Manufacturing process effects on fatigue design and optimization of automotive components: an analytical and experimental study

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A Dissertation

entitled

Manufacturing Process Effects on Fatigue Design and Optimization of Automotive Components – An Analytical and Experimental Study

by

Mehrdad Zoroufi

Submitted as partial fulfillment of the requirements for the Doctor of Philosophy in Engineering Science

Adviser: Dr. Ali Fatemi

Graduate School

The University of Toledo
December 2004
The University of Toledo

College of Engineering

I HEREBY RECOMMEND THAT THE DISSERTATION PREPARED UNDER MY SUPERVISION BY Mehrdad Zoroufi ENTITLED Manufacturing Process Effects on Fatigue Design and Optimization of Automotive Components—An Analytical and Experimental Study BE ACCEPTED IN PARTIAL FULFILLMENT OF THE REQUIREMENTS FOR THE DEGREE OF DOCTOR OF PHILOSOPHY IN ENGINEERING

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An Abstract of

Manufacturing Process Effects on Fatigue Design and Optimization of Automotive Components – An Analytical and Experimental Study

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December 2004

Numerous fatigue-critical parts could be found in ground vehicles, and time-varying loads have always challenged automotive designers. Fatigue design and life assessment of these components are essentially influenced by the material used and manufacturing processes chosen. Exploring the design criteria and optimization potentials with respect to manufacturing processes is vital to the industry.

This study was aimed at developing general procedures for fatigue analysis and optimization of safety-critical automotive components with manufacturing considerations. A literature survey was conducted, specimen and component tests were performed, and finite element stress analysis and durability and optimization evaluations of similar components produced by different manufacturing technologies were made to achieve the objectives. The typical example component chosen was a vehicle steering knuckle made of three competing materials and manufacturing processes including forged steel, cast aluminum and cast iron.

In the literature survey, manufacturing processes were studied and compared with focus on mechanical behavior. The methods used in the literature for fatigue life evaluation and prediction of automotive components, as well as for optimization studies with respect to geometry, material and manufacturing aspects were also reviewed. Specimen strain-
controlled tests were conducted to obtain material monotonic and cyclic deformation and fatigue properties. Components’ fatigue behaviors were investigated via constant-amplitude load-controlled fatigue tests. Comparisons of materials monotonic and fatigue properties, and components’ fatigue behaviors were made for competing material and manufacturing processes. In terms of structural performance and durability, based on both material testing and component evaluation, forged steel was found superior to cast iron which in turn was found superior to cast aluminum.

Finite element models of the components were analyzed, using linear and nonlinear stress analyses. The nominal stress and local stress and strain approaches were employed to assess durability of the components. Experimental and analytical stress and fatigue life results were compared to evaluate the validity of the analytical approaches. The strength and shortages of the applied models and alternative analyses were also investigated. It was concluded that the local life prediction approaches in combination with either nonlinear finite element analysis results, or linear finite element analysis results corrected for local plasticity, yielded satisfactory predictions.

A procedure was developed to optimize forged automotive components for weight reduction and cost savings with fatigue strength as the key performance indicator. By considering geometry variations, alternative materials and manufacturing process parameters as design variables, the example part was optimized. Guidelines were developed and limitations were identified for the optimization procedure. Although the optimization results showed limited changes for the particular example component, the approach that was followed is applicable to other forged components. It was emphasized that geometrical optimization of manufactured components could only be realistic and practical if other important parameters like material, manufacturability, and cost are taken into account.
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Nomenclature

\( a \) material characteristic length
\( A_0, A_f \) initial, final cross section area
\( b \) fatigue strength exponent
\( c \) fatigue ductility exponent
\( e \) engineering strain
\( e^* \) modified nominal strain
\( E \) modulus of elasticity
\( E' \) midlife cyclic modulus of elasticity
\( \%EL \) percent elongation
\( HB, HRB, HRC, HV \) Brinell, Rockwell B-Scale, Rockwell C-Scale, Vickers hardness number
\( K \) strength coefficient
\( K' \) cyclic strength coefficient
\( K_f \) fatigue notch factor
\( K_t \) elastic stress concentration factor
\( K_{eq} \) equivalent stress concentration factor
\( L_0, L_f \) initial, final specimen gage length
\( M \) moment
\( M_a \) moment amplitude
\( n \) strain hardening exponent
\( n' \) cyclic strain hardening exponent
\( N_{50\%}, (N)_{10\%}, (N)_{50\%} \) number of cycles to midlife, to 10% load drop, to 50% load drop
\( N_f \) cycles to failure
\( P \) load
\( P_f \) load at fracture
\( P_p \) plastic limit load
\( P_u \) ultimate load
\( \%RA \) percent reduction in area
\( r \) radius of the notch
\( S \) engineering stress, nominal stress
\( S^* \) modified nominal stress
\( S_a \) nominal stress amplitude
\( S_f \) fatigue strength
\( S_{Nf} \) equivalent completely reversed nominal stress amplitude
\( S_u \) ultimate strength
\( S_y \) yield strength
\( S'_y \) cyclic yield strength
\( t_i, t_f \) initial, final specimen thickness
\( w_i, w_f \) initial, final specimen width
\( \Delta \varepsilon \) strain range
\( \Delta \varepsilon_e \) elastic strain range
\( \Delta \varepsilon_p \) plastic strain range
\( \Delta \sigma \) stress range
\( \varepsilon \) total strain
\( \varepsilon_1, \varepsilon_2, \varepsilon_3 \) maximum, middle, minimum principal strain
\( \varepsilon_a \) strain amplitude
\( \varepsilon^{pl} \) elastic-plastic notch equivalent strain
\( \varepsilon^{eq} \) elastically calculated notch equivalent strain
\( \varepsilon \) elastic strain
\( \varepsilon_f \) true fracture ductility
\( \varepsilon_f' \) fatigue ductility coefficient
\( \varepsilon_m \) mean strain
\( \varepsilon_p \) plastic strain
\( \varepsilon_x, \varepsilon_y, \varepsilon_z, \varepsilon_{xy}, \varepsilon_{xz}, \varepsilon_{yz} \) local components of strain tensor
\( \varepsilon_{VM} \) local von Mises strain
\( \gamma_{max} \) local maximum shear strain
\( \sigma \) true stress, local stress
\( \sigma_1, \sigma_2, \sigma_3 \) maximum, middle, minimum principal stress
\( \sigma_a \) local stress amplitude
\( \sigma_{1a}, \sigma_{2a}, \sigma_{3a} \) local elastic principal stresses
\( \sigma^{pl} \) elastic-plastic notch equivalent stress
\( \sigma^{eq} \) elastically calculated notch equivalent stress
\( \sigma_f \) true fracture strength
\( \sigma_f' \) fatigue strength coefficient
\( \sigma_{max} \) local maximum stress
\( \sigma_m \) local mean stress
\( \sigma_{Nf} \) equivalent completely reversed local stress amplitude
\( \sigma_{VM} \) local von Mises stress
\( \sigma_x, \sigma_y, \sigma_z, \tau_{xy}, \tau_{xz}, \tau_{yz} \) local components of stress tensor
\( \tau_{max} \) local maximum shear stress
Chapter One

1 Introduction

Manufacturing processes face major competitions in automotive industry to produce lighter, cheaper and more efficient components that exhibit more precise dimensions, need less machining and require less part processing. Technology leaders follow two main routes to improve these processes; development and integration of currently available technologies, and invention of new technologies optimized with respect to various design or commercial aspects. The value of the know-how depends on a process’s ability to usefully differentiate its capabilities from those of its competitors. Material mechanical properties and manufacturing parameters play decisive roles and the weaknesses and strengths of each manufacturing process need to be available to designers in these respects, to enable them to choose the optimum choice for the specific component and application.

In automotive industry, designers have a wide range of materials and processes to select from. Steel and aluminum forgings and castings, cast irons, and powder forgings have found broad applications in automotive safety-critical systems. The competition is particularly acute in the chassis, and it is not unusual to find a range of different materials and manufacturing technologies employed within modern chassis components.

Many safety-critical components in the vehicle experience static as well as time-varying loadings, and obviously they undergo the latter during a major portion of their service life. However, material selection for these components made by various manufacturing techniques is often based on monotonic rather than cyclic properties. Fatigue
behavior is, therefore, a key consideration in design and performance evaluation of automotive components, and to address the issue effectively and economically, engineers need to model and design for mechanical fatigue early in the product design stage.

In automotive design, durability evaluation of components based on experimental assessments is time-consuming and expensive, so analytical approaches that include limited number of component verification tests have gained more attention. A problem that arises at the fatigue design stage of such components is the transferability of data from smooth specimens to the component. The component geometry, loading, and manufacturing process parameters such as surface conditions often deviate from that of the specimen investigated and neither a nominal stress nor a notch factor can be defined in most cases.

In automotive industry, the significant increase of the demand for lighter, more fuel efficient vehicles, reduced design-testing iterations, and satisfactory reliability level has promoted the adoption of optimum materials and components. The studies on optimization have focused on three aspects; geometry optimization, alternative materials and manufacturing process modifications. From the viewpoint of geometry and considering the boundary conditions, an optimized component could be obtained using different methods of geometry optimization. On the other hand, alternative materials are being put into trial and manufacturing processes are being re-evaluated to achieve lighter, cheaper and more efficient components.

1.1 Motivation

This research was motivated by a practical need to assess and compare fatigue performance of components produced by competing manufacturing processes, to develop a general durability assessment methodology for automotive chassis (and similar) components,
and to implement an optimization methodology that incorporates structural durability performance, material properties, manufacturing and cost considerations for such components.

In automotive industry, the evaluation of manufacturing processes with regards to structural performance is often performed simply based on monotonic rather than cyclic or fatigue characteristics. However, the stress-strain behavior obtained from a monotonic tension or compression test can be quite different from that obtained under cyclic loading. Although fatigue is one of the main parameters that differentiate among competing manufacturing processes and one of the major responsible phenomena that cause mechanical failure in components, it is often overlooked by design engineers, resulting in inefficient design methods and many over-designed parts that lack optimization potential. Accordingly, a comparison of competing manufacturing processes regarding fatigue performance that also considers cost of manufacturing was one of the main motivations of this study.

The analytical approach combined with a limited number of component testing reduces design cycle time due to reduced testing, allows inexpensive evaluation of changes in geometry, material, loading and manufacturing process through performance simulation, and finally provides evaluation techniques for product optimization and failure analysis. An advantage of the limited number of component tests is that the effects of material, manufacturing process parameters, and geometry are inherently accounted for, even though synergistically. This dissertation was partially motivated to implement the principles of this analytical approach to fatigue life assessment of automotive components. In this respect, developing an optimization methodology for these components, as an effective tool for design modification, becomes feasible.
The common practice in optimization of automotive components in the automotive industry is to focus on geometry, material or manufacturing parameters individually. An approach that incorporates these three main aspects, even though challenging, has the advantage of enabling the designer to evaluate a design in a comprehensive manner, to investigate the cost parameters and to determine practical optimization potentials. Therefore, the third motivating concept for this study was the need to investigate optimization of safety-critical automotive components considering the combined effects of geometry, material and manufacturing parameters and subsequently, cost.

1.2 Objectives

The overall objectives of this research program were:

1. To assess fatigue life and compare fatigue performance of competing manufacturing processes;
2. To develop a general durability assessment methodology for safety-critical automotive components;
3. To develop a method for efficient and reliable optimization of such components that satisfies performance criteria and considers geometry, material, manufacturing parameters and costs.

1.3 Scope of this Study and Overview of the Dissertation

This study involves several main topics; these are 1) a background study on forging and its competing manufacturing processes, and vehicle engine and chassis components that are produced by these competing processes discussed in Chapter 2, 2) a literature review that focuses on comparison of competing manufacturing processes, and durability assessment
and optimization of automotive components also presented in Chapter 2, 3) experimental work including specimen and component testing, and 4) analytical work including durability assessment and optimization analysis. In light of the high volume of forged steel vehicle components, the forging process was considered as the base for comparison of competing manufacturing processes.

Vehicle steering knuckles of three materials/processes were selected as the example parts for this study. These included forged steel SAE Grade 11V37 steering knuckle of the rear suspension of a 4-cylinder sedan weighing 2.4 kg, cast aluminum ASTM A356-T6 steering knuckle of front suspension of a 6-cylinder minivan weighing 2.4 kg, and cast iron ASTM A536 Grade 65-45-12 steering knuckle of the front suspension of a 4-cylinder sedan weighing 4.7 kg. Figure 1-1 shows the three components.

1.3.1 Experimental Work

Strain-controlled monotonic and fatigue tests of specimens made of forged steel, cast aluminum and cast iron steering knuckles based on ASTM standard test methods and recommended practices were conducted. From these experiments, monotonic as well as baseline cyclic deformation and fatigue properties of the three materials were obtained. The data obtained made it possible to compare deformation response, fatigue performance, and failure mechanisms of the base materials, without introducing the effects and interaction of complex design parameters. In addition, the required baseline data for life prediction analysis to predict component fatigue life and performance under actual service loading conditions became available. Chapter 3 discusses the specimen testing methods, results and comparisons of monotonic and fatigue behaviors of the three materials.
Load-control component tests for the forged steel and cast aluminum steering knuckles were conducted. Such data provides a direct comparison between fatigue performances of the components made of competing processes. The comparison inherently includes design effects such as surface finish, component size, residual stress, and stress concentration. In addition, the component test results made it possible to verify the analytical durability assessment. Chapter 5 explains the pre-test analysis, test configuration, and post-test analysis related to component testing.

1.3.2 Analytical Work

The analytical work in this study consisted of finite element analysis (FEA), durability assessment and optimization analysis. Linear and nonlinear finite element analyses of the steering knuckles were conducted to obtain critical locations of, and stress and strain distributions of each component. Details of geometry generation, boundary conditions, mesh specification, nonlinear material model, model solution and post processing are detailed in Chapter 4.

A general life prediction methodology for the subject components was developed and is described in Chapter 6. Material monotonic and cyclic data and results of the FEA were used in life prediction methods applicable to safety-critical automotive components. The strengths and shortages of each method are discussed.

Based on the results of analyses and testing performed, an analytical optimization study of the forged steel steering knuckle was performed. Such optimization sought to minimize weight and manufacturing costs while maintaining or improving fatigue strength of the component by targeting geometry, material and manufacturing parameters. The optimization methodology, manufacturing considerations and alternative materials are
discussed in Chapter 7 and recommendations are made on the optimized geometry. Consequently, Chapter 8 summarizes the results of this study.
Figure 1-1  From left forged steel, cast aluminum and cast iron steering knuckles selected as example parts for this study.
Chapter Two

2 Background and Literature Review

The background study includes two main parts. The first part provides an introduction to forging process and its parameters as a process that possesses high volume in automotive components. The second part gives an overview of vehicle power train and suspension components that are the subject of competition among manufacturing processes, and the service conditions of steering knuckle as the example part of this study. The literature review covers a number of studies on comparison of competing manufacturing processes with a focus on material mechanical behavior, life prediction methods implemented in automotive components, and studies on optimization of steering knuckle and similar components with manufacturing and material considerations. A brief report on the literature review is also presented in Fatemi and Zoroufi (2002).

2.1 Forging Process and the Influencing Parameters

Bulk metal forming, which could also generally be called forging, denotes a family of processes by which plastic deformation of the work piece is carried out by compressive forces. In automotive industry, forged components are commonly found at points of alternating and impact stresses such as steering knuckles, spindles, kingpins, axle beams and shafts, torsion bars, ball studs, idler arms, pitman arms and steering arms. Another common application is in the power train, where crank shafts, camshafts, connecting rods, transmission shafts and gears, differential gears, clutch hubs and universal joints are often
forged. Although typically forged from carbon or low-alloy steel, other materials such as aluminum and high strength low-alloy (microalloyed) steels are being increasingly used in forged automotive applications.

