

Multiaxial fatigue analysis with LS-DYNA®

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Abstract

Fatigue life is an important dimensioning criterion within product development. Several tools and software are today available and are widely used for fatigue assessment within the CAE process. To further improve the capabilities for integrated fatigue analysis in LS-DYNA, a time domain fatigue solver has been developed and implemented by LST (an ANSYS company), as a complement to the already existing frequency domain fatigue solvers. As of coming releases of LS-DYNA, different options for fatigue analyses will be available, based on the results from general load cases and structures including e.g. non-linearities, non-proportional and multiaxial loading conditions.

The time domain fatigue analysis can be based on stress or strain results from all time domain solvers (implicit, explicit, thermal, FSI, etc.) in LS-DYNA. The stress or strain state of the elements is usually three dimensional, especially for the parts under multiaxial loading cases like bending or twisting. However, the standard procedure to obtain the SN curve or EN curve is based on nominal stress or strain of the specimen, which is a scalar not a tensor.

*Several options to deal with the multiaxial stress state for fatigue analysis have been implemented in LS-DYNA (keyword *FATIGUE_MULTIAXIAL). They include*

- 1. Running fatigue analysis based on an equivalent stress index (e.g. von Mises stress);*
- 2. Running fatigue analysis on multiple planes and picking the highest damage ratio across the planes as the fatigue damage ratio of the element;*
- 3. Locating a critical plane first and projecting the whole stress history to the critical plane and then running fatigue analysis on the critical plane.*

Several examples are given in this paper, to discuss the different options for multiaxial fatigue analysis, including a crankshaft model and a cylinder bar model with a groove. Validation has been performed by comparing the simulation results from simple test cases to analytical results from the same load cases. Also, a comparison of the fatigue analysis results from LS-DYNA to the results from the fatigue postprocessing module mFAT (a plug-in to the post-processor META) is presented in this paper.

Introduction

Fatigue analysis is critical to the design and optimization of structures and parts involving metal components. To meet the requirements from users for running fatigue and durability analysis, a fatigue analysis module is needed in LS-DYNA.

Starting from R7 version of LS-DYNA 971, a series of frequency domain fatigue analysis methods have been implemented in LS-DYNA, to run fatigue and durability analysis based on random vibration

(*FREQUENCY_DOMAIN_RANDOM_VIBRATION_FATIGUE), and steady state dynamics (*FREQUENCY_SSD_FATIGUE). These methods are based on Miner's rule of cumulative damage ratio and the S-N fatigue curves of the materials (*MAT_ADD_FAIGUE). They are valid for the cases of linear and elastic deformation when stress level is low.

In the case of nonlinear and elasto-plastic deformation, a time domain fatigue analysis is preferred. Recently a new keyword *FATIGUE was implemented to LS-DYNA, to run fatigue analysis in time domain. The time domain fatigue analysis can be based on stress cycle (S-N curve) or strain cycle (E-N curve). This method is based on Rainflow counting of stress or strain cycles. More details of the Rainflow counting can be found in Section 3 of the reference [1], as well as in many other public resources.

The stress or strain state of a material is usually multiaxial. A typical example is a crankshaft assembly subjected to twisting or torque loading. However, the material's SN curve or EN curve is usually obtained by uniaxial testing where the S and E are nominal stress and strain. To apply the uniaxial SN or EN curve on the multiaxial fatigue analysis, some manipulation or simplification of the stress tensor is required. In LS-DYNA, several options for multiaxial fatigue analysis have been implemented. They include

1. Running fatigue analysis based on an equivalent stress index (e.g. von Mises stress);
2. Running fatigue analysis on multiple planes and picking the highest damage ratio across the planes as the fatigue damage ratio of the element;
3. Locating a critical plane first and projecting the whole stress history to the critical plane and then running fatigue analysis on the critical plane

More details of the three options will be given in following sections.

For benchmarking of the new time domain fatigue solver in LS-DYNA, some examples are given in this paper. For two of the examples, the results by LS-DYNA were compared with the corresponding results from the fatigue tool mFAT [2] (a plug in to the META [3] post-processor of BETA CAE). Most of the description and examples are based on using stress for fatigue analysis, but the same techniques can be used for strain-based fatigue analysis too.

Multiaxial fatigue analysis method

Keyword

The multiaxial fatigue analysis is activated by the keyword *FATIGUE_MULTIAXIAL, in addition to the keyword *FATIGUE. More details on the two keywords can be found in the LS-DYNA Keyword Users' manual [4]. The keyword *FATIGUE_MULTIAXIAL has only 1 card:

Card 1	1	2	3	4	5	6	7	8
Variable	MAXIAL	NPLANE						
Type	I	I						
Default	0	18						

VARIABLE**DESCRIPTION****MAXIAL**

Multiaxial fatigue analysis criterion:

EQ.0: fatigue analysis using equivalent stress or strain index
(defined by INDEX in *FATIGUE)

EQ.1: fatigue analysis on multiple planes

EQ.2: fatigue analysis on critical plane which is determined
by the highest 1st principal stress or strain**NPLANE**

Number of planes for fatigue analysis (for MAXIAL = 1 only)

Equivalent stress index method

The first method (MAXIAL = 0) to deal with the multiaxial fatigue problem is to use an appropriate scalar stress invariant in the fatigue analysis. The commonly used scalar stress invariant includes: von Mises stress, maximum principal stress, maximum shear stress, etc. These stress invariants can be computed with the 6 stress components (3 normal stresses and 3 shear stresses). The main issue with this approach is that the change of principal directions of stresses during the transient procedure cannot be considered.

Multiple-plane method

The second method (MAXIAL = 1) to deal with the multiaxial fatigue problem is to project the stress tensor to multiple planes and run an individual fatigue analysis with the normal stress on each plane. After that, the maximum value is picked from the fatigue damage ratios on the multiple planes and this maximum value is designated as the accumulative fatigue damage ratio of that element. The number of planes is defined by NPLANE. The default value of NPLANE is 18, which means there is a 20° angle between two adjacent planes. This method is more applicable to shell elements with plane stress state. Another issue is that this method can be expensive in computational cost, since the fatigue analysis needs to be performed on multiple planes.

