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VIBRATION BASED FATIGUE ANALYSES OF HELICOPTER SUPPORT STRUCTURE WITH DIFFERENT METHODS

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ABSTRACT

In this study, fatigue life calculations were carried out under vibration load of aluminum support carrying an electronic device integrated onto a military helicopter using different prediction methods. Harmonic (frequency) response analyzes have been performed to obtain the transfer functions of the structure. Following this analysis, fatigue analyzes in the time domain and then in the frequency domain were performed for the aluminum carrier. The effect of mean stress in fatigue analyzes was included. Simulation of the shaker test and analysis of the application of the multi-axis vibration were investigated. In the analysis of the shaker test, the vibration data collected on the platform was also accelerated according to the MIL STD 810G military standard so that the damage accumulated remains the same. In the time domain analyzes, classical linear superposition method and modal dynamic analysis methods were applied. The fatigue analysis results were compared and interpreted.

INTRODUCTION

Mechanical systems are usually subject to repeated dynamic loads. Under these loads, it is not necessary for the stress levels to exceed the yield level in order to cause damage to the material. Under variable stress, engineering materials wear off over time. In other words, damage can occur if the structure is exposed to a sufficient number of repetitive loads.

There are two types of fatigue approaches in the literature. The first one is the "safe-life" approach. Determination of fatigue life according to this approach is applied in this study. In this approach on formation damage is not permitted. Since damage is not allowed to occur, the part is replaced when it has not reached its intended life or has reached the specified life. For example, in Sikorsky Blackhawk (S-70) and Apache helicopters, the minimum life span of critical parts for fatigue is estimated at 4500-5000 flight hours. The second approach and the relatively new approach to fatigue is the concept of damage tolerance / "fail-safe" approach. This approach is based on the principle that the damage is determined and controlled by non-destructive inspection methods without reaching a critical level.

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METHOD

There are two basic methods for fatigue calculations using the finite element method. These are methods that operate in the time domain and in the frequency domain. For the time domain signals, rain flow cycle counting methods are used. However, instead of using the cycle counting method in the frequency domain, the signal's Probability Density Functions are used for this purpose (Figure 1). The process used for fatigue calculations in the finite element method should be as outlined in Figure 2 below:

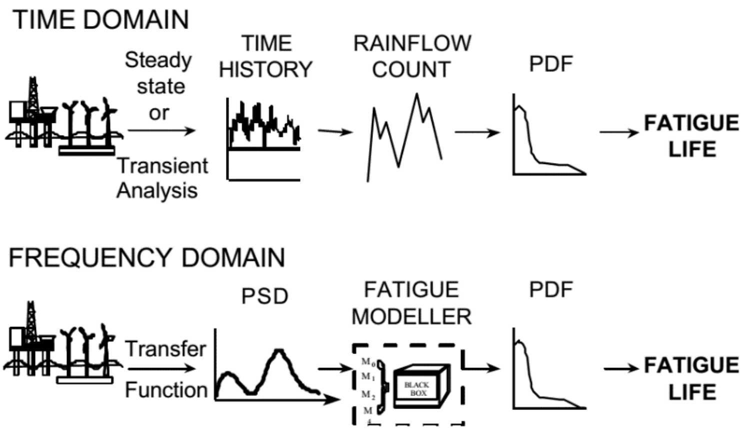

Figure 1: Differences in fatigue analysis in time and frequency domain [Bishop,1999]

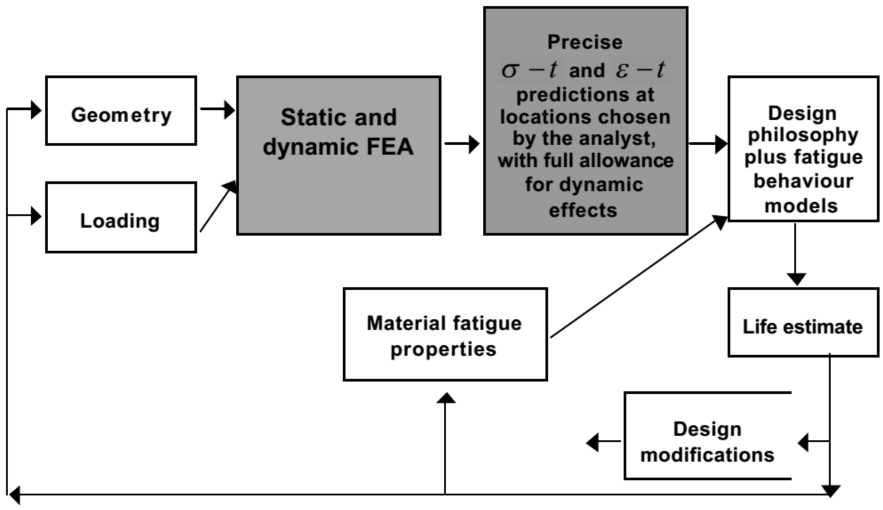

Figure 2: Finite Element Method Fatigue Analyzes [Bishop & Sherratt, 2000]

The fatigue analyzes carried out in the time domain and the frequency domain for the support structure in this study are shown in Figure 3 below. The direct transient dynamic analysis method (time domain transient analysis) is not considered and practiced as a practical method because of the long time required for this analysis. As an alternative to this method, a modal transient dynamic analysis method is proposed.