Figure 2-1 shows the sequences involved in forging process. In this process, a slug or billet is prepared, or the workpiece is preformed by shearing, sawing, or cutting off, either cold or hot. If necessary, the surfaces are cleaned by such means as shot blasting. For hot forging, the workpiece is heated in a furnace or by induction and, if necessary, descaled after heating. Descaling may also occur during the initial stages of forging when the scale, which is usually brittle, falls off during plastic deformation of the part. For hot forging, the dies are preheated and lubricated and for cold forging, the blank is lubricated.

The workpiece is forged in appropriate dies and in the proper sequence. The excess material such as flash is removed by trimming, machining or grinding. The forging is cleaned, its dimensions are checked and, if necessary, it is totally or partly machined to final dimensions and tolerances. Additional operations such as heat treating are performed to obtain the desired mechanical and metallurgical properties. Any finishing operations that may be required are conducted too. Finally the forging is inspected for any internal and external defects or imperfections. Different forging methods exercised are discussed in the following.

2.1.1 Forging Processes

The methods contemporarily practiced for forging process are usually categorized as open-die forging, impression-die forging, and miscellaneous forging operations including cold forging, ring forging, etc.
**Open-Die Forging**

Open-die forging, also called upsetting, is performed when a workpiece is placed on a lower die and its height is reduced by the downward movement of the top die. Friction between the end faces of the workpiece prevents the free, lateral spread of the ends of the workpiece and results in a barrel shape. Open-die forging is distinguished by the fact that the metal is never completely confined or restrained, and that the dies used are rather simple and universal. All types of hammers or presses may be used in open-die forging. Forgings are made by this method if (a) the forging is too large to be produced in closed dies; under open dies it is produced in many steps by forging only a part of it in each step, or (b) the quantity required is too small to justify the cost of complex closed dies.

Practically all of forgeable ferrous and non-ferrous alloys can be open-die forged, including age-hardening superalloys and corrosion-resistant refractory alloys. Open-die shape capability is wide and this process can produce step shafts, hollows cylindrical in shape, ring-like parts, and contour-formed metal shells like pressure vessels. Multiple open-die forging operations can be combined to produce the required shape, or these forging methods can be tailored to attain the proper amount of total deformation and optimum grain-flow structure, thereby maximizing property enhancement and ultimate performance for a particular application. Forging an integral gear blank and hub, for example, may entail multiple drawing or solid forging operations, and then upsetting. Similarly, blanks for rings may be prepared by upsetting an ingot, then piercing the center, prior to forging the ring.

**Impression-Die Forging**

In the simplest form of impression-die forging, commonly referred to as closed-die forging, a cylindrical or rectangular workpiece is placed in the bottom die. The dies contain no provision for controlling the flow of excess material. Figure 2-2 shows impression-forged
connecting rods along with the matching dies. As the two dies are brought together, the workpiece undergoes plastic deformation until its enlarged sides touch the sidewalls of the die impressions. At this point a small amount of material begins to flow outside of the die impressions, forming flash. In the further course of the die approach, this flash is thinned gradually. As a consequence, it cools rapidly and presents increased resistance to deformation. In this sense, the flash becomes a part of the tool and helps to build up high pressure inside the bulk of the workpiece. This pressure can aid material flow into parts of the impression previously unfilled so that, at the end of the stroke, the die impressions are nearly filled with the workpiece material.

Impression-die forging can produce a variety of three dimensional shapes with a wide range in weight. Because metal flow is restricted by the die contours, this process can yield more complex shapes and closer tolerances than open-die forging processes. Additional flexibility in forming symmetrical and non-symmetrical shapes comes from various pre-forming operations (sometimes bending) prior to forging in finisher dies. Part geometries range from simple spherical shapes, block-like rectangular solids, and disc-like configurations to components with thin and long sections that incorporate thin webs and relatively high vertical projections like ribs and bosses. Although many parts are generally symmetrical, others incorporate all sorts of design elements (flanges, protrusions, holes, cavities, pockets, etc.) that combine to make the forging very non-symmetrical. In addition, parts can be bent or curved in one or several planes, whether they are basically longitudinal, equi-dimensional or flat.

Most engineering metals and alloys can be forged via conventional impression-die processes, among them carbon and alloy steels, tool steels, and stainless, aluminum and copper alloys, and certain titanium alloys. Strain-rate and temperature-sensitive materials
(magnesium, nickel-based alloys, refractory alloys and some titanium alloys) may require more sophisticated forging processes or special equipment for forging in impression dies.

**Miscellaneous Forging Operations**

Most forging is done as hot work, at temperatures up to 1300°C; however, a variation of impression-die forging is cold forging. The temperature of metals being cold forged may range from room temperature to several hundred degrees. Cold forging encompasses many processes: bending, cold drawing, cold heading, coining, extrusion, punching, thread rolling and more, to produce a diverse range of part shapes. These include various shaft-like components, cup-shaped geometries, hollow parts with stems and shafts, all kinds of headed and bent configurations, as well as combinations. Material options range from low alloy and carbon steels to stainless steels, selected aluminum alloys, brass and bronze. There are times when warm forging practices are selected over cold forging especially for higher carbon grades of steel or where annealing can be eliminated.

Seamless rolled ring forging is another forging process and is typically performed by punching a hole in a thick, round piece of metal (creating a donut shape), and then rolling and squeezing (or in some cases pounding) the donut into a thin ring. Rings forged by the seamless ring rolling process can weigh less than 0.5 kg up to 150 tons, while outer diameters range from a few centimeters up to 10 m in diameter. Other miscellaneous forging operations include isothermal and hot die forging, rolling, radial forging, etc.

### 2.1.2 Important Factors in Forging Process

**Forgeability**

The forgeability of a metal can be defined as its capability to undergo deformation by forging without cracking or defects. This definition can be expanded to include the flow
strength of the metal. Thus a material with good forgeability is one that can be shaped with low forces without cracking. A commonly used test of forgeability is to upset a solid cylindrical specimen and observe any cracking on the barreled surface. The greater the deformation prior to cracking, the greater the forgeability of the metal (Kalpakjian and Schmid, 2000). If notch sensitivity of the material is high, surface defects will affect the results by causing premature cracking.

In another test of forgeability called hot-twist test, a round specimen is twisted continuously in the same direction until it fails. The test is performed at various temperatures, and the number of turns that each specimen undergoes before failure is observed. The optimal forging temperature is then determined. This test is particularly useful in determining the forgeability of steels, although upsetting tests can also be used for that purpose. Small changes in the composition of or impurities in the metal can have a significant effect on forgeability.

Hydrostatic pressure has a significant beneficial effect on the ductility of metals and nonmetallic materials. Experiments have indicated that the results of room-temperature forgeability tests are improved, i.e. cracking takes place at higher strain levels, if the tests are conducted in an environment of high hydrostatic pressure. Based on the results of various tests and observations, the forgeability of several metals and alloys is determined and shown in Table 2-1. It is based on such considerations as the ductility and strength of the metal, forging temperature required, frictional behavior and quality of the forging obtained.

**Lubrication**

During the deformation phase of a conventional hot forging operation in which the die temperature is much less than the billet temperature, the lubricant provides three basic effects (Byrer et al., 1985); lubricity, ensuring that the correct coefficient of friction is
maintained between the hot billet and the tooling; physical barrier, preventing physical contact between billet and dies; and thermal insulation, retarding the rate of heat transfer between the hot metal and the dies.

The degree of metal flow affects all of these effects. The manner in which a part is forged can also affect the proper functioning of a lubrication system. Lubricants can be classified into four groups, although a lubricant compound can be a blend of the first two (or the first three) categories to obtain the desired performance or to improve performance; Solids, meltable pigments, organic chemicals and water solubles.

**Surface Finish**

Surface finish of the forging depends on the effectiveness of the lubricant, preparation of the blank, die surface finish, and die wear. As described by Rice (1997), in addition to bulk effects, forgings subjected to cyclic or repeated service loading are significantly influenced by surface conditions. Many forgings are used in service with some or the entire as-forged surface remaining. It follows that the fatigue life is reduced by features such as forging laps, folds, cracks and other imperfections that can act as crack initiation sites. Surface metallurgical features can also be important. For example, steels may show decarburization and other alloys exhibit changes in chemistry and phase distribution resulting from the heating and cooling cycles associated with forging and heat treatment.

**Heat Treatment**

Heat treatment involves specific controlled thermal cycles of heating and cooling to improve one or more properties of the forged part. The primary objectives of such treatment may be to relieve internal stresses (as in tempering and stress relieving), control distortion, optimize the depth of hardening (hardenability), or develop the final specification
for mechanical or physical properties, because the forged component must meet performance engineering design requirements and ensure safety and reliability in service.

Underlying the selection of the proper treatment are cost considerations that must factor in the variables of forging shape and size, metal composition, machining, and final properties. For example, it would be wasteful of time and energy to reheat a forging when cooling from forging heat would satisfy the specification, or to add a temperature cycle when normalizing suffices.

**Anisotropy**

During the forging process, nonuniformities in alloy chemistry, second-phase particles, inclusions, and the crystalline grains themselves are aligned in the directions of the greatest metal flow. The directional pattern of the crystals following working is known as the grain-flow pattern. Grain flow produces directional characteristics in properties such as strength, ductility, and resistance to impact and fatigue. The grains could be oriented in the forged part to lie in the direction that maximum strength is needed. The important anisotropies are in ductility, notched-bar impact strength, and fatigue. The resistance to stress corrosion cracking can be highly directional, for example, in high-strength aluminum alloys. The die sequence and the designs of the dies may be altered in order to control the grain-flow pattern. Forging and lubrication techniques can also be used for this purpose (Byrer et al., 1985).

### 2.2 Vehicle Engine and Suspension

For a typical motor vehicle, the main competition among manufacturing processes is in the components of power train and suspension system. Typical forged components used in vehicles are crankshaft, connecting rod, camshaft, and suspension components such as
control arm, steering knuckle and wheel hub, as shown in Figure 2-3. In order to have a better understanding of the vehicle components, the technical features of vehicle engine and suspension system components and the steering knuckle in particular, are briefly reviewed here.

### 2.2.1 Vehicle Engine

The main components of a typical engine, as shown in Figure 2-4, are cylinder block and crankcase, pistons and rings, connecting rods and bearings, crankshaft assembly and bearings, cylinder head and gasket, camshaft, valve train and timing drive, and engine support mountings. Among these components, manufacturing processes compete over power train components, mainly crankshaft, transmission shaft, camshaft and connecting rod. Most vehicle engine components operate at certain speeds for very high number of cycles.

### 2.2.2 Suspension System

Suspension system consists of the springs and related parts intermediate between the wheels and the frame, subframe, or side rails of a unitized body. The suspension supports the weight of the upper part of a vehicle on its axles and wheels, allows the vehicle to travel over irregular surfaces with a minimum of up-and-down body movement, and allows the vehicle to corner with minimum roll or loss of traction between the tires and the road. Four types of springs used in automotive suspension are coil, leaf, torsion bar, and air spring.

In a typical suspension system for a vehicle with front-engine and front-wheel-drive (Figure 2-5), the weight of the vehicle applies an initial compression to the coil springs. When the tires and wheels encounter irregularities in the road, the springs further compress
or expand to absorb most of the shock. The suspension at the rear wheels is usually simpler than for the front wheels, which require multiple-point attachments so the wheels can move up and down while swinging from side to side for steering.

A telescoping hydraulic damper, known as a shock absorber, is mounted separately or in the strut at each wheel to restrain spring movement and prevent prolonged spring oscillations. The shock absorber contains a piston that moves in a cylinder as the wheel moves up and down with respect to the vehicle body or frame. As the piston moves, it forces a fluid through an orifice, imposing a restraint on the spring. Spring-loaded valves open to permit quicker flow of the fluid if fluid pressure rises high enough, as it may when rapid wheel movements take place. As categorized by Birch (2000), there are essentially five types of suspensions used on cars, pickups and trucks:

**Short-Long Arm (SLA) Suspension:** This is the typical RWD car’s front suspension. It consists of two control arms (a short upper arm and a longer lower arm), a steering knuckle with spindle, and the necessary bushings and ball joints (Figure 2-6). The outer ends of both control arms connect to the steering knuckle, which includes the spindle, through a ball joint.

**Multilink Suspension:** Some SLA designs have evolved so the steering knuckle has become taller, about to the top of the tire, and the spring is strut mounted over the shock absorber, as shown in Figure 2-7.

**MacPherson Strut Suspension:** This arrangement has no upper control arm or upper ball joint. The steering knuckle connects to a spring and shock absorber assembly, which is the strut. The upper end of this assembly connects to the car body through a pivot-damper unit. A lower control arm is used; it serves the same purpose as the lower arm of an SLA suspension (Figure 2-8).
**Solid Axles Suspension:** This kind of suspension is commonly used on trucks, 4WDs, and some pickups because of their simpler, stronger, and less expensive construction. In addition, they have traditionally been used for rear suspension. A solid axle is simply a strong, solid beam of steel (usually I shaped) with a kingpin at each end to connect to the steering knuckle (Figure 2-9).

**Swing Axles Suspension:** Ford Motor Company has used this kind of suspension on its pickups, 4WDs, and light trucks. Twin I-beam axles combine some of the sturdiness and simplicity of a solid axle with some of the improved ride and handling characteristics of an independent suspension. A twin I-beam axle is a compromise between these two suspension types (Figure 2-10).

### 2.2.3 Steering Knuckle

Suspensions use various links, arms, and joints to allow the wheels to move freely up and down; front suspensions also have to allow the front wheels to turn. All suspensions must provide transverse (or side to side) as well as longitudinal (front-to-back) wheel support. Steering knuckle/spindle assembly, which might be two separate parts attached together or one complete part, is one of these links. Its geometry depends on the type of suspension.

Figure 2-6 shows the assembly of the steering knuckle and spindle on an SLA, which is the typical rear-wheel-drive car’s front suspension. The Multilink Suspension, in which the long curved steering knuckle and angled upper control arm allow its use in areas of limited size, is shown in Figure 2-7. The steering knuckle for a MacPherson Strut Suspension, which has no upper control arm or upper ball joint, is shown in Figure 2-8. In the Solid Axle Suspension, a beam of steel is connected to the steering knuckle, as shown in Figure 2-9.
The arrangement of steering knuckle/spindle in the Swing Axle Suspension could be seen in Figure 2-10. In spite of different configurations of the steering knuckle/spindle assembly for each type of vehicle suspension, the assembly is intended to play a common role in all type, and that is to accommodate the service loading.

Haeg (1997) investigated the simulation of steering system dynamics and studied steering system inputs and responses. As he states, there are many factors that influence steer motions and forces in a vehicle suspension. These range from the kinematics of the suspension itself to its response to severe off-center lateral impact loading such as an oblique curb strike event. It must be remembered that even axles that are not steered, still experience steer axis-related forces during normal vehicle operation.

Most forces are imparted to the vehicle suspension via the tire patch. Figure 2-11 shows a static lateral force input at the tire patch center. The primary lateral reaction paths are through the upper and lower horizontal restraints. Very little force is reacted by the steering system if the static lateral force input vector intersects the line of the steer axis. Figure 2-11 also shows static longitudinal force applied at the tire patch center. When the tire patch or face of the tire encounters resistance or longitudinal load, it is resolved into a force and a moment about the axis of rotation of the tire/wheel (spindle center). The force is resolved at the centerline of the spindle itself. This moment (torque) can be reacted by the drive-train or brakes, or it can rotationally accelerate the tire.

In most suspensions the steer axis is well inboard (towards the center of the vehicle), therefore longitudinal force reacted at the spindle center imparts a moment about the steer axis. The steer moment is reacted via opposing forces (a couple) at the horizontal restraints and the tie rod end, as illustrated in Figure 2-12. The tie rod forces are reacted back through the steering gear to the body or frame. The distribution of these reaction forces is a function
of the relative stiffness of the various mounts and components, the steer angle, and loading of the suspension on the opposite side of the vehicle.

