

Update on the ANSYS Fatigue Module

Analysis determines if parts can withstand cyclic loading over their lifetime.

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While many parts may work well initially, they often fail in service due to fatigue failure caused by repeated cyclic loading. Characterizing the capability of a material to survive the many cycles a component may experience during its lifetime is the aim of fatigue analysis. In a general sense, fatigue analysis has three main methods: strain life, stress life and fracture mechanics, and the first two are available within the ANSYS Fatigue Module.

Strain life typically is concerned with crack initiation, whereas stress life involves total life and does not distinguish between initiation and propagation. In terms of cycles, strain life typically deals with a relatively low number of cycles and therefore addresses low cycle fatigue (LCF), but it works with high numbers of cycles as well. Stress life is based on SN curves (stress-cycle curves), and it has traditionally dealt with relatively high numbers of cycles; therefore, it addresses high cycle fatigue (HCF), inclusive of infinite life.

Types of Cyclic Loading

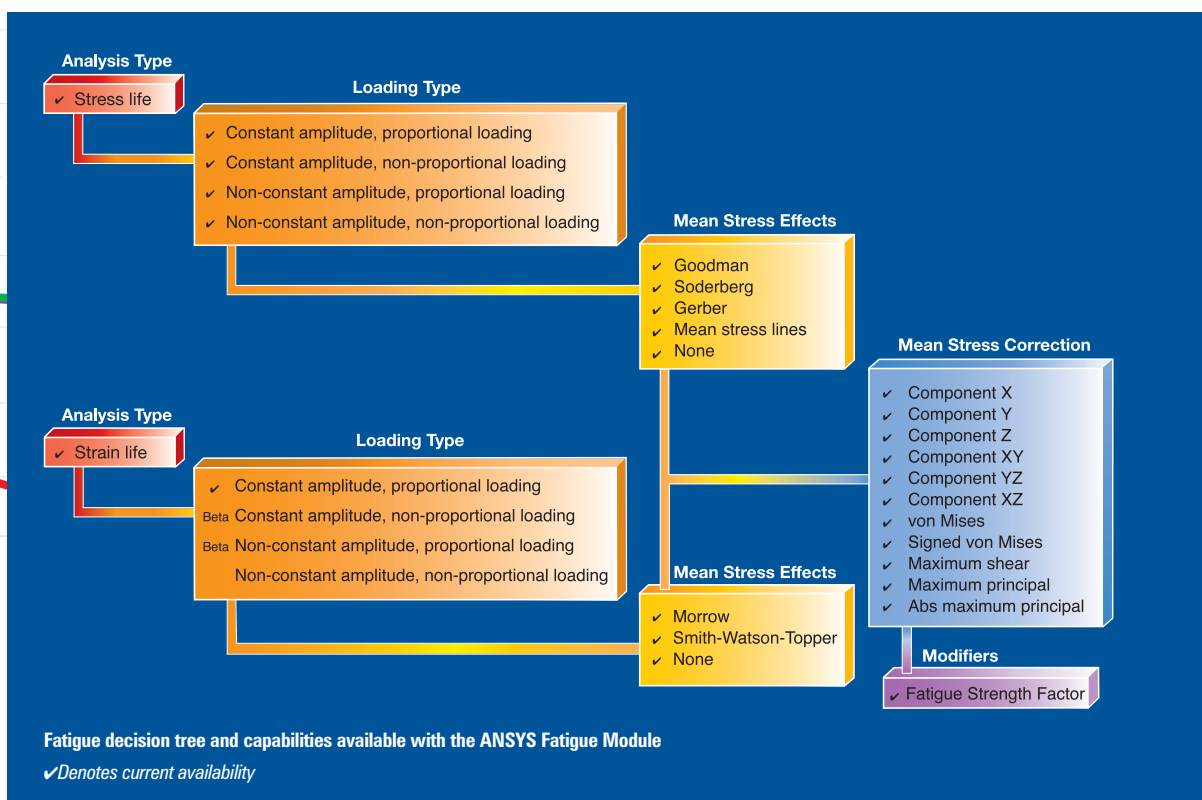
Unlike static stress, which is analyzed with calculations for a single stress state, fatigue damage occurs when stress at a point changes over time. There are essentially four classes of fatigue loading, with the ANSYS Fatigue Module currently supporting the first three:

- Constant amplitude, proportional loading
- Constant amplitude, non-proportional loading
- Non-constant amplitude, proportional loading
- Non-constant amplitude, non-proportional loading

In these descriptions, the amplitude identifier is readily understood: Is the loading a variant of a sine wave with a single load ratio, or does the loading vary, perhaps erratically, with the load ratio changing with time? The second identifier, proportionality, describes whether the changing load causes the principal stress axes to change. If the principal stress axes do not change, then it is proportional loading. If the principal stress axes do change, then the cycles cannot be counted simply and it is non-proportional loading.

