Load Identification of a Suspension Assembly Using True-Load Self Transducer Generation

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Abstract

The performance of a structural design significantly depends upon the assumptions made on input load. In order to estimate the input load, during the design and development stage of the suspension assembly of a BAJA car, designers and analysts invest immense amount of time and effort to formulate the mathematical model of the design. These theoretical formulations may include idealization errors which can affect the performance of the car as a final product. Due to the errors associated with the assumption of design load, several components might have more weight or may have less strength than needed. This discrepancy between the assumed input load (lab or theoretical studies) and the actual load from the environment can be eliminated by performing a real life testing process using load recovery methodology. Commercial load cells exist in industry to give engineers insight to understanding the complex real world loading of their structures. A significant limitation to the use of load cells is that the structure needs to be modified to accept the load cell and not all desired loading degrees of freedom can be measured. The testing procedure followed in this paper replaced load cells with strain gauges and used strain response in conjunction with a correlation matrix from FEA to estimate the true value of input load under real life conditions. The suspension assembly itself will act as a transducer by converting the load into change in electrical resistance of attached strain gauges. The true load acting on the suspension assembly can be estimated from the strain response recorded from the suspension components. Strain gauge placement was determined using True-Load software which creates a correlation matrix relating to the strain response at the gauge locations due to user defined unit load cases. The strain gauge measurements together with the correlation matrix calculated by True-Load give the best estimate of actual load. After determining the true load, the designers redesigned the structural components that can guarantee a better performance in real life situations. The major objectives of this work are to enhance the total performance of the car by designing components to its optimum performance and to identify the true load acting on the rear suspension components.

Introduction

Technical assumptions and analytical decisions are a vital part of engineering designs. During the design and development stage of a structural component, engineers usually assume the design load, based either on prior experience or on available experimental data. It is obvious that any miscalculation or incorrect assumption can severely affect the structural integrity under design, or mitigate the economic feasibility of the design. There are several conventional methodologies followed to avoid these problems. 1) Continuous monitoring or maintenance techniques can be exercised on the structure after fabrication to avoid any potential failure. 2) A high design allowance can be used to ensure that the working load falls inside the assumed load limit. Both of these methodologies are conventional and have the disadvantages mentioned in the beginning. An unconventional methodology, which can easily overcome the disadvantages of conventional methods, is to design the structure after identifying the true value of operational loads. By performing this methodology, the engineer can assure quality and reliability. There are two different methods to identify the true value of operational loads. First method is to directly identify the load using a load cell, and the second method is to estimate the operational loads using structural strain response.

While performing a technical comparison between the two methods, the second method shows several advantages over the first method. The following table compares the advantages of using structural response from sensors, such as strain gauge to computationally estimate loads from structural response over directly using a load cell to calculate operational loads.

Table 1. Chart comparers between using structural response from a sensor over a load cell to calculate operational loads.

Load Cells	Structural Response from Strain Sensors
Not all types of loads can be measured	All type of loads such as thermal, seismic, and wind, etc can be measured
May need some structural modification for installation of load cells.	No structural modification is needed.
Will affect the original response due to mass and stiffness changes of the structural modifications and presence of load cells.	Will not affect structural response since structure is not modified.
Limited access to load cases	Unlimited access to load cases

From Table 1 it is clear that in order to perform a reliable and accurate design load calculation it is advantageous to use structural response from strain measurement. Several studies have been performed in the field of load identification using sensors. Hillary and Ewins [1] employed sensors on structures to estimate sinusoidal loads acting on a test cantilever beam using least square method. Later, Busby and Trujillo [2] designed the true operational load identification methodology as a problem of error between the measured structural response and response predicted from the design model. They defined this issue as a pure minimization problem which concentrates on reducing the error between the above mentioned factors. Time domain techniques are the most recent and accepted methods in the field of load identification.

In 1992, Carne et al.[3] proposed a methodology that is known as the Sum of Weighted Acceleration Technique (SWAT) that computes the input loads by summing the weight scaled measured accelerations. Steltzner and Kammer [4] developed a technique using an Inverse Structural Filter (ISF) that processes the structural response data and returns an estimate of the operational loads. Law et al. [5] developed a finite element analysis based recovery methodology to recover the true operational value of moving loads acting on a structure. Hashemi and Kargarnovin [6] created an objective function that was developed as the difference between analytical and measured responses and structured the whole problem as a pure optimization problem. Hiray et al. [7] applied the strain response collected from a motorcycle main frame into the matrix inverse calculations and finite element analysis to estimate the operational loads.