2.3 Comparison of Manufacturing Processes

Various mechanical and metallurgical properties, environmental considerations, and above all cost competitiveness aspects are the main driving forces shaping the future direction of the manufacturing processes. In the ground vehicle industry conventional forged components include crankshaft, camshaft, connecting rod, piston crown, steering lever, suspension arm, steering knuckle, wheel hub, drive flange and axle beam. Some of these components are also manufactured by die-casting, and more recently by powder forging and composite technologies. Powder metallurgy (P/M) processes using sintering, which can offer net-shaped products, have also been used, though having limitations for large parts and parts with challenging geometries. Composites offer lightweight and directional properties, but are comparatively expensive.

Mechanical properties of the manufactured component are influenced by its manufacturing process. As an example, the forging process is investigated in this study to identify the process parameters that affect mechanical properties and material behavior of the forged component and the result is a chart shown in Figure 2-13. The chart has four columns; the first column includes the process influential parameters. Each parameter and its connector lines are coded with the same color. The second column entails the mechanical and metallurgical parameters that play a bridge role between the process and mechanical properties. Defects, surface finish, residual stresses generated in the component and the microstructure of the workpiece material are those bridge parameters affected by the process. The third and fourth columns cover the general mechanical properties and the
breakdown of monotonic, cyclic and fatigue properties, respectively, of the workpiece material that are subsequently affected.

A number of selected examples from literature are provided in the following to indicate the behavior comparison of competing manufacturing processes.

**Casting vs. Forging**

Gunnarson et al. (1987) investigated replacing conventional forged quenched and tempered (Q&T) steel with precipitation hardened pearlitic-ferritic cast steels for connecting rod, steering knuckle, crank shaft, control arm and other automotive components. They also compared toughness and machinability characteristics of forging versus casting components. They observed insufficient toughness but higher machinability for cast components. For the case of the vehicle steering knuckle, equivalent fatigue strength was found for the cast steering knuckle, but only with increased spindle diameter and reduced length, as compared with the forged steel steering knuckle.

The positive trend for the application of cast components is mainly due to lower cost incentives. However, weaker mechanical properties of cast components due to a wide variety of flaws and their low ductility have always been a matter of concern. Houshito et al. (1989) performed a feasibility study on the application of high strength ductile iron to automotive chassis parts, namely steering knuckles. They state that while the shape of forged part is limited by the manufacturing process, the shape of casting part can be optimized by balancing the stress distribution. They intended to reduce weight and cost of the steering knuckle by replacing the forged part by a cast part with optimized shape and comparable strength. In this regard, stress and rigidity analyses and fatigue and impact experiments were conducted on the steering knuckle. They concluded that the Young’s modulus of castings
was lower than that of forging by 20%, and fatigue strength of the cast steering knuckle was lower than the original forged steering knuckle by 23%.

Fatigue crack growth resistance can also be an important consideration, when evaluating fatigue performance. Fatigue crack growth behavior of Q&T steels, which are often used to produce forged products, is compared to cast steel in Figure 2-14 (Rice, 1997). The Q&T 4140 steel exhibits superior fatigue crack growth resistance compared to the cast SAE 0030 steel. Machined bars and plates may be more susceptible to fatigue and stress corrosion because machining cuts material grain pattern. When a fabricated component has a high number of inches of weld in critical stress area, a greater chance exists that problems in the weld itself or from micro structural defects in the neighboring heat affected zone will cause failure in the field. High-stress welded joints are generally less capable in overloads or cyclic fatigue than junctions formed in the forging or metal casting process.

Traditionally, forged components have been produced from heat treated carbon and low alloy steels. Although heat-treated steels are still widely used, air cooled forging steels are becoming increasingly popular. Through a reduction in energy consumption, fewer process steps and lower inventories, these forgings can offer significant cost savings. Cristinacce et al. (1998) provide some recent examples of the range of components produced from air-cooled forging steels. An air-cooled 0.53%C steel was used in the production of hubs and spindles, where an approximately 400% increase in hardness was achieved compared to the heat-treated forgings. In another case, a swivel hub was redesigned as a forging in place of steel casting that had been proposed originally. The steel casting exhibited unacceptable distortion of the steering arm in heat treatment, surplus material leading to high machining cost, and excessive weight affecting the unsprung mass of the suspension design. The redesigned swivel hub was forged and control air-cooled. The results of mechanical tests on
both the forging and the casting are given in Table 2-2. The forging had superior strength, ductility, and hardness, compared to the heat-treated casting. The use of the forging also resulted in lower weight by 21%, better dimensional control, less machining, and avoidance of heat treatment costs.

van Bennekom and Wilke (2003) compared forged and cast stainless steels for physical, mechanical and corrosive properties. They concluded that “the forged components exhibited better corrosion resistance, better mechanical properties, superior machinability and surface finish and were more suitable for non-destructive testing. However, more complex components could be obtained by castings and the cost per component produced was lower. For the specific case of fatigue properties, steel cleanliness, uniformity of the grain size and microstructure, being free from segregation, and being free from slag and other metallic or non-metallic inclusions were included as factors that affected the fatigue resistance of cast and forged steel components. The effect of surface finish was ignored, since most components were machined to attain the final dimensions and tolerance after casting or forging.” It should, however, be noted that machined forged surfaces have superior fatigue resistance as compared to machined castings. This is because casting porosities, which significantly affect fatigue behavior of castings, are still present after machining.

van Bennekom and Wilke (2003) also added that “the cleanliness of forgings was superior to the cleanliness of similar components that have been cast. The same applied to the presence of slag and other metallic or non metallic inclusions. The grain size of forgings was substantially finer than that of castings since the coarser solidification structure has been destroyed and refined by the hot forming operation. Finer grain sizes resulted in a substantial increase in the fatigue strength of the stainless steel. The fatigue properties were also
considered to depend to a large extent on grain orientation effects. During the solidification process in castings, the grains structure was highly orientated in the transverse direction to the surface since solidification started at the surface and then propagates inwards via the growth of coarse columnar grains. This grain orientation was particularly detrimental from a fatigue point of view and as such the forged components in which the grains were aligned with the profile of the forging displayed superior fatigue properties.

**P/M Forging vs. Conventional Forging**

P/M offers high precision and low weight tolerances leading to less machining operations in comparison to classically forged or cast components. On the other hand, fatigue resistance and toughness are not generally as good as steel forgings (Esper and Sonsino, 1994). Jang et al. (2000) conducted a study on powder materials and production processes by producing the clutch disk spline hub of automobile, to replace the existing forged component. They also investigated mechanical properties and microstructure along with the performance of torsional durability test of three types of powder materials. They concluded that one of the produced P/M samples, which is a diffusion alloy powder and is treated with carburizing -tempering, performed better in torsion durability tests and wear resistance than that of existing forged steel component. They also concluded that toughness could be improved to the same level as forged metal, if powder metal is sintered and treated with adequate condition. There was no mention of cost and comparison with the forged part, however.

In an investigation about P/M connecting rods, Whittaker (2001) states that powder forged connecting rods are around 30% more expensive than the drop forged product at the as-formed blank stage. However, the machining cost savings compared to a conventionally split drop forging are sufficiently high to create a cost advantage of around 10% for the
powder forged product as a fully machined connecting rod. In the meanwhile, the recent introduction of fracture splitting of drop forged steel connecting rods has reversed this cost advantage, as the finished machined cost of drop forged fracture-split steel connecting rods is 3% lower than fracture split powder forged one, but 19% higher than the single press/sinter connecting rods. Another study by Repgen (1998) believes that the fracture-cracking technology reduces production cost of connecting rods by 25% compared to the conventional forging.

A Comparative study was performed by Afzal and Fatemi (2004) on fatigue behavior of forged steel and P/M connecting rods including strain-controlled specimen tests and component load-controlled bench tests. Specimen testing showed long-life fatigue strength defined at 10^6 cycles of the forged steel to be 27% higher than that of the P/M, as shown in Figure 2-15. This resulted in about an order of magnitude longer life for the forged steel, as compared with the P/M. Forged steel and P/M connecting rod bench test results as shown in Figure 2-16 indicated that the forged steel connecting rod exhibits 37% higher fatigue strength, as compared with the powder metal connecting rod. This increased strength resulted in about two orders of magnitude longer life for the forged steel connecting rod. The difference in fatigue performance between the two connecting rods increased with longer lives.

In general, the main advantages of P/M parts are reduction of waste material as well as machining operations and low unit cost when mass-produced; while their main disadvantages are high cost of dies, typically lower physical properties, higher cost of materials, limitations on the design, and the limited range of materials which can be used.
Thixoforming vs. Forging

Thixoforming has come to play an increasing role as an alternative to forging, especially in obtaining high-strength aluminum components for lightweight automotive designs. In this process a semi-solid metal is injected into a closed die. Thixo-formed aluminum components are often intended to replace steel forgings to form near-net shape components, or nodular cast iron components to reduce the solidification shrinkage. Hirt et al. (1997) used a pilot thixoforming system to redesign, thixoform and test an aluminum steering knuckle, as a thin walled structural component subject to high loads, and to compare the product with the original steel-forged steering knuckle. They combined steering knuckle thick-walled and thin-walled areas with stiffening ribs and undercuts to gain necessary yield strength, fracture toughness and stiffness. As a result, the weight of the new part was 50% below that of conventional forged steel design, despite identical functional capabilities.

There are numerous challenges for applying this technology to carbon and alloy steels, however, according to the Steel Industry Technology Roadmap (AISI, 2002). These include the high melting points and relatively small differences between liquidus and solidus temperatures and the need for advanced mold/die materials to withstand the high temperature & pressures. This makes the process more costly than conventional processes. Process controls for steel cleanliness, chemical uniformity and microstructure consistency throughout the section are also needed.

Comparisons of Multiple Processes

The influence of surface roughness and defects on fatigue life for various manufacturing processes is a key consideration in durability performance. Processes such as P/M have sometimes been more appealing, if they require less machining after production
(Blarasin and Giunti, 1997; Rice, 1997). Surface effects also include differences in microstructure, chemical composition, and residual stresses. Figure 2-17 shows a comparison of fatigue strength in various manufacturing processes for a front suspension arm. Even though for the hot forged steel surface defects of blank surfaces reduce fatigue strength by 30% from that of the machined surface, the fatigue strength is still considerably better than that of the nodular cast iron arm.

Surface enhancement processes like shot peening and shot blasting can affect fatigue performance of components. In the previously-mentioned study by Afzal and Fatemi (2004) on fatigue behavior of forged steel and P/M connecting rods, it was found that in the bench test of the P/M connecting rods the crack originated either at the surface or subsurface, while for the forged steel connecting rods cracks started subsurface. It was emphasized that for the forged steel connecting rod, the S-N approach predictions are reasonable if the predictions are based on smooth rather than forged surface finish. This was attributed to the fact that the beneficial compressive residual stresses on the surface from the shot blasting process nullify the detrimental effect of forged surface finish.

Strain hardening of the surface layer has a strong influence on fatigue behavior and depends on the depth of the deformed layer. Surface decarburization can occur after hot forging, which can cause different defects on the surface layer, reduce strain hardening of the surface layer, and consequently reduce the fatigue strength. Excessive strain hardening resulting from large deformations can also produce cracking and flaking of the surface and significantly reduce the fatigue strength.

Each manufacturing process is capable of producing a part to a certain surface finish and tolerance range without extra expenditure. Figure 2-18 compares the surface roughness and tolerance range for a number of manufacturing processes. It shows that forging is
superior to sand casting and even die-casting in terms of producing parts to some characteristic tolerance and surface finish, but inferior in comparison with machining processes. Therefore, forgings and castings generally require additional finishing operations, such as heat-treating, to modify properties, and then machining to obtain accurate finished dimensions. For forged components, the cost of machining is typically 50% of the total cost of the part (Naylor, 1998). Surface defects can be the source of fatigue failures, and they may lead to such other problems as corrosion and wear during the service life of the component. The forging process tends to reduce surface porosity and discontinuities (and may close up small internal cavities). Surface porosity and discontinuities occasionally appear on steel castings and require weld repair.

To compare directional properties, cast and powder forged parts do not exhibit grain flow or directional strength. Because hot working refines grain pattern and imparts high strength, ductility and resistance properties, forged products have lower possibility of internal defects compared to castings and they are manufactured without the added costs for tighter process controls and inspection that are required for casting. However, tensile strength, elongation and impact properties in forged products decrease in the transverse direction(s). Thus forgings are anisotropic. In castings, the metal is typically isotropic with similar properties in all directions, although flow lines, porosity and several other casting defects can show up depending on casting practice.

Figure 2-19 shows the variation in impact strength, yield strength and percent elongation with direction. While the properties for the cast part remains constant, the impact strength, reduction of area, and percent elongation change significantly from longitudinal to transverse direction in the forged part, though tensile strength and yield strength remain nearly constant. Additionally, the service conditions of the components must be carefully
evaluated. If the loading is uniaxial along the longitudinal axis, then the directionality of the forging is an advantage. As the stresses become multiaxial, directionality becomes more complex. In comparison to welding, selective heating and non-uniform cooling that occur in welding can yield undesirable metallurgical properties like inconsistent grain structure that could rarely be found in an appropriately forged component. In use, a welded seam may act as a metallurgical notch that can lead to part failure.

Although forging and casting processes are capable of offering favorable shape complexity, the level of dimensional tolerances drops as the component becomes more complex. P/M offers good dimensional control and, in many instances, results in reduction of machining and finishing operations; in this way it reduces scrap and waste and saves energy. But the nature of P/M process imposes limitations on part size and shape complexity. A problem with P/M parts is lack of proper density at sharp corners or stress concentration areas, and for larger parts due to the limitations of the process. This is even more prominent in more geometrically complex shapes.

Production quantity is another major factor. A cost comparison for a typical automotive component (connecting rod) is provided in Figure 2-20. All other factors being the same and depending on the number of pieces required, manufacturing a certain part by, for example, expandable mold casting may well be more economical than doing so by forging and on the other hand, for large quantities forging is more economical.

To conclude, forged products are compared to competitor cast and powder-forged products with respect to a number of important mechanical properties and manufacturing specifications in Table 2-3. Forged products exhibit higher strength, ductility and toughness compared to casting and P/M parts and the fatigue crack growth resistance of the forged parts is superior. Directional strength, that could be a favorable property if used properly,
only exists in forging. Internal defects exist in forged and cast products, while the oxygen trapped in the powder during the compacting process of P/M parts results in porosity and induces a detrimental effect on mechanical properties. From the economic perspective, the P/M process has higher initial tooling cost. This process generates the least material waste among the three processes, though this advantage can be overshadowed by higher cost of raw material and the non-uniform distribution of density in P/M products. Surface finish is another important factor where forged parts have better specifications compared to the other two processes. Forgings have good response to surface enhancement processes; for instance, the surface defects in forgings may be reduced and/or made harmless by shot blasting process. Choosing the right process for manufacturing a component needs thorough acquaintance with different aspects including part’s design criteria, service conditions, and economic aspects.

2.4 Durability Assessment of Automotive Components

This part of the literature review is dedicated to the procedures for designing automotive components for durability and fatigue. The results of selected studies on determination of local stresses and strains, notch analysis, force and moment measurements, multiaxial stress/strain, fatigue failure diagnosis and analysis guidelines, and component fatigue test are briefly discussed. A number of methods and models implemented in durability assessment of steering knuckle or similar fatigue-critical automotive components are explained. A summary of the procedures of fatigue life prediction methods implemented in literature for automotive components is provided in Appendix A.

The flow chart in Figure 2-21 shows the fatigue design process of components. In this design process the designer gathers input data including geometry, loading history,
material properties, and environmental parameters. Implementing the design criteria, the configuration, material and manufacturing processes of the component are selected. Stress and strain analysis enables evaluating the critical locations of the designed component under the assigned loading condition. Generally four fatigue life analysis models are commonly used; the nominal stress-life (S-N) model, the local strain-life (ε-N) model, the fatigue crack growth \((da/dN-\Delta K)\) model, and the two-stage model, which combines the second and third model to incorporate both macroscopic fatigue crack formation (nucleation) and fatigue crack growth. While nominal stress approach may fall short due to complexity of geometry and loading in many cases, local stress or strain approach has been popularly used in the automotive industry. When designing components subjected to occasional overloads, particularly for notched components where cyclic plastic deformation can be significant, cyclic ductility plays an important role. This is typical of suspension components, such as steering knuckle, that are considered as fatigue critical parts.

