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Weight Reduction of Structural Vibration Isolation Hydro-Mount Bracket through Design Analysis and Use of Advanced High Strength Steels

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ABSTRACT

Vibration Isolation Mounts are designed to isolate noise, reduce resonant vibration and lower structure-borne noise. The external mounting bracket that encases the fluid filled inner mount is usually made from relatively thick (~3 mm) high strength low alloy steel and has critical structural and durability requirements. In this study, a practical approach combining FEA and physical testing was used effectively in component design with complex functional requirements. A new lighter weight configuration with ~ 8% weight savings was validated to pass all the structural and durability requirements.

INTRODUCTION

An ideal mounting system has two conflicting requirements; to be stiff for comfort and soft for acoustic reasons. These demands cannot be met simultaneously with conventional mounts. Hydraulic mounts were introduced in the 1970s and have been standard for almost all engines and power-trains in passenger cars since the 1990s. Combining highly elastic rubber (with favorable acoustic properties) with hydraulic devices (damping vibrations) generates the damping effect at a selected frequency within the mount (1-3) A typical mount in a passenger car is shown in Figure 1 below:

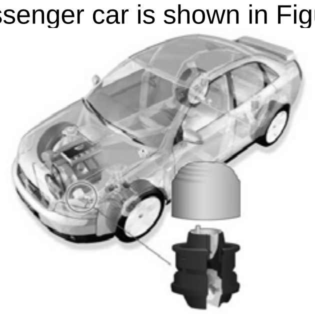

Figure 1: Typical hydro-mount shown in vehicle position (http://www.trelleborg.com)

During development, the hydro-mount undergoes extensive virtual and real life testing to optimize product performance. Advanced computer software is used to simulate different configurations and their performance in various driving maneuvers. Simulations are used to analyze, for example, damping of vibrations when the engine is switched on/off or idling, as well as torque reactions in different gears.

The steel components of the hydro-mounts are commonly stamped from 3-4 mm HSLA hot rolled steel sheet, although other fabrications, such as castings, are sometimes used. The molded rubber steel-cased inner mount is inserted into the hydro-mount housing, often with substantial press-fit. The hydraulic fluid (ethylene glycol) is injected in the final assembly process with requisite pressure.

The hydro-mount bracket has several functional requirements. As a critical component in the vibration isolation system, the hydro-mount bracket has to adequately protect the fluid filled rubberized inner mount from shock loads in vehicle service as well as meet long term durability performance expectations. At the same time, it must support the weight of the engine and transmission and the in-service loads.

CURRENT HYDRO MOUNT BRACKET DESIGN & PERFORMANCE REQUIREMENTS

The current hydro-mount bracket design is shown below, comprising a deep drawn can two legs and an upper bracket with mounting holes that are fixed to the vehicle frame. The inner-mount which is fluid filled is inserted into the outer bracket assembly as shown in Figure 2.

Figure 2: Current hydro-mount assembly with the inner mount press fit into the outer bracket.

STATIC LOAD REQUIREMENTS

Table 1 shows the maximum service loads that the hydro-mount bracket was expected to withstand without exceeding a Von Mises stress level greater than 70% of material yield strength.

Table 1: Static Load Requirements

Direction	Load (KN)	Failure Criterion
Z+	14.3	70% Y.S
Z-	14.3	70% Y.S
X+	11.7	70% Y.S
X-	11.7	70% Y.S
Y+	7.8	70% Y.S
Y-	7.8	70% Y.S

DURABILITY REQUIREMENTS

Durability test requirements vary upon application. In some cases, samples are subjected to vector loading conditions at a given frequency for a specified number of cycles. For the evaluation of these samples, a vector load condition was applied based on proprietary data received from the OEM. Each sample had to pass 400,000 cycles at a specified cyclical load and frequency.

MATERIAL SPECIFICATIONS

All the components in the hydro-mount bracket assembly were specified as 3 mm thick HSLA 350 MPa material. The rubber is a highly damped and highly elastic compound typical of hydro-mount applications.

STRUCTURAL FEA BASED PRODUCT DESIGN & DEVELOPMENT

STATIC ANALYSIS - BASELINE

Baseline static analysis was conducted using the commercial finite element analysis code (LS-DYNA explicit) using ETA-VPG as the preprocessor to set up the model. All components in the assembly (can, legs, and upper bracket) were modeled as solid elements. Although it is common to model sheet metal parts as shells in forming analysis, these components are fairly thick (3 mm) and for structural analysis we felt that it was better to model the geometry as solids. The physical welding of the can to the legs and brackets was also modeled explicitly in this study as shown in Figure 3.