Dhingra, Hunter and Gupta [8] extended an algorithm based on D-optimization (Determinant optimization) that can identify optimal sensor locations including their local orientations which in together can give the most accurate load estimates from structural response. D-optimal methods utilize sequential-exchange algorithm to select optimum sensor locations. Hunter, based on his research and experience, developed the True-Load software that enables engineers to use D-optimization technique for any structural components to identify true value of operational loads. This software was used in this test event predominantly.

The structure under investigation in this paper is the suspension assembly of a SAE BAJA Car. The suspension assembly includes suspension links, upright and a suspension tab. The suspension links and tabs were made of Chromoly 4130 Steel and the upright was made of Aluminum. The designers had invested huge amount of time and effort to formulate a mathematical model of design to estimate input load. Even though designers have a rough notion about operational loads, assurance on reliability was unable to be guaranteed. This lack of confidence can be easily eliminated by performing an unconventional testing process. True value of operational loads can be computed from strain responses and these structural responses can be measured from strain gauges placed on the suspension assembly. In order to perform the test three design and analysis software were used. 1. Pro-E Creo, 2. Abaqus/CAE and 3. True-Load. Following section will explain the theoretical background behind the True-Load software and estimating operational loads from strain response.

Theoretical Background

The applicability of the approach outlined by Dhingra, et al, requires the structure to behave linearly under the event of interest. The structure may behave non-linearly prior to or after the event of interest. The term linear in this context refers to structures whose flexure is sufficiently small (e.g. sin $(\theta) \approx \theta$). In addition, and most importantly, the strain response can be thought of as being proportional to the applied load. With these conditions being satisfied, a general expression may be written representing this proportionality: Eqn. (1) is a linear relationship between the applied load cases [F] in the finite element model and the resulting strains [ε] as retrieved from the finite element model at prescribed locations and orientations. The term [C] is the matrix of proportionality that needs to be determined through standard linear algebra manipulations. The strain matrix [ε] will have dimensions of *n* loads by *m* gauges. Each row of the strain matrix will consist of the strain as measured from the finite element model to the corresponding load case. Each column of the strain matrix corresponds to a specific location and orientation on the finite element model. These locations are determined through the D-optimal approach outlined in Dhingra, et al [8]. In essence, these are virtual uniaxial strain gauges. The loading matrix [F] has dimensions of *m* loads by *m* loads. The approach outlined in Dhingra, et al and applied in the True-Load software uses a diagonal representation of the [F] matrix.

$$[F] = [\varepsilon][C] \tag{1}$$

Specifically, the [F] matrix is represented as:

$$[F] = \begin{bmatrix} F_1 & \cdots & 0 \\ \vdots & \ddots & \vdots \\ 0 & \cdots & F_m \end{bmatrix} \tag{2}$$

Furthermore, the values for the entries in each of the elements of the [F] matrix can be thought of as scale factors for the applied load cases. When a load case is turned "on" the corresponding value in the matrix becomes 1. Conversely when the load case is turned "off" the corresponding value becomes 0. Following this convention, when diagonal terms of the F matrix are simply represented by the value of 1, each load case is being activated independently of the other load cases. This then allows the F matrix to be represented as the identity matrix [I]. This simplifies equation [1] to be of the form:

$$[\varepsilon]][C] = [I] \tag{3}$$

This expression will give rise to the pseudo inverse relationship to determine the value of the [C] matrix. The pseudo inverse is used since the strain matrix [ε] in general will not be square.

