FE-Based Wheel Fatigue Analysis Using MSC.FATIGUE

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ABSTRACT

A validation exercise is currently underway at BF Goodrich to evaluate the fatigue life of a rotating wheel using a new "WHEELS" capability within MSC.FATIGUE. BF Goodrich will shortly be using this capability to automate fatigue calculations for rotating aircraft wheels subject to vertical and lateral loads. As part of a validation exercise, an FE model of the wheel was analysed by applying a bearing roller load around the inner surface of the wheel hub at 15° increments. A linear static analysis was conducted at each increment to produce a stress tensor history (σ_x, σ_y, and τ_{xy}) for all surface nodes. Due to the nature of a rolling wheel, it was expected that the principal stresses and their directions would vary for each increment. For this reason, the stress tensors were also rotated on the model surface through 360°, at 10° increments, to calculate the components of the principal stresses in those directions. All stress data was run through an S-N fatigue life analysis with no mean stress correction. Although not complete, the results are expected to show contour plots of fatigue life and fatigue damage for all nodes at the worst (most damaging) surface angle. MSC.Software envisions that this new capability can also be applied to any rotating body, especially automotive wheels.

INTRODUCTION

BF Goodrich Aerospace is a world leader in the development and advancement of technologies for the design and manufacture of aircraft wheels and brakes. As part of a commitment to utilising new technology, BF Goodrich and MSC.Software have joined forces to create a new analysis solution technique for the prediction of fatigue life on wheels using MSC.Fatigue. The new "WHEELS" capability within MSC.Fatigue will allow BF Goodrich to conduct fatigue analyses on wheels for a variety of loading conditions utilising their own, unique database of material properties. As part of a validation exercise, this new capability is being tested on an A320 aircraft nose wheel FE model. Although the validation exercise is not fully completed, this paper explains the analysis process in detail and its application to the automotive industry.

NOSE WHEEL MODEL

Figure 1 is an aircraft nose wheel solid FE model that has been simplified to remove detailed machining features to facilitate the FE meshing. For the validation exercise, only the hub area of the inner half of the wheel was used (as indicated by the arrow on Figure 1). The hub area was modelled using 12049 solid parabolic tetrahedron elements. The elements representing the hub were given the material property of 2014-T6 aluminium. The elements representing the bearing cup were given the

material property of high strength steel. A Timken Bearing Select Analysis was conducted on the bearing to determine the roller loads based on the rated ground load of the wheel. The bearing roller loads were transmitted from the surface of the bearing cup to the inner bore surface of the hub through 1905 contact elements.
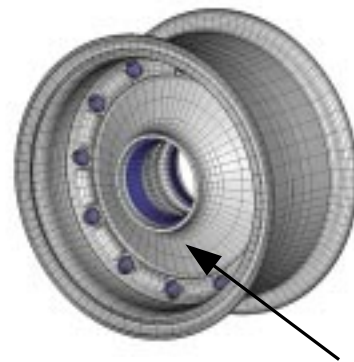

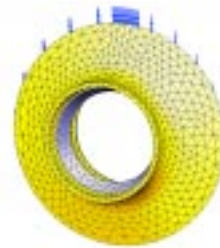Figure 1 Aircraft nose wheel solid model.

Figure 2 Hub model with reaction loads.

To counteract the radial component of the bearing roller loads, a distributed traction load was applied in the corresponding radial direction to the outer diameter web surface of the model (this is indicated by small arrows on the outer hub surface in Figure 2).

The distribution of the bearing roller loads was rotated 180° around the bearing cup surface in increments of 15° in order to model the wheel in rotation. A static analysis was conducted at each 15° increment using SCRC's I-DEAS internal solver, "Model Solution". This analysis produced 13 load sets representing the movement of the bearing roller loads around the inner bore of the hub as the wheel rotated through 180°. The model and load sets were read into MSC.Fatigue through the "universal file" reader capability.