Critical plane method

Instead of running fatigue analysis on multiple evenly spaced planes, one can try to locate a critical plane for the whole stress history for each element and project the stress tensor onto this “fixed” plane for the entire time history. Consequently, the fatigue analysis needs to be performed only once on each element and this can gain big saving in computational cost, comparing with the multiple-plane method. A detailed description of the method to locate the critical plane can be found in Section 3 of the paper [1]. This critical plane method is activated by MAXIAL = 2.

Example 1: A plate with predefined notch

Model

The model used in this section is shown in Figure 1. It is composed of 2039 shell elements. Elastic material is assumed for this model. The material properties are shown in Table 1.

Table 1. Assumed material data for the plate model

Material	Density	Young's modulus	Poisson's ratio
Steel	7800 kg/m ³	210 GPa	0.3

As shown in Figure 1, the plate is constrained (6 dof each node) at the edge of the hole and is loaded by cyclic nodal force by one node on the plate. The nodal force lasts for 10000.0 seconds and is defined as

$$F = \sin(22.0\pi t) \quad (1)$$

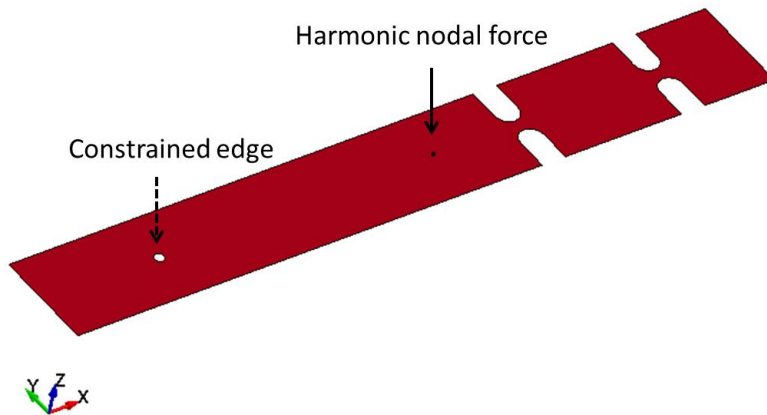


Figure 1. A plate subjected to harmonic nodal force excitation

Due to the out of plane loading, the plate will be subjected to cyclic bending stresses which is three-dimensional. The three different multiaxial fatigue analysis methods were used for this model and fringe plot of their results are shown in Figures 2-4. For the first method (equivalent stress index method), Signed von mises stress was used as the equivalent stress index; For the second method (multiple-plane method), 36, 72 and 180 planes were used separately for the computation of the cumulative damage ratio.

As can be seen from Figures 2-4, the resulting distribution of the cumulative damage ratio from these three different methods are very similar, and the maximum values are close (from 1.369 to 1.410).