$$[C] = [[\varepsilon^T][\varepsilon]]^{-1}[\varepsilon]^T \tag{4}$$

The [C] matrix is the load proportionality matrix. It specifically is the proportionality relationship between strains [ε] and loads [F]. The stability of [C] is dependent upon the inverse of the dot product of the strain matrix. A nearly infinite number of configurations of the [C] matrix exist, but there are a limited number of [C] matrices that behave “well”. “Well” behaving [C] matrices will be relatively insensitive to signal noise and gauge placement accuracy. In order for [C] to be stable, the determinant of the strain matrix [ε] needs to be maximum. A key attribute of the strain matrix [ε] is the condition number of the strain matrix [ε]. Heuristic evidence has shown that a very stable strain matrix [ε] has a condition number of 10 or less. Acceptable strain matrices have condition numbers of 50 or less. Systems that exhibit a condition number of 50 or more should be reexamined for suitability of load cases and candidate gauge locations. Large condition numbers indicate that the system of load cases and strain gauge locations does not have sufficient linear independence. Following section will explain the projection of the above mentioned theory to the suspension assembly problem using True-Load software.

Suspension Assembly Unit Load Cases and Test Procedure

The testing process starts with a model, designed for manufacturing. Fig .1 illustrates the whole testing process and is described in this paper.

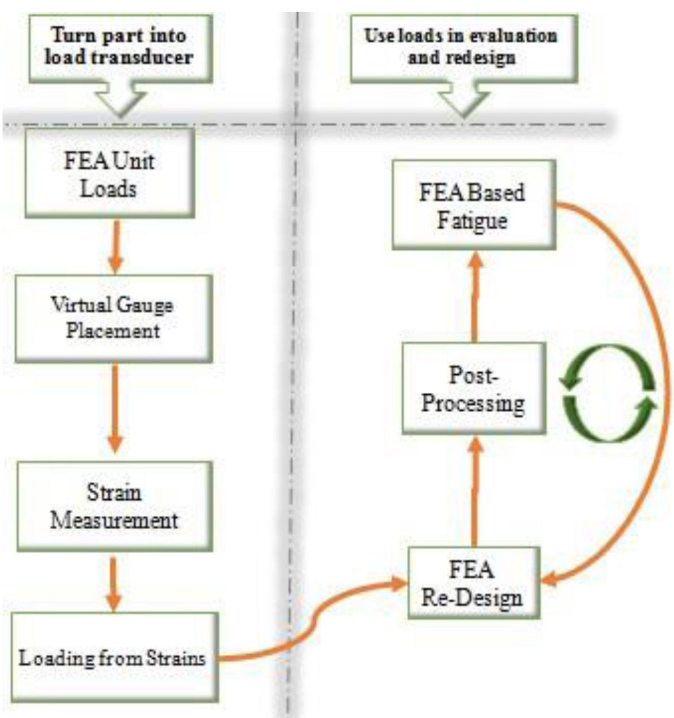

Figure 1. Product Development and Test Cycle using True-Load

Once the design was finished and the prototype was ready, the suspension assembly design model was imported into FEA software. The design software used in this testing process was ProE Creo and the FEA software used is Abaqus/CAE. The model was properly formulated for boundary conditions and for other constraints including weldments and joints. The designer at this point should think about the incoming loads that need to be known during the design stage, and the possible locations of acting loads. Any complex load can be decomposed into combination of load vectors using the principles of mechanics. Each possible acting load is counted as a load case. The suspension assembly had a total of nine load cases and is shown in Fig. 2.

The process started with applying loads of unit magnitude at these load cases one after another. The unit loads on the structure need to of sufficient magnitude such that the strain response is large enough such that the subsequent mathematical operations on the strain data have numerical significance in digital computers. The unit load in this problem was chosen to be 100lb. Fig. 2 demonstrates the unit load cases for the suspension assembly. Each yellow arrow shows a load case. It is important to remember that after the testing, the designers will be able to recover the loads for these nine-unit load cases as a time series of operational loads. If there exist more locations where the loads are to be identified, more load cases must be included in the model. This experimental load measurement process is highly related to influence coefficient method in which the designer know the strain produced for a unit load and can calculate the load produced by an actual strain.