Constant amplitude, proportional loading is the classic “back of the envelope” calculation describing whether the load has a constant maximum value or continually varies with time. Loading is of constant amplitude because only one set of FE stress results along with a loading ratio is required to calculate the alternating and mean values. The loading ratio is defined as the ratio of the second load to the first load ($LR = L_2/L_1$). Loading is proportional since only one set of FE results is needed. (Principal stress axes do not change over time.) Common types of constant amplitude loading are fully reversed (Apply a load, then apply an equal and opposite load; a load ratio of -1.) and zero-based. (Apply a load then remove it; a load ratio of 0.) Since loading is proportional, studying a single set of FE results can identify critical fatigue locations. Likewise, since there are only two loadings, no cycle counting or cumulative damage calculations need to be done.

Constant amplitude, non-proportional loading examines exactly two load cases that need not be related by a scale factor. The loading is of constant amplitude but non-proportional, since principal stress or strain axes are free to change between the two load sets. No cycle counting needs to be done. But since



the loading is non-proportional, the critical fatigue location may occur at a spatial location that is not easily identifiable by looking at either base loading stress state. This type of fatigue loading can describe common fatigue loadings, such as alternating between two distinct load cases (for example, bending load and torsional load) or cases in which loading is proportional but results are not. This happens under conditions in which changing the direction or magnitude of loads causes a change in the model's relative stress distribution. This may be important in situations with nonlinear contact, compression-only surfaces or bolt loads.

Non-constant amplitude, proportional loading also needs only one set of FE results. But instead of using a single load ratio to calculate alternating and mean values, the load ratio varies over time. Think of this as coupling FE analysis with strain-gauge results collected over a given time interval. Since loading is proportional, the critical fatigue location can be found by looking at a single set of FE results. However, the fatigue loading that causes the maximum damage cannot be seen easily. Thus, cumulative damage calculations (including cycle counting such as rainflow and damage summation such as Miner's rule) need to be done to determine the total amount of fatigue damage and the cycle combinations that cause the damage. Cycle counting is a means to reduce a complex load history into a number of events, which can be compared to the available constant amplitude test data.

Non-constant amplitude, proportional loading within the ANSYS Fatigue Module uses a "quick counting" technique to substantially reduce runtime and memory. In quick counting, alternating and mean stresses are sorted into bins before partial damage is

calculated. Without quick counting, data is not sorted into bins until after partial damages are found. The accuracy of quick counting usually is very good if a proper number of bins are used when counting. For stress life, another available option when conducting a variable amplitude fatigue analysis is the ability to set the value used for infinite life. In constant amplitude loading, if the alternating stress is lower than the lowest alternating stress on the fatigue curve, the fatigue tool will use the life at the last point. This provides for an added level of safety because many materials do not exhibit an endurance limit. However, in non-constant amplitude loading, cycles with very small alternating stresses may be present and may incorrectly predict too much damage if the number of the small stress cycles is high enough. To help control this, the user can set the infinite life value that will be used if the alternating stress is beyond the limit of the SN curve. Setting a higher value will make small stress cycles less damaging if they occur many times. The rainflow and damage matrix results can be helpful in determining the effects of small stress cycles in your loading history.

Non-constant amplitude, non-proportional loading is the most general case and is similar to constant amplitude, non-proportional loading. But in this loading class, there are more than two different stress cases involved that have no relation to one another. Not only is the spatial location of critical fatigue life unknown, but also unknown is what combination of loads causes the most damage. Thus, more advanced cycle counting is required, such as path-independent peak methods or multiaxial critical plane methods. Currently, the ANSYS Fatigue Module does not support this type of fatigue loading.