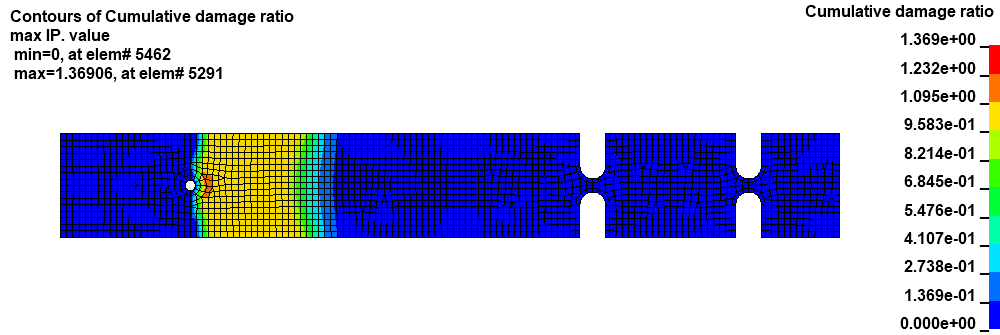


Figure 2. Cumulative damage ratio for the plate, with MAXIAL = 0

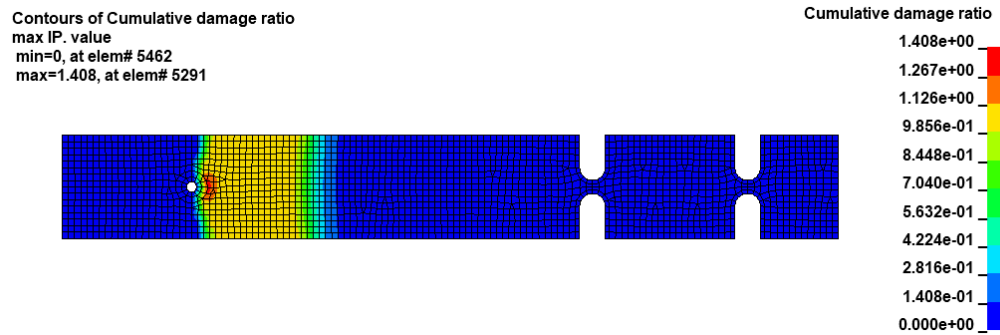


Figure 3. Cumulative damage ratio for the plate, with MAXIAL = 1 and NPLAE = 36

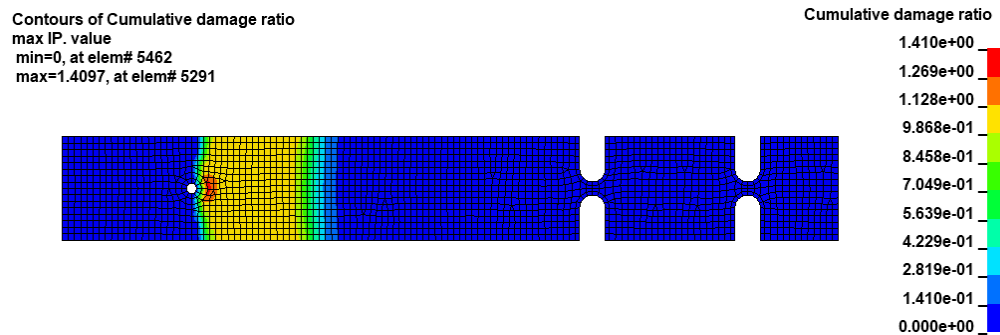


Figure 4. Cumulative damage ratio for the plate, with MAXIAL = 2

The maximum values of cumulative damage ratio (D) in the model, by different methods, are presented in Table 2. The cumulative damage ratios by the critical plane method and the multiple plane method are higher than that by the equivalent stress index method, as the principal stress direction may change with time. Increasing the number of planes when using the multiaxial option (MAXIAL = 1) enhances the solution accuracy and the value of the fatigue damage converges.

Table 2. Maximum cumulative damage ratio D of the plate, by different multiaxial fatigue analysis methods

MAXIAL	NPLANE	Max. D	Element showing max D
0		1.36906	5291
1	36	1.4080	5291
1	72	1.40847	5291
1	180	1.40846	5291
2		1.4097	5291

Example 2: A crankshaft assembly

Model

In this Section, an example of a fatigue analysis of a crankshaft assembly, see Figure 5, is presented. This example has previously [1] been used to demonstrate the capabilities of the fatigue tool mFAT [2], which is a plug-in to the post-processor META [3] of BETA CAE. The crankshaft assembly represents a typical small-size (bore \varnothing 75 mm, stroke 74 mm, total displacement 1.3 L) engine. Only one cylinder is included in the model, and the engine block and cylinder liner are represented by displacement boundary conditions on the crankshaft bearings and the piston, respectively. The FE-model consists mostly of solid elements (1st and 2nd order tets, 1st order hexa). In order to evaluate stresses at the surfaces, the solids were covered with thin membrane elements. Since the model was developed for demonstrational purposes only, a rather coarse mesh was used, and stresses in critical areas would most likely not be satisfactory resolved for a proper fatigue analysis. The total model size is 4E5 elements.

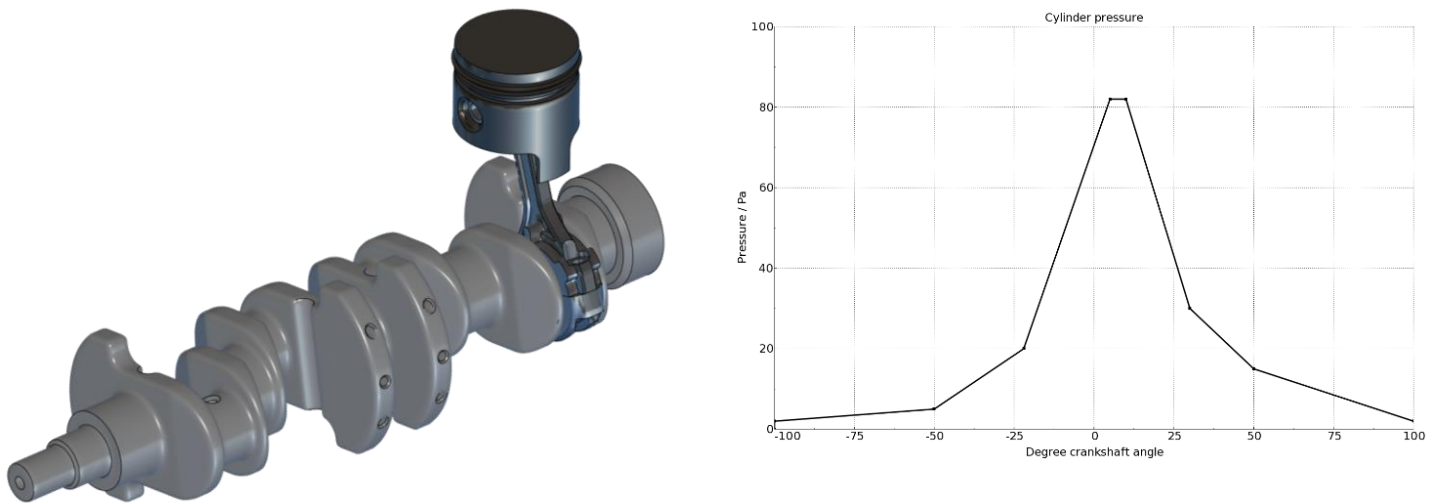


Figure 5. Left image: The crankshaft assembly consists of the crankshaft, conrod, piston pin and piston. Right image: The assumed cylinder pressure curve.

A quasi-static analysis (neglecting crankshaft dynamics) of the work phase (compression and expansion) was performed using the implicit solver of LS-DYNA. A distributed pressure was applied to the top face of the cylinder, following a simplified cylinder pressure curve according to the right image of Figure 5, and a prescribed rotation (totally 200°) was applied to flywheel end of the crankshafts. Contacts in bearings etc. were included using Mortar contacts. Assumed linear elastic material data according to Table 3 was applied to the different parts.

Table 3. Assumed material data for the different parts of the crankshaft assembly

Part	Material	Young's modulus	S _u	N _u	m ₁	m ₂
Crankshaft	Steel	210 GPa	400 MPa	2.E6	5	11
Conrod	Steel	210 GPa	600 MPa	2.E6	6	13
Piston pin	Steel	210 GPa	800 MPa	2.E6	6	13
Piston	Aluminum	70 GPa	100 MPa	1.E6	4	9

The effective stress at two states of the analysis is shown in Figure 6.

