Chapter 8

ASSESSMENT

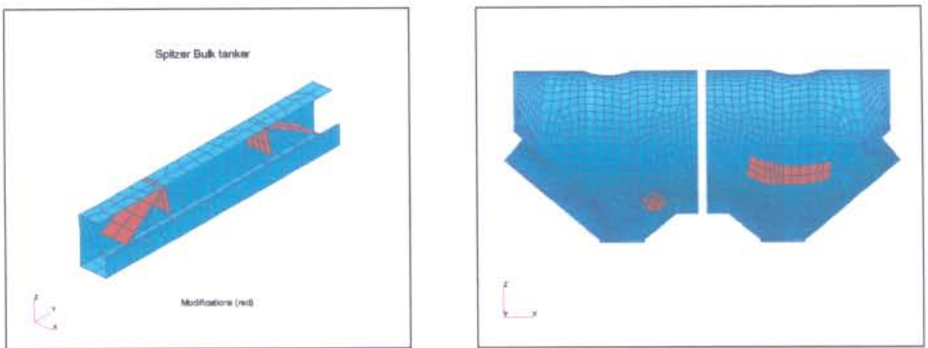
8.1 Scope

The finite element model enables the engineer, using measurements, to calculate Fatigue Equivalent Static loads for a vehicle structure. The real power of the finite element analysis lies,however, in the ability to assess the structure using the FESL. This chapter deals with the assesments of the finite element analyses.

8.2 Aluminum Dry-bulk Tanker

The results of the analysis performed with PERMAS were evaluated in two parts. High stress areas on the Bulk tanker were identified. These high stress areas were addressed with modifications on the specific part in subsequent analysis (refer to figure 8.1). The first three high stress areas were identified as:

- Certain areas on the tank vessel.
- The kingpin structure, specifically, the rear cross-member.
- At the top of the front boom.



(a) Crossmember (b) Doublers

Figure 8.1: Bulk tanker - Modifications

The high stress areas on the tank vessel were addressed by inserting doublers at strategic positions on the vessel (refer to figure 8.1). Various finite element analyses were done to verify the effectiveness of the additional doublers. An inner bracing was also developed to help alleviate the stresses. The results were plotted for evaluation (refer to figure 8.2).