The load history of an automotive component is typically variable amplitude and, for instance in the case of suspension components, complex load spectrums exist. Choosing a proper damage model is, therefore, the next step in fatigue design that accounts for the cumulative effects of the cycles. The fatigue life calculated for the component is verified at the next step by component or vehicle test and the component’s configuration, material and manufacturing processes are modified in an iterative process to achieve the optimum design.

The methods for component durability assessment could be categorized into experimental and numerical procedures as shown in Figure 2-22. While, historically, the numerical pre-dimensioning was followed by experimental optimization of particular components and experimental proof-out of a system consisting of different components, the present industrial trend interacts these phases of product development with each other by
simultaneous engineering, Figure 2-23, in order to reduce time. This procedure can deliver a reliable design only if the numerical assessment considers service experience and is accompanied by experimental verification. The experimental methods comprise material testing (stress or strain controlled), strain analysis and component testing (uni- or multiaxial); numerical methods consist of stress/strain analysis by finite or boundary elements, the simulation of dynamic system behavior with regard to service stresses, the simulation of component strength properties and fatigue life calculations (Berger et al., 2002).

Blarasin and Farsetti (1989) adopted a model that utilizes the fatigue properties of the material from specimen testing, nominal load history and notch factors to predict fatigue life of similar steering knuckles made from Q&T and micro-alloyed steels. They assumed that the fatigue behavior of the component is equivalent to that of the smooth specimen with respect to material properties, metallurgical structure and surface conditions, subject to a stress history corresponding to that acting in the critical zone of the component. The local behavior of the material was described by utilizing the cyclic stress-strain curve. The damage parameter utilized was the Smith-Watson-Topper (SWT) parameter. Component bench tests were performed by subjecting the component to a time history of the primary load. The results showed that predictions over- and under-estimate fatigue life for Q&T and micro-alloyed steel steering knuckles, respectively. They attributed the discrepancy to an underestimate of the stress concentration factor value as well as to differences between the microstructures of the components (where in the critical region, in a matrix of tempered martensite quantities of ferrite were found at the austenitic grain boundaries) and the fatigue specimens (characterized by homogeneous tempered martensite). Other sources of this discrepancy could be the component critical location stress states and gradient, surface condition, residual stresses, etc.
Conle and Mousseau (1991) used vehicle simulation and finite element (FE) results to generate fatigue life contours for chassis components using automotive proving ground load history results combined with computational techniques. They concluded that the combination of vehicle dynamics modeling, finite-element analysis, and fatigue analysis is a viable technique for the fatigue design of automotive components.

In an application of the local strain approach, Lee et al. (1995) developed a methodology to quantitatively assess fatigue lives of automotive structures and to identify critical and non-damaging areas for design enhancement and weight reduction. An MS-3760A cast iron steering knuckle was the example component of this study. The methodology combines the load-time history file with results from elastic finite element analysis (FEA) to estimate fatigue lives. The load- and moment-time histories of the wheel (Figure 2-24) showed the nature of loading on a typical steering knuckle, that includes occasional overloads. The differences between the fatigue lives observed in bench tests and predicted lives in the inelastic range were found to be factors of 3.9 and 1.4 of the fatigue life with reliability of 50% and confidence level of 50% for fore/aft and lateral loading tests, respectively.

Complexity of loading, plastic deformation due to overloads, and the influence of manufacturing process parameters in fatigue assessment of the steering knuckle are emphasized by Diboine (1996). It is indicated that the loading condition in service is non-proportional multiaxial, variable amplitude and much more complex than for engine components. In his work, the loading was simplified to two block loadings called normal and accidental, whose levels were defined such that the same damage values from the track load recordings with respect to the loading conditions were obtained. The accidental block was found to represent only a few percent of the total number of cycles, but is important to
fatigue analysis as it is usually very close to plastic loading. The commercial software tools used for this component gave less accurate predictions when the loading is a mixture of elasto-plastic and fully elastic cycles. Modelling the residual stresses under variable amplitude loading, the surface finish effects, the manufacturing process, and formulating fatigue criteria for both intermediate and infinite lives with provision for overloads are named as a number of problems not fully solved.

A practiced procedure in durability analysis is described in a review paper by Conle and Chu (1997) and features using strain-life results from material tests, simulating three-dimensional stress-strain models and multiaxial deformation paths to assess fatigue damage, and searching for a critical plane for the most damaging direction. After the complex load history is reduced to a uniaxial (elastic) stress history for each critical element, a Neuber plasticity correction method is used to correct for plastic behavior.

Devlukia and Bargmann (1997) conducted fatigue assessment of a suspension arm using deterministic and probabilistic approaches. The strength reduction effect due to surface roughness was accounted for by representing the surface as a collection of notches and making use of Neuber's rule. The strength reduction effects due to the surface roughness were assumed to be similar under constant and variable amplitude loading. It was concluded that residual stress demonstrated a more pronounced effect under constant amplitude loading as compared to variable amplitude loading. Cumulative damage under variable amplitude loading sequence of long duration on simple specimens was non-conservative by a factor of about two as compared to measured data. The prediction of component lives based on specimen data and strain-life method was conservative by a factor of about two.
Sonsino, Kaufmann, Foth and Jauch (1997) performed constant and variable amplitude fatigue tests on induction-hardened automatic transmission shafts made from Ck 35 mod SAE 1038 steel, in order to evaluate safety reserves of the shafts and to apply several selected methods to assess fatigue life, to compare these methods and to gain insights for future component development. They concluded that the required data and criteria in order to make a reliable prediction of fatigue life are:

- S-N curve of the component on the basis of local stresses/strains including the influences of geometry, material, surface condition and residual stresses on fatigue behavior;
- actual local material states (e.g. hardened or non) at the locations that are critical for failure;
- component and material mean stress modification model for the critical location;
- cyclic stress-strain curves for the particular material states at the critical failure locations; and
- local (equivalent) strains or stresses in the critical area.

It was concluded that methods based on local stresses and strains (such as SWT parameter and component-related Haigh diagram) predict fatigue life better than the methods based on nominal loads.

Taylor (1997) proposed a technique for the prediction of fatigue failure in the presence of stress concentrations. The technique, called “crack modeling”, predicts fatigue failure in the presence of stress concentration through a modification of linear elastic fracture mechanics (LEFM). FEA was used in conjunction with a modeling exercise in order to extend the method to include bodies of arbitrary shape subjected to any set of loads. The method was first tested using standard notch geometries (blunt and sharp notches in beams),
where accurate predictions of fatigue limit could be achieved. It was then applied to an industrial problem, giving a reasonably close prediction of high-cycle fatigue behavior for an automotive crankshaft. It was concluded that LEFM can be extended to predict the fatigue behavior of bodies containing notches of standard geometry, instead of cracks. The method is postulated to require only simple mechanical property data (the material fatigue limit and stress-intensity threshold) and to use only linear-elastic FE modeling. Fracture mechanics theory was used without the need to specifically model the presence of a crack and uses far-field elastic stresses to infer behavior in the region of a stress concentration.

To verify this crack modeling method for torsional loading, Taylor, Zhou, Ciepalowicz and Devlkia (1999) described the analysis of fatigue failure in an automotive crankshaft as a result of loading under test conditions in bending and torsion. They concluded that the crack-modeling method, previously used for the analysis of the crankshaft under bending loads, could be extended to consider torsional loading. The model was claimed to correctly predict the fatigue limit and also the crack initiation point and growth direction. Also, the method was expected to be successful in analyzing any combination of bending and torsion applied in-phase. Out-of-phase loading was not considered. Moreover, predictions were not significantly affected by refinement of the FE mesh, even though this had a strong effect on the value of the hot-spot stress.

Due to the significance of determining assembly stresses in aluminum knuckles, Taylor et al. (1999) proposed analytical and numerical methods based on mechanics of materials principles and FEA, respectively, for determining the stress distributions in steering knuckle/tapered stud assemblies. The results showed that plane stress solutions for the assembly radial and hoop stresses in a steering knuckle boss provide good correlation to the FEA solutions for given draw distances. Furthermore, the effects of the coefficient of
friction do not significantly affect the hoop and radial stress distributions in the steering knuckle.

Witter et al. (1999) converted a steering knuckle into a 6-DOF transducer to be able to estimate the operating wheel translation force and moment inputs to a Mercury Sable steering knuckle. To reach this goal, a 6-DOF load cell was used to provide an estimate of the six forces and moments’ inputs at a point on the plate bolted to the steering knuckle. They concluded that the calibration matrix did not vary significantly with suspension height, but did vary significantly with large steering angles. Moreover, the strain gage responses were sensitive to moments. It was suggested to place the vehicle on a 4-poster and apply forces and moments through the 4-poster exciter to prevent the effect of un-suspended vehicle array calibration problems. The results of their analysis was compared with two cornering tests on the same design, showing an 11% error in forces and moments with respect to the physical test results conducted on the prototypes.

Local strain approach was used to analyze fatigue behavior of a hardened and tempered forged vehicle axle steering arm made of 41Cr4 low alloyed steel by Savaidis (2001). The elastic-plastic strain-time path was evaluated using the strain-time sequence measured at the failure critical location, taking into account the Masing and memory behavior. The SWT damage parameter and a parameter developed by means of elastic-plastic fracture mechanics (J-integral) based on a semi-elliptical surface micro-crack were used for estimation of damage caused by the local stress-strain path. Introducing a factor to describe the decrease of the endurance limit stress due to surface roughness, it was shown that the experimentally determined fatigue lives agreed well with those calculated for the critical location using the parameter based on J-integral.
Multiaxial fatigue is an important factor that should be considered in the fatigue analysis of components. Kocabicak and Firat (2001) proposed a bi-axial load-notch strain approximation for proportional loading to estimate the fatigue life of a passenger car wheel during the cornering fatigue test under plane stress conditions. In their work the elasto-plastic strain components were calculated analytically using the total deformation theory of plasticity. In addition, the input for the load–notch strain analysis was the measured or calculated plastic strain state at the notch together with the material stabilized cyclic stress-strain curve evaluated from unnotched axial specimens. The damage accumulation was based on the Palmgren-Miner rule.

One of the objectives of the present study was to conduct component testing on steering knuckle in order to obtain component fatigue performance. The configuration of the test machine, the control mode, fixturing of the steering knuckle, simulation of the loads and moments applied to the component in service, and testing frequency are among the considerations to conduct a well-simulated component test from which reliable data could be obtained. To reach this goal, a number of studies, which include component testing on steering knuckle and other similar automotive parts, were reviewed. These include studies by Lee (1986), Osuzu et al. (1986), Gunnarson et al. (1987), Blarasin and Farsetti (1989), Houshito et al. (1989) and Witter et al. (1999) on vehicle steering knuckle and the study by Savaidis (2001) on steering arm. Table 2-4 shows a summary of the test configurations used in these studies. In addition, brief descriptions of testing details are provided.
2.5 Optimization with Material, Manufacturing and Cost Considerations

The results of literature review for the optimization of steering knuckle and similar automotive components are discussed in this section. The focus of the review, as well as the focus of the optimization task in this study has been on more of a global view of the optimization problem including geometry optimization, alternative materials, and manufacturing process modifications and costs. In this respect, apart from issues like mathematical shape optimization, those studies were more emphasized that either include manufacturing and alternative material considerations, or would lend to that purpose. In this regard, the first set of the review is dedicated to shape optimization of automotive components. The second set relates to manufacturing and cost considerations in optimization.

Size and Shape Optimization

Botkin (1991) used a shape design modeling with three-dimensional mesh generation to model and optimize vehicle suspension components, namely suspension arm and steering knuckle. In the model the control arm geometry was assembled from two parts: boss and arm, and similarly the steering knuckle consists of boss, slab and hub (see Figure 2-25). A preliminary set of design primitives were developed which can be assembled into complete solid models. The resulting models were associated with design variables. It was stated that the mesh generation was capable of discretizing 3-D surfaces and solids into triangles and tetrahedral, respectively. It was concluded that the most accurate FEA solution results were required on the surface of a part and the curved faces, edges and vertices were necessary to be precisely represented.
Lee et al. (1995) presented the design process for a proposed weight-reduced steering knuckle. In their work, the non-damaging areas were identified by the life-contour plots of the original steering knuckle and a new design for weight reduction was proposed. Then, the original and proposed finite element steering knuckle models were loaded and constrained. They found that the proposed steering knuckle, being lighter, should survive the entire vehicle test field schedule by having fatigue damage values within acceptable limit. The percent reduction in weight due to this optimization was not mentioned.

The application of shape optimization to a steering knuckle using thermal shape vectors was presented by Krishna and Fetc ho (1998). Since the steering knuckle was not uniform in shape, it was cumbersome to generate shape vectors, which define the possible variations in the shape of the component, using the traditional loads and displacement methods. So thermal loading was used to generate shape vectors. For this purpose, the baseline model was assumed to be at a temperature of zero degree (Kelvin) and a temperature raise of 100 degrees was applied to twelve different regions of the steering knuckle, independently. These were the regions where shape was allowed to be modified. A shape optimization problem was defined; \textit{Minimize (objective):} volume of the structure, \textit{Subject to (Constraints):} maximum von Mises stresses less than or equal to material ultimate tensile strength. No safety factor was considered for the constraint. The redesigned steering knuckle offers a weight reduction of 7.6 percent. A shortcoming of such analysis is the fact that the real service loads applied to the component are not static, and considering ultimate tensile strength as constraint may not be reasonable when cyclic loading exists. However, the analysis method can be implemented by using cyclic rather than static stresses and fatigue strength instead of ultimate tensile strength. This issue was not discussed in this study.
Krishna (2001) investigated the application of shape optimization techniques to reduce the weight of a steering knuckle of a heavy truck suspension. A rigid body dynamic analysis of the suspension axle was carried out and steering knuckle load cases were obtained. Five different worst load cases were identified. A baseline FEA of the model showed that some parts of the steering knuckle experienced low stresses during all the five load cases. Shape vectors, which are the possible shape variations that the steering knuckle could undergo to reduce weight, were generated by numerical interpolation method. To generate the shape vectors, a group of elements were first identified and then a domain element was created to enclose these elements. All the nodes of the domain element and the other regular elements were grouped together as a node set. Perturbation vectors were identified to specify how the deformation is to be affected. The shape optimization problem was defined as follows – Minimize (Objective): Weight of the steering knuckle. Subject to (Constraints): Maximum von Mises stresses less than or equal to limiting value for each of the five load cases. The Sequential Linear Programming method as recommended by MSC/NASTRAN was used to solve the optimization problem and give the directions for changes. The shape optimization took 18 iterations for convergence. The objective (weight) was brought down by 12.7%, which was considered to be an appreciable saving both in material cost and final product cost. It was concluded that shape optimization techniques can be used effectively to bring down the weight of a complicated component like a steering knuckle, and multiple load cases can be handled by considering them separately. Figure 2-26 shows the superposed baseline and optimized steering knuckles of the study by Krishna (2001).

Ferreira et al. (2003) investigated the basic concepts of structural optimization and design for automotive durability. A trailing arm bracket seam welded to the reinforcement
bracket and bolted to vehicle body rail, was the case study. The optimization criteria were defined in terms of size optimization: component thicknesses (sheet metal components) were calculated in order to use a minimal amount of material and to provide maximum durability. For this purpose, a standard structural optimization procedure using Altair Optistruct (a general purpose finite element software and optimization package) was conducted with volume of the bracket as objective function, maximum admissible von Mises stress less than fatigue limit of the material (SAE 1010) as constraint, and thickness of trailing arm bracket, reinforcement and welds as design variables. The optimization problem converged after 7 iterations and 40 percent reduction in the volume was achieved. Then the optimized components were submitted to fatigue calculation in a full vehicle finite element model using FDynam (in house Ford software for durability calculation). The durability results showed that the bracket reinforcement was overloaded in the full vehicle model with complete load history. The solution was to increase the thickness of the components using high strength steel (SAEJ1392050) for the bracket reinforcement. This revised thickness and material configuration achieved the durability requirement for the components.