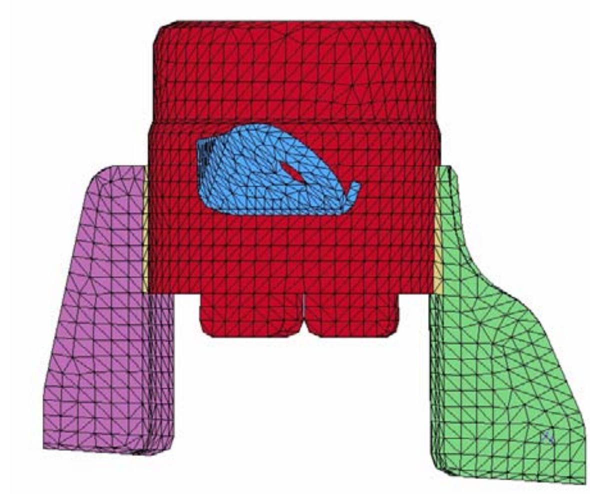

Figure 3: Finite element analysis model for static loading simulations.

The loads were applied, as shown in Figure 4, using a linear ramping to maximum value of 14.3 KN in the negative Z direction and stress results were taken at 11 ms for comparison of different assembly configurations.

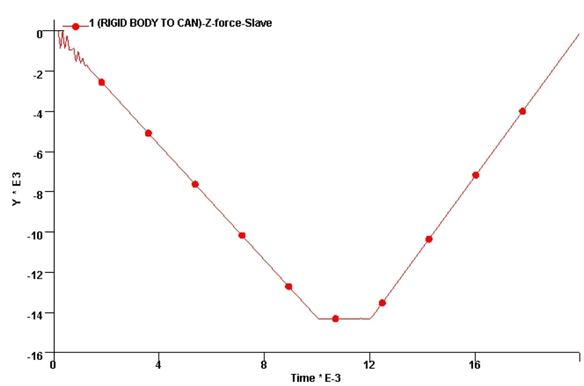

Figure 4: Load profile used in FEA analysis. Maximum applied load is held for 2 ms at minus 14.3 KN

A Von Mises stress map is shown in Figure 5 for the Z-loading condition of -14.3 KN. The maximum VM stress is ~160 MPa and is less than 50% of the initial material yield strength of HSLA 350 MPa material. The stresses on the legs are only 35% of the material yield strength making them the target components for weight reduction.

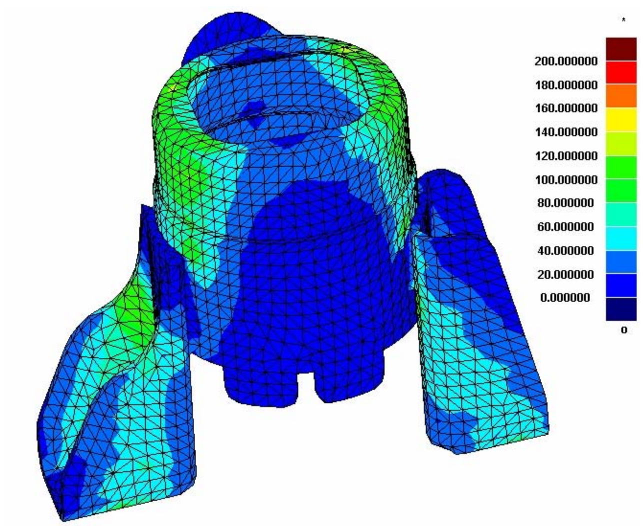

Figure 5: Von Mises stress map. showing that maximum stress is less than 50% of the original yield strength of the HSLA 350 material

ALTERNATE PRODUCT CONFIGURATIONS

The different assembled product configurations considered in the study are shown below in Table 2.

Table 2: Alternate Product Configurations

Configuration	Can	Legs	Weight Savings
Baseline	3 mm HSLA	Original Design 3 mm HSLA	
TC1	3 mm HSLA	New Legs, V1 3 mm HSLA	12%
TC2	3 mm HSLA	New Legs, V1 3 mm AHSS	12%
TC3	3 mm HSLA	New Legs, V1 2.3 mm AHSS	17%
TC4	3 mm HSLA	New Legs, V2 3 mm HSLA	8%

The geometry of the new leg design used for alternate product configurations (TC1-TC3) is shown in Figure 6.