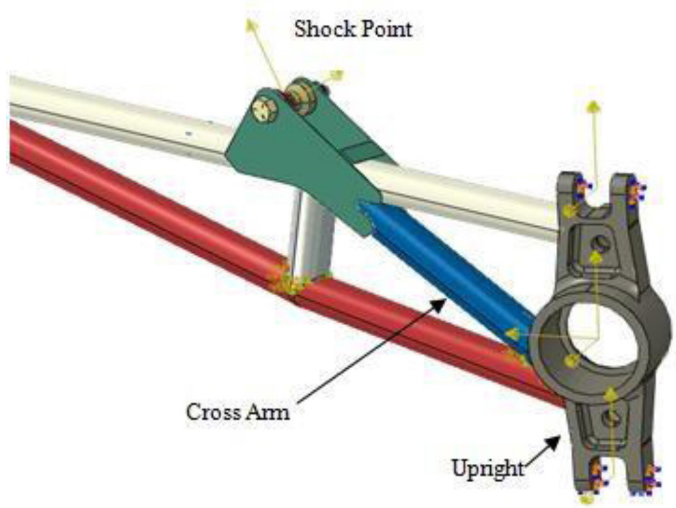

Figure 2. Suspension assembly design model with nine-unit load cases.

Load Table

Row	ODB Step	Frame	Frame Description
1	Unit-Loads	1 -	Load Case: SHOCK 100LBF-FX
2	Unit-Loads	2 -	Load Case: SHOCK 100LBF-FY
3	Unit-Loads	3 -	Load Case: UPRIGHTBOTTOM-100LBF-FY
4	Unit-Loads	4 -	Load Case: UPRIGHTBOTTOM100LBF-FX
5	Unit-Loads	5 -	Load Case: UPRIGHTCENTRE100LBF-FX
6	Unit-Loads	6 -	Load Case: UPRIGHTCENTRE100LBF-FY
7	Unit-Loads	7 -	Load Case: UPRIGHTCENTRE100LBF-FZ
8	Unit-Loads	8 -	Load Case: UPRIGHTTOP100LBF-FX
9	Unit-Loads	9 -	Load Case: UPRIGHTTOP100LBF-FY

Figure 3. List of nine-unit load cases applied on the suspension assembly.

After solving the problem in Abaqus/CAE for nine-unit load cases, the Abaqus results file was processed into the True-Load/Pre-Test software. The Pre-Test software will show the location and local orientation of strain gauges to be placed. Since there are infinite number of locations on a structure, it is important to place the strain gauges at locations where the load estimates are as accurate as possible. There are statistical studies [9] that proves that the location of strain gauge placement can significantly influence the load estimation process.

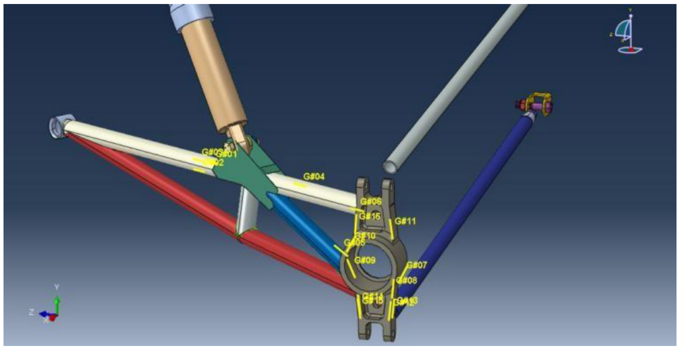

Figure 4. Virtual strain gauge locations from True-Load Pretest software.

Gauge Number	Part Instance	Element Label	Ang Deg
G1	FORWARDLINK_PIPE_1_MIR-1	117	195
G2	FORWARDLINK_PIPE_1_MIR-1	3911	135
G3	FORWARDLINK_PIPE_1_MIR-1	107	225
G4	FORWARDLINK_PIPE_1_MIR-1	1990	14
G5	FORWARDLINK_PIPE_7_MIR-1	405	70
G6	FORWARDLINK_PIPE_1_MIR-1	3732	165
G7	UPRIGHT_V12-1-1	39746	80
G8	UPRIGHT_V12-1-1	36462	263
G9	UPRIGHT_V12-1-1	38467	89
G10	UPRIGHT_V12-1-1	38157	90
G11	UPRIGHT_V12-1-1	40459	110
G12	UPRIGHT_V12-1-1	39080	96
G13	UPRIGHT_V12-1-1	39290	90
G14	UPRIGHT_V12-1-1	38919	90
G15	UPRIGHT_V12-1-1	39039	110
G16	UPRIGHT_V12-1-1	40282	98

Figure 5. List of sixteen virtual strain gauge locations from True-Load Pretest software.