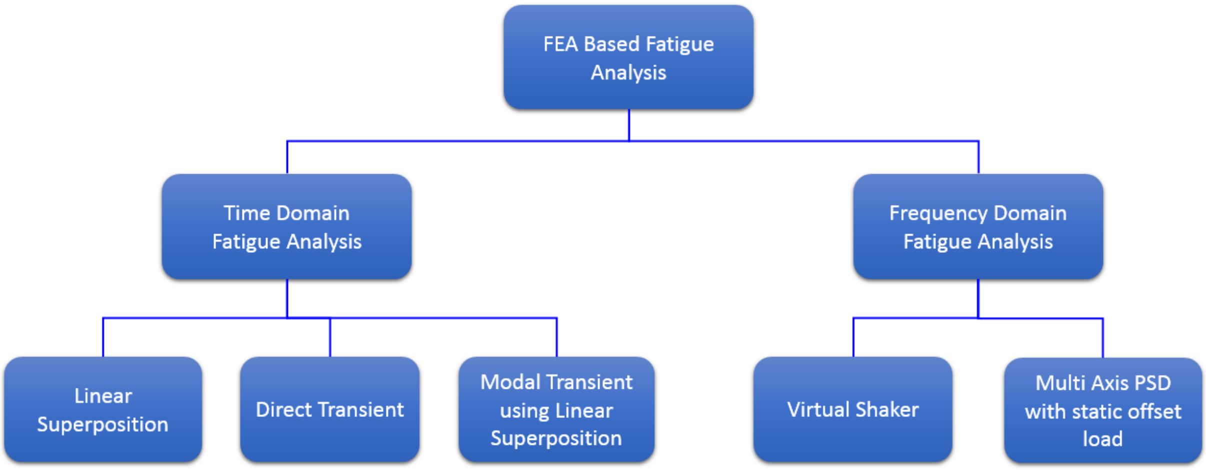

Figure 3: Different fatigue analysis types performed by means of finite element method in the frequency and time domain

APPLICATION

The Aluminum 2024-T3 support structure in this study is designed for a mission equipment integrated onto a military helicopter. Helicopter is shown in Figure 4 below.

Figure 4: Example of integrated helicopter [Sikorsky,2016]

The finite element model of the support was created using Altair Hypermesh software. The finite element model consists of 26551 nodes, 25450 elements and 157242 degrees of freedom. The model has 6 CONM2 point mass elements, 24723 CQUAD shell elements, 721 TRIA3 shell elements and 207 rigid elements (RBE2).

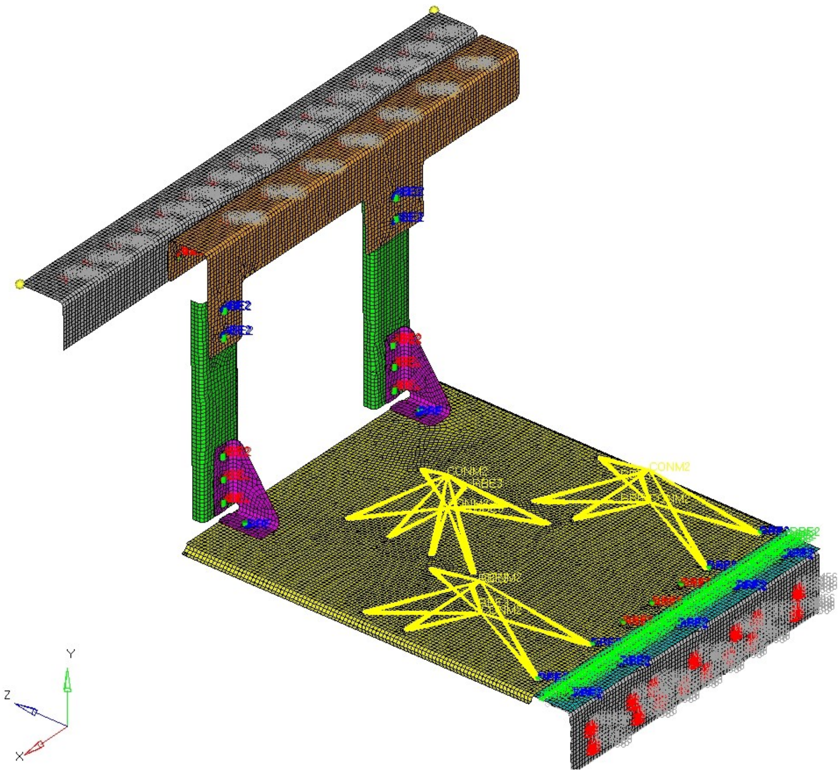

Figure 5: Finite element model of aluminum support carrying electronic equipment

The vibration data used in frequency based fatigue analyzes is defined as power spectral density graph. At the bottom, the vibration spectrum representing the region where the support structure is located on the helicopter is shown. When this spectrum is examined, it is seen that dominant drive frequencies are 12,7 Hz and 17 Hz (blade passage frequency). The natural frequency of the designed support structure must not coincide with these drive frequencies.

