NVH and Random Vibration Fatigue Analysis of a Landing Gear's Leg for an Un-Manned Aerial Vehicle Using LS-DYNA®

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Abstract

The present work concerns the new capability of LS-DYNA® in solving Noise, Vibration and Harshness (NVH) and fatigue based on Power Spectrum Density (PSD) analysis. These new capability includes random vibration and high-cycle fatigue analysis in a random vibration environment.

In this analysis, the cumulative damage ratio of a landing gear's leg for an Un-Manned Aerial Vehicle (UAV) is computed using material S-N (Stress-Number of cycles) fatigue curve. Dirlik method is used for the analysis of life time as it is proven to provide accurate results for large number of applications, both in automotive and aerospace industry. It is also compared to other methods that have been developed in LS-DYNA® as well. The input acceleration PSD data are provided through measurements.

1 Introduction

Most real fatigue loadings are random processes in respect of frequency and stress amplitude. Estimating fatigue damage with Power Spectrum Density (PSD) was first proposed by Rice [1] in 1954. The stress power spectra density (PSD) represents the frequency domain approach input into the fatigue. This is a scalar function that describes how the power of the time signal is distributed among frequencies [2]. Mathematically this function can be obtained by using a Fourier transform of the stress time history's autocorrelation function, and its area represents the signal's standard deviation. It is clear that PSD is the most complete and concise representation of a random process [3]. Finite element analysis is very efficient in taking the PSD of applied loads and determining the PSD of the resulting stresses at various points in the structure. A method of taking the PSD of stress and calculating fatigue lives therefore has attractions. Much of the early work on fatigue analysis from PSD's was carried out by NASA in order to determine the fatigue damage caused by vibration and buffeting of space vehicles.
Fatigue analysis can be conducted in both time and frequency domain. In time domain, rain-flow counting algorithm is usually used to calculate the number of cycles at each stress and strain levels based on the time history. Due to random load in many cases, a description in frequency domain is more efficient and hence using a statistical method is the most appropriate approach.
There are seven frequency domain fatigue analysis methods have been implemented in LS-DYNA. All of them are based on Palmgren-Miner's rule of cumulative damage ratio:

$$E[D] = \sum_i \frac{n_i}{N_i} \tag{1}$$

Where *E[D]* is the expected damage ratio, n_i is the number of cycles at stress level S_i, and N_i is the number of cycles for failure at stress level S_i, given by material's S-N curve.
To get n_i from the PSD of the random stress response and further compute *E[D]*, a variety of statistical methods have been used to predict Probability Density Function (PDF) and rainflow count, namely: Dirlik, Steinberg, Narrow Band,Wirsching, Chaudhury and Dover, Tunna and Hancock method.

2 Dirlik method

The Dirlik method [4][5], devised in 1985, approximates the cycle- amplitude distribution by using a combination of one exponential and two Rayleigh probability densities. It is based on numerical simulations of the time histories for two different groups of spectra. This method has long been considered to be one of the best methods and has already been subject to modifications, e.g., for the inclusion of the temperature effect, by Zalaznik and Nagode [6]. The rainflow cycle amplitude probability density, p_a estimate is given by:

$$p_a(S) = \frac{1}{\sqrt{m_0}} \left[\frac{G_1}{Q} e^{\frac{-Z}{Q}} + \frac{G_2 Z}{R^2} e^{\frac{-Z^2}{2R^2}} + G_3 Z e^{\frac{-Z^2}{2}}\right] \quad (2)$$

where Z is the normalized amplitude and x_m is the mean frequency, as defined by the author of the method [4]:

$$Z = \frac{S}{\sqrt{m_0}} \qquad x_m = \frac{m_1}{m_0} \left(\frac{m_2}{m_4}\right)^{\frac{1}{2}} \quad (3)$$

For a fatigue analysis the moments up to m_4 are normally used. The even moments represent the variance σ_X^2 of the random process X and its derivatives:

$$\sigma_X^2 = m_0 \qquad \sigma_{\dot{X}}^2 = m_2 \quad (4)$$

and the parameters G1 to G3, R and Q are defined as:

$$G_1 = \frac{2\left(x_m - \alpha_2^2\right)}{1+\alpha_2^2} \qquad G_2 = \frac{1-\alpha_2-G_1+G_1^2}{1-R}$$
$$G_3 = 1 - G_1 - G_2 \qquad R = \frac{\alpha_2 - x_m - G_1^2}{1-\alpha_2-G_1+G_1^2} \quad (5)$$
$$Q = \frac{1{,}25\ (\alpha_2 - G_3 - G_2 R)}{G_1}$$

where α_i is used for spectral width estimation and it has the general form:

$$\overline{D}^{DK} = C^{-1} v_p m_0^{\frac{k}{2}} \left[G_1 Q^k \Gamma(1+k) + \left(\sqrt{2}\right)^k \Gamma\left(1+\frac{k}{2}\right)(G_2|R|^k + G_3)\right] \quad (7)$$

Where $\Gamma\ (\cdot)$ the Euler gamma function.