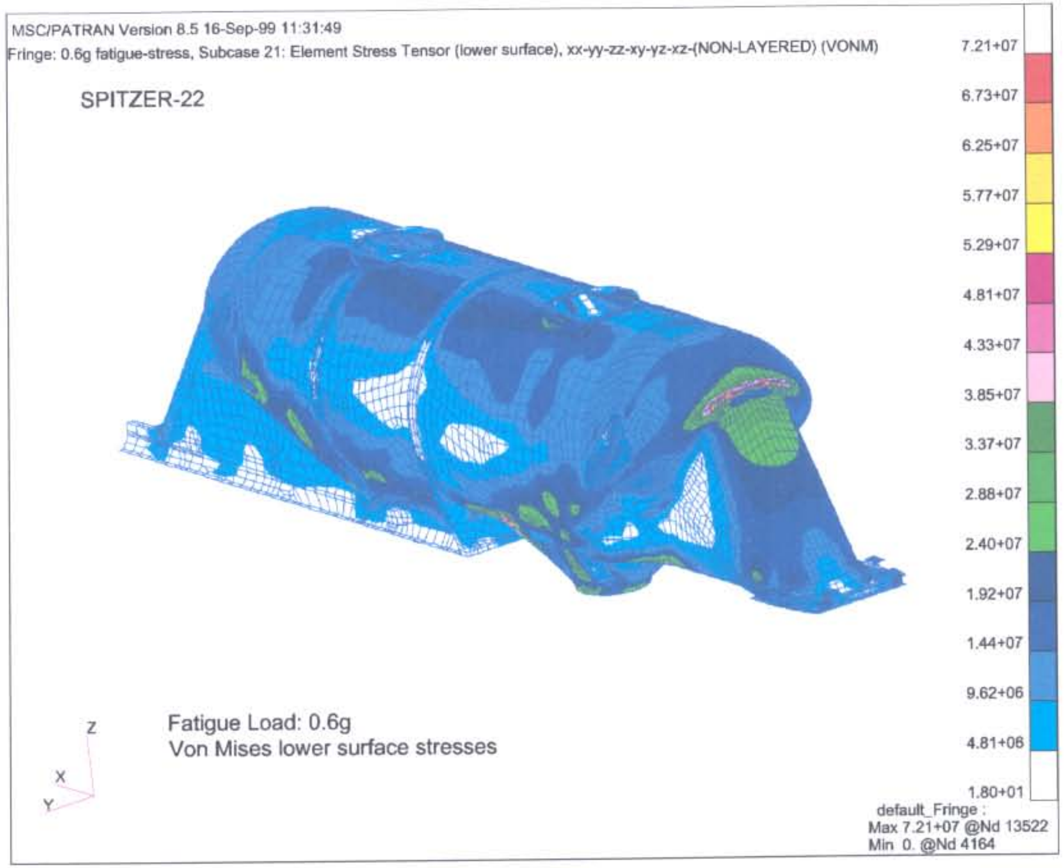


Figure 8.2: Bulk tanker - von Mises stresses

8.3 Sub-frame of a pick-up truck

The static analysis, using the calculated quasi-static fatigue load, indicated that the chassis and the sub-frame would last the design life-time of 200 000km. The finite element analysis of the chassis and sub-frame indicated that no stresses would effect the fatigue strength of the two structures. Stresses of no more than 75 MPa were calculated near the welds on the chassis (refer to figure 8.3). These stresses are well below the weld classifications of the fatigue code [15], and the welded parts would therefore experience no damage during their 200 000km life-time. The sub-frame displayed stresses at some of the welds of approximately 95 MPa. The weld classification of this type of weld is, however, 100 MPa. These welds would therefore last for the designed life-time of 200 000km (refer to figure 8.4).