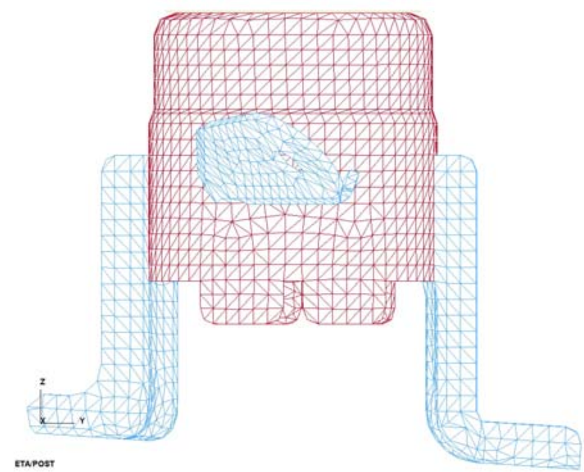

Figure 6: Alternate product configuration with weight-saving potential of 12% without material substitution

A Von Mises stress map is shown in Figure 7 below for the 3 mm HSLA material with the stress contour range same as before (0 to 200 MPa). From a static loading requirement perspective, the new legs easily meet the maximum stress criterion of 70% of original yield strength.

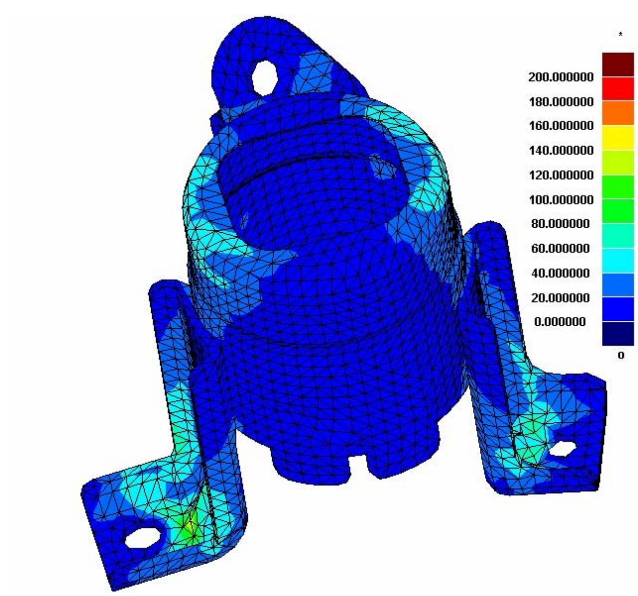

Figure 7: Von Mises stress map showing that the lighter legs meet the static loading requirements

EXPLORING USE OF ADVANCED HIGH STRENGTH STEELS FOR HYDRO-MOUNT BRACKET APPLICATION

The material currently used for this application is produced to meet ASTM A1011, Grade 340 hot rolled pickled sheet. The chemistry used is typical of HSLA grades using columbium as a strengthening alloy. This grade has been producing this part consistently since die development was completed.

Alternate product configurations using AHSS were considered in this study as indicated in Table 3 (TC2, TC3). Dual Phase steels have a microstructure of ferrite with regions of martensite acting as the strengthener (4,5). The martensitic structure is produced by alloy additions, often high manganese with titanium, vanadium or columbium, combined with heat treatment. The ferrite regions deform easily, making the steel more formable. Initial strength is determined by the fraction of martensite present. Further strengthening comes from the degree of work hardening of the ferrite. Dual Phase steels often have further strength gains from bake hardening. Dual Phase welding practices are similar to HSLA, facilitating substitution in welded assemblies compared to more complex AHSS steels.

Advanced high strength steel samples were supplied by different steel sources (Table 3). While AHSS has been applied successfully to frame rail components so far, there is little prior work regarding applicability of these grades to interior components. Of particular interest in this study was possible weight reduction using thinner AHSS steel for the legs, and to determine whether the AHSS grades would perform better under cyclic fatigue loading.

Table 3: Mechanical Properties of Sample Materials

Sample	Thick (mm)	Yield Strength (MPa)	Tensile Strength (MPa)	Total El. (%)	n value
340 HSLA	3	375	451	25	0.14
AHSS-1	2.3	439	628	24	0.17
AHSS-2	3	428	654	23	0.18

Modified new legs were produced AHSS steels at 3 mm and 2.3 mm thicknesses. Welded assemblies in different configurations (3mm HSLA can with 3 mm original design legs and with new legs at 2.3 mm and 3.0 mm dual phase steels) were produced for subsequent static and durability evaluation.