Lee and Lee (2003) presented optimization design methodologies in the design stages of a cast aluminum control arm for a suspension. Using topology optimization, the optimal layout and the reinforcement structure were obtained, and then the detail designs were carried out using shape optimization for structural rigidity and strength. The baseline of structural safety was considered to be yield strength and ultimate strength. In comparison with a stamped steel control arm, the mass reduction was 50 percent and the structural rigidity and static strength were improved up to 40 percent. Even though a control arm undergoes cyclic load history, fatigue strength was not mentioned in this study.
Material and Manufacturing Considerations

Application of forged parts made of microalloyed steels to automobile parts is becoming increasingly common due to their superior properties and cost reductions compared to conventional Q&T carbon steels. Microalloyed steels do not require heat treatment and, therefore, no additional machining for correcting distortions due to heat treatment after forging is necessary. The fatigue properties and toughness of microalloyed steel forgings have been demonstrated to fit for purpose. However, compared with heat treated low alloy steels their fracture toughness may be somewhat lower, even though still significantly superior to castings.

Farsetti and Blarasin (1988) investigated the possibility of replacing forged Q&T steels by forged microalloyed steels of appropriate composition and microstructure. They concluded that replacing Q&T steels with microalloyed steels is possible with the following considerations; for mechanical components that can be made from 800 MPa class steels, microalloyed steels could be used. With low-C high-Mn steels, satisfactory strength values can be attained, keeping good toughness properties. The higher-strength class microalloyed steel lends itself to an increase in strength and toughness by optimizing the micro structural parameters. Small ferrite grain and presence of bainitic phase will increase strength of microalloyed steels significantly.

Following the same goal, Kuratomi et al. (1990) developed lightweight connecting rods based on fatigue resistance analysis of microalloyed steels. Rotating bending fatigue tests on smooth and notched specimens as well as component buckling and load-controlled fatigue tests were conducted. Figure 2-27 shows the fatigue test results obtained with actual connecting rods made of SV40CL1 microalloyed steel (0.4% carbon steel) and S40C Q&T (equivalent to SAE 1040) steel. It was concluded that connecting rods made of the forged
microalloyed steel exhibit 25% higher fatigue limit than the similar forged Q&T steel and are 10% lighter in weight. In addition, in selecting high fatigue strength materials for connecting rods, they emphasized considering notch sensitivity because actual parts have small notches on their surface. They added that when the grain size became larger with increasing temperature, notch sensitivity of fatigue properties was not observed. They attributed this to the relation between the effective grain size and notch depth in a way that when the effective grain size was the same as or larger than one-half the notch depth, notch sensitivity decreased.

Material type and manufacturing process considerations are two key factors in reducing component cost. To reduce component cost, Repgen (1998) modified a connecting rod with respect to material and manufacturing process. For the material modification, a direct comparison of steel-forged versus hot formed powder metal connecting rods with similar cross section showed a 21 percent higher fatigue strength level for the forged part. To reach a considerable cost reduction for connecting rods it was found necessary to analyze the total component costs consisting of rough part and machining. It was noted that steel forged rough parts have the advantage of low material and production costs. With the development of fracture splitting the connecting rods a total cost reduction up to 25% was achieved compared to conventionally designed connecting rods. Further steps were taken by improving the forging technology to produce an optimized rough part with regards to tolerances that reduce the machining costs.

Nägele et al. (2000) reported the optimization of the process flow between bulk forming of parts and the subsequent machining processes of the forgings. Four case studies were presented in which the design of the part was modified in order to optimize the whole manufacturing process. The production process of a drive shaft flange (Figure 2-28a) that
was originally designed to be machining was switched to cold forming (forging). By doing this, cost was reduced by replacing expensive machining processing methods such as spline slotting. In addition, mechanical properties in the root of the spline teeth were enhanced by as much as 25-30% due to grain flow orientation, and weight and scrap reduction were achieved by optimum material utilization and process considerations.

For the case of an output shaft of the study by Nägele et al. (2000), Figure 2-28b, the original design required a hot forging with a cold calibration process where the excess material is removed at room temperature. To optimize the process, the cold calibration process was eliminated and replaced by a subsequent machining process in-house. This machining process incorporated not only the standard turning operation, but also a deep-hole drilling, induction hardening, cold rolling for a spline to standard and several threads. The greatest potential for cost savings was mainly found in technically advanced processing and forming operations. It was demonstrated that the emphasis needs to be on the manufacturing process as a whole and not just sectors of the manufacturing requirement. This example proved to be approximately 20% less expensive than the original intention of manufacturing for this component. The cost savings were attributed to elimination of the cold calibration requirements, same source forming and machining that allows a greater material utilization, which in turn improves die life, scrap and rework requirements, and delivering the component assemble-ready to the customer with no further processing required.

Another case of the same study by Nägele et al. (2000) involved a flanged cylinder. Figure 2-28c illustrates a warm-formed workpiece and the part after machining. The production process included warm forging and turning. An improved manufacturing process was developed where the forging process was modified and elimination of machining the
head was implemented. This was achieved by an additional cold calibration operation of the head in the area of the pierced elongated holes. A chamfer was added to the part which eliminated the formation of fins, reducing the machining time considerably, and eliminated a deburring operation. In addition to this, the shaft diameter was reduced which also helped to minimize the machining time. These changes did increase the cost of the forming operation, but the total part cost was reduced by 9% compared to the original design practices.

The final case of the study by Nägele et al. (2000) was a link shaft as shown in Figure 2-28d, in which a machined component (left) was replaced by a formed part (right). The original design started with a hot forged preform that required several subsequent machining operations. An optimal design was proposed for this component, in which the preform was warm formed in a three stage process with an additional cold calibration operation for the outside diameters. A final cold forming of the external spline to print specification produced the spline as assemble-ready. Finally the part was machined in a dual spindle lathe with two separate setups. The material usage for this component was 24% less and the weight reduction was 10.3% less material. The cost saving to produce this part with the new design amounted to 51.2%. In addition to these reductions, the drive shaft was also able to be re-designed due to the size reduction of this link shaft which was an overall reduction in weight of the transmission in whole.

The key aspects of the study by Nägele et al. (2000) were to demonstrate the significance of combining the forming and machining operations to be optimized by one supplier, and how this can be more cost effective than the alternative of simply delivering a formed part which then has to be subsequently machined. Also, it was emphasized that an intelligent combination of individual manufacturing processes may lead to the reduction of
individual processes and this leads to an increased cost effectiveness and competitiveness of
the component.

Flashless forging is introduced as an efficient method to reduce manufacturing cost. In
conventional hot forging of connecting rods, the material wasted to the flash accounts for
approximately 20-40% of the original workpiece (Vazquez and Altan, 2000). In order to
reduce the cost of forged products, the forging must be performed in a closed cavity to
obtain near-net or net shape parts. In flashless forging, the volume distribution of the
preform\(^1\) must be accurately controlled to avoid overloading the dies and to fill the cavity.
Additionally, the preform must be simple enough to be mass-produced. A study by Vazquez
and Altan (2000) deals with the preform design for flashless forging of a connecting rod and
introduces a new tooling concept for forging of complex parts with a controlled amount of
flash. A hot forging tooling was developed that would allow the forging of a connecting rod
with a controlled amount of flash. It was established that 5% material waste for connecting
rod may be reasonable under the present production conditions. The tooling was also
targeted to be simple enough to be used in mass production.

To improve fatigue performance of forged components, reducing surface defects
and/or their effects play an important role. One way to achieve this goal is to induce surface
compressive residual stresses. Applying localized inelastic deformation through processes
such as shot peening or surface rolling, are the common available methods of inducing
residual stresses. Fifty percent greater fatigue strength has been reported in rolled threads,
compared with cut or ground threads made of high strength steel (Stephens et al., 2000).
Figure 2-29 illustrates the beneficial effect of shot peening on fatigue resistance of gears.

\(^1\) Preform is the forging operation in which stock is preformed or shaped to a predetermined size and contour
prior to subsequent die forging operations; the operation may involve drawing, bending, flattening, edging,
fullering, rolling, or upsetting.
Thermal processes can also induce residual stresses. For example, in surface hardening of steel by processes such as induction hardening, carburizing, or nitriding, in addition to the hard surface produced, a beneficial compressive residual stress is also created on the surface. This compressive residual stress can very effectively prevent the formation of cracks.

To estimate machining process quality and cost, Nicolaou et al. (2002) presented a method for formulating a model for first estimating quality, cost and cutting fluid wastewater treatment impacts of two machining operations (end milling and drilling), and then for tradeoff decision making. A case study of an automotive steering knuckle was presented, where decision variables include material choice (cast iron versus aluminum), feed rate, cutting speed and wet versus dry machining. The results of Nicolaou et al. study was used to specify the cost attributes of the steering knuckle of the present study and is discussed in more detail in Section 7.1.

Manufacturing considerations in structural optimization is addressed in a study by Schramm et al. (2002). One of the techniques described is a manufacturing constraint for topology optimization. In this technique a draw direction is defined to open up the design into one direction only. This would allow for better casting, forging, and machining manufacturability. As an example, an engine mount bracket was optimized. A new layout with a reduced mass for a given design was sought. The new design should have had equal or better strength characteristics than the existing design. The mass of the initial design, which was manufactured from cast aluminum, was 950 g. Figure 2-30 shows the original part. The design space was defined by the space of the existing design. The following six load cases were of interest: Drive off forward; drive off backwards; driving into a pot hole; driving out of a pot hole; loads from an attached belt transmission; transportation of the engine block for assembly. The result of the topology optimization (Figure 2-30) was
transformed into a new design that was further improved using shape optimization. The final design (Figure 2-30) has a structural mass of 730 g, with deformations equal to or less than the initial design and stresses less than the allowable stress.

Shenoy (2004) and Shenoy and Fatemi (2005) investigated weight and cost reduction opportunities of a mid-size sedan forged steel connecting rod. Cost was reduced by replacing the conventional forged steel material with crackable C-70 forged steel. They concluded that fatigue strength was the most significant design driving factor in optimization of the component. Geometry optimization was performed under maximum engine operating speed and maximum gas load (service worst case conditions) and the optimized connecting rod satisfied all the constraints defined. The optimized geometry, while feasible to be manufactured, was 10% lighter and 25% less expensive than the conventional forged steel connecting rod, despite the lower strength of C-70 steel compared to the conventional forged steel.
Table 2-1  Classification of metals in decreasing order of forgeability (Kalpakjian and Schmid, 2000).

<table>
<thead>
<tr>
<th>Metal or Alloy</th>
<th>Approximate Range of Hot Forging Temperature (°C)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Aluminum alloys</td>
<td>400–550</td>
</tr>
<tr>
<td>Magnesium alloys</td>
<td>250–350</td>
</tr>
<tr>
<td>Copper alloys</td>
<td>600–900</td>
</tr>
<tr>
<td>Carbon and low–alloy steels</td>
<td>850–1150</td>
</tr>
<tr>
<td>Martensitic stainless steels</td>
<td>1100–1250</td>
</tr>
<tr>
<td>Austenitic stainless steels</td>
<td>1100–1250</td>
</tr>
<tr>
<td>Titanium alloys</td>
<td>700–950</td>
</tr>
<tr>
<td>Iron-base superalloys</td>
<td>1050–1180</td>
</tr>
<tr>
<td>Cobalt-base superalloys</td>
<td>1180–1250</td>
</tr>
<tr>
<td>Tantalum alloys</td>
<td>1050–1350</td>
</tr>
<tr>
<td>Molybdenum alloys</td>
<td>1150–1350</td>
</tr>
<tr>
<td>Nickel-base superalloys</td>
<td>1050–1200</td>
</tr>
<tr>
<td>Tungsten alloys</td>
<td>1200–1300</td>
</tr>
</tbody>
</table>

Table 2-2  Properties of cast and forged steels swivel hub (Cristinacce et al., 1998).

<table>
<thead>
<tr>
<th></th>
<th>C</th>
<th>Si</th>
<th>Mn</th>
<th>S</th>
<th>V</th>
<th>0.2% PS Lower $S_y$ (N/mm$^2$)</th>
<th>$S_u$ (N/mm$^2$)</th>
<th>$E_l$ (%)</th>
<th>$R/A$ (%)</th>
<th>3mm $U$ (J)</th>
<th>$H_v$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Casting</td>
<td>0.31</td>
<td>0.45</td>
<td>1.34</td>
<td>0.014</td>
<td>-</td>
<td>440</td>
<td>655</td>
<td>15-25</td>
<td>63</td>
<td>190-205</td>
<td></td>
</tr>
<tr>
<td>(wide arm)</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Forging</td>
<td>0.39</td>
<td>0.26</td>
<td>1.28</td>
<td>0.075</td>
<td>0.099</td>
<td>669</td>
<td>969</td>
<td>17</td>
<td>46.6</td>
<td>9</td>
<td>290-300</td>
</tr>
<tr>
<td>(wide arm)</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>
### Table 2-3 Summary of characteristic comparison between manufacturing processes.

<table>
<thead>
<tr>
<th>Property</th>
<th>Forging</th>
<th>Casting</th>
<th>Powder Metallurgy</th>
</tr>
</thead>
<tbody>
<tr>
<td>Strength</td>
<td>high</td>
<td>medium</td>
<td>medium</td>
</tr>
<tr>
<td>Ductility</td>
<td>high</td>
<td>low</td>
<td>low</td>
</tr>
<tr>
<td>Toughness</td>
<td>high</td>
<td>medium</td>
<td>medium</td>
</tr>
<tr>
<td>Fatigue crack growth resistance</td>
<td>good</td>
<td>poor</td>
<td>fair</td>
</tr>
<tr>
<td>Directional strength capability</td>
<td>yes</td>
<td>none</td>
<td>none</td>
</tr>
<tr>
<td>Heat treatment response</td>
<td>good</td>
<td>requires close control</td>
<td>good</td>
</tr>
<tr>
<td>Internal defects</td>
<td>possible</td>
<td>many</td>
<td>possible</td>
</tr>
<tr>
<td>Production volume</td>
<td>high</td>
<td>high</td>
<td>moderate to high</td>
</tr>
<tr>
<td>Production rate</td>
<td>high</td>
<td>low (sand casting) to high (die casting)</td>
<td>high</td>
</tr>
<tr>
<td>Initial tooling cost</td>
<td>medium</td>
<td>low (sand casting) to medium (die casting)</td>
<td>high</td>
</tr>
<tr>
<td>Production cost</td>
<td>low</td>
<td>low</td>
<td>low</td>
</tr>
<tr>
<td>Material waste</td>
<td>yes</td>
<td>yes</td>
<td>limited</td>
</tr>
<tr>
<td>Shape complexity</td>
<td>limited</td>
<td>limited</td>
<td>limited</td>
</tr>
<tr>
<td>Dimensional versatility</td>
<td>high</td>
<td>limited</td>
<td>limited</td>
</tr>
<tr>
<td>Dimensional accuracy</td>
<td>medium</td>
<td>medium</td>
<td>high</td>
</tr>
<tr>
<td>Surface finish</td>
<td>good to poor</td>
<td>poor</td>
<td>fair</td>
</tr>
<tr>
<td>Material versatility</td>
<td>high</td>
<td>limited</td>
<td>limited</td>
</tr>
</tbody>
</table>
Table 2-4  Test setup for steering knuckle and other relevant automotive components.