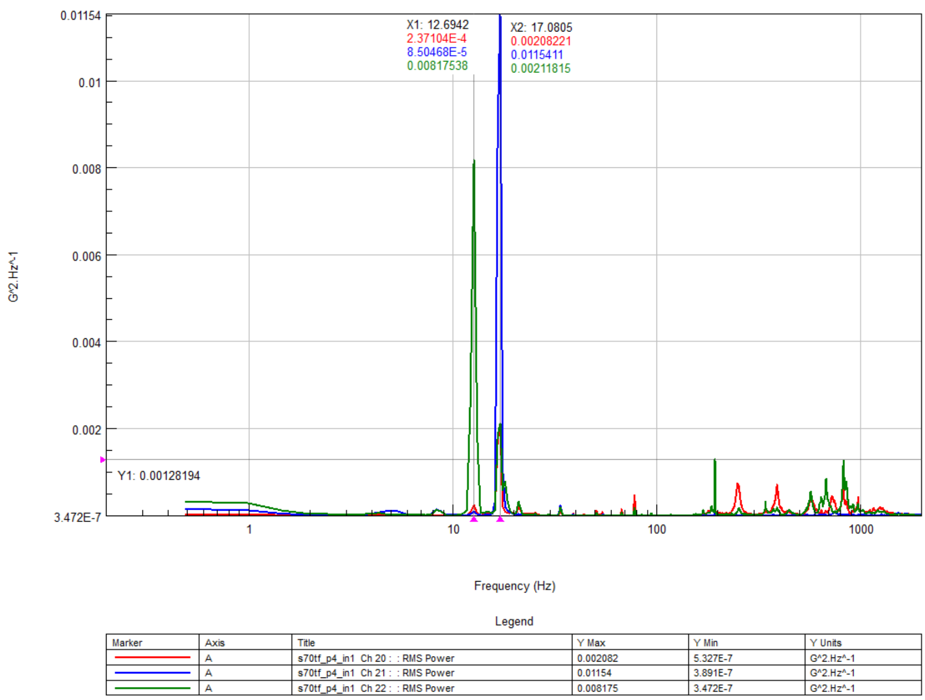

Figure 6: The power spectral density (PSD) of the vibration data measured

Frequency Response Analyses

The way to figure out whether a sufficient number of modes in the frequency response analysis is used to predict the behavior of the structure is to look at the total effective mass for each degree of freedom (Table 1). The total effective mass table shows the active masses at 6 degrees of freedom for each mode. Ideally, the sum of the modal effective masses at each degree of freedom is at least 90% of the total mass.

Table 1: Modal Effective Mass

MODAL EFFECTIVE MASS FOR SUBCASE 1
RIGID BODY MODES BASED ON REFERENCE POINT AT ORIGIN OF BASIC COORDINATE SYSTEM

Mode	Frequency	X-TRANS	Y-TRANS	Z-TRANS	X-ROTAT	Y-ROTAT	Z-ROTAT
1	2.874E+01	5.129E-08	1.985E-03	1.752E-05	1.480E+02	4.741E-03	1.081E-01
2	4.428E+01	9.831E-05	3.898E-07	1.126E-09	1.621E-03	9.057E+00	4.163E+00
3	5.944E+01	3.401E-08	5.680E-05	4.170E-06	1.917E+01	4.112E-02	9.609E-02
4	9.647E+01	3.692E-05	3.444E-05	1.095E-04	2.880E+01	7.748E-01	1.196E+01
5	1.058E+02	4.954E-07	7.639E-07	2.295E-06	5.080E-01	2.881E+00	2.570E-02
6	1.113E+02	1.580E-03	3.227E-07	9.402E-07	8.044E-04	4.990E+01	5.238E+02
7	1.300E+02	1.076E-05	1.790E-04	9.130E-05	1.096E+02	2.375E-01	3.634E+00
8	1.550E+02	9.586E-05	1.742E-07	2.385E-08	4.254E-02	8.486E-03	2.133E+01
9	2.041E+02	3.103E-11	4.072E-05	2.152E-03	5.883E+02	8.804E-02	6.645E-03
10	2.680E+02	3.270E-06	7.272E-07	2.954E-05	4.023E+00	5.711E-01	1.594E+00
11	2.829E+02	1.882E-06	1.422E-05	2.784E-05	5.649E+00	9.264E-01	1.981E-01
12	2.958E+02	6.304E-07	8.265E-07	1.397E-05	1.403E+00	5.882E-03	4.206E-01
13	3.041E+02	1.342E-05	8.931E-07	9.898E-06	1.968E+00	3.744E+00	4.605E+00
14	3.210E+02	9.722E-08	2.459E-05	2.044E-05	5.940E+00	1.079E-01	5.718E-03
15	3.836E+02	3.472E-07	1.540E-06	4.495E-06	1.884E+00	4.860E-03	1.180E-03
16	4.170E+02	2.967E-08	1.615E-05	1.126E-05	1.476E+00	1.514E-02	4.318E-02
17	4.335E+02	6.605E-07	3.299E-06	1.365E-06	5.412E-01	2.110E-01	1.330E-01
.	.	.	.	.	.	.	.
.	.	.	.	.	.	.	.
52	1.402E+03	7.825E-05	7.055E-07	3.160E-07	1.512E-02	9.582E+00	2.480E+01
53	1.431E+03	2.984E-06	3.175E-06	4.454E-07	3.320E-02	3.665E-01	9.759E-01
54	1.465E+03	1.367E-05	7.976E-07	3.403E-07	2.757E-02	1.877E+00	4.487E+00
55	1.488E+03	3.659E-05	3.294E-06	3.427E-07	1.087E-02	5.106E+00	1.294E+01
56	1.518E+03	8.155E-06	2.314E-06	7.447E-07	4.205E-02	1.151E+00	2.442E+00
57	1.588E+03	5.687E-06	1.825E-06	2.080E-08	1.873E-04	5.360E-01	1.299E+00
58	1.595E+03	2.847E-06	6.953E-06	2.198E-07	6.527E-03	1.377E-01	1.057E+00
59	1.605E+03	7.427E-07	2.913E-06	2.526E-07	1.872E-02	2.561E-02	2.416E-01
60	1.650E+03	4.091E-06	2.737E-06	5.454E-08	1.336E-04	7.610E-01	2.237E+00
61	1.685E+03	7.751E-07	9.502E-07	7.729E-07	4.904E-02	4.637E-02	1.273E-01
62	1.734E+03	6.688E-06	2.698E-06	2.898E-07	1.657E-02	7.082E-01	1.461E+00
63	1.781E+03	1.069E-08	6.837E-06	2.491E-06	1.745E-01	6.870E-02	4.388E-02
64	1.799E+03	2.224E-08	5.064E-08	4.699E-07	3.434E-02	4.556E-03	4.515E-04
65	1.834E+03	4.720E-11	5.381E-07	6.224E-08	5.907E-03	1.792E-02	7.817E-05
66	1.848E+03	1.467E-06	9.788E-07	8.445E-07	4.066E-02	2.658E-01	5.444E-01
67	1.855E+03	6.903E-07	9.075E-07	2.257E-06	1.177E-01	9.307E-02	1.590E-01
68	1.862E+03	5.012E-08	2.340E-06	1.162E-06	4.542E-02	7.596E-03	1.927E-03
69	1.887E+03	1.742E-08	2.374E-08	6.851E-08	6.019E-03	2.984E-03	1.560E-02
70	1.914E+03	8.194E-07	5.233E-09	3.822E-09	1.764E-05	1.051E-01	3.202E-01
71	1.938E+03	8.393E-08	2.808E-09	6.699E-11	2.580E-03	6.202E-03	3.869E-02
72	1.960E+03	2.182E-11	1.258E-07	3.196E-07	9.230E-02	2.333E-02	4.762E-03
SUBCASE TOTAL		2.439E-03	2.549E-03	2.626E-03	9.374E+02	1.525E+02	7.710E+02