The mathematics of the pseudo inverse (Eqn. 4) requires the number of strain gauges be greater than or equal the number of load cases. Good practice for applying this methodology indicates the number of gauges be 1.5 to 2 times the number of load cases. For this problem the number of strain gauges used was sixteen. Fig. 4 shows the optimum strain gauge locations and Fig. 5 lists the same. Also, it is important to note that the [C] matrix of Eqn. 4 is formed during the identification of strain gauge location and is saved in the True-Load software for further computation of operational loads. The strain data used in Eqn. 4 were from the virtual strain gauges placed on the FEA model for all unit load cases. As mentioned before the True-Load software will employ D-optimization method and sequential exchange algorithm to identify the potential strain gauge location and local orientation. Since the process is conducted in finite element software, the gauge placing locations are shown in element numbers as shown in Fig. 5. Once the gauge locations are known, the following step in this testing procedure is to place strain gauges on the structure and is explained in the next section.

In Situ Instrumentation

In this testing process, the structure itself will act as its own transducer and can be called as a self-transducer. The strain gauges that were placed on the suspension assembly will convert the incoming load signals to strain signals. The strain gauges used in this experiment were CEA-13-250UW-350/P2 series from MicroMeasurements. Strain gauge locations from the True-Load software were manually marked by using micro gauges, vernier calipers and height gauges. Fig. 6 and Fig. 7 shows the instrumented suspension links and upright respectively.

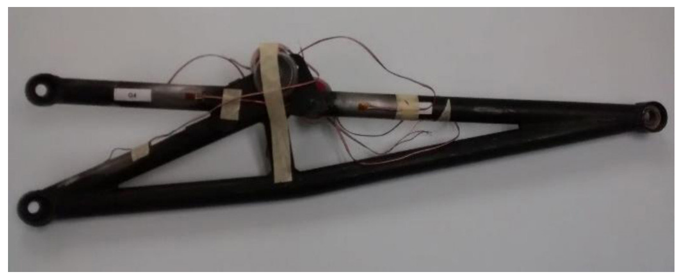

Figure 6. Instrumented suspension link and tab. The suspension link had six strain gauges.

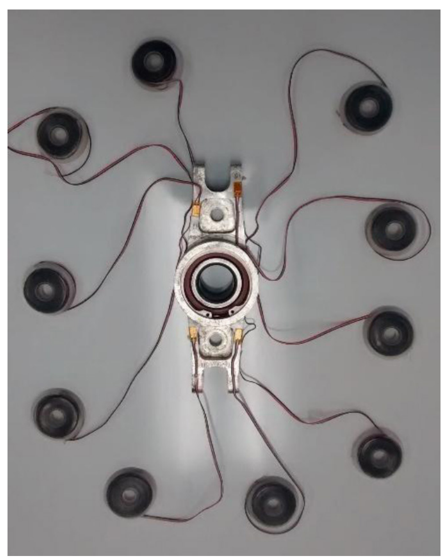

Figure 7. Instrumented suspension upright. The upright had ten strain gauges.

Once the instrumentation is done, an efficient Data Acquisition System is required to collect real time strain data. In this problem, NI 9237 DAQ cards were used to collect strain response. The DAQ was programmed by using NI Systems Lab-View software. Fig. 8 demonstrates the flow of signal in this test event.

Fig. 8 clearly demonstrates the strain data flow from strain gauge to a final storage place. The strain gauge was connected to a connector which acts as a junction between NI 9237 DAQ and strain gauge. From the connector the data is transmitted to NI 9237 DAQ by using

RJ 50 cables. A total of four data cards can be installed into the NI cDAQ-9174 chassis and each data card can handle four strain gauges, which exactly matches the number of strain gauges used which was sixteen. The chassis was electrically powered by an external battery source and was also connected to a portable computer. The Lab-View program thus will be able to collect real time data while performing test runs.