3 Frequency domain random vibration in LS-DYNA®

The random vibration capability of LS-DYNA® originated from Boeing's in-house vibroacoustic code N-FEARA [7]. This feature computes the dynamic response of structures exposed to vibration or structural-acoustic coupling based on a known source. Various excitations and acoustic environments can be considered, including base excitations, correlated or non-correlated acoustic waves such as plane wave, progressive wave, reverberant wave, turbulent boundary layer, etc. [8]
A new keyword ***FREQUENCY_DOMAIN_RANDOM_VIBRATION** has been introduced in LS_DYNA® to perform random vibration analysis since version 971 R5 [7]. Through the keyword, user provides information about the location, direction, range of frequencies for the random excitation and response area can be given as node, set of nodes, set of segments, or part. The direction of load can be in any of the x, y, z directions or given as a vector by using ***DEFINE_VECTOR**. Load curve IDs for the PSD loads in random computation are also specified under the keyword [9].
The feature of random vibration fatigue is implemented as an option of the keyword ***FREQUENCY_DOMAIN_RANDOM_VIBRATION**, as it is a natural extension of the random analysis procedure. The method for performing fatigue analysis is defined by the parameter **MFTG** in card 1.

An additional card (Card 6) is needed when the ***{FATIGUE}*** option appears in the keyword. This card defines the parts or elements where the fatigue analysis is needed, the material's S-N fatigue curve ID, and some other options. The exposure time is defined by the parameter **TEXPOS** in Card 4.
Since modal analysis is the first step for running this feature, the keywords ***CONTROL_IMPLICIT_GENERAL and *CONTROL_IMPLICIT_EIGENVALUE** must be included in the input. Some other keywords related to implicit solution may also be needed, depending on the type of analysis.
The results are given in binary plot file d3ftg which is accessible to LS-PREPOSR®. Five plot states are included in d3ftg:

State 1: Cumulative damage ratio
State 2: expected fatigue life
State 3: Zero-crossing frequency
State 4: Peak-crossing frequency
State 5: Irregularity factor

Ringeval A. and Huang Y. [9] give in their paper several examples to demonstrate the effectiveness of the random vibration fatigue analysis feature with LS-DYNA®. One of those examples is a simple cantilever aluminium beam [10] subjected to base accelerations. The numerical values are compared with the experimental results and different fatigue failure theories are used to predict fatigue life. That example is considered as a benchmark for the capability of LS-DYNA® to handle random vibration fatigue analysis efficiently as the authors compare their results with those found in the literature on the same subject. Also, the same problem has been simulated with ANSYS® and RADIOSS® BULK and the maximum root mean square (RMS) stress computed by LS-DYNA® is in good agreement with the results obtained by the other commercial software.

4 Landing gear design

The landing gear is a structure that supports an aircraft on ground and allows it to taxi, take off, and land [11]. In fact, landing gear design tends to have several interferences with the aircraft structural design. Nowadays the weight of landing gear has become an important factor. Efforts are being made to reduce the weight of the aircraft and consequently increase the payload. Design of landing gears takes the effect of static and dynamic loads as well as the impact loading. This paper presents analysis for prediction of the service life for a light landing gear with a thickness optimized to only 8 millimeters. This design optimization where carried by Albahkali E. and Alqhtani M. [12]. Dimensions of the landing gear are shown in Fig. 1.