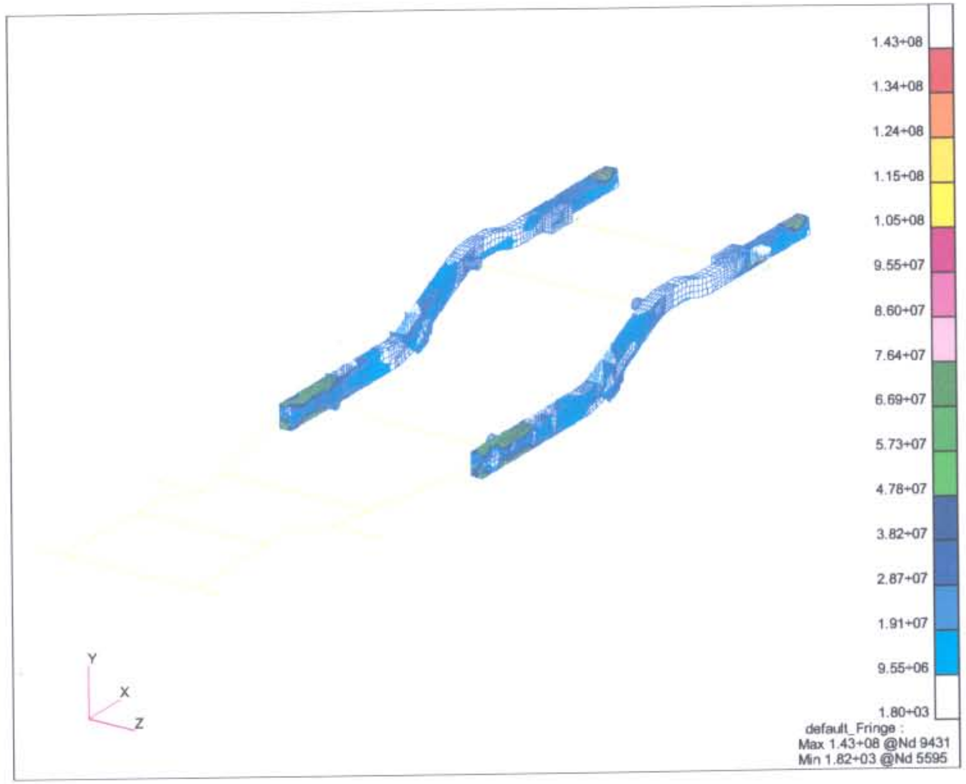


Figure 8.3: Sub-frame: Chassis stresses

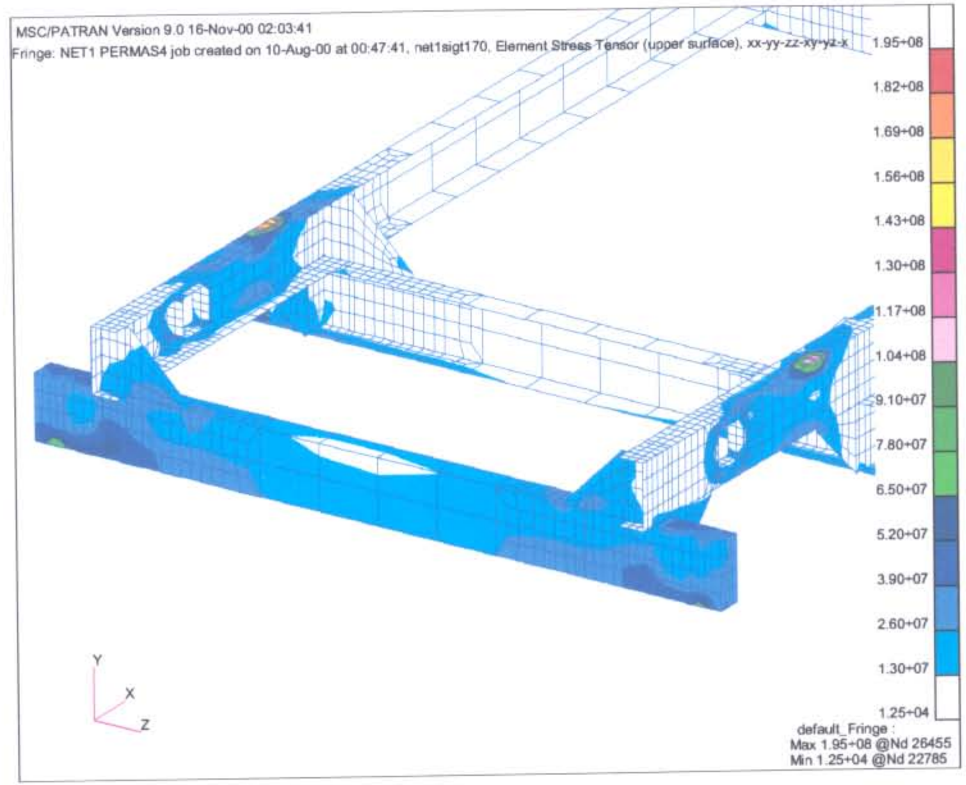


Figure 8.4: Sub-frame: stresses

8.4 Suspension bracket of a large passenger bus

The FE model was subjected to a load of $2kN$. The finite element model consequently showed that the base plate experiences stresses normal to the angled plate in the region of 32 MPa (refer to figure 8.5). A *fatigue equivalent load* of 56.21kN was applied to the model (refer to table 6.7). The new stresses are approximately in the region of 899 MPa. It should be noted that these stresses are fatigue equivalent *stresses*, and the bracket would not actually experience such stresses. According to the ECCS fatigue code [15], the weld that experiences these stresses is a weld with a classification of either 71MPa or 36MPa (an equivalent force of $4.44kN$ and $2.25kN$). A stressed member that falls in the 71 MPa weld classification would endure a nominal stress of 71 MPa for 2×10^6 cycles (refer to section 2.4.3, page 31). The welds on the suspension bracket would fall in the 71 MPa weld category. This is however an optimistic assumption, considering the quality of welds seen on the provided specimen. As mentioned before, the bracket experiences equivalent fatigue stresses of more than 900MPa, while it can only withstand a stress of 71MPa to endure for 2×10^6 cycles (or 3 000 000km). The life time prediction can, however, be calculated for the equivalent fatigue stresses with the stress-life equation 2.6, page 21. According to the fatigue and finite element analysis, using the values from table 6.7 ($56.21kN$, 927MPa), it is predicted that the left suspension bracket would only last approximately 890 cycles or $\pm 1\,350km$ before *crack initiation* would start. The predicted life of the right suspension bracket, using $9.36kN$ and 149.76MPa (refer to table 6.7), would be approximately 213 000 cycles or $\pm$ 320 000 km. These calculations show a close correlation with the actual lifetime of the bracket.