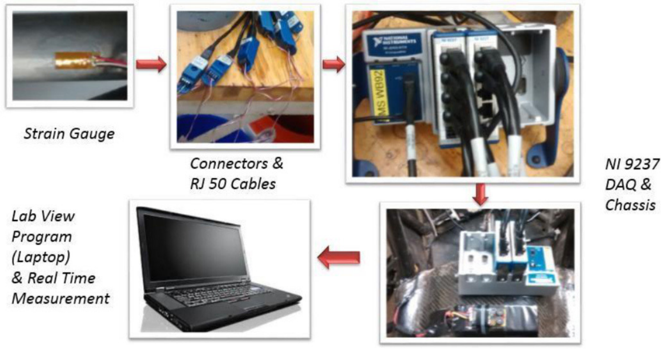

Figure 8. Strain measurement data flow for instrumented suspension assembly.

Field Testing and Correlation

Proving ground testing is usually performed as a combination of extreme operational situations. In this problem, a slanting obstacle was fabricated to facilitate tough riding. From prior experience it was known that extreme loads were acting on the suspension during high jumps, hard turns, and sudden braking. So, based on this information definite number of trails was performed on various speeds ranging from 10 mph to 30 mph. Fig. 9 shows the fully instrumented suspension and the vehicle and Fig. 10 shows the SAE BAJA car under high jump testing.

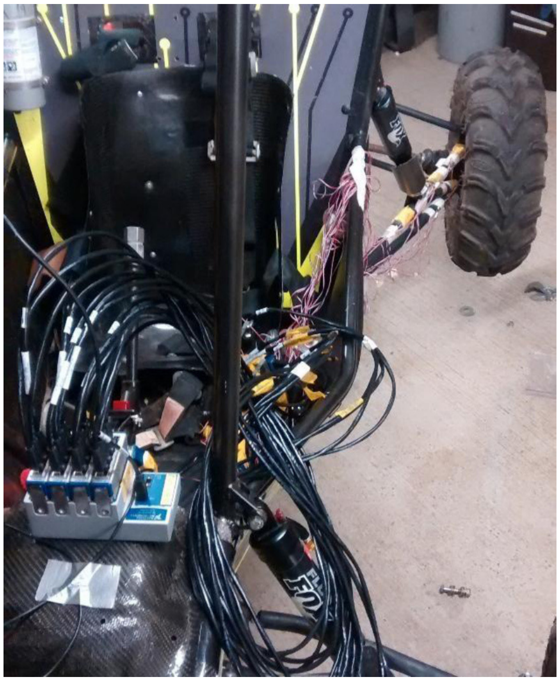

Figure 9. Instrumented vehicle with Data Acquisition System

Figure 10. Instrumented vehicle under high jump testing, 10 mph

After performing the test events, designers were concentrating into events that gave high strain. Out of all the events, a jump trial made at 10 mph and a hard turn event made at 20 mph were chosen for further analysis. The strain measurement from all the 16 gauges were plotted (Fig. 10 and Fig. 11) against the simulated strain from FEA and an excellent match was achieved.

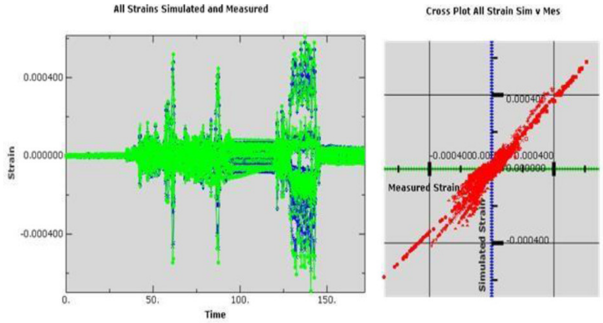

Figure 10. Strain plot correlation between measured strain and simulated strain for hard turn event made at 20 mph.

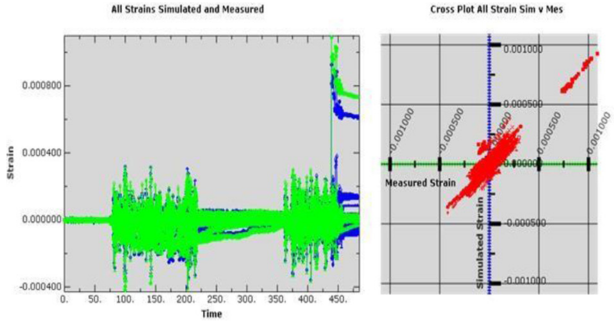

Figure 11. Strain plot correlation between measured strain and simulated strain for jump even made at 10 mph.