In this example, it is not practically possible to increase the percentage of the X-axis further, so an 86% value is considered sufficient. In this case, it is appropriate to include the first 72 modes in the frequency response analyzes (Table 2).

Table 2: Total Mass Ratios of Modal Effective Masses

Number of Modes	X axis	Y axis	Z axis
72	% 86	% 90	% 92
100	% 86	% 94	% 92
115	% 86	% 95	% 92

Accelerated Vibration Test Data

Vibration profiles that are applied in endurance tests performed with the shaker are accelerated test profiles with equivalent damage. For this reason, it is necessary to accelerate the vibrational loads measured on the aircraft before the fatigue analyzes are performed. In this study, the empirical method specified in the MIL STD 810G standard was used for the exaggeration. Accordingly, there is a relationship shown below in Equation 1 between the laboratory test duration and the service life of the part to be tested.

$$\frac{t_2}{t_1} = \left[\frac{W(f)_1}{W(f)_2}\right]^{m/2} \tag{1}$$

In this equation,

t_2: Time corresponding to fatigue life (flight time)

t_1: Laboratory test duration

$W(f)_1$: The amplified (endurance) power spectral density for laboratory tests g^2/Hz

$W(f)_2$: The (operational) power spectral density at service conditions g^2/Hz

m: Coefficient associated with the fatigue life curve

Here the ratio $E = \frac{W(f)_1}{W(f)_2}$ is called the "exaggeration factor". It is recommended that this value does not exceed 2 for random vibration loads, according to the MIL STD 810G standard. Values of 5 to 8 are used generally for m. However, in the same standard it is also recommended to use a value of 6 for m for helicopters (Category 14). Furthermore it is stated that the laboratory vibration profiles given here (4 hours / axis) represent 2500 flight hours [MIL STD 810 G, 2008]. Accordingly, the exaggeration factor obtained is 8.55 (Table 3). The accelerated laboratory test time obtained for this exaggeration factor is 19.2 hours.

Table 3: Exaggeration factor used to accelerate vibration data for laboratory environment

	E
810 G General	2
810 G Rotary wing Aircraft	8.55
LALANNE [Lalanne,1997]	4

Fatigue Analyzes Performed with Simultaneous Application of Vibration Loads (Multi-axial)

It is important to understand the relationship of the axes to each other when the vibration loads are applied simultaneously (the actual state). The way of establishing this relationship is to obtain Cross Power Spectrum which contains phase information between inputs.

Equation 2 below can be used for n number of loads in the general case. Where $W_{ab}(f)$ is the spectrum of the vibration load (PSD and cross power spectral densities), *H(f)* is the transfer function, and $G_{zz}(f)$ is the stress response spectrum

$$G_{zz}(f) = \sum_{a=1}^{n}\sum_{b=1}^{n} H_a(f).H_b^*(f).\mathrm{W}_{ab}(f) \qquad (2)$$

If a triaxial accelerometer data is considered, Equation 2 yields Equation 3

$$G_{zz}(f) = \sum_{a=1}^{3}\sum_{b=1}^{3} H_a(f).H_b^*(f).\mathrm{W}_{ab}(f) \qquad (3)$$

$$= \sum_{a=1}^{3}\left[H_a(f).H_1(f).\mathrm{W}_{a1}(f) + H_a(f).H_2(f).\mathrm{W}_{a2}(f) + H_a(f).H_3(f).\mathrm{W}_{a3}(f)\right]$$

$$= \left[H_1(f).H_1(f)^*.\underline{\mathrm{W}_{11}(f)} + H_1(f).H_2(f)^*.\underline{\mathrm{W}_{12}(f)} + H_1(f).H_3(f)^*.\underline{\mathrm{W}_{13}(f)}\right] +$$
$$\left[H_2(f).H_1(f)^*.\underline{\mathrm{W}_{21}(f)} + H_2(f).H_2(f)^*.\underline{\mathrm{W}_{22}(f)} + H_2(f).H_3(f)^*.\underline{\mathrm{W}_{23}(f)}\right] +$$
$$\left[H_3(f).H_1(f)^*.\underline{\mathrm{W}_{31}(f)} + H_3(f).H_2(f)^*.\underline{\mathrm{W}_{32}(f)} + H_3(f).H_3(f)^*.\underline{\mathrm{W}_{33}(f)}\right]$$

Time Domain Fatigue Analyzes

<u>Linear Superposition Method:</u> The classical linear superposition method is one of the classical fatigue analysis methods performed in the time domain. In this method, static analysis is first performed with a unit load of 1g (vibration-based fatigue is examined in this example). Then, the results of the stresses obtained for the unit load (stress transfer function) are multiplied by the vibrational loads and the stress results for each of the elements on the finite element model are obtained with respect to time. The histogram of the stresses is then subtracted using the rainflow cycle counting method. In the last stage, Miner's linear damage theory is used to obtain the damage per second or the corresponding lifetime information interms of repeats (Fig. 7). Absolute maximum principal stress is usually used in fatigue life calculation.