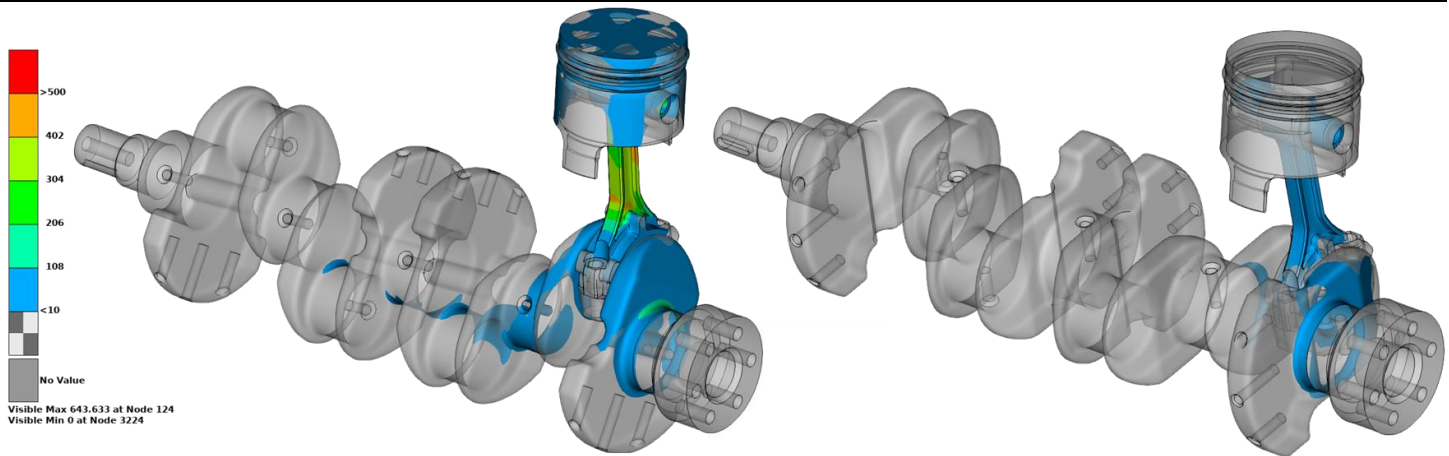


Figure 6. The fringe plots show the effective stress in the crankshaft model at two positions during the analysis, from 10 (blue) to 500 MPa (red). The left image shows the configuration with maximum stress, while the right image shows the configuration during the expansion phase. Areas with stress below 10 MPa are shown as semi-transparent.

Fatigue analysis results

Based on the FE-analysis of the crankshaft motion, fatigue analyses were performed using mFAT and LS-DYNA. In mFAT, a simple critical plane approach is used: it is assumed that the normal to the critical plane, for each element respectively, is given by the direction corresponding to the principal stress with the maximum absolute value during the analysis. Then the stress history is projected onto this direction, and fatigue damage is computed using rainflow count and linear damage accumulation.

In LS-DYNA, the fatigue analysis was activated by the `*FATIGUE_ELOUT` keyword. The multiaxial fatigue option `MAXIAL = 2` on the `*FATIGUE_MULTIAXIAL` – keyword was active, resulting in a similar fatigue evaluation as used in mFAT; fatigue analysis on a critical plane which is determined by the maximum absolute value of the principal stress.

For mFAT, the fatigue data is input using the fatigue limit S_u and the cycle limit N_u and two slopes of the S-N curve, see Figure . LS-DYNA offers more flexibility for assigning fatigue data to materials using either curves or exponents, by the keyword `*MAT_ADD_FATIGUE`.

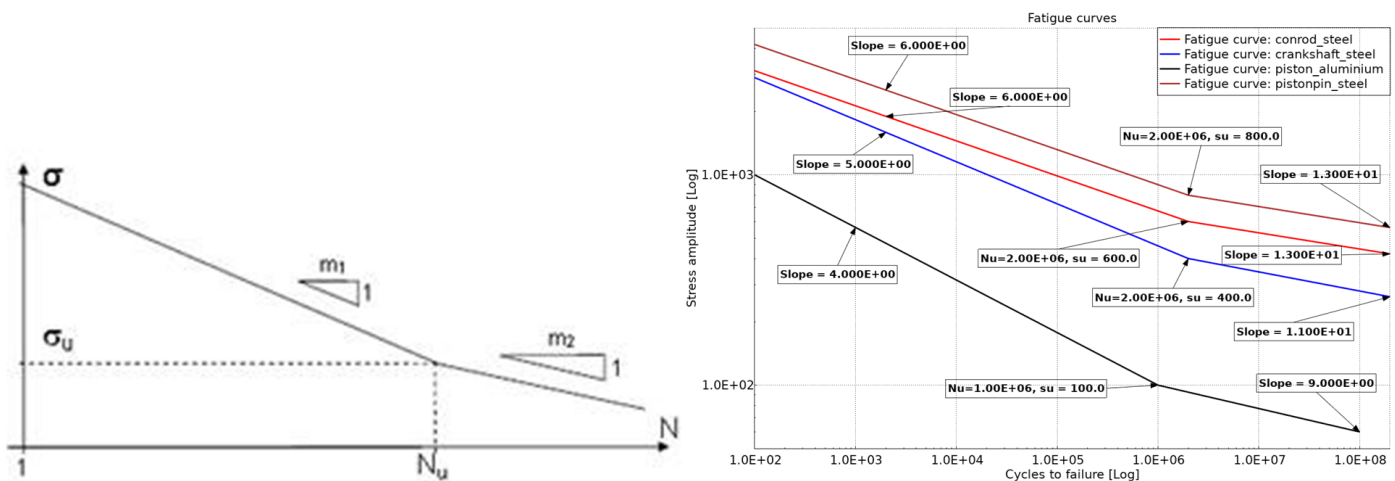


Figure 7. The left image shows the parameters (S_u , N_u , m_1 , m_2) determine the S-N curve used for damage calculation used in mFAT. The right image shows the S-N curves applied for the fatigue analyses of the crankshaft assembly.