PROCEDURE

MSC.Fatigue is sold as a package that works with MSC.Patran or as a "Pre & Post" module that runs independent of MSC.Patran. For this exercise, the Pre & Post module was used although the GUI forms, described on the following pages, do not change regardless of the selected start up method.

Figure 3 shows the general setup and parameters form that was used to enter specific information about the model. The form was modified (from the standard setup form) to select the new "**Wheels"** analysis capability as the default solution type. The 3 buttons located in the "Specific Setup Forms:" area were used to display modified forms relevant to the Wheels analysis type. Selecting the *Solution Params ...* button displayed the modified solution parameters form shown in Figure 4.

Due to the nature of a rolling wheel, it was expected that the principal stresses and their directions would vary with each 15° increment of the bearing roller loads. For that reason, stress tensor information calculated at each node was rotated in-plane through 360° in 10° increments. This additional calculation was conducted by selecting the new stress combination toggle, "**Surface Angle**", and entering an in-plane rotation angle increment of 10°. For this analysis, only the surface nodes were considered where the biaxial stress state produced 3 independent components (σ_x, σ_y, and τ_{xy}). Using these components in the equations below produced principal stress components σ_θ and τ_θ where θ represented the in-plane rotation angle.

$$\sigma_\theta = \frac{\left(\sigma_x + \sigma_y\right)}{2} + \frac{\left(\sigma_x - \sigma_y\right)}{2}\cos 2\theta + \tau_{xy}\sin 2\theta$$

$$\tau_\theta = \frac{\left(\sigma_x - \sigma_y\right)}{2}\sin 2\theta - \tau_{xy}\cos 2\theta$$

General setup parameters indicating the load set results are stresses in KSI.

Area for jobname and title.

Specific information required to conduct a fatigue analysis.

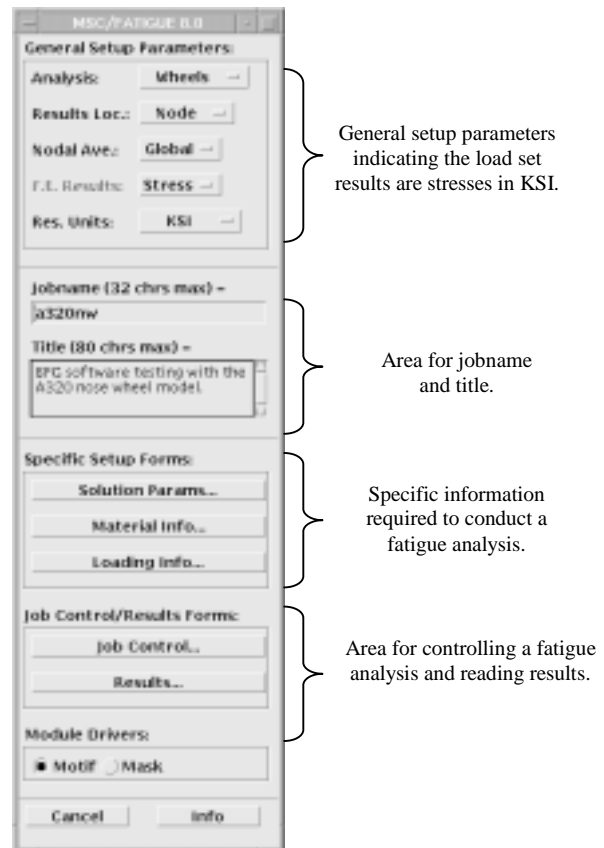Area for controlling a fatigue analysis and reading results.

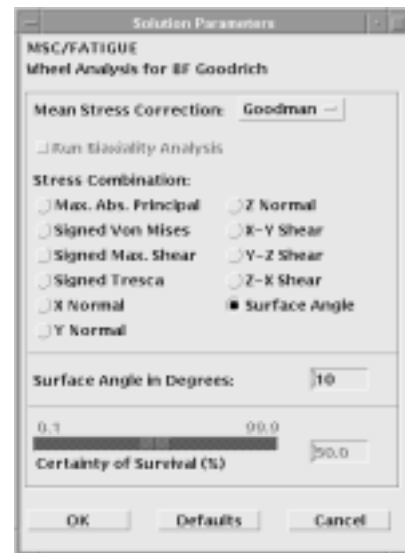Figure 3 General Setup and Parameters form.