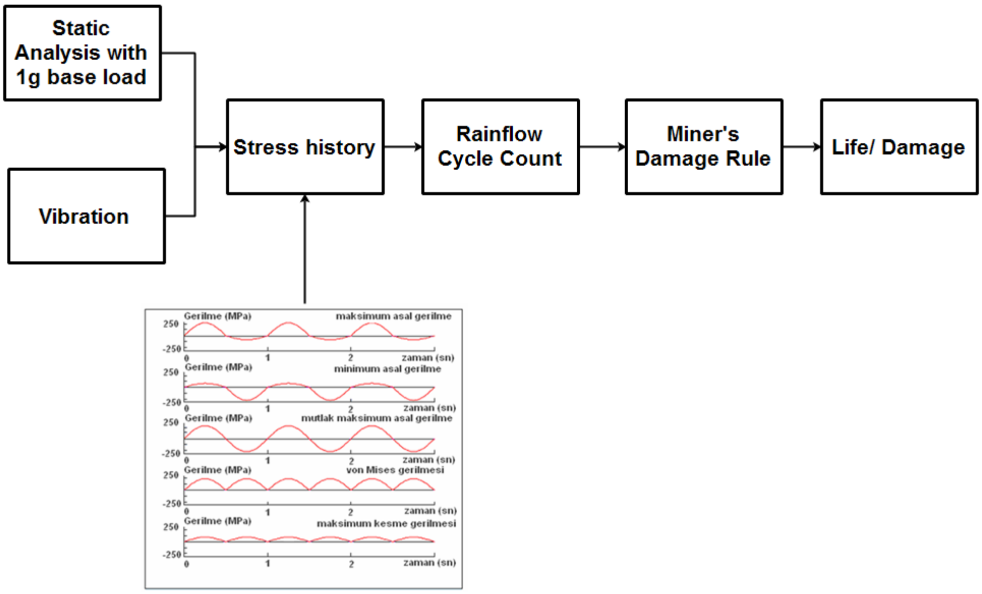

Figure 7: General flow of fatigue method in time domain

If the loading is multi-axis, the analyzes given in Fig. 7 are repeated for each three axes, and the obtained stress values are summed before the fatigue life calculation (Fig. 8).

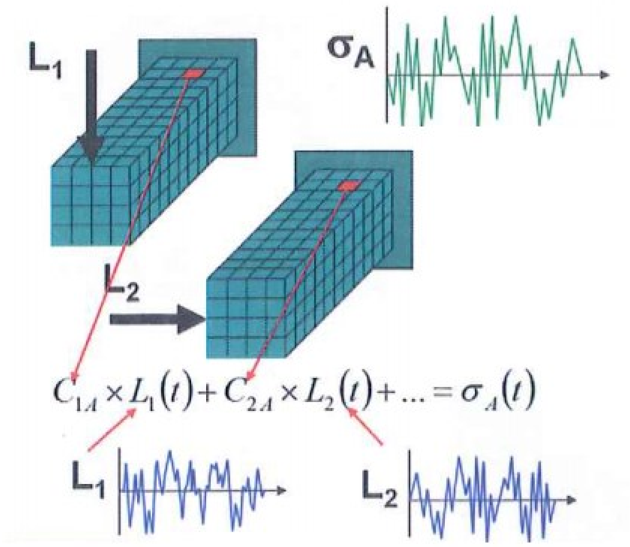

Figure 8: The accumulation of different loads in the time-domain fatigue method [ncode, 2010]

The data from two different flight tests performed in this study were used. Resultant acceleration values (Figure 10) were obtained by adding inertial acceleration loads measured from the center of gravity of the helicopter to the measured vibration accelerations in one of the other tests (Figure 9). The purpose of adding vibration accelerations to maneuvering accelerations is to include the effect of the mean stress on the fatigue analyzes

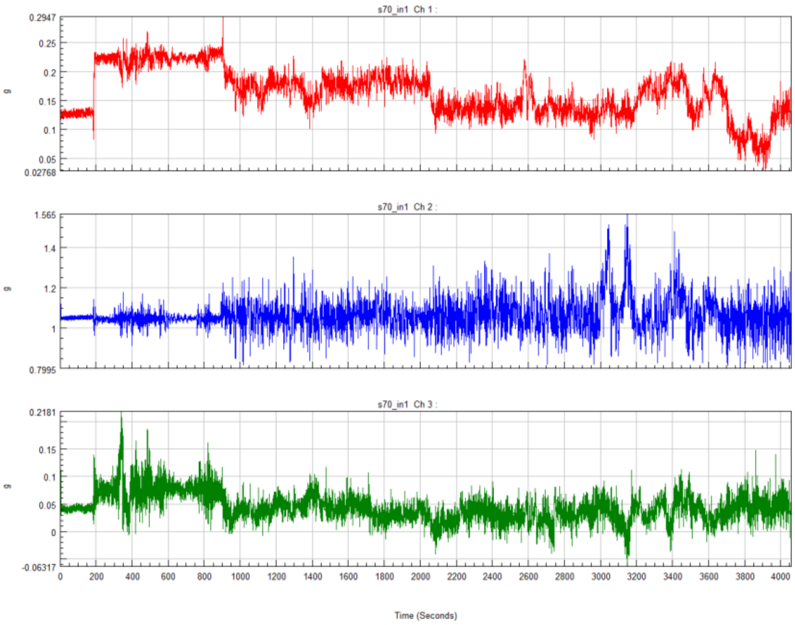

Figure 9: Time-dependent variation of maneuver (static) accelerations measured at the center of gravity of the helicopter at a normal flight (red: longitudinal / blue: vertical / green: sidewards)