EXPERIMENTAL VERIFICATION OF STATIC AND DURABILITY PERFORMANCE

STATIC COMPRESSION TESTING

Static compression load tests were performed on the as-welded assemblies. These checked if the parts met the 14.3 kN z-load compression requirement, checked weld strength as a potential failure source, determined what was the actual crush load the part could support before buckling, and compared those results with the model's predicted behavior. The assemblies were mounted in a fixture that duplicated the vehicle mounting points, then loaded in compression using a Satec hydraulic test frame. Figure 8 shows the test setup with the TC4 product configuration. The bottom legs and the upper bracket are locked and compression load is applied on top of the can mount.

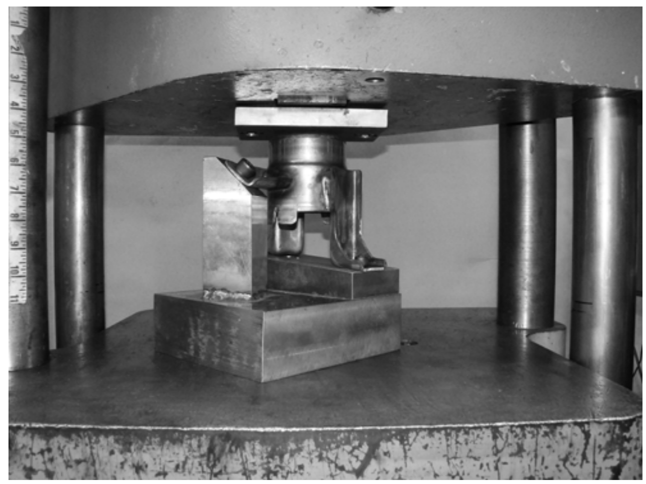

Figure 8: Test setup for static compression testing shown with TC4 product configuration.

Static test results are presented in Table 4 for the different product configurations. All specimens passed the 14.3 KN Z-load compression requirement without any visible distortion.

Table 4: Static Test Results

Product Configuration	14.3 KN Z load	Buckling Load, KN	FEA Prediction, KN
Baseline	Passed	158	143
TC1	Passed	130	NA
TC2	Passed	NA	NA
TC3	Passed	NA	NA
TC4	Passed	NA	NA

Product configurations TC1 and TC2 were loaded until buckling. The original design held an average load of 158 kN before onset of buckling; the first modified design held 130 kN. Figure 9 shows buckling on the top of the can for the baseline product configuration.

Figure 9: Baseline test configuration sample showing buckling on the top of the can.

Finite element analysis correctly predicted the failure mode and buckling load, showing good correlation to the test results as shown in figures 10, 11 for the baseline product configuration.

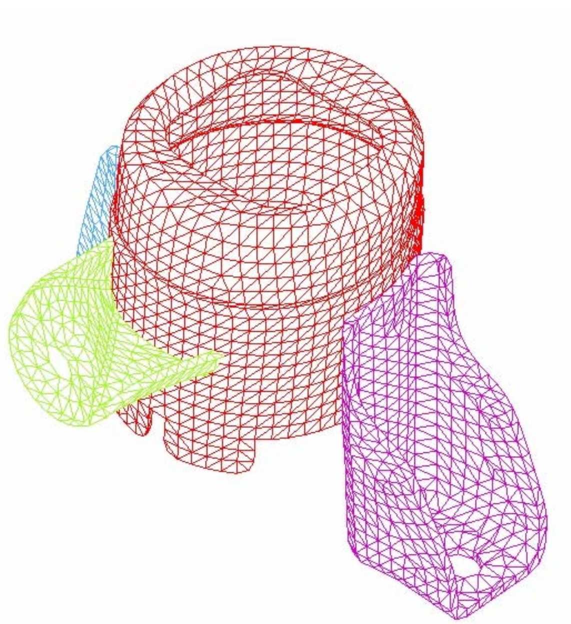

Figure 10: FEA results for the baseline test configuration sample showing buckling on the top of the can.