Loading Application and System Redesign

The measured strain from all the channels can be used to estimate the operational loads. The load computation is performed in TrueLoad/Post-Test software by using Eqn. 1. Correlation matrix is already saved from the Pretest and by using the measured strain the true value of operational loads can be estimated. The load estimated from Jump test and hard turn test is shown in Fig. 12 and Fig. 13 respectively.

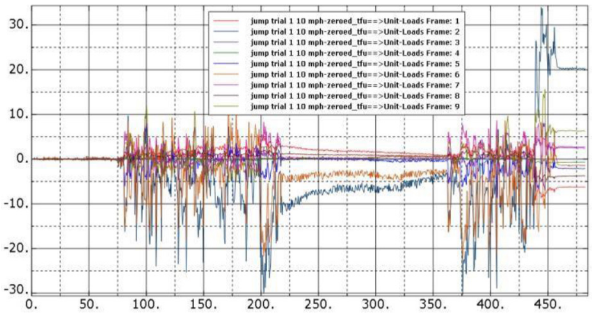

Figure 12. Load estimated from jump event made at 10 mph. The x-axis is time in seconds and the y-axis is load scaled down by 100 lb.

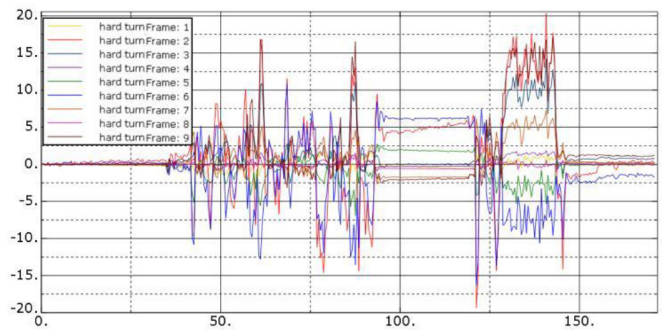

Figure 13. Load estimated from hard turn event made at 20 mph. The x-axis is time in seconds and the y-axis is load scaled down by 100 lb.

From the load estimated, it is very much clear that more operational loads are acting on the structure during jumps. The correlation plots in Figures 10 and 11 show that calculated loads in Figures 12 and 13 produce accurate strains at the strain gauge locations for the entire time history. This confidence in understanding the loads on the suspension can now be used for a wide variety of analysis which includes calculation of displacement, stress and strain from the FEA program. One advanced usage of loading time histories is to perform fatigue calculation. For this example, the fatigue calculations were performed by commercial software, fe-safe®. The desire was to understand areas of over design as an opportunity for redesign and weight savings. The following paragraphs will show the use of loads estimated from test events for fatigue analysis with fe-safe® software.

Once the loads from different test events got estimated, it is necessary to analyze the results to see the most loaded time frame. From Fig. 12 it is clear that the structure was loaded to its maximum at time 219 seconds during the jump test. Considering the load calculated at this time frame as the design load, a simple static analysis was performed for the same structure, but with a lesser thickness wall. The initial wall thickness was 0.049 inches and was reduced to 0.035 inches. From the static event performed, it was clear that the structure can withstand the maximum operational load and the von-Mises stress was below the yield limit of the material used (4130 Steel).

Once the structure passes the simple static analysis, the next step in this testing process is to perform a durability analysis. Fe-safe® software was used in this project to perform the durability test. The intention behind this analysis is to assure, that the structure with a reduced wall thickness can withstand the estimated true operational loads. Along with adding values to the integrity of the structure, this step will ensure safety to the end user of the product.

The first step while performing a durability analysis to create a duty cycle based on experience. A duty cycle will comprise of events that can replicate the true operational situations. Since all events in this testing process were designed for specific tasks, hypothetical duty cycle of events was made based on Table 2.

Table 2. Duty cycle generation based on performed events and repeats.