Figure 4 Solution Parameters form.

Figure 5 shows the material information form which was displayed after selecting the *Material info ...* button on the general setup form. The material form provided a means of specifying the material properties for the FE model. For this exercise, the material chosen was 2014-T6 aluminium that matched the wheel hub material. The "Reduction Factor" and "Residual Stress" were factors specific to this BF Goodrich analysis and were set to 1.0 and 0.0, respectively. The "Region" column was used to

specify which elements in the model had the material properties of 2014-T6. In this case, all hub elements were chosen. There was no need to specify the bearing cup material as the fatigue analysis was only conducted on the surface of the hub.
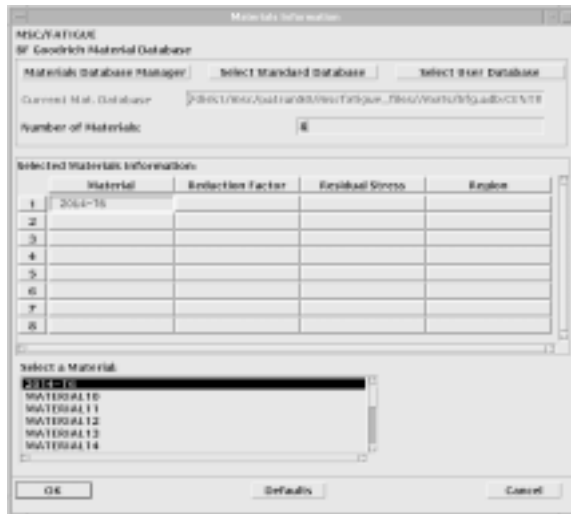

Figure 5 Material Information form

In the future, BF Goodrich can add further materials to the database by using the material database editor which is spawned by selecting the **Materials Database Manager** button located on the top left of the form.

Figure 6a & 6b show the two forms required to enter the loading information into the fatigue analysis. The Loading Information form (6a), was displayed by selecting the ***Loading Info …*** button on the main setup form. The Get/Filter Results form (6b), was displayed by selecting the **Get/Filter Results …** button on the loading information form. This button and other databoxes appear when a Loading Condition ID cell is selected.

Figure 6a Loading Information form.

Figure 6b Get/Filter Results form

Figure 6b shows the 13 load sets as listed on the Get/filter Results form. For this validation exercise, the stress results from all 13 load sets were selected and used to produce a single loading condition as shown in the first cell of the Loading Condition ID column (referred to as B.C. 2, ST FEA (1-13)) on the Loading Information form. Figure 7 is an example of the stress results calculated for one load set.

In the next cell, under Design Life, a value of 40000 miles was entered. This value represents the required life of this wheel under the loading condition specified. The next cell, under Loading Factor, allowed for a linear scaling of the stress values and was set to 1.0. The values calculated for fatigue damage (life) were based on the inputs provided in "X repeats of the loading = Y miles" databoxes located at the top of the Loading Information form. In this analysis, the circumference of the aircraft wheel (with tire) was 18 inches. Therefore, 560 revolutions of the wheel covered 1 mile. Although the bearing loads were only rotated around the inner bore of the hub through 180° (which represents 1 "repeat" of the loading), it was sufficient, in this case, to pick up the max and min stresses in the cycle thus making the remaining 180° redundant. Therefore, based on a design life of 40000 miles the hub must be able to withstand 2.24E7 (560 x 40000) repeats of the stress loading in order to successfully pass the fatigue life criteria. In fact, because of the statistical nature of a fatigue