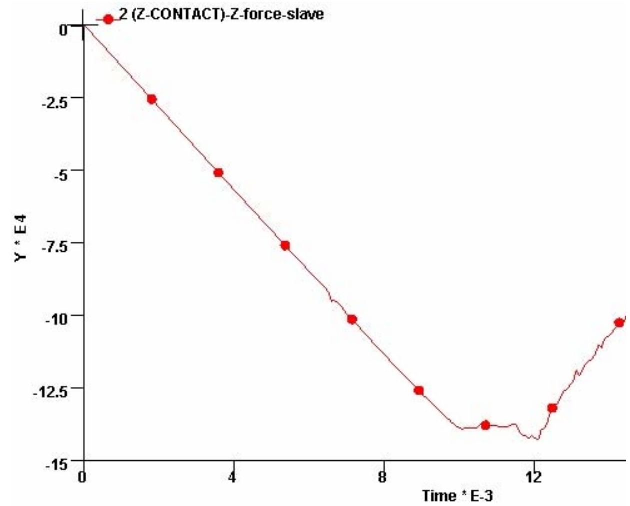

Figure 11: FEA results for the baseline test configuration sample showing predicted buckling load of 143 KN compared to experimental measurement of 158 KN.

DURABILITY TESTING

Durability testing was performed using the loading profile defined earlier. Figure 12 shows the approximate loading direction for the TC4 product configuration. The durability requirement for the bracket is to be free of cracks or breaks after 400,000 cycles.

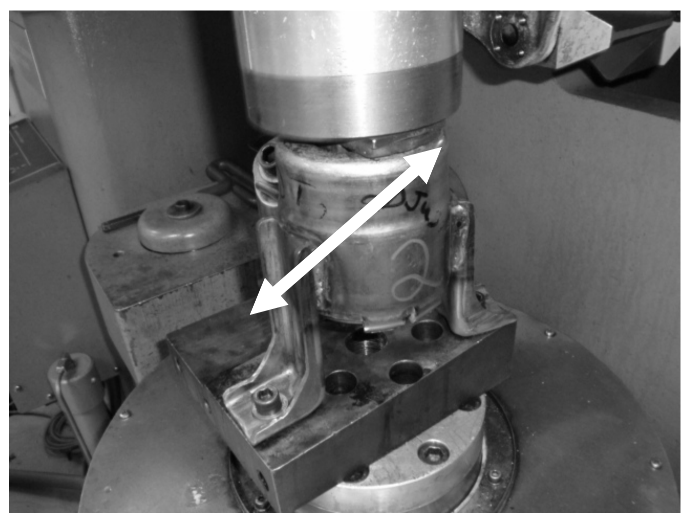

Figure 12: Set up and cyclic loading direction for durability evaluation with the TC4 specimen.

Durability test results are presented in Table 5 for the various product configurations. The baseline configuration and TC4 were the only ones to pass the durability test requirement of 400,000 cycles under imposed cyclic loading conditions.

Table 5: Durability Test Results

Product Configuration	Durability Cycles	Pass/Fail
Baseline	400,000	Pass
TC1	322,000	Fail
TC2	232,745	Fail
TC3	NA	Fail
TC4	400,000	Pass

Interestingly the HSLA sample TC1 performed better than the corresponding AHSS sample, TC2. A possible reason lies in the higher n value for AHSS steels which increases strength level substantially after forming, resulting in lower fatigue life. Figure 13 below shows the failure mode for the TC2 specimen.

Figure 13: Failure mode after 232,745 cycles with TC2 specimen made from AHSS steel.

While the original leg design passed the durability test, the first version of the new legs (TC1-TC3) did not quite meet the 400,000 cycle requirement with failure occurring at the 90 degree bend location. It was apparent that there was a stress concentration at the bend corners, so a modified version of the legs (TC4) was produced as shown in Figure 12 with additional material reinforcement at the corners. The test sampled was examined after the durability testing. No cracks or breaks were found in the bracket after 400,000 cycles, as shown in Figure 14.

Figure 14: TC4 specimen with additional material reinforcement at the corners after passing 400,000 cycle requirement in the durability test

DISCUSSION AND FUTURE WORK

This work demonstrates a practical approach combining FEA and physical testing that can be used effectively in component design with complex functional requirements including static and durability response. FEA in static analysis is quite well developed and usually forms the backbone in design of interior components. For the most part, much of the static analysis done in Industry is performed assuming linear elastic behavior of the material. In several cases when loading is beyond the elastic limit the stress predictions based on elastic material considerably over-predicts the stress in the component resulting in rejection of the design. We would recommend modeling of the component as an elastic-plastic material for more accurate stress prediction in such cases.