Mean Stress Correction

Once you have made the decision on the type of fatigue analysis to perform — stress life or strain life — and have determined your loading type, the next decision is whether to apply a mean stress correction. Cyclic fatigue properties of a material often are obtained from completely reversed, constant amplitude tests. Actual components seldom experience this pure type of loading, since some mean stress usually is present. If the loading is other than fully reversed, a mean stress exists and may be accounted for.

For stress life, if experimental data at different mean stresses or r -ratios exist, mean stress can be accounted for directly through interpolation between material curves. If experimental data is not available, several empirical options may be chosen — including Gerber, Goodman and Soderberg theories, which use static material properties (yield stress, tensile strength) along with SN data to account for any mean stress. In general, most experimental data fall between the Goodman and Gerber theories, with the Soderberg theory usually being overly conservative. The Goodman theory can be a good choice for brittle materials and the Gerber theory a good choice for ductile materials. The Gerber theory treats negative and positive mean stresses the same, whereas Goodman and Soderberg do not apply any correction for negative mean stresses. This is because, although a compressive mean stress can retard fatigue crack growth, ignoring a negative mean is usually more conservative. Of course, the option of no mean stress correction also is available. The mean stress effects graph (below) provides a graphical representation.

For strain life, the ANSYS Fatigue Module has a variety of mean stress correction methods, including Morrow, Smith-Watson-Topper (SWT) and no mean

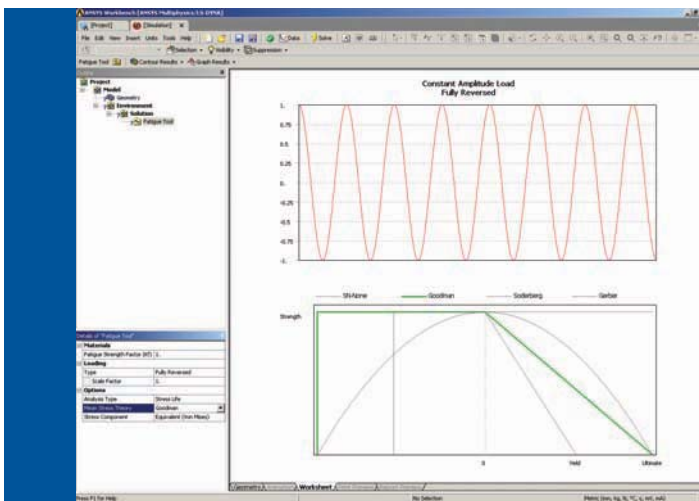
stress effects. In Morrow's method, the elastic term in the strain life equation is modified by the mean stress. This modification is consistent with observations that the mean stress effects are significant at low values of plastic strain (in which elastic strain dominates) and that mean stress has little effect at shorter life (in which plastic strains dominate). Unfortunately, it incorrectly predicts that the ratio of elastic to plastic strain is dependent on mean stress, which is not true. Smith, Watson and Topper suggested a different equation to account for the presence of mean stresses. It has the limitation that it is undefined for negative maximum stresses. The physical interpretation of this is that no fatigue damage occurs unless tension is present at some point during the loading. Of course, the option of no mean stress correction also is available.

Correction Factors

Two more decisions need to be made before fatigue results can be calculated: which multiaxial stress correction to use and whether to use a fatigue strength factor. Experimental test data is mostly uniaxial, whereas FE results are usually multiaxial. At some point, stress must be converted from a multiaxial stress state to a uniaxial one. Von Mises, max shear, maximum principal stress or any of the component stresses can be used to compare against the experimental uniaxial stress value. A “signed” von Mises stress may be chosen in which the von Mises stress takes the sign of the largest absolute principal stress. This is useful to identify any compressive mean stresses, since several of the mean stress theories treat positive and negative mean stresses differently.

Fatigue material property tests usually are conducted under very specific and controlled conditions. If the service part conditions differ from the as-tested conditions, modification factors can be applied to try to account for the difference. The fatigue alternating stress usually is divided by this modification factor and can be found in design handbooks. (Dividing the alternating stress is equivalent to multiplying the fatigue strength by K_f .) Fatigue strength factor (K_f) reduces the fatigue strength and must be less than one. Note that this factor is applied to the alternating stress only and does not affect the mean stress.