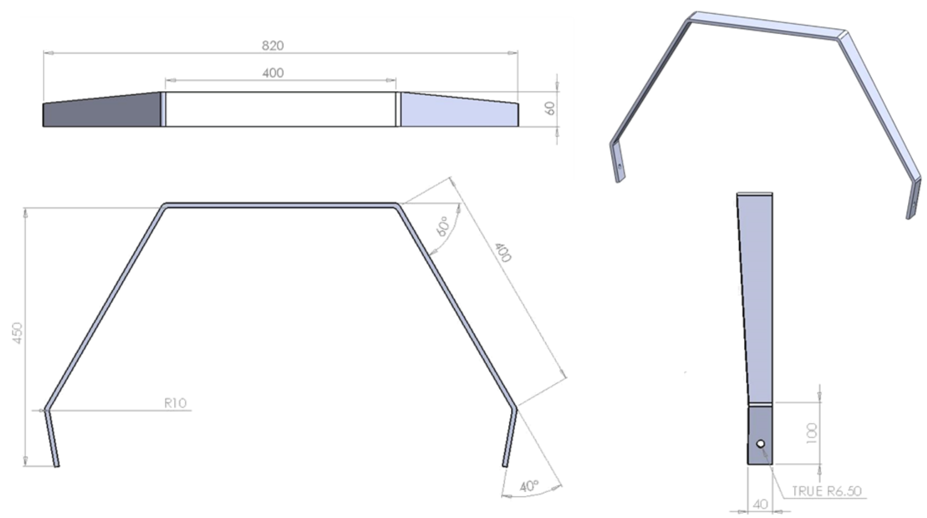

Fig. 1: Geometry details of the landing gear (all dimensions are in millimeter)

As shown in Fig.2, the landing gear is clamped to the fuselage in its middle horizontal part. The two wheels attached to its ends are weighing 0.651 kg each.

Fig.2: Landing gear

During flight, the excitation from propeller engine mainly exerts on the clamped area in the vertical direction. The measured input acceleration PSD is shown in Fig. 3.

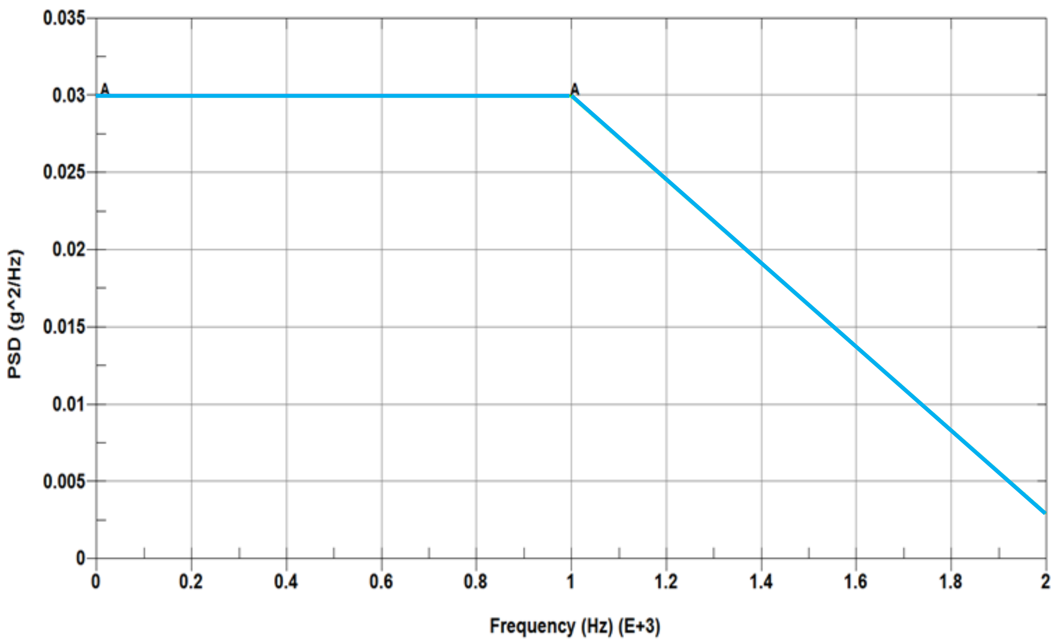

Fig. 3: Input acceleration PSD

The landing gear is made from AA7076-T6 aluminum alloy with density ρ = 2700 kg/m^3, Young's modulus E = 70,000 MPa and Poisson's ratio v = 0.33, its S-N fatigue curve, Fig. 4, is selected according to the European Standard Eurocode 9 [13] with a reference fatigue strength $\Delta\sigma$ = 71 MPa at 2.10^6 cycles and a single inverse constant slope m = 7.0.