In order to be able to compare the fatigue analysis results in this case, the option $LCID=-3$ was used for *MAT_ADD_FATIGUE and the mFAT parameters were converted to the corresponding LS-DYNA input. The fatigue damage results from LS-DYNA and mFAT are compared in Figure 8, Figure 9 and 10. Both methods indicate the same critical areas of each part respectively, and the fatigue damage results are similar in magnitude.

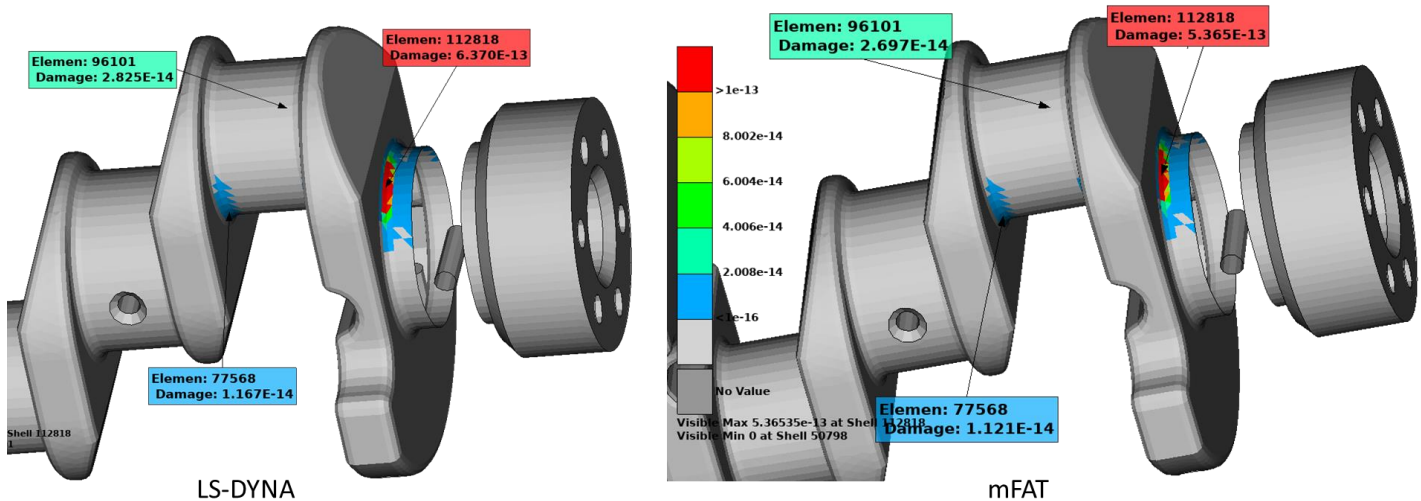


Figure 8. Comparison of the fatigue damage results for the crankshaft from LS-DYNA (left image) and mFAT (right image).

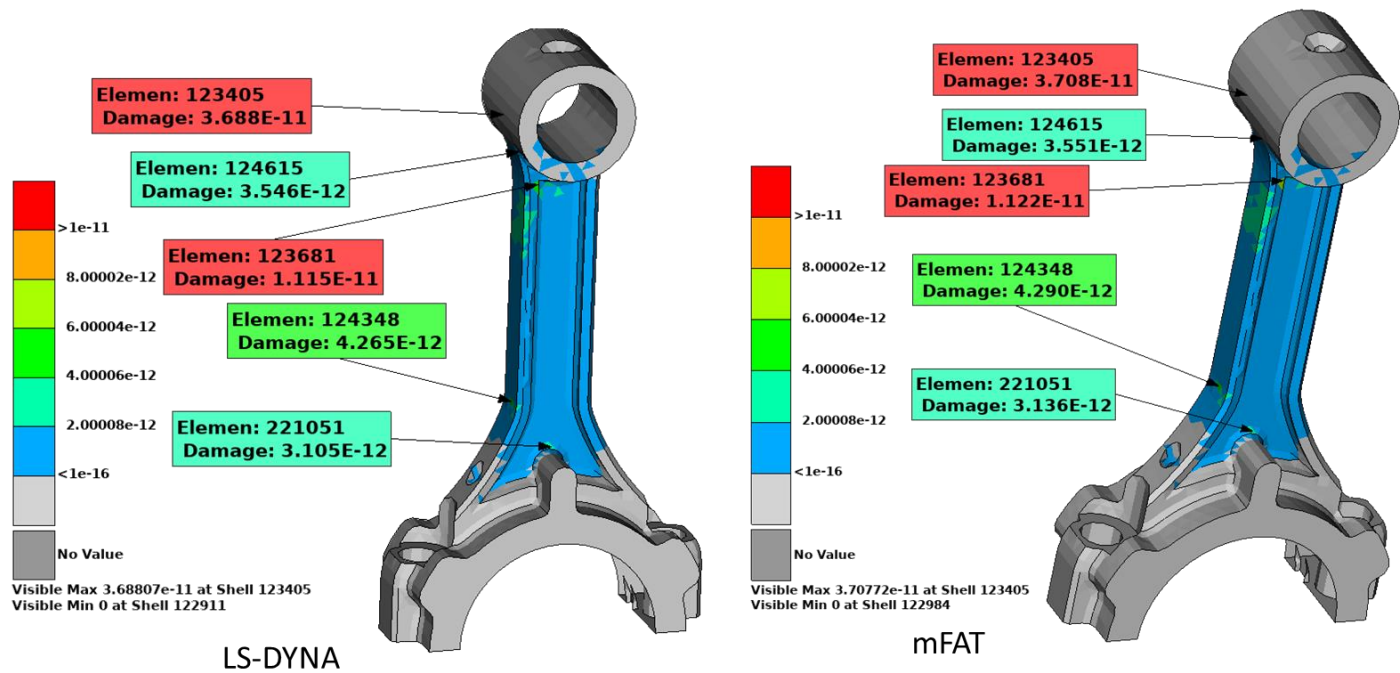


Figure 9. Comparison of the fatigue damage results for the conrod from LS-DYNA (left image) and mFAT (right image).