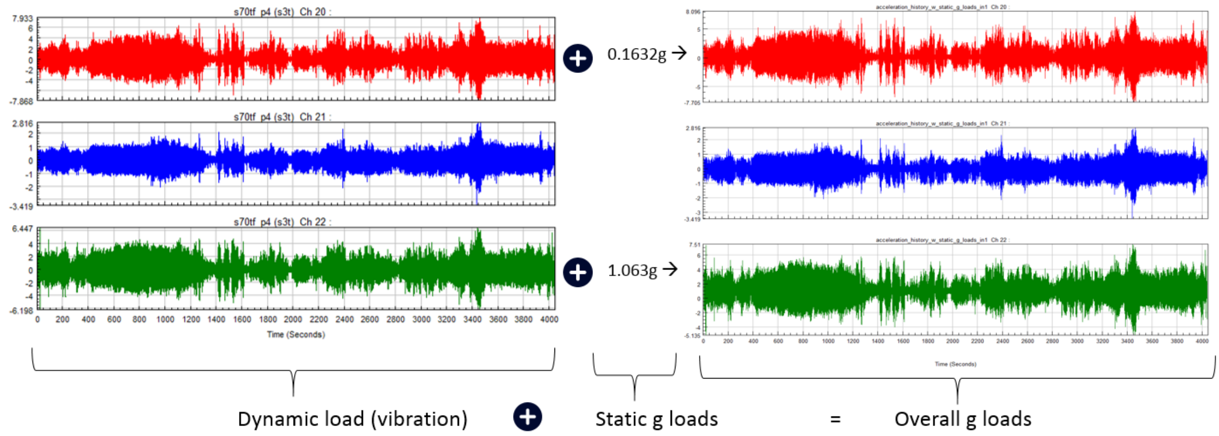

Figure 10: Total acceleration over a normal flight by gathering dynamic and static accelerations

Modal Based Dynamic Analysis Method: The stress values obtained by the linear superposition method may not be very accurate when the potential of driving the natural frequencies of the load is high. In such cases, transient dynamic analyzes must be used in the fatigue calculations in the time domain. In case of using dynamic analysis, there are problems such as solution time in large models and storage of large files. In order to overcome these problems, modal based transient dynamic analysis is considered to be used in fatigue calculations. In this method (Figure 11), the response of the structure can be obtained much more quickly through the structural modes

The terms $\phi_1(\mathrm{t})$, $\phi_2(\mathrm{t}),\ldots,\phi_n(\mathrm{t})$ denoted below are modal participation factors derived from modal based transient dynamic analysis; $\sigma_{1A}, \sigma_{2A}, \ldots, \sigma_{nA}$ are the modal stresses resulting from modal analysis; N is the number of modes used in modal-based transient dynamic analyzes (in this example, the first 72 modes are used).

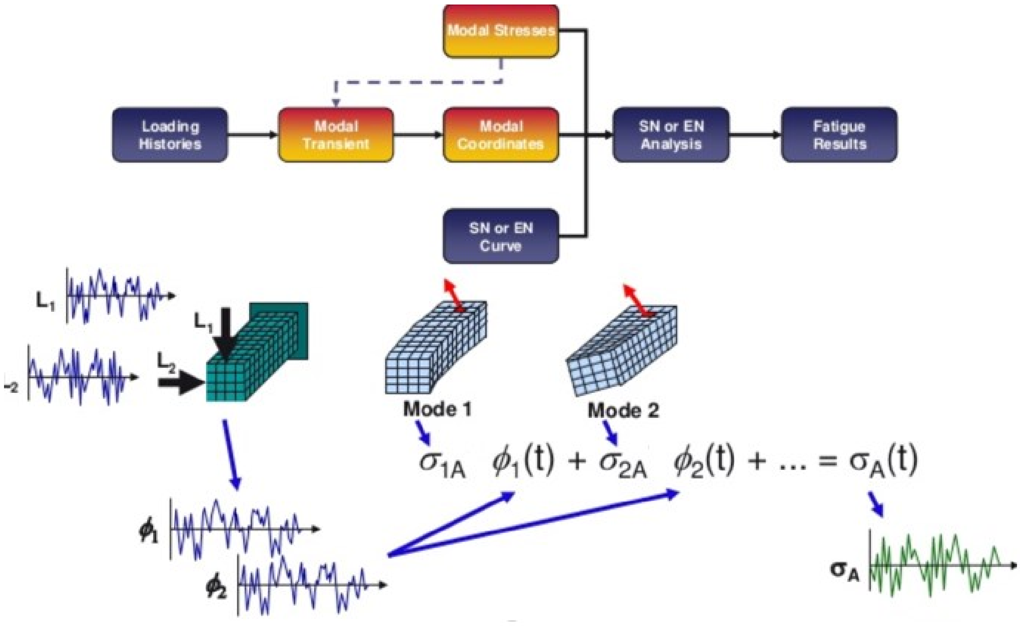

Figure 11: Fatigue Analyzes Using Modal Superposition Method [Vervoort, 2011]

It is possible to shorten a long vibration data before fatigue analysis and to reduce the solution time. In the example shown in Figure 12 below, the signal for fatigue analysis is simplified prior to analysis.

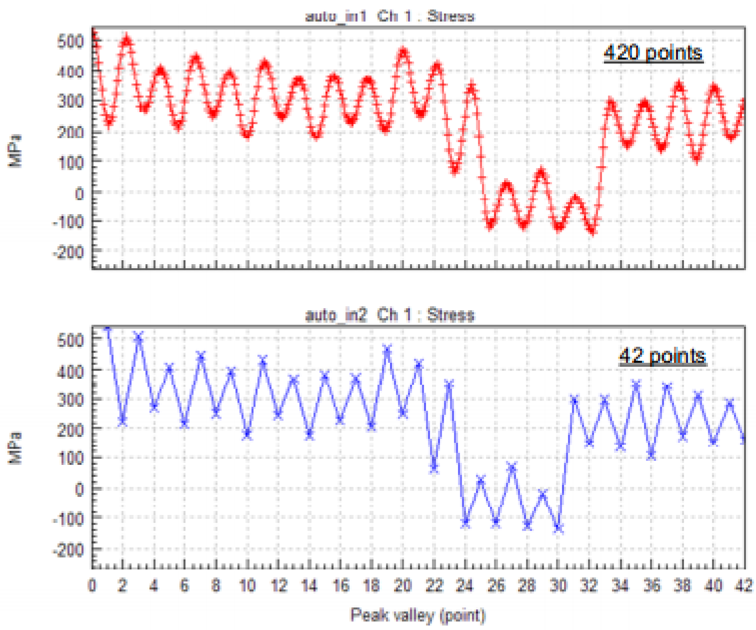

Figure 12: Simplification of load input in time-domain fatigue analysis with "Peak Valley" method [Halfpenny,2006]