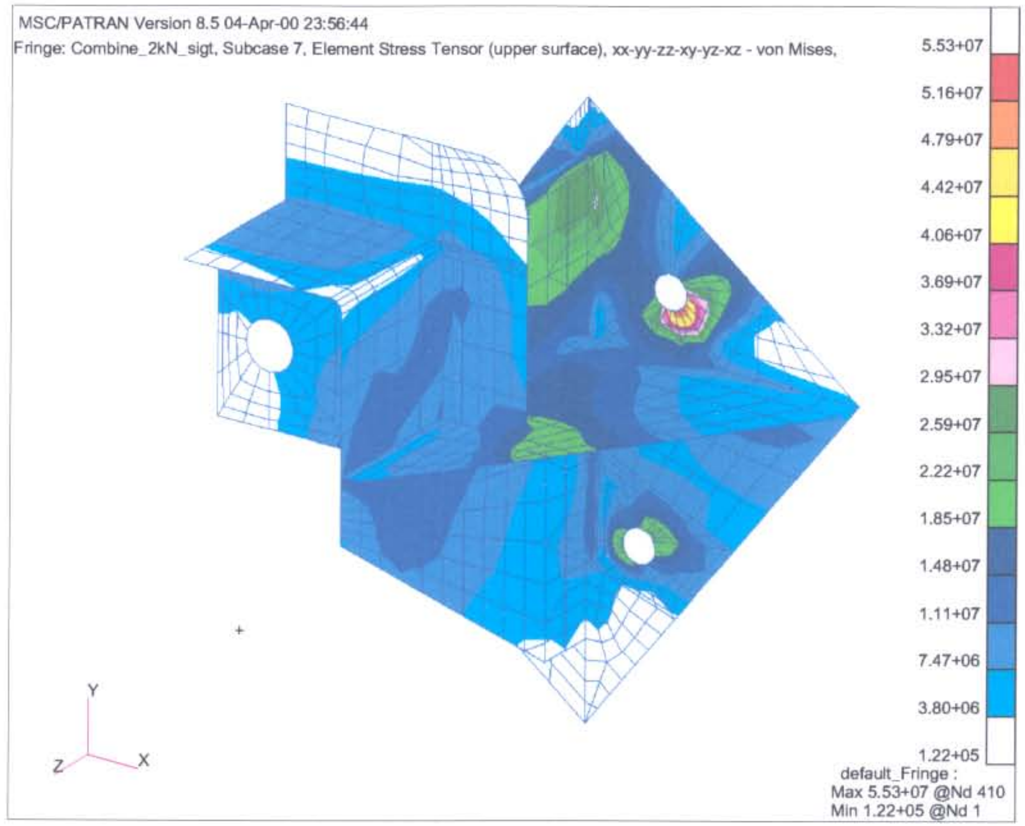


Figure 8.5: Stress results of FEA - bus bracket

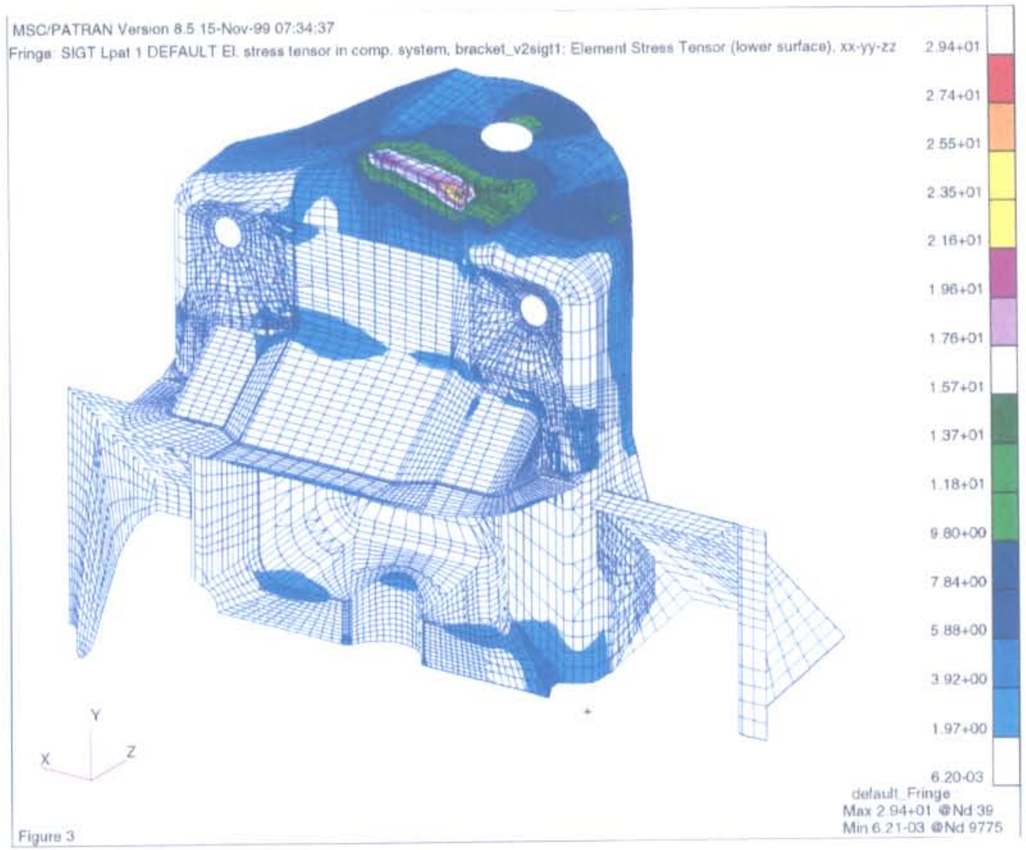


Figure 8.6: Results of FEA: 4x4 pick-up truck bracket

8.5 Suspension bracket of a 4x4 pick-up truck

The front suspension brackets of a 4x4 pick-up truck experienced unacceptable failures during durability testing. Measurements were performed to obtain the loading induced on the shock absorber bracket on the durability test route. A finite element analysis of the original design bracket was performed, resulting in the conclusion that failure is initiated at a stress concentration on a corner between the mounting and back faces of the bracket (refer to figure 8.6). Using the measured data in combination with the finite element results, the fatigue criterion for a modified design was derived. Two iterations of design modifications were performed. Firstly, a gusset welded to the mounting face and the back face of the bracket was modelled and assessed. It became clear from these results that any welding onto the mounting face of the bracket would not achieve the required life expectancy, due to the reduced fatigue strength of a weld. The second design iteration, consisting of an additional U-gusset, welded only to the back face of the bracket, was assessed and found to meet the fatigue criterion. Stresses in the area of the weld were found to be less than 4 MPa and the peak stresses away from the welds on the bracket less than 10 MPa (refer to figure 8.6).

Durability rig testing performed on a baseline (original design) and a prototype modified design specimen, confirmed that the modified design would survive more than 200 000 km of durability route testing. Refer to figures 6.3, 6.4, page 78, 79.

8.6 Closure

In this chapter, the various case-studies were assessed, using the finite element method in conjunction with the static fatigue equivalent load theory. The fatigue load determined for the aluminium bulk tanker, predicted failures on the structure where problems were experienced by the manufacturer. The finite element method enabled the engineers to address the fatigue problems accurately and economically. The sub-frame on the pick-up truck indicated how powerful the static fatigue equivalent load methodology is during the initial design stages of a vehicle. Various iterations could be performed on the design, without using expensive dynamic analyses and a multitude of testing vehicles. The suspension bracket of the large bus and the 4x4 pick-up truck showed the relative accuracy of the static equivalent fatigue load. During the progress of these case-studies, fatigue life predictions were made and verified against actual field data.