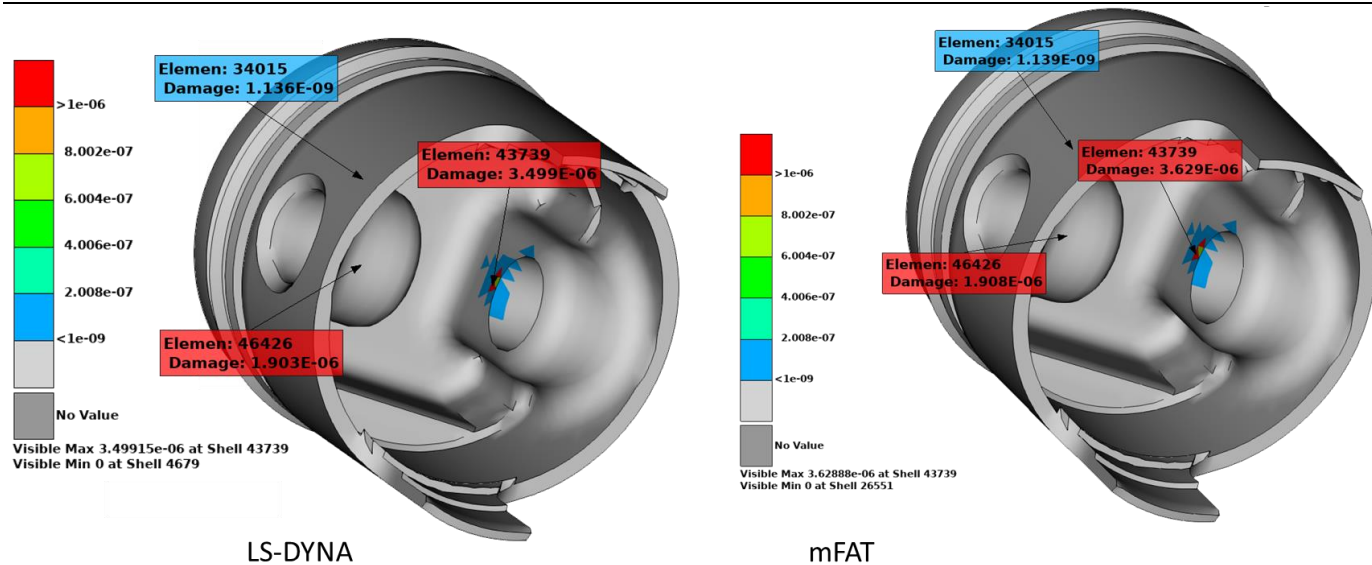


Figure 10. Comparison of the fatigue damage results for the piston from LS-DYNA (left image) and mFAT (right image).

Finally, the maximum fatigue damage results per part are summarized in Table 44. Good agreement is found between the two fatigue analysis methods. This demonstrates that LS-DYNA, by introduction of the *FATIGUE keywords, can offer even broader applications to strength and durability analyses of complex assemblies.

Table 4. Comparison of the maximum fatigue damage ratio D per part from LS-DYNA and mFAT

Part	D by LS-DYNA	D by mFAT
Crankshaft	6.37E-13	5.37E-13
Conrod	3.69E-12	3.71E-12
Piston	3.50E-6	3.63E-6

Example 3: Bar with a groove

In this section, a simple notched bar specimen was used. This benchmark example concerns the cycle counting algorithm and the accumulated damage. The simulated results were compared to analytical calculations where traditional rain flow count and the Palmgren-Miner rule of damage accumulation was used and to results obtained from using mFAT [2], a fatigue plug-in tool in METApst [3]. The keyword features tested were: *FATIGUE_ELOUT using RESTRT=1, *FATIGUE_LOADSTEP, *FATIGUE_MULTIAXIAL and *FATIGUE_MEAN_STRESS_CORRECTION. However, due to paper size restrictions, this section concerns only benchmarking of the underlying fatigue cycle counting scheme and the *FATIGUE_LOADSTEP keyword.

Model

The FE-model used is shown in Figure 11. It includes 64845 nodes, 63360 solid elements. A cyclic pressure load was applied to one end of the bar while the other end was constrained in all d.o.f.'s.

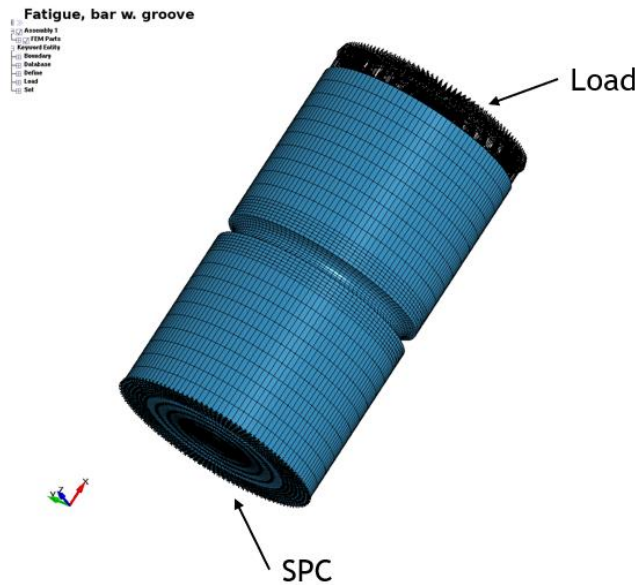


Figure 11. FE-model of bar with a groove.

Fatigue data

The S-N curve used in the benchmark is shown in Figure 12 below. The parameters m_1 and m_2 are used to define the S-N curve in mFAT.