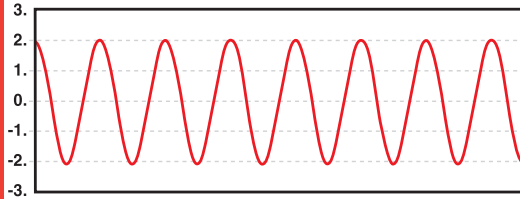
This concludes the input decisions required to perform either a stress life or a strain life fatigue analysis. Once the fatigue calculation has been performed, there are a variety of results available that depend on the type of fatigue analysis performed. Some provide contour plots of a specific result, while others give supplemental information about the critical location. We will explore fatigue results in the next issue of *ANSYS Solutions*. ■



Mean stress effects

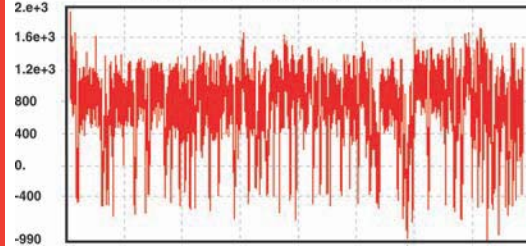
Types of Loadings

Constant Amplitude Load – Fully Reversed



Proportional

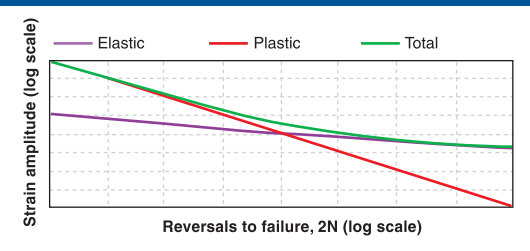
Non-Constant Amplitude Load – History Data



Non-proportional

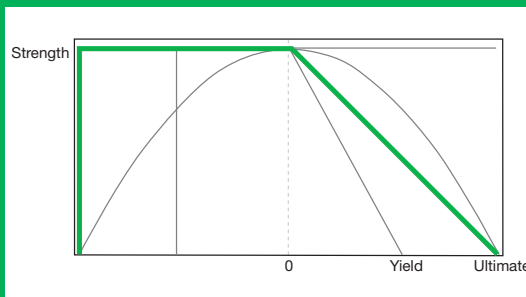
Types of Mean Stress Corrections

Strain Life

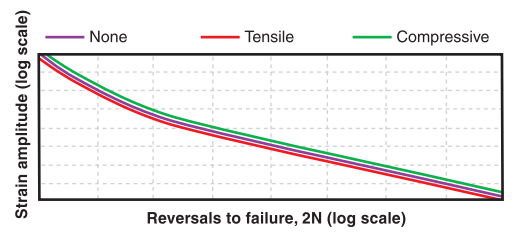


No mean stress correction

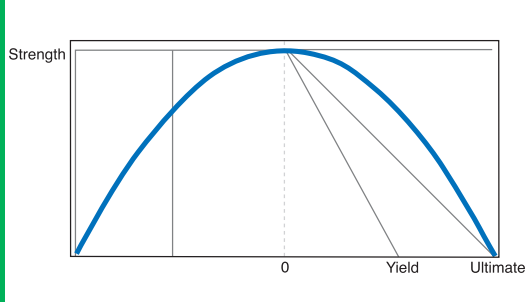
Stress Life



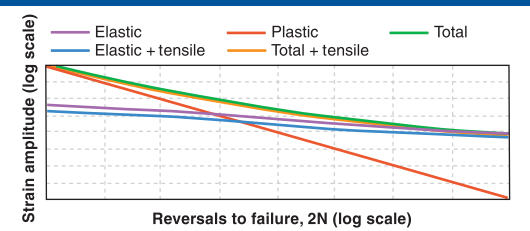
Goodman mean stress correction



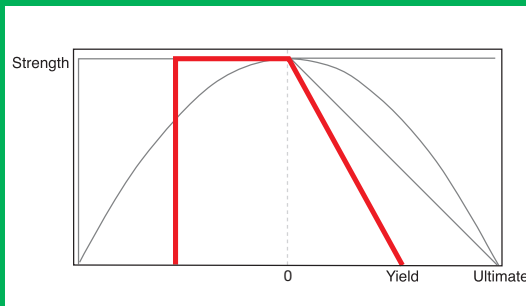
SWT mean stress correction



Gerber mean stress correction



Morrow mean stress correction



Soderberg mean stress correction