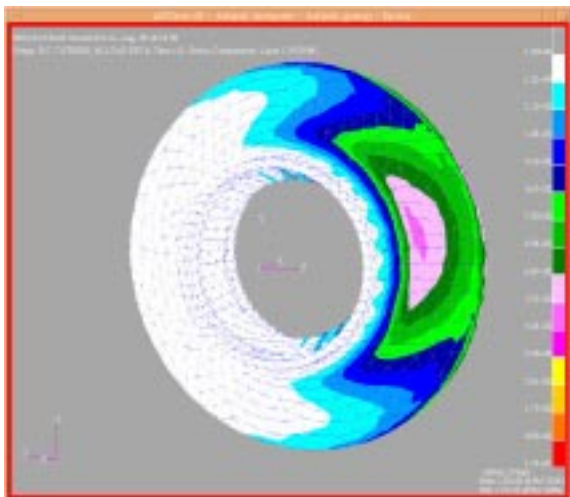analysis it would be far better for the hub to withstand 3 times this value.

Figure 7 Stress results from 1 of 13 load sets.

BACKGROUND TO AN S-N FATIGUE CALCULATION

The starting point for any fatigue analysis is the response of a structure (FE model) to the input loading and is usually expressed as a stress or strain time history.   If the time history was made up of constant amplitude stress or strain cycles then the fatigue design could be accomplished by referring to a typical S-N diagram. However, because real signals rarely conform to this ideal constant amplitude situation, an empirical approach is used for calculating the damage caused by stress signals of variable amplitude. Despite its limitations, the Palmgren-Miner rule is generally used for this purpose.

This linear relationship assumes that the damage caused by parts of a stress signal with a particular range can be calculated and accumulated to the total damage separately from that caused by other ranges. A ratio is calculated for each stress range, equal to the number of actual cycles at a particular stress range, n, divided by the allowable number of cycles to failure at that stress, N, (obtained from the S-N curve). Failure is assumed to occur when the sum of these ratios, for all stress ranges, equals 1.0.

If the response time history is irregular with time, as shown in the Figure 8, then *rainflow* cycle counting is widely used to decompose the irregular time history into equivalent sets of block loading. The numbers of cycles in each block are usually recorded in a stress range histogram. This can then be used in the Palmgren Miner calculation. An example of the way rainflow ranges are extracted from a time signal is given in [1].

S-N Relationship

A traditional S-N curve as shown in Figure 9 is used to model the material properties of the components being analyzed. This simply shows that, under constant amplitude cyclic loading, a linear relationship exists between cycles to failure N and applied stress range S when plotted on log-log paper. There are two alternative ways of defining this relationship, as given below.
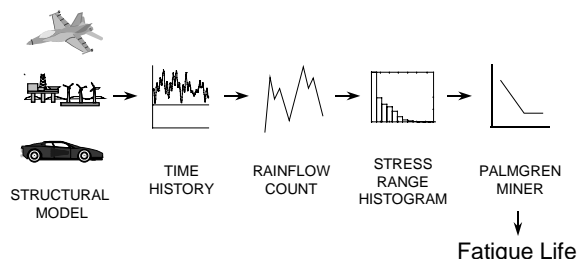
$$NS^m = K \qquad N^{-b}S = SRI1$$
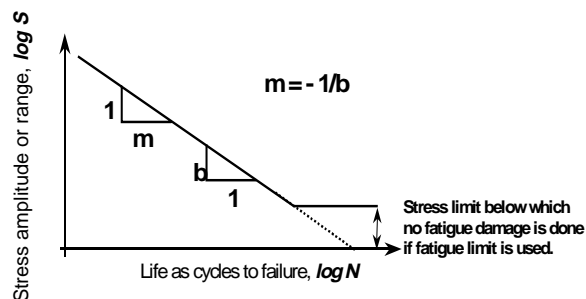