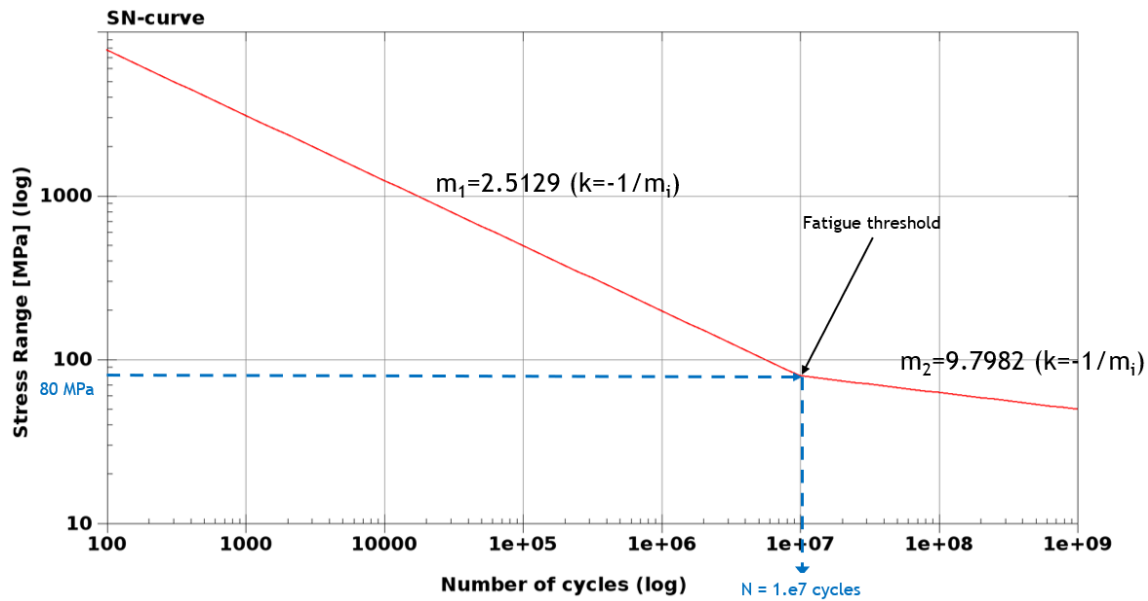
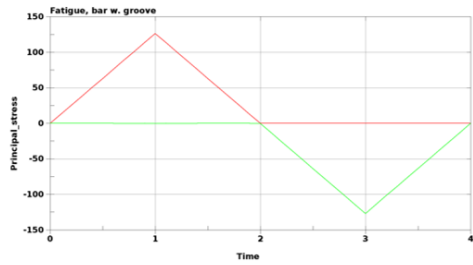


Figure 12. S-N curve used for the benchmarks

Checking of the fatigue cycle counting algorithm

Pulsating and alternating loads were used when running this benchmark. The results were compared to analytical calculations (where cycle counting was made on elout data) and mFAT results. The loading and cycle counting are shown in Figure 13.

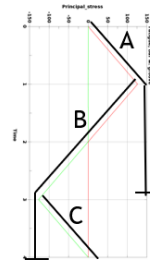


$$S_A = 127 \text{ MPa}$$

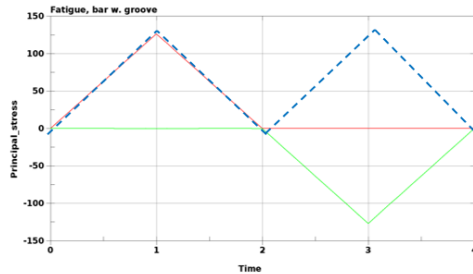
$$S_R = 254 \text{ MPa}$$

$$S_M = 0 \text{ MPa}$$

$$R = -1$$



$$D_{ALT} = \sum \frac{n_i}{N_i} = \frac{0.5}{N_A} + \frac{0.5}{N_B} + \frac{0.5}{N_C}$$

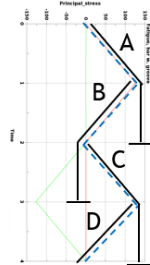


$$S_A = 63.5 \text{ MPa}$$

$$S_R = 127 \text{ MPa}$$

$$S_M = 63.5 \text{ MPa}$$

$$R = 0$$



$$D_{PULS} = \sum \frac{n_i}{N_i} = \frac{1}{N_{AB}} + \frac{1}{N_{CD}} = \frac{2}{N_{AB}}$$

Figure 13. Fully reversed, constant amplitude load (upper) and constant amplitude, pulsating load (lower).

The results comparison is shown in Table 5. Note that the analytically calculated results include reading of an S-N curve which is not all that exact. Also, mFAT is always using a multiaxial fatigue approach based on the Principal stress. Still, the results from the above, simple exercise show very good agreement and should indicate that the cycle counting in the LS-DYNA package works as intended.

Table 5. Results from fatigue cycle counting exercise.

Case	End time	Load range	Rainflow count cycles	D, LS-DYNA	D, mFAT	D, analytical
1	2.0 Pulsating	0,1,0	1 cycle, 1 reversal	3.176E-7	3.188E-7	3.159E-7
2	4.0 Pulsating	0,1,0,1,0	2 cycles	6.351E-7	6.376E-7	6.318E-7
3	4.0 alternating	0,1,0,-1,0	1 cycle, 2 reversals	1.225E-6	1.229E-6	1.211E-6
4	4.0 alternating	0,0.5,0,-0.5,0	1 cycle, 2 reversals	1.691E-7	1.698E-7	1.680E-7