In addition, this chapter also showed how the various structures were analysed and how the structural fatigue problems were solved. The following chapter shall present the conclusion of this study.

Chapter 9

CONCLUSION

The present study aimed to provide an in-depth explanation of the Fatigue Equivalent Static Load methodology. Chapter 3 provided a formulation of the fatigue equivalent static load methodology. Through the use of four case-studies, the method was thoroughly described. Chapter 5 discussed the methods employed to determine the input loads and measurements to the vehicle structures. The fatigue calculations performed on the measurement data were discussed in Chapter 6. Chapter 7 discussed the finite element structural analyses that were performed. The assessment of each of the case-studies was discussed in Chapter 8. The fatigue equivalent static load method has considerable advantages above other methods currently being used in the industry:

1. The fatigue equivalent static load method is flexible enough to be successfully deployed using various methods of input loads. To calculate the static fatigue load, a *relative damage value* is needed at a certain position on the structure. The case-studies in this thesis made exclusive use of time domain strain gauge measurements to ultimately obtain the damage value. However, various other methods can also be used to obtain the damage value (for instance, frequency domain measurements, dynamic analyses and virtual simulations).

2. A Fatigue Equivalent Static Load is structurally *independent*. Theoret-

ically, if two different vehicles are travelling over the same terrain, the resulting FESL analysis would yield exactly the same fatigue load. This is of course only true is the vehicles exhibit similiar dynamic characteristics. Similiar dynamic characteristics would be exhibited by vehicles of approximate the same size and mass (and therefore similiar suspension systems). The Fatigue Equivalent Static Load of two 40 ton trucks would therefore be very similiar, although the structures differ from each other. A design team can therefore use a previously obtained fatigue load to evaluate a new structure without any costly measurement exercises.

3. The fatigue equivalent static load can be incorporated in design codes. This will provide an even more cost-effective method to design and manufacture structures for fatigue loads.

4. The fatigue analysis performed with the use of a FESL is applicable to the complete structure. Positions on a structure that are very difficult to access for measurement purposes can therefore be easily evaluated using a finite element analysis.

The Fatigue Equivalent Static Load method does have a few disadvantages. It is therefore recommended that when this methodology is employed, the following points are considered:

1. The main disadvantage of the FESL method is its inability to address fatigue failures caused by dynamic vibrations. The fatigue load, obtained using the FESL method, subjects a structure to a static deformation. The static deformation would often also correspond to the first global mode shape (eg. chassis under vertical inertial load). The FESL method would therefore take into account dynamic response induced at the first mode. Quasi-static response and excitation frequencies much lower than natural frequencies would also be taken in account. Most of the fatigue related problems that would occur on a structure can be attributed to dynamic response lower that the first mode of bending category. However, loads that excite the structure at higher frequencies would not be

accounted for in the FESL load. An excellent example is shown with the loads calculated on the dry-bulk tanker (see section 6.2, page 66).

The methodology explained by Olofsson et al is especially suited for fatigue analyses caused by dynamic vibrations (see subsection 2.5.5, page 39). Incidently, Olofsson makes use of a very similar methodology to obtain his results (refer to [27]). The Fatigue Damage Response Spectrum (FDRS) method can therefore be successfully combined with the FESL method. The disadvantage of dynamic loads present in higher order bending modes would therefore be solved.

2. Another disadvantage of the FESL method is the use of finite element analysis. Although the finite element analysis procedure is a well established computer aided engineering (CAE) tool, the cost of such an analysis is still more expensive than using, for instance, hand calculations. In addition, a finite element analysis must be done by a person that is experienced and well trained in the use of this tool. The costs of finite element packages are, however, steadily declining. It should be noted that a structure designed with the use of finite element analyses would necessarily yield a much more structurally effective design.

In conclusion, the Fatigue Equivalent Static Load method is an engineering tool that delivers accurate and reliable results to one of the most common structural problems experienced by the industry. The FESL method must however, as with any newly developed engineering tool, be rigorously verified to ensure the robustness of this technique. Complex fatigue loads for vehicle structures do not need to be the main stumbling block of the mechanical design engineer.