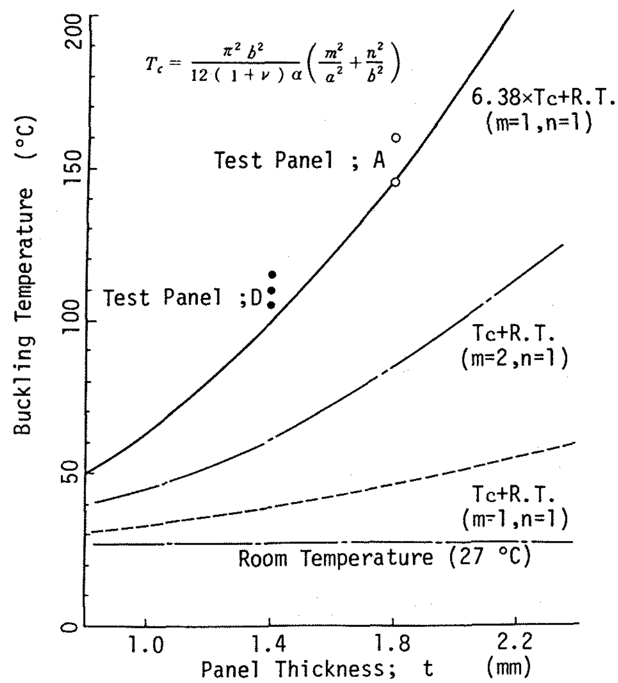

Figure 11. Thermal Buckling Temp.(=Tc)

The present numerical analysis is comfirmed to give conservative estimates for entire frequency range.

The overall R.M.S. and mean values of strain response versus temperature show linear trend as is depicted in Figure 14. It is interesting to note that those trends look quite opposite: the mean value decreases, while the R.M.S. strain increases as temperature rises. Since overall strain levels are low, less than 30 micro-strain, and no oil canning type vibration seems to have occured, significant damage to the structure can not be expected from R.T to 200 °C.

The correlation between the analysis and experiment is shown Figure 15, where the analytical result is about 10 times larger than experimental results. The difference between the analysis and experiment is also exemplified in Figure 16, which present numerical simulation procedure.

VI Acoustic Fatigue Test

Acoustic fatigue test at R.T and 150 °C, including preliminary tests have been conducted at NAL Acoustic Test Stand whose general layout, control and measurement systems are summarized by the authors. (Reference 6)

Table 2 summarized the acoustic fatigue test result carried out so far in NAL acoustic fatigue test stand. It was scheduled that the first half of the specified test hours is allocated for the recalled Spec. Test and the second half to the Accelerated Test. Although acoustic fatigue test hours for panels (A),(B),(D) and (E) amount to 140 hours, while for (C) is only 60 hours. This is because in many flight modes, the fore flap remains extracted beneath the wing and, thus, its upper surface is less exposed to noise and hot gas.

In the Spec. Test, the noise input has the spectrum shown in Figure 2.