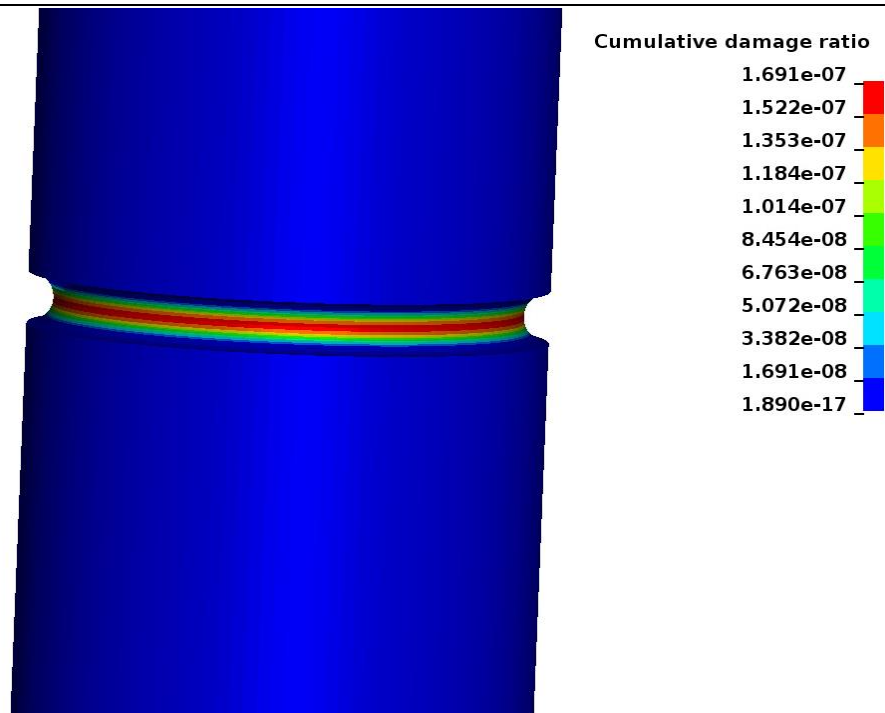
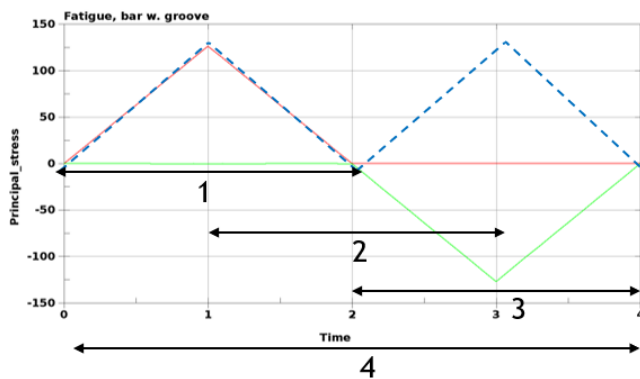


Figure 14. Accumulated fatigue damage for case 4.

Checking of the *FATIGUE_LOADSTEP functionality

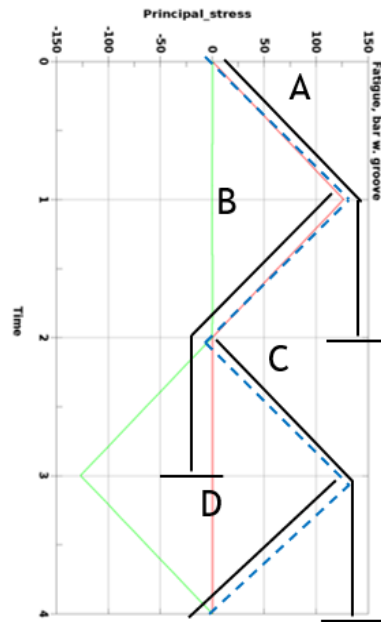
The definitions of the load cases used for checking the load step function are shown in Figure 15 and the cycle counting for the sequence is shown in Figure 16.



Case no.	Start at time	End at time
1	0	2
2	1	3
3	2	4
4	0	4

Figure 15. Load step definitions.

The Rainflow counting is performed on elout time-history data as shown in Figure 16.



$$D_{PULS} = \sum \frac{n_i}{N_i} = \frac{1}{N_{AB}} + \frac{1}{N_{CD}} = \frac{2}{N_{AB}}$$

Figure 16. Analytical rainflow counting and damage accumulation.

The benchmark results are shown in Table 6. As seen the agreement to analytical calculation results is good.

Table 6. Results from the load step functionality benchmark.

Case No.	D, LS-DYNA	D, analytical	D, mFAT
1	3.176E-7	3.159E-7	3.176E-7
2	3.176E-7	3.159E-7	3.176E-7
3	3.176E-7	3.159E-7	3.176E-7
4	6.351E-7	6.318E-7	6.351E-7

As can be seen from the comparison, *FATIGUE_{OPTION} in LS-DYNA shows good agreement to analytically calculated results and to mFAT results. The cycle counting seems to follow the rule of a classical fatigue cycle counting algorithm well, the calculated accumulated damage shows good agreement to Palmgren-Miner and the loadstep functionality yield results close to the analytical solutions and are therefore judged as reliable and accurate.

Summary

This paper provides a brief review of multiaxial fatigue analysis with LS-DYNA. The methods and options of the multiaxial fatigue analysis are introduced. Several examples are also included to demonstrate the effectiveness of the implemented methods. Comparison of LS-DYNA fatigue analysis results with fatigue results by other software and by analytical solution is also included. The new keyword *FATIGUE_LOADSTEP was also checked in the last example. It is confirmed that LS-DYNA fatigue analysis results have a good match with the results given by other software, and the results given by analytical solutions when available.

The future development includes possibly more advanced fatigue model (for example the Dang-Van criterion), as well as fatigue analysis on spot welds, HAZ and squeezing zone, etc.

References

- [1] Jonsson, A., Sjöberg, M., and Grenwald, J., mFAT – a basic fatigue module for μ ETA – Post, 6th International BETA Conference, Thessaloniki 2015, internet source: https://www.beta-cae.com/events/c6pdf/4C_2_DYNAMORE.pdf
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