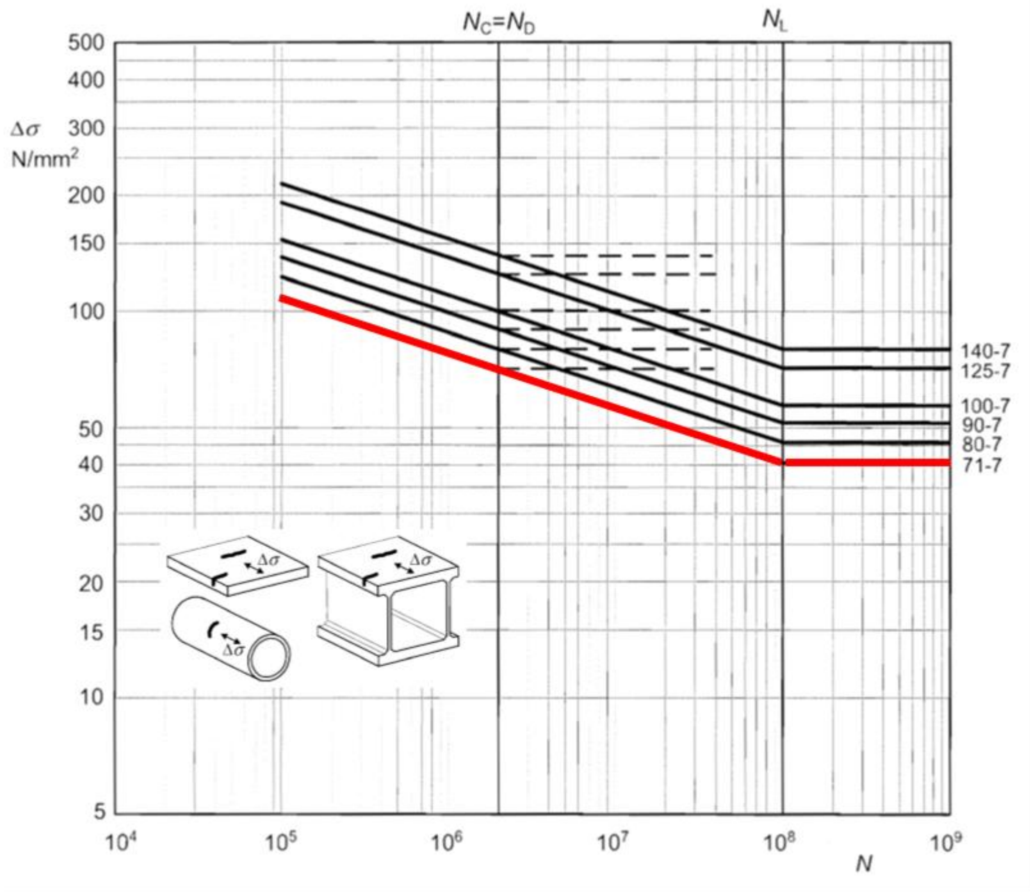

Fig. 4: S-N fatigue curve used for the landing gear material

5 Analysis steps

The model is meshed with 9073 nodes and 8576 shell elements (fully integrated linear DK quadrilateral and triangular shell). The two wheels are represented by 28 mass elements with 0.0465 kg each. They distributed uniformly at the circumference of the two holes. The upper face is subjected to base Acceleration Spectral Density for the range of frequency 0-2000 Hz. A constant damping ratio 0.035 is adopted. The landing gear is exposed to the random vibration load for 45 minutes (2700 seconds) which is equal to the flight time.

The first 10 natural modes are required for the eigenvalue analysis. The keyword ***CONTROL_SHELL, INTGRD=1** is used to apply Lobatto's integration rule with three integration points through the thickness of the landing gear to get the stress on the shell surface. The keyword ***SECTION_SHELL, SHRF,** shear factor which scales the transverse shear stress, is suggested to have a value equal to 5/6 by the developer. For shell element with implicit solution the invariant node numbering, **INN** is equal to 2 in the keyword ***CONTROL_ACCURACY**.

The parameter **SNLIMT** in card 4 is set to 2 to give infinite fatigue life for stress lower than the lowest stress on S-N curve (40.6 MPa).

6 Results

The response stress PSD measured at the critical point shows that two natural frequencies are excited by the input loading in the range 10-2000 Hz (Fig. 5), the first natural frequency (11.711 Hz) and the fifth natural frequency (105.77 Hz). When examine the model with higher input acceleration PSD load (g^2/Hz = 0.1), the eighth natural frequency (287.244 Hz) is excited also. Table 1 summarizes the fatigue life obtained by the numerical predictions methods used in LS-DYNA®.