<table>
<thead>
<tr>
<th>Author(s)/Year</th>
<th>Component(s)</th>
<th>Test Type(s)</th>
<th>Component Test Configuration</th>
</tr>
</thead>
<tbody>
<tr>
<td>Lee/1986</td>
<td>Steering knuckle/other components</td>
<td>Load-controlled component fatigue tests,</td>
<td>A steering knuckle consisted of wheel stud, carrier tube, and steering arm. The characteristic</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Vehicle field tests</td>
<td>load amplitudes were 4.0 kN for wheel stud and carrier tube (test A) and 4.5 kN for steering arm</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>(test B). Acceptable criteria were no crack initiation and no permanent deformation during the test until $2 \times 10^5$ cycles. Different grips were developed for tests A and B.</td>
</tr>
<tr>
<td>Author(s)/Year</td>
<td>Component(s)</td>
<td>Test Type(s)</td>
<td>Component Test Configuration</td>
</tr>
<tr>
<td>---------------</td>
<td>------------------------------------------</td>
<td>------------------------------------------------------</td>
<td>------------------------------</td>
</tr>
<tr>
<td>Osuzu, et al. / 1986</td>
<td>Axle spindle, Knuckle arm</td>
<td>Specimen tensile and impact tests, Load-controlled component fatigue tests</td>
<td>Impact tests were conducted on complete axle spindle forgings using a drop weight test procedure. Fatigue tests were conducted on forged axle spindles.</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Gunnarson et al. / 1987</td>
<td>Steering knuckle/other components</td>
<td>Load-control component fatigue test</td>
<td>Schematic of fatigue testing is shown below:</td>
</tr>
</tbody>
</table>

![Axle spindle](image)

Drop weight impact test of axle spindle

Fatigue test setup of knuckle.
<table>
<thead>
<tr>
<th>Author(s)/Year</th>
<th>Component(s)</th>
<th>Test Type(s)</th>
<th>Component Test Configuration</th>
</tr>
</thead>
<tbody>
<tr>
<td>Blarasin and</td>
<td>Steering knuckle</td>
<td>Load-control component fatigue tests, Specimen strain-controlled fatigue</td>
<td>Fatigue bench tests were carried out on components mounted as they would be on a vehicle, the</td>
</tr>
<tr>
<td>Farsetti/1989</td>
<td></td>
<td>tests, Component road tests</td>
<td>articulated joints being substituted by appropriate flexible strip joints.</td>
</tr>
</tbody>
</table>

Locations of strain gages

Layout of the steering knuckle test bench.
<table>
<thead>
<tr>
<th>Author(s)/Year</th>
<th>Component(s)</th>
<th>Test Type(s)</th>
<th>Component Test Configuration</th>
</tr>
</thead>
<tbody>
<tr>
<td>Houshito et al. /1989</td>
<td>Steering knuckle</td>
<td>Tensile tests, Load-control component fatigue tests, Impact tests</td>
<td>Vertical, lateral and longitudinal load tests were conducted as shown below:</td>
</tr>
</tbody>
</table>

![Diagram of load tests](image)
<table>
<thead>
<tr>
<th>Author(s)/Year</th>
<th>Component(s)</th>
<th>Test Type(s)</th>
<th>Component Test Configuration</th>
</tr>
</thead>
<tbody>
<tr>
<td>Witter et al. /1999</td>
<td>Steering knuckle</td>
<td>Force/moment simulation on a vehicle</td>
<td>A small triangular plate was constructed and bolted to the steering knuckle at the locations shown. The strain-gaged steering knuckle and plate were reattached to the vehicle.</td>
</tr>
<tr>
<td>Savaidis/2001</td>
<td>Steering arm</td>
<td>Specimen strain-controlled fatigue tests, Load control component fatigue bench tests</td>
<td>The load was induced in one side of the component via hydraulic actuator while its other side was born by a block gear shaft positioned in the lug of steering arm.</td>
</tr>
</tbody>
</table>

Steering knuckle instrumented with 18 ICP strain gages.

Steering arm and load configuration.
Figure 2-1  Flow chart of the forging process (Kalpakjian and Schmid, 2000).

Figure 2-2  (a) Stages in impression-die forging of a connecting rod for an internal combustion engine. (b) Fullering and (c) edging operations to distribute the material when pre-shaping the blank for forging (Kalpakjian and Schmid, 2000).
Figure 2-3  Typical vehicle components that are subject to competition among manufacturing processes (Cristinacce et al., 1998).

Figure 2-4  Components of the engine block.
Figure 2-5 Front-wheel-drive car with MacPherson-strut front suspension and strut-type independent rear suspension (Saturn Corp.).

Figure 2-6 Arrangement for SLA suspension. The steering knuckle/spindle assembly in SLA suspension supports the wheels and attaches to the control arm with ball and socket joints. The control arm attaches to the frame of the vehicle through rubber bushings to help isolate noise and vibration between the road and the body (Halderman and Mitchel, 2000).
Figure 2-7  Multilink double-wishbone suspension uses a spring and damper unit. The long curved steering knuckle and very angled upper control arm allow its use in areas of limited size, Courtesy of American Honda Motor Co., Inc. (Birch, 2000).

Figure 2-8  Steering knuckle in strut suspension. In this type of suspension, the strut rod is the longitudinal support to prevent front-to-back wheel movement, Courtesy of Ford Motor Company (Birch, 2000).
Figure 2-9  Spindle assembly in a solid axle suspension, Courtesy of Ford Motor Company (Birch, 2000).

Figure 2-10  A twin I-beam front suspension. This particular swing axle design uses two ball joints to connect the steering knuckle to the axle, Courtesy of Ford Motor Company (Birch, 2000).
Figure 2-11  Static lateral input (left) and static longitudinal input (Haeg, 1997).

Figure 2-12  Steering knuckle assembly force reactions with longitudinal input isometric view (left) and plan view (Haeg, 1997).
Figure 2-13  Forging process parameters, the manufacturing influenced parameters and the effect on mechanical properties. Each parameter and its connector lines are coded with the same color.
Figure 2-14  Constant amplitude fatigue crack growth behavior of Q&T vs. cast steels (Rice, 1997).

Figure 2-15  Superimposed plots of true stress amplitude versus reversals to failure for forged steel and powder metal materials used in connecting rods (Afzal and Fatemi, 2004).
Figure 2-16  Experimental stress amplitude vs. cycles to failure for forged steel and powder metal connecting rods (Afzal and Fatemi, 2004).

Figure 2-17  Changes in fatigue performance of vehicle front suspension arm due to surface defects from forged and cast manufacturing processes (Blarasin and Giunti, 1997).
Figure 2-18  Capability of each manufacturing process of producing parts to some characteristic tolerance and surface finish under typical conditions (Schey, 2000).

Figure 2-19  Influence of forging reduction on anisotropy for a 0.35% carbon wrought steel. Properties for a 0.35% carbon cast steel are shown in the graph by a star (*) for purposes of comparison (Blair and Monroe, 2002).
Figure 2-20  Relative unit costs of a small connecting rod made by various forging and casting processes (Kalpakjian and Schmid, 2000).

Figure 2-21  Fatigue design flow chart (Stephens et al., 2000).
Figure 2-22  Methods of structural durability assessment (Berger et al., 2002).

Figure 2-23  Timing of past (top) and current product development periods (Berger et al., 2002).
Figure 2-24 Wheel loads and moments on the test road indicating the existence of overloads are part of the service history (Lee et al., 1995).
Figure 2-25 Design model (left) and meshes of the steering knuckle (Botkin, 1991).

Figure 2-26 Baseline superposed over optimized steering knuckle (Krishna, 2001).

**Result:**
Weight Reduction – 12.7%

Note:  
- Baseline  
- Optimized
Figure 2-27  Fatigue strength of microalloyed (SV40CL1) and Q&T (S40C) forged connecting rods (Kuratomi et al., 1990).

Figure 2-28  Case studies to optimize manufacturing processes of automotive components: (a) drive shaft flange, (b) output shaft, (c) flanged cylinder, and (d) link shaft (Nägele et al., 2000).
Figure 2-29  Effect of shot-peening on fatigue behavior of carburized gears (Stephens et al., 2000).

Figure 2-30  Original design (left), topology optimization results (middle) and final design of an engine mount bracket (Schramm et al., 2002).
Chapter Three

3 Material Fatigue Behavior and Comparisons

Material mechanical properties can be used to compare material and/or processes. Such properties are also often necessary as inputs for different analyses like life prediction and optimization. Modulus of elasticity, yield and tensile strengths, and ductility and strain hardening properties are obtained from tensile test. These properties are used to evaluate material’s tensile behavior under static or quasi-static loading conditions during elastic and plastic deformations.

In design situations where cyclic loading is involved, considering monotonic strength and ductility as the design variables is not sufficient to obtain a safe and reliable design, due to introduction of cyclic softening or hardening. In such cases, to which many automotive components are subject to them, considering cyclic deformation of the material is also essential.

From strain-controlled fatigue tests, cyclic deformation and fatigue properties are extracted. Cyclic modulus of elasticity represents cyclic stiffness prior to cyclic yielding. Cyclic yield strength shows the yield point of the material in cyclic loading. Cyclic strength coefficient and cyclic strain hardening exponent are the properties that help the designer to investigate cyclic deformation after yielding, as well as to compare the cyclic and monotonic deformation characteristics while the material plastically deforms.
Fatigue strength coefficient and exponent represent S-N behavior of the material often used to evaluate long-life fatigue performance, while fatigue ductility coefficient and exponent are properties used to describe ε-N behavior to evaluate low-cycle fatigue performance.

In the specimen testing program of this study as discussed in this chapter, it was intended to obtain monotonic, cyclic and fatigue properties of forged steel 11V37, cast aluminum A356-T6, and cast iron 65-45-12, and consequently compare them with respect to those properties. The monotonic properties included are stiffness, yield and ultimate strengths, percent elongation, percent reduction in area, strength coefficient, strain hardening exponent, and true fracture strength and ductility. The cyclic deformation parameters consist of cyclic stiffness, cyclic yield strength, cyclic strength coefficient, and cyclic strain hardening exponent. In addition, cyclic deformation characteristics such as hardening and softening were also investigated. The fatigue properties include fatigue limit, fatigue strength coefficient, fatigue strength exponent, fatigue ductility coefficient and fatigue ductility exponent. These properties were used to characterize and compare stress-life and strain-life material behaviors.

3.1 Experimental Program

3.1.1 Material and Specimen Fabrication

The materials were in the form of vehicle steering knuckles made of forged steel 11V37, cast aluminum A356-T6, and cast iron 65-45-12. Although utilized in three different vehicles, the three steering knuckles were selected from relatively similar-engine-size vehicles to make the comparisons.
Identical flat plate specimens with rectangular cross section were prepared out of the received steering knuckles for the monotonic and fatigue tests. The specimen configuration and dimensions are shown in Figure 3-1. This configuration was chosen such that the gage section length could be minimized to prevent buckling. A detailed finite element analysis (FEA) was performed to evaluate stress concentration at the radius. The FEA results indicate the stress concentration factor of about 1.05 for the specimen.

All specimens were machined in the Mechanical, Industrial, and Manufacturing Engineering Machine Shop at the University of Toledo. Figure 3-2 shows the steering knuckles of the three investigated materials, from which the specimens were machined. Specimens in three geometrical orientations were made from the forged steel steering knuckle to investigate the effect of directionality (see Figure 3-2). For cast aluminum and cast iron steering knuckles, since the material properties are independent of geometrical orientation, the specimens were machined from the hub and one of the arms, respectively.

The specimens were initially rough cut to a rectangular strip out of the vehicle steering knuckles using a milling machine and then inserted into a fixture for cutting the required geometry in another CNC milling machine. Using the CNC milling machine, final machining was performed to achieve the tolerable dimensions specified on the specimen drawings.

After machining, specimens were checked carefully for flatness on a machinist’s stone-flat surface. Any specimen with camber exceeding 0.1 mm (0.004 in) from end to end was rejected. The specimen gage section edges in the thickness and width directions were polished, the polishing marks coinciding with the specimen’s longitudinal direction. The polished surfaces were carefully examined to ensure complete removal of machining marks in the test section.
3.1.2 Testing Equipment

An INSTRON 8801 closed-loop servo-hydraulic axial load frame in conjunction with a Fast-Track 8800 digital servo-controller was used to conduct the tests. The calibration of this system was verified prior to beginning the test program. The load cell used had a capacity of 50 kN (11 klb). In order to achieve the best alignment of the specimens, two stops were designed and mounted on the hydraulically operated universal wedge grips with flat faces, during all tests. These stops helped to align the specimen’s ends in series with the load train. Any twisting of the specimens was avoided by using precisely machined blocks to prevent any relative rotation of the grips.

Total strain was controlled for all tests using an extensometer rated as ASTM class B1 (ASTM E83-02, 2002). The calibration of the extensometer was verified using displacement apparatus containing a micrometer barrel in divisions of 0.0001 in. The extensometer had a gage length of 0.2362 in (6 mm) and was capable of measuring strains up to 10%. In order to protect the specimen’s surface from the knife-edges of the extensometer, M-coat D mixture was used to “cushion” the attachment. The extensometer was carefully positioned at the center section of the specimen uniform gage section.

All tests were conducted at room temperature and were monitored using a digital thermometer. In order to minimize temperature effects upon the extensometer and load cell calibrations, fluctuations were maintained within ± 2 °C (± 3.6 °F) as required by ASTM Standard E606 (1998). Also, the relative humidity of the air was monitored using a precision hydrometer.

Significant effort was put forth to align the load train (load cell, grips, specimen, and actuator). Misalignment can result from both tilt and offset between the central lines of the load train components. According to ASTM Standard E606 (1998), the maximum bending
strains should not exceed 5% of the minimum axial strain range imposed during any test program. For this study, the minimum axial strain range was 0.0025 in/in. Therefore, the maximum allowable bending strain was 125 microstrains. ASTM Standard E1012 (1999), Type A, Method 1 was followed to verify specimen alignment. A 0.25 in × 0.25 in square cross-section bar with eight strain gages was used and the maximum bending strain was much smaller than that allowed by the ASTM standard.

3.2 Test Methods and Procedures

3.2.1 Monotonic Tension Tests

All monotonic tests in this study were performed using test methods specified by ASTM Standard E8-02 (2002). Two specimens for forged steel and one for each of cast iron and cast aluminum were used to obtain the monotonic properties. Due to the limitations of the extensometer, strain-control was used only up to 10% strain. Unless the specimen failed prior to this level, displacement-control was used after this point until fracture. A stress versus strain plot was obtained automatically for each test.

For the elastic and initial yield region (0% to 0.5% strain), a strain rate of 0.003125 mm/mm/min was chosen. This strain rate was three-quarters of the maximum allowable rate specified by ASTM Standard E8-02 (2002) for the initial yield region. After yielding (0.5% to 10% strain), the strain rate was increased by a factor of three (i.e., 0.0094 mm/mm/min). After the extensometer was removed, a displacement rate of 0.215 mm/min was used. This displacement rate provided approximately the same strain rate as that used prior to switching control modes.

After the tension tests were concluded, the broken specimens were carefully reassembled. The final gage lengths of the fractured specimens were measured with a
Vernier caliper having divisions of 0.001 in. Using an optical comparator with 10x magnification and divisions of 0.001 in, the final cross section dimensions were measured. It should be noted that prior to the test, the initial cross section was measured with this same instrument.

### 3.2.2 Constant Amplitude Fatigue Tests

All constant amplitude fatigue tests in this study were performed according to ASTM Standard E606 (1998). It is recommended by this standard that at least 10 specimens be used to generate the fatigue properties. For this study, three sets of tests for forged steel (in three geometrical orientations) and one set for each of cast aluminum and cast iron were conducted.

For forged steel, a total of 29 tests were conducted, in order to obtain the fatigue data as well as to investigate directionality effects. Specimens of three geometrical directions were tested at 6 different strain amplitudes ranging from 0.15% to 0.7%, with a total of 11, 8 and 10 specimens in directions A, B and C respectively. Figure 3-2 shows a schematic of the forged steering knuckle and the above-mentioned directions. For cast aluminum, a total of 14 fatigue tests at 8 strain amplitude levels ranging from 0.125% to 0.7% and for cast iron, a total of 10 fatigue tests at 5 strain levels ranging from 0.13% to 0.5% were conducted.