Duty Cycle		
Event	**Time (s)**	**Repeats (Numbers)**
Hard Brake Trial	124.032	30
Hard Turn Trial	170.904	40
Jump Trial	482.412	20
Time for 1 Duty Cycle		5.6126 *Hours*

After presenting the designed duty cycle to the fe-safe® software, the durability analysis was performed for suspension assembly with 2 wall thickness. 1. Arm wall thickness of 0.049 inches and 2. Arm wall thickness of 0.035 inches. Fig. 14 and Fig. 15 show the results for separate durability analysis for suspension arms with different wall thickness.

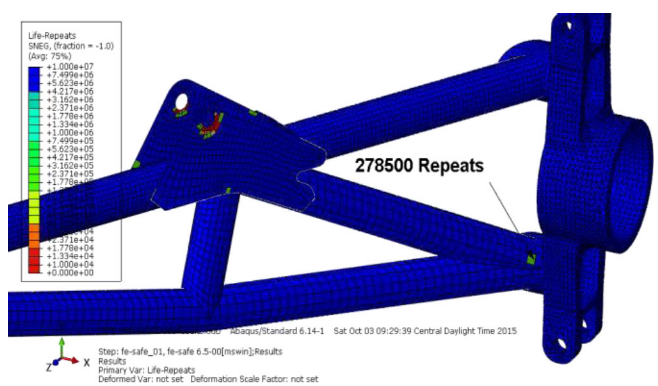

Figure 14. Result from durability analysis for suspension arm with a wall thickness of 0.049 inches.

It is clear from the analysis result shown in Fig. 14 that the structure will withstand the designed load cycle and even the most stress concentrated and sensitive region can go nearly 278500 repeats. A pure analytical decision was made at this point to keep the thickness of the cross arm (see Fig. 2) which was subjected to the most stress concentration. Hence, in the re-design analysis, all arms will have a reduced wall thickness of 0.035 inches excluding the cross arm. Fig. 15 will show the result gained for the re-designed structure under the same duty cycle.

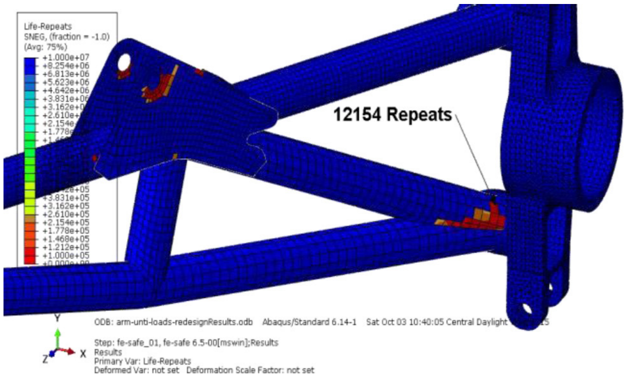

Figure 15. Result from durability analysis for suspension arm with a wall thickness of 0.035 inches for all arms excluding cross arm.

From Fig. 15 it is clear that the structure even with reduced thickness can endure 12154 repeats. Considering the purpose of an SAE Car and its total life, the redesigned suspension assembly will be a recommended design for manufacturing. Table 3 shows detailed the results including the weight saved by the redesign work.

Table 3. Final results comparison between original design and redesigned suspension assembly.

	Duty Cycle Repeats	Endurance Time (Days)	Weight (lbf)
Original Design	278500	65130	4.91
Redesign	12154	2842	3.51
	Weight Savings		1.39
			28.5%

The initial mass of the structure was 4.91 lb and was reduced to 3.51 lb when using a wall of lower thickness, which directly reduces the mass by 28.5%.

Conclusions

Identification of true operational loads is a vital part in a structural design industry. In this paper, the suspension assembly of a SAE BAJA car is subjected to test to identify the true operational load values. By using the True-Load software exact strain gauge placing locations for specific load cases were identified. From the structural strain response collected from predefined events, exact loads were estimated by using True-Load/ Post-Test software. Gaining the support from the results of a static structural and durability analysis the thickness of the structure was reduced and a weight reduction of 28.5% was achieved.

The work presented in this paper will assist designers in commercial automotive industry to have a calculated approach towards their product under development. By estimating the true load, the pattern of real life situation in which their product is going to be released can be studied. Finite element simulations can be performed against this true load pattern to develop an optimal product. Since understanding the real life situation is the key in any development project, in commercial or non-commercial level, by using the technique described in this paper, a better product can be developed.

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