As shown in Table 1, the results depend on the method used to interpret the RMS results. Steinberg, Chaudhury and Dover and Hancock methods give close results but conservative. Wirsching method gives less conservative results. Tunna's prediction is completely off. Narrow Band's prediction is too conservative comparing to the other methods. Fig. 6 and Fig. 7 show contour plot for the cumulative damage ratio and the RMS of S_x stress at the critical point using Dirlik method respectively.

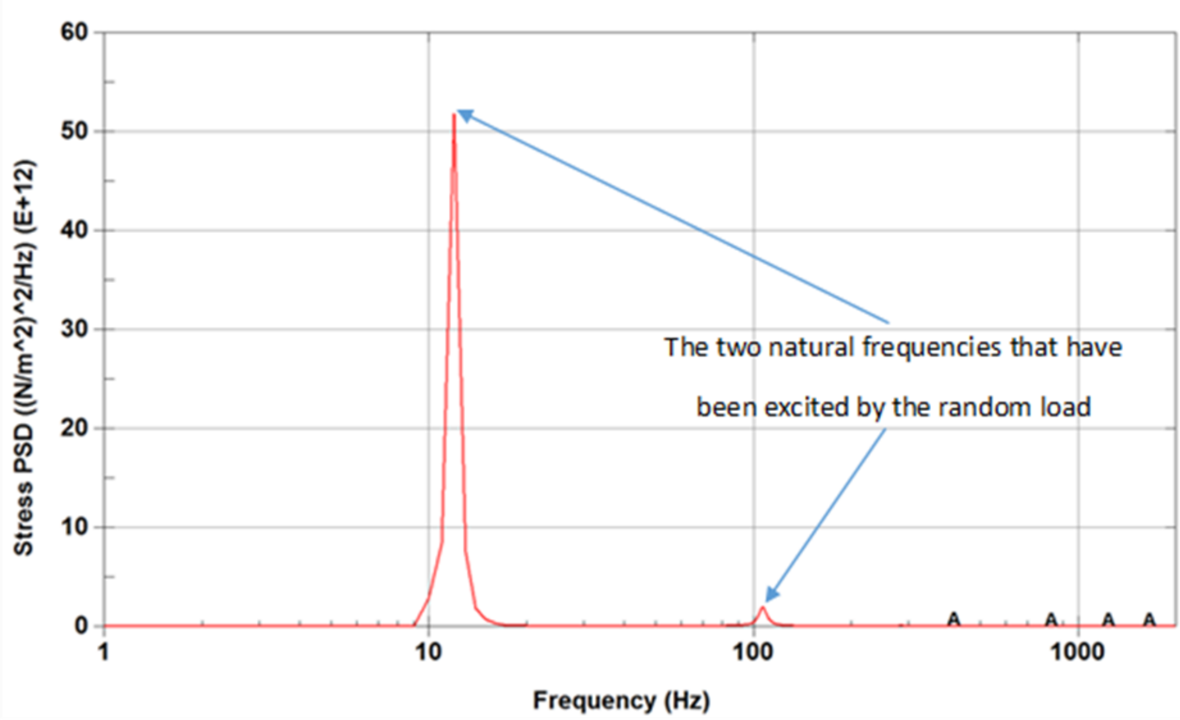

Fig. 5: Stress PSD at critical point

Method used	Fatigue life (Hrs.)	Cumulative fatigue ratio
Steinberg	994.1	0.000754
Dirlik	3037.6	0.000246
Narrow Band	386.7	0.001939
Wirsching	1389.9	0.000539
Chaudhury and Dover	1035.1	0.000724
Tunna	67021.7	0.000011
Hancock	1040.0	0.000721

Table 1: Fatigue life obtained by the numerical predictions methods used in LS-DYNA®

Based on Dirlik method results, the service life for the landing gear will be equal to 3037.6 Hrs. or 4050 flight trips.