There were two control modes used for these tests; strain-control and load-control. Strain-control was used in all tests, except for: 1) a number of tests that were switched to load-control due to stabilized cyclic behavior during strain-control mode. For these tests, strain-control was used initially to determine the stabilized load. Then load-control was used for the remainder of the test. The reason for the change in control mode was due to frequency limitations of the extensometer. 2) A number of tests conducted in load-control...
mode due to negligible predicted amount of plastic deformation. And 3) run-out tests that were conducted in load-control mode. For the strain-control tests, the applied frequencies ranged from 0.1 Hz to 2 Hz. For the load-control tests including run-out tests, the frequency was increased to up to 30 Hz in order to shorten the overall test duration. All tests were conducted using a triangular waveform.

Strain amplitudes larger than 0.7% were not possible due to specimen buckling limitation. Instron LCF and SAX software tools were used for strain-control and load-control tests, respectively. During each strain-control test, the total strain was recorded using the extensometer output. Test data were automatically recorded, periodically, throughout each test.

### 3.3 Experimental Results, Observations, and Analysis

#### 3.3.1 Monotonic Deformation Behavior and Comparisons

The properties determined from monotonic tests were the following: modulus of elasticity ($E$), yield strength ($S_y$), ultimate tensile strength ($S_u$), percent elongation ($\%EL$), percent reduction in area ($\%RA$), true fracture strength ($\sigma_f$), true fracture ductility ($\varepsilon_f$), strength coefficient ($K$), and strain hardening exponent ($n$).

True stress ($\sigma$), true strain ($\varepsilon$), and true plastic strain ($\varepsilon_p$) were calculated from engineering stress ($S$) and engineering strain ($e$), according to the following relationships, which are based on constant volume assumption:

\[
\sigma = S(1 + e) \quad \text{(3-1a)}
\]

\[
\varepsilon = \ln(1 + e) \quad \text{(3-1b)}
\]

\[
\varepsilon_p = \varepsilon - \varepsilon_e = \varepsilon - \frac{\sigma}{E} \quad \text{(3-1c)}
\]
The true stress-true strain \((\sigma-\varepsilon)\) plot is often represented by the Ramberg-Osgood equation:

\[
e = \varepsilon_e + \varepsilon_p = \frac{\sigma}{E} + \left(\frac{\sigma}{K}\right)^{1/n}
\]  

(3-2)

The strength coefficient, \(K\), and strain hardening exponent, \(n\), are the intercept and slope of the best line fit to true stress \((\sigma)\) versus true plastic strain \((\varepsilon_p)\) data in log-log scale:

\[
\sigma = K (\varepsilon_p)^n
\]  

(3-3)

In accordance with ASTM Standard E739 (1998), when performing the least squares fit, the true plastic strain \((\varepsilon_p)\) is the independent variable and the stress \((\sigma)\) is the dependent variable. These plots for the two tests of forged steel and one test for each of cast aluminum and cast iron are shown in Figure 3-3(a), (b) and (c), respectively. To generate the \(K\) and \(n\) values, the range of data used in this figure was chosen according to the definition of discontinuous yielding specified in ASTM Standard E646 (2000). The data range used was between 1% and 9% total strain.

The true fracture strength, \(\sigma_f\), was calculated from:

\[
\sigma_f = \frac{P_f}{A_f} = \frac{P_f}{w_f t_f}
\]  

(3-4)

where \(P_f\) is the load at fracture, and \(w_f\) and \(t_f\) are the width and thickness at fracture, respectively. The true fracture ductility, \(\varepsilon_f\), was calculated from the relationship based on constant volume:

\[
\varepsilon_f = \ln \left(\frac{A_o}{A_f}\right) = \ln \left(\frac{1}{1 - RA}\right)
\]  

(3-5)

where \(A_f\) is the cross-sectional area at fracture, \(A_o\) is the original cross-sectional area, and \(RA\) is the reduction in area.
A summary of the monotonic properties for the three materials is provided in Table 3-1, including the ratios of each property with respect to that of forged steel. As discussed later in this report, since direction A is the primary loading direction of the component, it was selected to obtain monotonic data of forged steel. Table 3-2 summarizes the monotonic test results. Separate monotonic stress-strain curves are shown in Figure 3-4 for each material, and Figure 3-5 presents a direct comparison of the three materials’ monotonic stress-strain behavior.

Comparing $E$ values from the monotonic tests, the stiffness of cast aluminum and cast iron are 39% and 96% of forged steel's, respectively. The $E$ value of cast iron in the literature was found to be about 170 MPa, which is somewhat less than what was found in this study. This could be due to the characteristics of cast products, and parameters such as porosity and inclusions. To investigate the difference, further metallurgical and microstructural analyses are required that are beyond the scope of this investigation.

Ultimate tensile strength of forged steel is far above the other two materials, cast aluminum and cast iron being 37% and 57% of forged steel, respectively. The yield strength of cast aluminum and cast iron is also lower, 42% and 54% of forged steel, respectively. As a criterion to measure ductility, the percent elongation of cast aluminum and cast iron were found to be 24% and 48% of forged steel, respectively. This shows that cast aluminum and cast iron behave less ductile compared to the behavior of forged steel. This could also be observed from the fracture ductility results where true fracture ductility of cast aluminum and cast iron is about 23% and 59% of forged steel, respectively. Thus, with respect to strength and ductility, which are two major mechanical properties measured from monotonic loading, forged steel shows superiority to the two cast materials. As can be seen in Figure
3-5, strain hardening is more prominent for forged steel and cast iron, while cast aluminum behaves toward elastic-perfectly plastic behavior.

Although comparison of monotonic strength and ductility data as two major design parameters show superiority of forged steel, it should be noted that considering only monotonic properties for design purposes involving cyclic loading may lead to false conclusions due to introduction of cyclic softening and/or hardening.

### 3.3.2 Cyclic Deformation Behavior and Comparisons

#### 3.3.2.1 Transient Cyclic Response

Transient cyclic response describes the process of cyclic-induced change in deformation resistance of a material. Data obtained from constant amplitude strain-controlled fatigue tests were used to determine this response. Plots of stress amplitude variation versus applied number of cycles can indicate the degree of transient cyclic softening/hardening. Also, these plots show when cyclic stabilization occurs.

A composite plot of the transient cyclic response for the three materials studied is shown in Figure 3-6 and Figure 3-7. The transient response is normalized on the rectangular plot in Figure 3-6, and a semi-log plot is shown in Figure 3-7. While the first set of normalized plots present the response over the entire life, the second set magnifies the stress response over the early cycles, in order to facilitate observing the variation in stress. Even though multiple tests were conducted at each one of the strain amplitudes in most cases, data from one test at each one of the strain amplitudes tested are shown in these plots. For forged steel, the data plotted are for direction A only. These figures show some transient softening for forged steel, while cast iron and cast aluminum behave in a cyclically hardening manner.
3.3.2.2 Steady-State Cyclic Deformation

Another cyclic behavior of interest is the steady state or stable response. Data obtained from constant amplitude strain-controlled fatigue tests were also used to determine this response. The properties determined from the steady-state hysteresis loops are the following: cyclic strength coefficient \((K')\), cyclic strain hardening exponent \((n')\), and cyclic yield strength \((\delta_y')\). Half-life (midlife) hysteresis loops and data were used to obtain the stable cyclic properties.

Similar to monotonic behavior, the cyclic true stress-strain behavior can be characterized by the Ramberg-Osgood type equation:

\[
\frac{\Delta \epsilon}{2} = \frac{\Delta \epsilon_e}{2} + \frac{\Delta \epsilon_p}{2} = \frac{\Delta \sigma}{2E} + \left(\frac{\Delta \sigma}{2K'}\right)^{n'}
\] (3-6)

It should be noted that in Equation (3-6) and the other equations that follow, \(E\) is the monotonic modulus of elasticity, obtained from monotonic tests. However, since the value of \(E\) obtained from monotonic test of ductile cast iron was different from that published in the literature of ductile cast iron as discussed previously, the average \(E\) value from the first cycle of strain-control fatigue tests was used to calculate plastic strains for this material.

The cyclic strength coefficient, \(K'\), and cyclic strain hardening exponent, \(n'\), are the intercept and slope of the best line fit to true stress amplitude \((\Delta \sigma/2)\) versus true plastic strain amplitude \((\Delta \epsilon_p/2)\) data in log-log scale:

\[
\frac{\Delta \sigma}{2} = K' \left(\frac{\Delta \epsilon_p}{2}\right)^{n'}
\] (3-7)

In accordance with ASTM Standard E739 (1998), when performing the least squares fit, the true plastic strain amplitude \((\Delta \epsilon_p/2)\) is the independent variable and the stress
amplitude \( \frac{\Delta \sigma}{2} \) is the dependent variable. The true plastic strain amplitude was calculated by the following equation:

\[
\frac{\Delta \varepsilon_p}{2} = \frac{\Delta \varepsilon}{2} - \frac{\Delta \sigma}{2E}
\]  

(3-8)

These plots are shown in Figure 3-8. To generate the \( K' \) and \( n' \) values, the range of date used in the figure was chosen for \( 0.003 \leq \varepsilon_a \leq 0.007 \) for forged steel, \( 0.00375 \leq \varepsilon_a \leq 0.007 \) for cast aluminum, and \( 0.002 \leq \varepsilon_a \leq 0.005 \) for cast iron, based on the plastic strain range being considerable at these strain range domains.

The cyclic stress-strain curve reflects the resistance of a material to cyclic deformation and can be vastly different from the monotonic stress-strain curve. The cyclic stress-strain curve of each process is shown separately in Figure 3-9. It could be observed that the Ramberg-Osgood equation, Equation (3-6), provides good representation of cyclic deformation behavior. In addition, the forged steel curves for the three geometrical directions (Figure 3-9a) show that cyclic deformation curve is independent of direction. The results of two previous studies on fatigue properties of cast aluminum A356-T6 (Wigant and Stephens, 1987) and cast iron 65-45-12 (Meritor Automotive, 1997) are also illustrated in Figure 3-9b and Figure 3-9c, respectively. The curves from these two studies are similar to those found in this study. In Figure 3-10 the cyclic curves of the three materials are compared. The cyclic yield strength \( (\sigma'_y) \) of cast aluminum and cast iron is 54% and 75% of forged steel, and the cyclic strain hardening exponent \( (n') \) is 46% and 55% of forged steel, respectively. This shows the higher strength of forged steel against cyclic yielding, and its higher resistance to cyclic plastic deformation.

Superimposed plots of monotonic and cyclic curves for each material are shown in Figure 3-11. Figure 3-12 represents the same superimposed plot for all three materials. As
can be seen in Figure 3-12, forged steel has mixed mode cyclic behavior. Initially, this material cyclically softens, but then hardens at strain amplitudes larger than 0.54%. Cast aluminum and cast iron show cyclic hardening behavior by about 25% and 30%, respectively. Figure 3-13 shows a composite plot of the steady-state (midlife) hysteresis loops. The stable loops from only one test at each of the strain amplitudes are shown in this plot, even though at some strain levels more than one test was conducted.

3.3.3 Strain-Controlled Fatigue Behavior and Comparisons

Constant amplitude strain-controlled fatigue tests were performed to determine the strain-life curve. The following equation relates the true strain amplitude to the fatigue life:

\[
\frac{\Delta \epsilon}{2} = \frac{\Delta \epsilon_e}{2} + \frac{\Delta \epsilon_p}{2} = \frac{\sigma_f'}{E} \left(2N_f\right)^b + \epsilon_f' \left(2N_f\right)^c
\]

(3-9)

where \( \sigma_f' \) is the fatigue strength coefficient, \( b \) is the fatigue strength exponent, \( \epsilon_f' \) is the fatigue ductility coefficient, \( c \) is the fatigue ductility exponent, \( E \) is the monotonic modulus of elasticity, and \( 2N_f \) is the number of reversals to failure (which was defined at 50% load drop, as recommended by ASTM Standard E606 (1998)). A summary of the cyclic properties for the three materials is provided in Table 3-1. Table 3-3 to Table 3-5 provide the summary of the fatigue test results for directions A, B, and C of forged steel, and Table 3-6 and Table 3-7 provide these results for cast aluminum and cast iron, respectively.

The fatigue strength coefficient, \( \sigma_f' \), and fatigue strength exponent, \( b \), are the intercept and slope of the best line fit to true stress amplitude (\( \Delta \sigma/2 \)) versus reversals to failure (\( 2N_f \)) data in log-log scale:

\[
\frac{\Delta \sigma}{2} = \sigma_f' \left(2N_f\right)^b
\]

(3-10)
In accordance with ASTM Standard E739 (1998), when performing the least squares fit, the stress amplitude \((\Delta\sigma/2)\) is the independent variable and the reversals to failure \((2N)\) is the dependent variable. This plot for the three directions of forged steel is shown in Figure 3-14a. To generate the \(\sigma'_f\) and \(b\) values, the range of data used in this figure for directions A, B and C was chosen for \(0.00175 \leq \varepsilon_a \leq 0.007\). Table 3-8 compares fatigue constants for these three directions. It could be seen that direction A provides a longer fatigue life in the high-cycle region. It should be noted that this direction is the primary direction of stressing in the forged steel steering knuckle. Therefore properties generated in this direction are used as the basis for comparison with cast aluminum and cast iron.

Figure 3-15 presents the three materials S-N behavior and Figure 3-16 shows direct comparison of the three materials with respect to S-N behavior. Direction A of forged steel was used for this comparison. To generate the \(\sigma'_f\) and \(b\) values for cast aluminum and cast iron, the range of data used in these figures was chosen for \(0.0015 \leq \varepsilon_a \leq 0.007\) and \(0.0013 \leq \varepsilon_a \leq 0.005\), respectively. Comparison of long-life fatigue strength, \(S_f\), which is defined as the fatigue strength at \(10^6\) cycles, shows that the fatigue limit of cast aluminum and cast iron are 35% and 72% of forged steel, respectively. In addition, while the fatigue strength of forged steel at \(10^6\) cycles is expected to remain about constant at longer lives, fatigue strength of the two cast materials continuously drops with longer lives (see Figure 3-16). Figure 3-16, therefore, indicates significantly better S-N fatigue resistance of forged steel, as compared with the two cast materials.

The fatigue ductility coefficient, \(\varepsilon'_f\), and fatigue ductility exponent, \(\epsilon\), are the intercept and slope of the best line fit to calculated true plastic strain amplitude \((\Delta\varepsilon_p/2)\) versus reversals to failure \((2N)\) data in log-log scale:
\[
\left( \frac{\Delta \varepsilon_p}{2} \right)_{\text{calculated}} = \varepsilon'_f \left( 2N_f \right)^{\eta}
\]  

(3-11)

In accordance with ASTM Standard E739 (1998), when performing the least squares fit, the calculated true plastic strain amplitude \(\Delta \varepsilon_p/2\) is the independent variable and the reversals to failure \(2N_f\) is the dependent variable. Calculated true plastic strain amplitude was determined from Equation (3-8). This plot for the three directions of forged steel is shown in Figure 3-14b. To generate the \(\varepsilon'_f\) and \(\varepsilon\) values, the range of data used in this figure was chosen for \(0.003 \leq \varepsilon \leq 0.007\). Table 3-8 compares fatigue ductility constants for these three directions. Fatigue ductility parameters determine the short-life or low-cycle fatigue behavior of the material and its resistance against plastic deformation in that domain. Figure 3-14b shows that direction A provides better fatigue ductility than the other two directions.

Figure 3-17 presents the three materials plastic strain-life behavior and Figure 3-18 shows a direct comparison of them with this respect. The range of data used in these figures for cast aluminum and cast iron was chosen for \(0.00375 \leq \varepsilon \leq 0.007\) and \(0.002 \leq \varepsilon \leq 0.005\), respectively. Figure 3-18 shows the superiority of forged steel at short lives and to cyclic plastic deformation. In automotive design, cyclic ductility is a major concern when designing components subjected to occasional overloads, particularly for notched components, where significant plastic deformation can occur.