Figure 8. A standard S-N fatigue analysis

Figure 9. A typical S-N curve

Estimating fatigue life from a stress pdf

Once the stress range histogram has been converted into a stress range pdf then there is an elegant and efficient equation to describe the expected fatigue damage caused by this loading history.

$$E[D] = E[P]\frac{T}{k}\int_0^\infty S^m p(S)dS$$

In order to compute fatigue damage over the lifetime of the structure in seconds (T) the form of material (S-N) data must also be defined using the parameters k and m (or b and $SRI1$). In addition, the total number of cycles in time T must be determined from the number of peaks per second E[P]. If the damage caused in time T is greater than 1.0 then the structure is assumed to have failed. Or alternatively the fatigue life can be obtained by setting T = 1.0 and then finding the fatigue life in seconds from the above equation.

For the validation exercise, the 13 individual load set analyses were combined to generate 13 points on a stress-time history (i.e. the response time history, as discussed earlier). The stress time history was then subjected to the procedure laid out in Figure 8 in order to calculate the fatigue life (damage).

AUTOMOTIVE APPLICATION

This type of analysis can also be used on automotive wheels with the following considerations. If the wheel is treated as a solid disk, the loads can be applied in the same fashion as was done for this exercise. If a wheel has symmetry (between bolt holes or from the number of spokes) then the load can be applied to the symmetric portion of the model only.

As an example, BF Goodrich will be using the WHEELS fatigue tool on a 24 bolt main gear wheel. The wheel has symmetry between bolt holes (every 15°) and the applied load is a pressure load trace generated from transducers located between the wheel and tire. The pressure load was previously measured during several rotations of the wheel in a BF Goodrich testing laboratory and was shown to have 180° symmetry.

The analysis procedure for this wheel will require BF Goodrich to segment the pressure loading into 12 sections (minimum) representing 12 load sets applied to the 15° "wedge" shaped FE model. For each load application, a static FE analysis must be conducted and the results from all 12 analyses will produce a stress time history. MSC.Fatigue then conducts a standard S-N fatigue analysis on this time history.

Two important aspects must be considered before conducting a WHEELS analysis. First, it is important that the loading applied to the model contains the full maximum to minimum loading cycle (if not, it is possible that large, damaging stress cycles will be missed in the fatigue calculation). Secondly, the load set profile must be small enough to provide load continuity to the nodes from one load set to the next. As an example, refer at Figure 10.
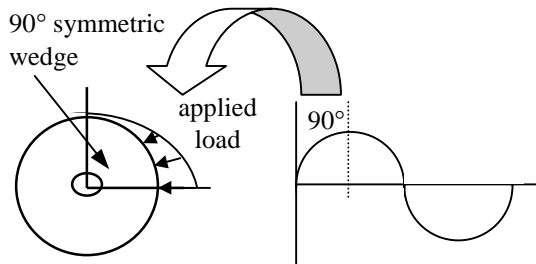
Imagine a wheel had symmetry that allowed it to be modelled as a 90° wedge and the loading applied to the wheel was a sinusoidal trace. A full analysis of that wheel would require 4 static FE analyses (minimum) with the sinusoidal load being rotated through 90° for each analysis. Unfortunately, this will lead to poor results since each node in the wedge does not undergo a smooth load transition from one load set to the next. Hence, most nodes will not be loaded with the full max/min loading cycle shown by the sinusoidal load curve since only 4 load sets are being used to represent the entire loading history.

Therefore, for best fatigue results, the loading profile should only be rotated around the wheel (or around the symmetric FE wedge) in the smallest increment possible that allows for good fatigue results yet not unduly overload the analysis operation. Using small increments also ensures that all nodes will experience the max/min loading profile as it rotates around the model.

ACKNOWLEDGEMENTS

REFERENCES

1. Bishop, N.W.M and Sherratt, F. (1989) *Fatigue life prediction from power spectral density data. Part 1 – traditional approaches and Part 2 – recent development.* Environmental Engineering. Vol 2.

Figure 10 Example of loading applied to a large symmetric section of a wheel.