The flight test data in this study is about 1 hour and the sampling rate is 4 kHz. It is not possible in practice to use this data in this way even in modal based analysis (4000 seconds * 4000 (data / second = 16 million data points)). For this reason, the 1-minute section in which the highest value of the data exists in the data is cut off and down sampled without removing the low frequency fatigue critical part.

RESULTS AND DISCUSSION

The results of the fatigue analysis carried out in the frequency domain are given in Fig. 13 and Fig. 14, respectively, below. Figure 13 shows the results of the analysis representing the shaker test. As a result of analysis, the lowest lifetime was 3.209E8 repeats, that is, 6.654E13 seconds.

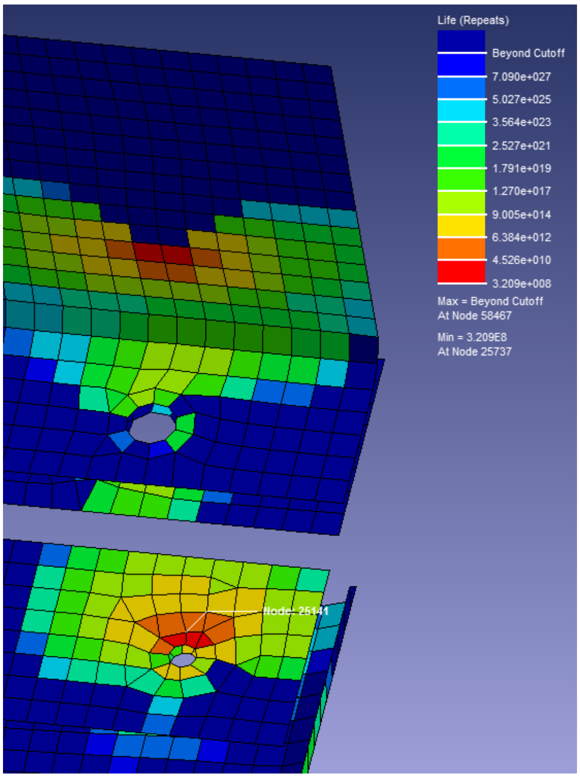

Figure 13: Fatigue analysis results representing the shaker test (frequency domain)

Fig. 14 shows the results of the analysis, which represents the real conditions, that is, the case where the vibration is applied simultaneously. As a result of analysis, the lowest life was obtained at 3.858E17 sec.

In fatigue analyzes performed with the linear superposition method (classical method) in the time domain, the lowest life was obtained as 9.04 E6 repeats, 3.62E10 seconds (Figure 15). In the fatigue analysis using time domain modal-based dynamic analysis, the lowest life was obtained as 9.72 E7 repeats and 5.83E9 seconds (Figure 16).

As a result of four different fatigue analyzes, the most critical points of fatigue were detected in the same regions. Time-domain fatigue analysis results are obtained more conservative than the frequency domain results.

The results of the time domain analysis are obtained closer to each other than those of the frequency domain analysis results. About 10^4 times difference was obtained for the lowest fatigue life between frequency domain analysis results. In addition, for frequency analysis purposes, it is also an interesting result that the shaker test analysis results have a lower life than the real conditions. The reason for this may be that when the loads are applied simultaneously, they might reduce the effects of each other. However, in the shaker test simulations, fatigue analysis for each axis are done separately and the damage effects are summed at the end according to duty cycle defined.

In the time domain analysis, since the natural frequencies of the structure are likely to be driven by the frequency content in the load, the results of modal-based transient dynamic analyzes need to be more reliably relied upon.

In this example, the applied load is shortened based on the highest part of the load. Rather, the time signal can be edited such that damage will be similar which is more accurate approach Moreover, experimental results can be compared with numerical results by performing experimental fatigue analyzes in the time domain with data obtained from strain gage.

Finally, it should be mentioned that the damping is a very important input which affects the results of fatigue analysis. In this case the modal damping is assumed for structure and it is assumed to be 3%. Instead of this assumption, it is suggested that the modal test of the support structure is performed so that damping can be modelled more accurately in the numerical analyses so that numerical fatigue results would be more accurate.

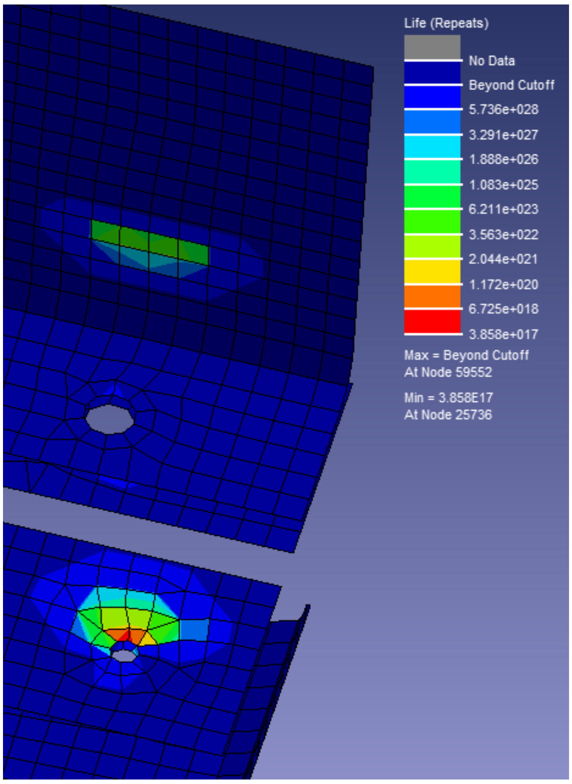

Figure 14: Fatigue analysis results of the multiaxis vibration (real conditions) (frequency domain)