In this work we suggest use of solid elements (rather than shells) in structural analysis for thick sheet metal components, particularly in the case involving welded assemblies. This facilitates modeling of the weld explicitly and allows the stresses in the weld to be correctly addressed. Figure 15 shows the weld region explicitly modeled as solid elements and assigned material properties of an HSLA 420 MPa steel as an example. Figure 16 shows Von Mises stress development in the weld region.

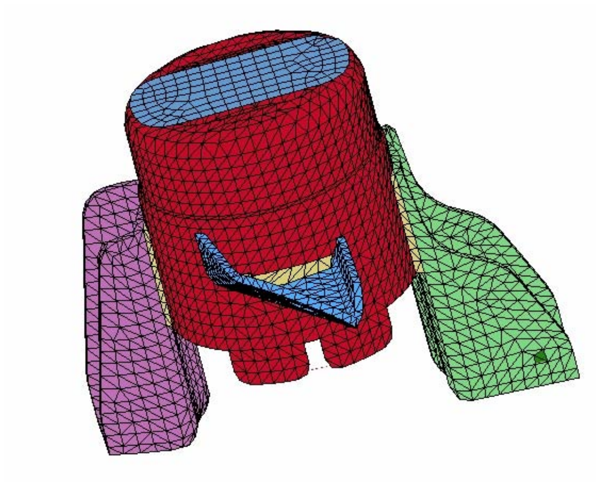

Figure 15: FEA analysis with the welds modeled as separate solid elements

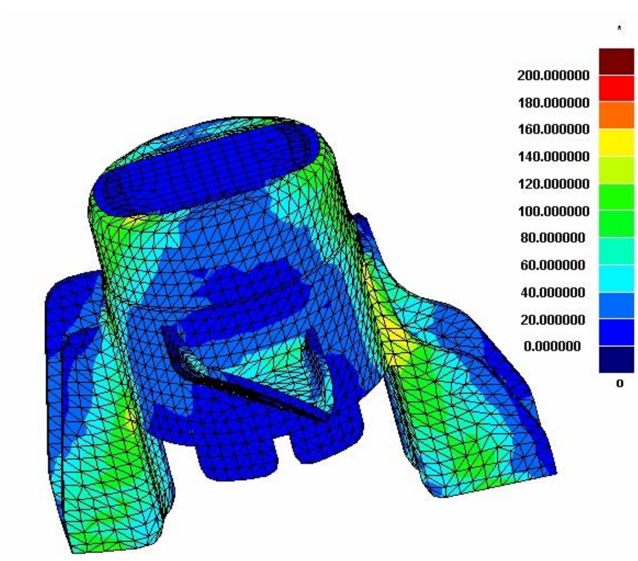

Figure 16: FEA analysis showing VM stress in the welded region

Another issue of importance in the design of components is the need to incorporate the effects of forming related strain hardening and thickness changes in subsequent structural analysis. For this particular component with the buckling occurrence at the top of the can, including hardening effects may not be appreciable since this region experiences very little cold work. A further constraint at this time is the lack of an effective interface between shells used for forming analysis and solids used in the structural analysis. Capturing the effect of strain hardening and thickness variations from the stamped part is ongoing future work on this project.

Durability was an important consideration in the design of the mount assembly. While we are aware of simulation software in this space, we do not believe that fatigue life predictions based on simulation are sufficiently reliable at this time particularly for AHSS steel which has a very complex microstructure. In this study durability results are based only on physical tests.

Finally, FEA was also very useful in process development in this work. There were manufacturing issues associated with insertion of the fluid-filled inner mount into the outer mount bracket that were successfully addressed. Due to the proprietary nature of this work, we are unable to share the benefits seen with application of FEA in process design.

CONCLUSION

A judicious blend of finite element analysis and physical durability testing has been applied in product and process development of a hydro-mount bracket assembly. The following conclusions apply:

1. Weight reduction of 8% was possible by just changing the design of the lower legs without any material substitution. Bracket assemblies with the new legs passed the stringent durability requirement of 400,000 cycles at load.

2. AHSS has been explored as a possible substitute material at lighter gages for further weight savings opportunities; however, initial durability testing results have not been too encouraging. It is postulated that the higher rate of strain hardening in AHSS steel may be responsible for lowering the fatigue life in this application.

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