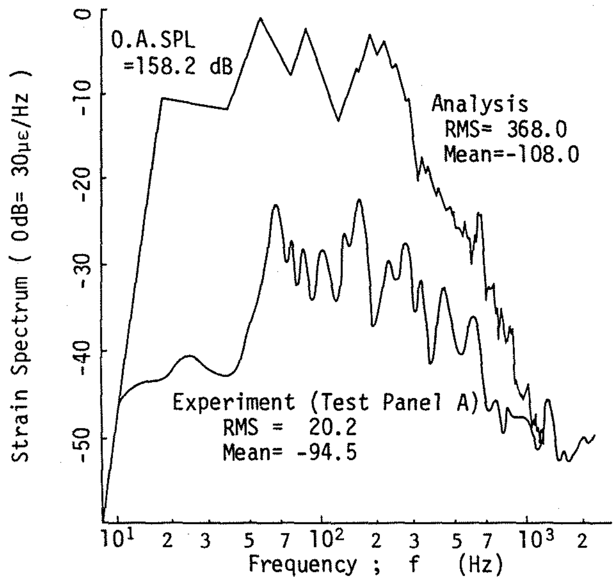

Figure 12. Strain Response at 150 °C

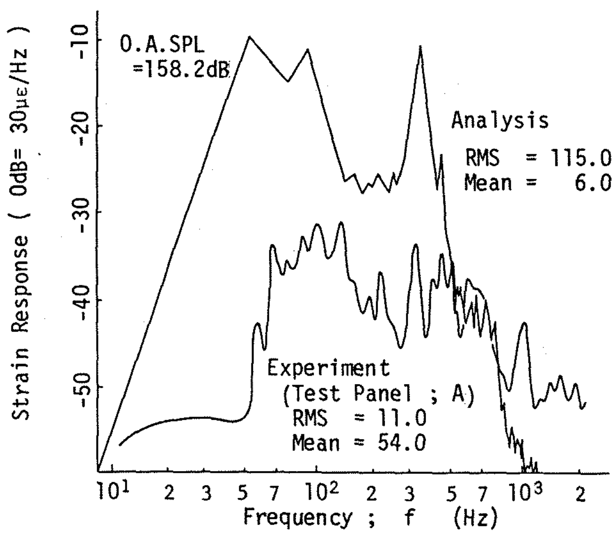

Figure 13. Strain Response at Room Temp.

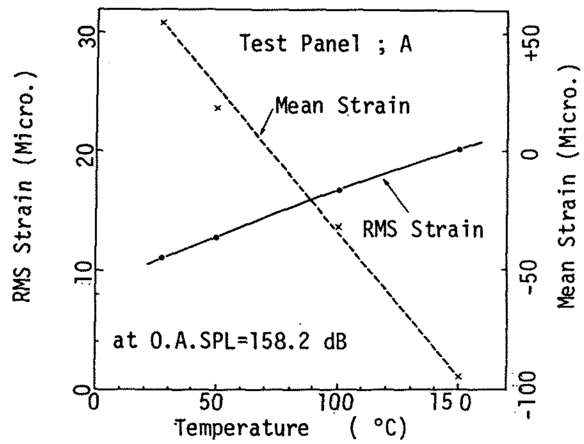

Figure 14. RMS & Mean Strain vs Temperature

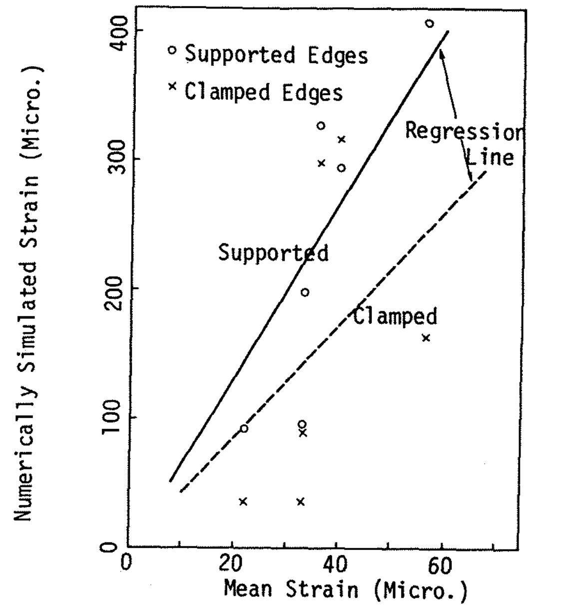

Figure 15. Correlation between Measured Strain and Numerically Simulated Strain at the under Rivet Line.

	Spec. Test			Acceleration Test							Total T
Panel	O.A.SPL (dB)	Temp. (°C)	t_s	Freq. Range	O.A.SPL (dB)	Temp. (°C)	St.G. RMS Ratio	a	t_a	t_e	$t_s + t_e$
A	160 / 163.5	150	71^h 26^m	300-500 (Hz)	156.4	150	100/70	5.3 / 4.66	42^h 05^m	223 / 195	294^h / 266*
B	160 / 159.9	R.T.	71; 26	"	155.2	R.T.	100/40	72 / 84.5	42: 05	3030 / 3551	3100 / 3621*
C	160 / 156.2	150-170	31: 37	600-800	156 / 155.7	150-170	51/41	2.8 / 1.134	12; 30	35 / 14:10	66:37 / 45:47*
D	160 / 158.1	150	22:17	400-630	154 / 154.9	150	90/60	6.6 / 6.97	33:08	218 / 230	240 / 252*
E	160 / 157.3	R.T.	22:17	"	154 / 153.7	R.T.	50/14	380 / 31.6	33:08	12600 / 1043	12600 / 1065*

* : Hour Obtained by Computor Data Processing. a : Acceleration Factor.

Table 2. Acoustic Fatigue Test Hour

In the Accelerated Test, the noise spectrum is adjusted so that the R.M.S. Strain response of the various test panels is pushed up as much as possible. Thus the Accelerated Test hours depend on the R.M.S. ratio. (=strain of Acc. Test/ strain of Spec. Test)

No damage has been found in all test panels.