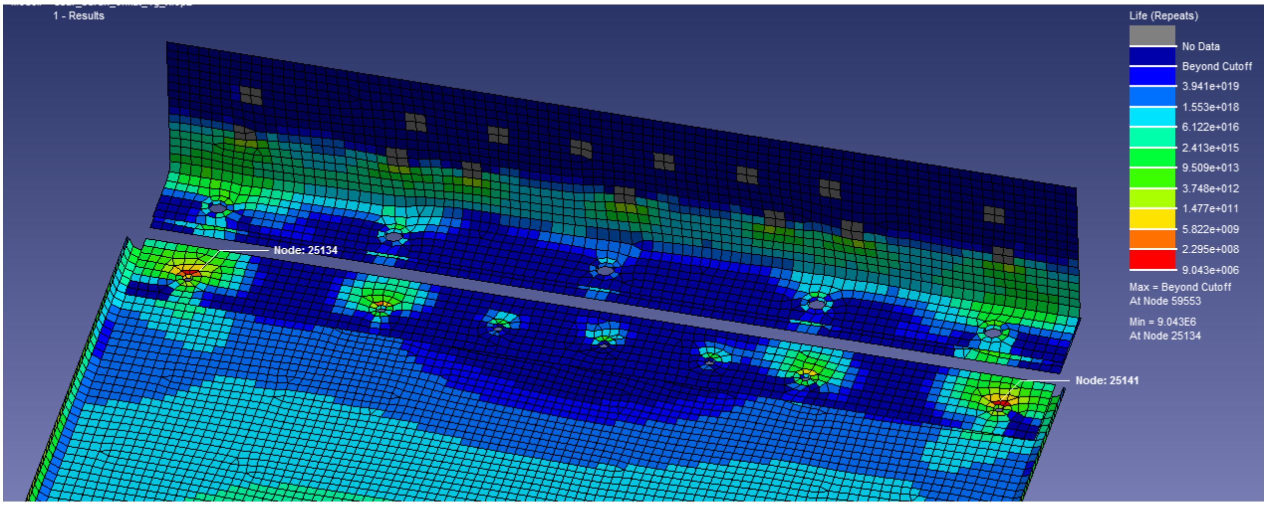

Figure 15: Fatigue Analysis Results of the linear superposition method (time domain)

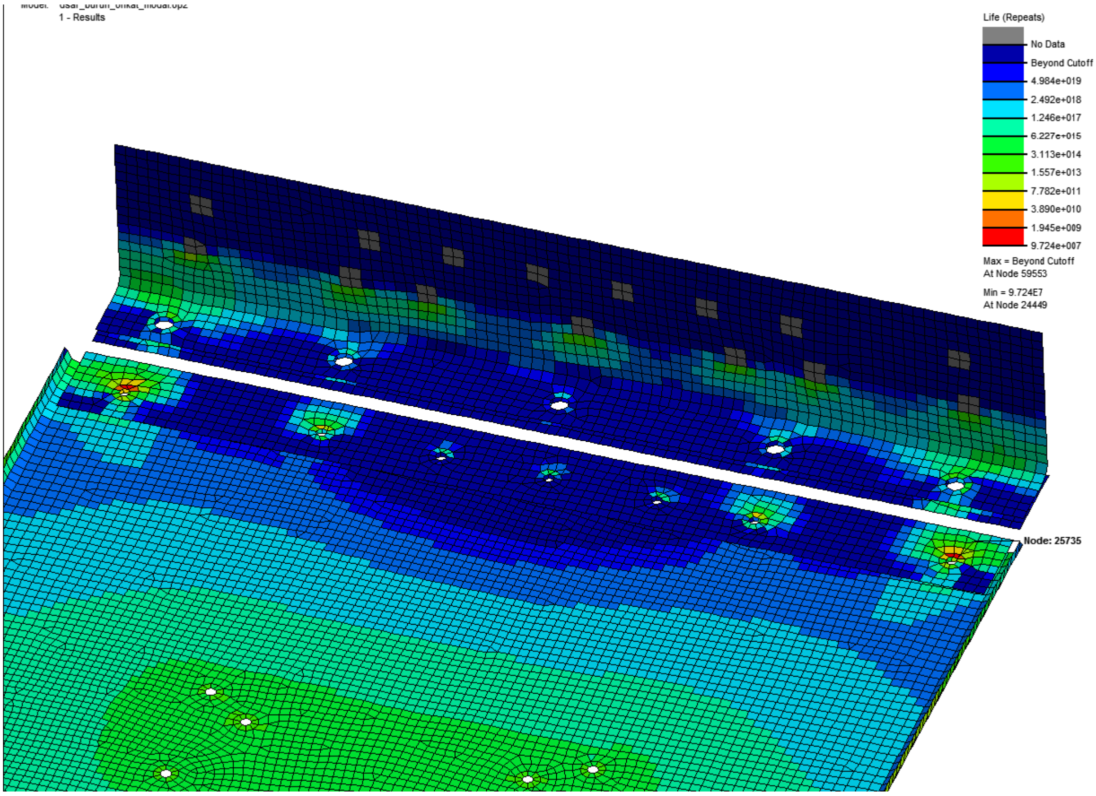

Figure 16: Fatigue analysis results of modal based transient dynamic analysis (time domain) method

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