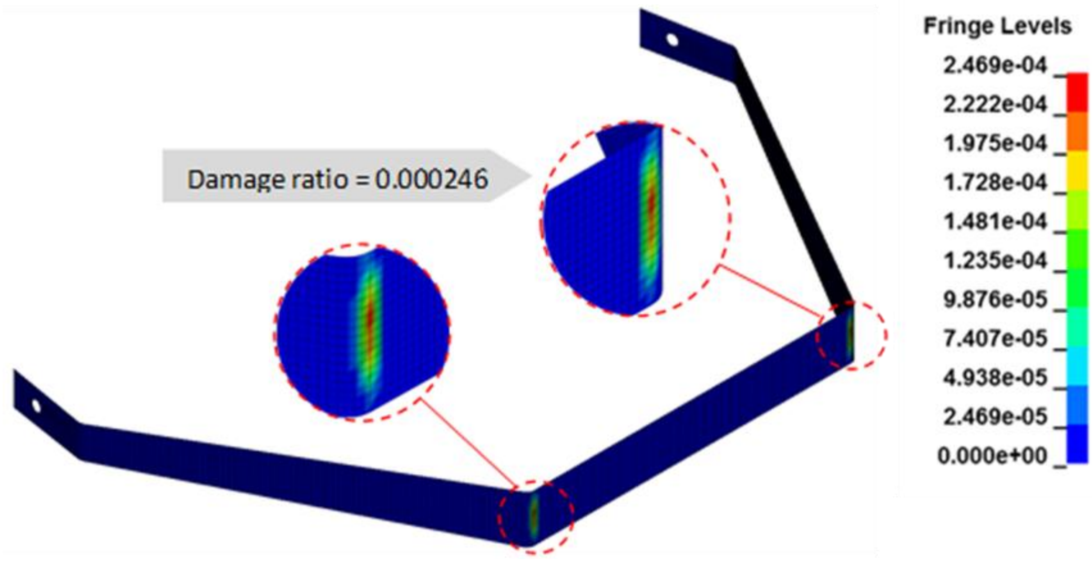

Fig. 6: Cumulative damage ratio by Dirlik method

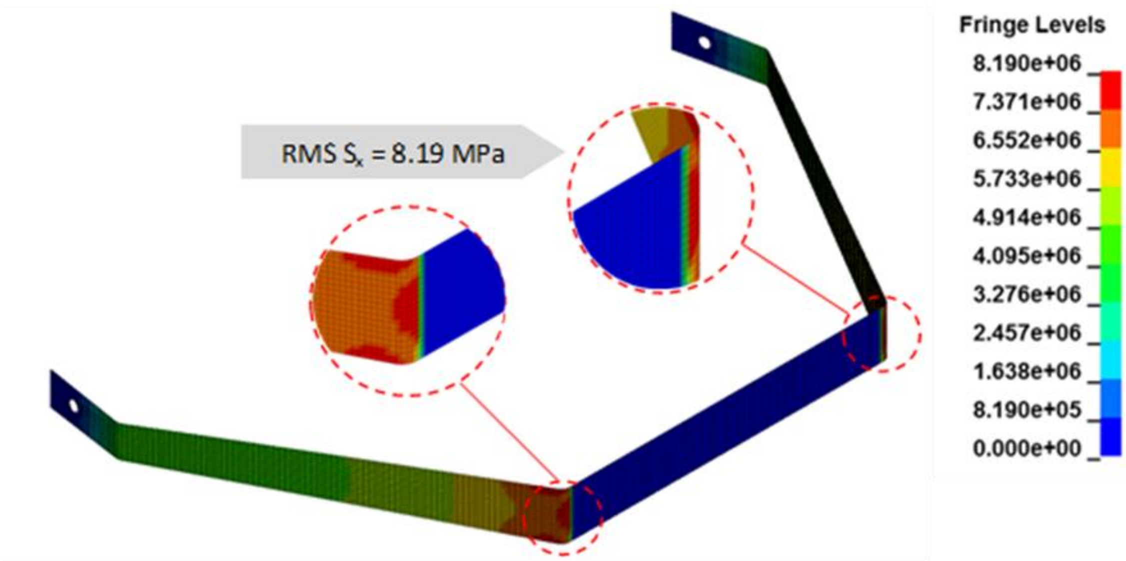

Fig. 7: RMS of S_x stress at the critical point using Dirlik method

7 Conclusion

This paper introduces a new feature of LS-DYNA® for solving frequency domain random vibration fatigue. The feature provides cumulative damage ratio calculation and fatigue life prediction for structures subjected to random vibration excitations based on various theories. Dirlik method which has long been considered to be one of the best methods is demonstrated. The service life for UAV landing gear is predicted using that new feature.

Due to weight constrain, as in the case of the landing gear design, dynamic and static analysis are not enough, it is important to conduct fatigue analysis also to predict the service life.

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