Finally Table 3 shows the fatigue life estimates by various methods. Although the method Ⅲ gives only one thousand hours of acoustic fatigue life, the method Ⅳ gives the incredibly long acoustic fatigue lives.

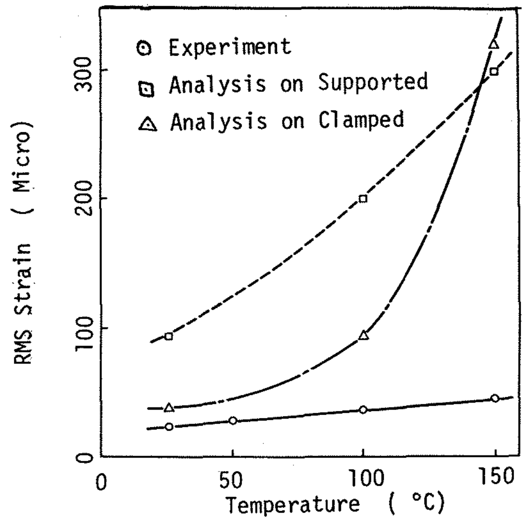

Figure 16. RMS Strain Trend versus Temperature

Ⅶ Conclusion

In spite of insufficient technical data available on acoustic fatigue life estimation at elvated temperatures, the designed flap structures have been successfully proved to possess adequate fatigue strength under the predicted noise and thermal loads. Moreover the panels with reduced thickness have also proved to have sufficient fatigue strength.

The preliminary experiments conducted prior to the fatigue test have provided useful information on resonant frequencies, mode shapes, thermal buckling temperatures and dynamic response spectra.

Most of the results have qualitatively good correlation with analytical estimates. However, there exist the quantitative disagreement between the measured response spectra and the numerically simulated ones, which apparently leads to the discrepancy of the two fatigue lives obtained from the respective method.

Acknowledgement

The authoers are indebted to Dr. K. Takeuchi, Head of 1st Airframe Division in NAL, for valuable suggestion during those tests.

		Spec. Test							Acceleration Test						
Test Panel	St.G. No.	Mean με	RMS με	Temp. C	Fatigue Life Estimate I	II	Ⅲ	Ⅳ	Mean με	RMS με	Temp. C	Fatigue Life Estimate I	II	Ⅲ	Ⅳ
A	9	54.0	11.1	R.T.			2.0 +9								
	9	11.8	12.3	49.0											
	9	-32.1	16.6	88.1											
	9	-94.5	20.2	144.6		5.1+5	2.2+4	9.0+10	-202.4	28.1	148.0		1.1+5	8.1+2	
B	2	35.1	11.0	R.T.	4.0+9				-19.2	28.5	R.T.	1.2+7			
	4	-10.1	11.3	"	3.4+9				-20.1	12.9	"	1.5+9		1.8+8	
	7	43.9	32.6	"	5.1+9			7.1+9							
C	2	-213.0	14.6	145.0		1.1+6			-220.0	15.0	148.4		1.0+6		
D	2	-98.0	17.8	147.0		1.0+6			-93.0	27.0	149.0		1.5+5		
E	3	33.0	4.1	R.T.	2.0+12				34.6	8.6	R.T.	2.1+10			
+n $=10^n$ Hours, Method I= NAL TR-620, II = AIAA Paper #73-994, Ⅲ = Simulation+Rheology Model Ⅳ = Experiment + Rheology Model.															

Table 3. Acoustic Fatigue Life Estimate

Reference

(1) E.S.D. Item 73011, "Damping in Acoustic Excited Structures",Jul. 1973.
E.S.D. Item 67028, "Estimation of the R.M.S. Stress in Skin Panel Subjected to Random Acoustic Loading",Oct. 1972

(2) Harradine, P.J., "Powered Lift -- Its Impact on YC-14 Materials and Structure", AIAA Paper #77-1231.

(3) Schneider, C.W. and Rudder,F.F., "Acoustic Fatigue Resistance of Aircraft Structures at Elevated Temperatures", AIAA Paper #73-994.

(4) Maekawa, S., "On the Sonic Fatigue Life Estimation of Skin Structures at Room and Elevated Temperatures", J. of Sound & Vib., 1982. 80(1), pp 41-59

(5) Maita, M. and Torisaki, T., "Acoustic Characteristics of the External Upper Surface Blowing Propulsive-Lift Configuration", AIAA Paper #1063.

(6) Fujimori, Y., Sano, M., Iida, S. and Egawa, K., "Preliminary Study for Acoustic Fatigue Test at Elevated Temperatures", NAL TM-433. April 1981.