The true total strain amplitude versus reversals to failure plot, based on Equation (3-9), for the three directions of forged steel is shown in Figure 3-14c. The effect of anisotropy and directionality of forged steel could be observed, where it is shown that direction A offers about twice more life than the other two directions. This superiority of direction A remains nearly the same over the whole life spectrum. As indicated earlier, the loading direction of the forged steering knuckle induces the primary stresses in this direction.
Figure 3-19 displays the strain-life curves (Equation 3-9), the elastic strain portions (Equation 3-10), the plastic strain portions (Equation 3-11), and superimposed fatigue data for each material. Wigant and Stephens (1987) also investigated the low-cycle fatigue behavior of A356-T6 cast aluminum. The results of their study are also shown in Figure 3-19b. Fatigue properties of 65-45-12 cast iron were also available from Meritor Automotive (1997) and superimposed in Figure 3-19c. These Figures show that the current study resulted in somewhat better fatigue behavior of cast aluminum and cast iron compared to the other two studies, in the low-cycle fatigue region. This could be due to a number of reasons such as microstructural variations, as well as degree of alignment of the test machines used.

Comparisons of strain-life fatigue behavior of the three materials, as shown in Figure 3-20, demonstrate the superiority of forged steel over cast aluminum and cast iron. Forged steel provides about a factor of 5 longer life in short-life regime compared to cast aluminum and cast iron. In high-cycle regime, forged steel results in more than an order of magnitude longer life than cast iron, and about a factor of 3 longer life than cast aluminum.

The so called Neuber stress $\left(\sqrt{\sigma_a \varepsilon_a E}\right)$ versus life, known as Neuber plot, is shown in Figure 3-21. This plot is useful when analyzing component geometries with stress concentrations, where the notch root fatigue behavior is a function of both local stress and strain. Therefore, rather than considering the individual effects of stress amplitude (Figure 3-16) or strain amplitude (Figure 3-20), a Neuber plot considers the combined effects of both stress and strain amplitudes. This plot shows forged steel to have about two orders of magnitude longer life than cast iron and about four orders of magnitude longer life than cast aluminum.
Table 3-1 Summary of mechanical properties and their comparative ratios (forged steel is taken as the base for ratio calculations).

<table>
<thead>
<tr>
<th>Monotonic Properties</th>
<th>Forged Steel 11V37</th>
<th>Cast Aluminum A356-T6</th>
<th>Cast Iron 65-45-12</th>
</tr>
</thead>
<tbody>
<tr>
<td>Modulus of elasticity, $E$, GPa (ksi)</td>
<td>201.5(29,231)</td>
<td>78.1(11,327)</td>
<td>193.0(27,991)</td>
</tr>
<tr>
<td>Ratio</td>
<td>1</td>
<td>0.39</td>
<td>0.96</td>
</tr>
<tr>
<td>Yield strength (0.2% offset), $S_{y}$, MPa (ksi)</td>
<td>556.2(80.7)</td>
<td>232.4(33.7)</td>
<td>300.0(43.5)</td>
</tr>
<tr>
<td>Ratio</td>
<td>1</td>
<td>0.42</td>
<td>0.54</td>
</tr>
<tr>
<td>Ultimate tensile strength, $S_u$, MPa (ksi)</td>
<td>821.2(119.1)</td>
<td>302.7(43.9)</td>
<td>471.2(68.3)</td>
</tr>
<tr>
<td>Ratio</td>
<td>1</td>
<td>0.37</td>
<td>0.57</td>
</tr>
<tr>
<td>Percent elongation, %EL (%)</td>
<td>21</td>
<td>5</td>
<td>10</td>
</tr>
<tr>
<td>Ratio</td>
<td>1</td>
<td>0.24</td>
<td>0.48</td>
</tr>
<tr>
<td>Percent reduction in area, %RA (%)</td>
<td>37</td>
<td>10</td>
<td>25</td>
</tr>
<tr>
<td>Ratio</td>
<td>1</td>
<td>0.27</td>
<td>0.68</td>
</tr>
<tr>
<td>Strength coefficient, $K$, MPa (ksi)</td>
<td>1,347.3(195.4)</td>
<td>417.8(60.6)</td>
<td>796.5(115.5)</td>
</tr>
<tr>
<td>Ratio</td>
<td>1</td>
<td>0.31</td>
<td>0.59</td>
</tr>
<tr>
<td>Strain hardening exponent, $n$</td>
<td>0.157</td>
<td>0.095</td>
<td>0.187</td>
</tr>
<tr>
<td>Ratio</td>
<td>1</td>
<td>0.6</td>
<td>1.19</td>
</tr>
<tr>
<td>True fracture strength, $\sigma_f$, MPa (ksi)</td>
<td>496(71.9)</td>
<td>301(43.7)</td>
<td>219.2(31.8)</td>
</tr>
<tr>
<td>Ratio</td>
<td>1</td>
<td>0.6</td>
<td>0.44</td>
</tr>
<tr>
<td>True fracture ductility, $\varepsilon_f$ (%)</td>
<td>47</td>
<td>10</td>
<td>28</td>
</tr>
<tr>
<td>Ratio</td>
<td>1</td>
<td>0.23</td>
<td>0.59</td>
</tr>
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<table>
<thead>
<tr>
<th>Cyclic and Fatigue Properties</th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Cyclic modulus of elasticity, $E'$, GPa (ksi)</td>
<td>194.9(28,267)</td>
<td>73.3(10,636)</td>
<td>169.4(24,568)</td>
</tr>
<tr>
<td>Ratio</td>
<td>1</td>
<td>0.38</td>
<td>0.87</td>
</tr>
<tr>
<td>Fatigue strength coefficient, $\sigma_{f'}$, MPa (ksi)</td>
<td>1,156.8(167.8)</td>
<td>665.9(96.6)</td>
<td>760.8(110.3)</td>
</tr>
<tr>
<td>Ratio</td>
<td>1</td>
<td>0.58</td>
<td>0.66</td>
</tr>
<tr>
<td>Fatigue strength exponent, $b$</td>
<td>-0.082</td>
<td>-0.117</td>
<td>-0.076</td>
</tr>
<tr>
<td>Ratio</td>
<td>1</td>
<td>1.42</td>
<td>0.92</td>
</tr>
<tr>
<td>Fatigue ductility coefficient, $\varepsilon_{f'}$</td>
<td>3.032</td>
<td>0.094</td>
<td>0.864</td>
</tr>
<tr>
<td>Ratio</td>
<td>1</td>
<td>0.03</td>
<td>0.28</td>
</tr>
<tr>
<td>Fatigue ductility exponent, $c$</td>
<td>-0.791</td>
<td>-0.610</td>
<td>-0.771</td>
</tr>
<tr>
<td>Ratio</td>
<td>1</td>
<td>0.77</td>
<td>0.97</td>
</tr>
<tr>
<td>Fatigue strength, $S_i$ @ 10$^6$ cycles, MPa (ksi)</td>
<td>352(51.0)</td>
<td>122(17.6)</td>
<td>253(36.6)</td>
</tr>
<tr>
<td>Ratio</td>
<td>1</td>
<td>0.35</td>
<td>0.72</td>
</tr>
<tr>
<td>Cyclic yield strength, $S_{y'}$, MPa (ksi)</td>
<td>541.2(78.5)</td>
<td>290.7(42.2)</td>
<td>407.3(59.1)</td>
</tr>
<tr>
<td>Ratio</td>
<td>1</td>
<td>0.54</td>
<td>0.75</td>
</tr>
<tr>
<td>Cyclic strength coefficient, $K'$, MPa (ksi)</td>
<td>1,269.5(184.1)</td>
<td>430.3(62.4)</td>
<td>649.1(94.1)</td>
</tr>
<tr>
<td>Ratio</td>
<td>1</td>
<td>0.34</td>
<td>0.51</td>
</tr>
<tr>
<td>Cyclic strain hardening exponent, $n'$</td>
<td>0.137</td>
<td>0.063</td>
<td>0.075</td>
</tr>
<tr>
<td>Ratio</td>
<td>1</td>
<td>0.46</td>
<td>0.55</td>
</tr>
</tbody>
</table>
Table 3-2  Summary of monotonic test results for forged steel 11V37, cast aluminum A356-T6, and cast iron 65-45-12.

<table>
<thead>
<tr>
<th>SPECIMEN ID</th>
<th>$t_0$, mm (in)</th>
<th>$w_0$, mm (in)</th>
<th>$t_f$, mm (in)</th>
<th>$w_f$, mm (in)</th>
<th>$L_o$, mm (in)</th>
<th>$L_f$, mm (in)</th>
<th>$E$, GPa (ksi)</th>
<th>$S_y$ (offset=0.2%), MPa (ksi)</th>
<th>$S_u$, MPa (ksi)</th>
<th>$K$, MPa (ksi)</th>
<th>$n$</th>
<th>%EL</th>
<th>%RA</th>
<th>$\varepsilon_f$</th>
<th>$\sigma_f$</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Forged Steel</strong></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
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<td></td>
<td></td>
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<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>FS2A3</td>
<td>2.57 (0.101)</td>
<td>2.53 (0.100)</td>
<td>2.24 (0.088)</td>
<td>1.92 (0.076)</td>
<td>6.00 (0.24)</td>
<td>7.23 (0.28)</td>
<td>193.5 (28058)</td>
<td>566.2 (82.1)</td>
<td>830.2 (120.4)</td>
<td>1,359.7 (197.2)</td>
<td>0.157</td>
<td>20%</td>
<td>34%</td>
<td>41%</td>
<td>469</td>
</tr>
<tr>
<td>FS2A4</td>
<td>2.53 (0.100)</td>
<td>2.53 (0.100)</td>
<td>1.98 (0.078)</td>
<td>1.91 (0.075)</td>
<td>6.00 (0.24)</td>
<td>7.24 (0.29)</td>
<td>209.6 (30404)</td>
<td>546.2 (79.2)</td>
<td>812.3 (117.8)</td>
<td>1,334.8 (193.6)</td>
<td>0.158</td>
<td>21%</td>
<td>41%</td>
<td>53%</td>
<td>523</td>
</tr>
<tr>
<td><strong>Average values</strong></td>
<td>201.5 (29231)</td>
<td>556.2 (80.7)</td>
<td>821.2 (119.1)</td>
<td>1,347.3 (195.4)</td>
<td>0.157</td>
<td>21%</td>
<td>37%</td>
<td>47%</td>
<td>496</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td><strong>Cast Aluminum</strong></td>
<td></td>
<td></td>
<td></td>
<td></td>
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<td></td>
<td></td>
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<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>CA1-1-3</td>
<td>2.58 (0.102)</td>
<td>2.59 (0.102)</td>
<td>2.44 (0.096)</td>
<td>2.46 (0.097)</td>
<td>6.00 (0.24)</td>
<td>6.32 (0.25)</td>
<td>78.1 (11327)</td>
<td>232.4 (33.7)</td>
<td>302.7 (43.9)</td>
<td>417.8 (60.6)</td>
<td>0.095</td>
<td>5%</td>
<td>10%</td>
<td>10%</td>
<td>301</td>
</tr>
<tr>
<td><strong>Cast Iron</strong></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
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<td></td>
<td></td>
</tr>
<tr>
<td>CI1-12</td>
<td>2.50 (0.098)</td>
<td>2.54 (0.100)</td>
<td>2.16 (0.085)</td>
<td>2.21 (0.087)</td>
<td>6.00 (0.236)</td>
<td>6.60 (0.260)</td>
<td>193.0 (27991)</td>
<td>300.0 (43.5)</td>
<td>471.2 (68.3)</td>
<td>796.5 (115.5)</td>
<td>0.187</td>
<td>10%</td>
<td>25%</td>
<td>28%</td>
<td>219.2</td>
</tr>
</tbody>
</table>
Table 3-3  Summary of constant amplitude completely reversed fatigue test results for forged steel 11V37- Direction A.

<table>
<thead>
<tr>
<th>Specimen ID</th>
<th>1st Cycle E Value GPa (ksi)</th>
<th>Test control mode</th>
<th>Test freq. Hz</th>
<th>Δε/2%</th>
<th>Δε/2 [a] (calculated) %</th>
<th>Δε/2 [a] (measured) %</th>
<th>Δσ/2 MPa (ksi)</th>
<th>σm MPa (ksi)</th>
<th>2(N50%) reversals</th>
<th>2(N5)% reversals</th>
<th>2(N50)% reversals</th>
<th>Failure location[e]</th>
</tr>
</thead>
<tbody>
<tr>
<td>FS1A1</td>
<td>196.9 (28,550)</td>
<td>strain</td>
<td>0.10</td>
<td>0.698%</td>
<td>0.403%</td>
<td>0.377%</td>
<td>592.8 (86.0)</td>
<td>-4.1</td>
<td>2,048</td>
<td>3,798</td>
<td>4,044</td>
<td>IGL</td>
</tr>
<tr>
<td>FS2A1</td>
<td>189.9 (27,542)</td>
<td>strain</td>
<td>0.50</td>
<td>0.499%</td>
<td>0.231%</td>
<td>0.200%</td>
<td>539.5 (78.2)</td>
<td>-3.9</td>
<td>4,096</td>
<td>9,008</td>
<td>10,716</td>
<td>IGL</td>
</tr>
<tr>
<td>FS3A3</td>
<td>208.6 (30,252)</td>
<td>strain</td>
<td>0.50</td>
<td>0.499%</td>
<td>0.218%</td>
<td>0.204%</td>
<td>565.5 (82.0)</td>
<td>-3.5</td>
<td>4,096</td>
<td>7,752</td>
<td>8,354</td>
<td>IGL</td>
</tr>
<tr>
<td>FS1A5</td>
<td>189.7 (27,515)</td>
<td>strain</td>
<td>1.20</td>
<td>0.300%</td>
<td>0.067%</td>
<td>0.048%</td>
<td>468.9 (68.0)</td>
<td>45.2</td>
<td>16,384</td>
<td>41,418</td>
<td>46,392</td>
<td>IGL</td>
</tr>
<tr>
<td>FS3A6</td>
<td>193.6 (28,073)</td>
<td>strain</td>
<td>1.20</td>
<td>0.300%</td>
<td>0.070%</td>
<td>0.070%</td>
<td>463.6 (67.2)</td>
<td>4.8</td>
<td>16,384</td>
<td>33,810</td>
<td>34,308</td>
<td>IGL</td>
</tr>
<tr>
<td>FS1A3</td>
<td>195.6 (28,362)</td>
<td>load</td>
<td>2.00</td>
<td>0.200%</td>
<td>0.009%</td>
<td>0.007%</td>
<td>385.4 (55.9)</td>
<td>14.6</td>
<td>4,096</td>
<td>661,904</td>
<td>661,914</td>
<td>IGL</td>
</tr>
<tr>
<td>FS3A4</td>
<td>206.2 (29,901)</td>
<td>load</td>
<td>1.80</td>
<td>0.200%</td>
<td>0.004%</td>
<td>0.006%</td>
<td>395.5 (57.4)</td>
<td>13.0</td>
<td>4,096</td>
<td>893,294</td>
<td>893,316</td>
<td>IGL</td>
</tr>
<tr>
<td>FS2A2</td>
<td>220.5 (31,976)</td>
<td>load</td>
<td>2.00</td>
<td>0.175%</td>
<td>0.002%</td>
<td>0.000%</td>
<td>347.1 (50.3)</td>
<td>30.9</td>
<td>33,992</td>
<td>471,934</td>
<td>471,944</td>
<td>IGL</td>
</tr>
<tr>
<td>FS3A8</td>
<td>216.6 (31,417)</td>
<td>load</td>
<td>30.00</td>
<td>0.175%</td>
<td>0.000%</td>
<td>0.000%</td>
<td>352.7 (51.2)</td>
<td>0.0</td>
<td>-</td>
<td>-</td>
<td>1,922,906</td>
<td>OGIT</td>
</tr>
<tr>
<td>FS3A7</td>
<td>187.5 (27,194)</td>
<td>load</td>
<td>30.00</td>
<td>0.175%</td>
<td>0.000%</td>
<td>0.000%</td>
<td>352.7 (51.2)</td>
<td>0.0</td>
<td>-</td>
<td>-</td>
<td>&gt;10^7</td>
<td>not failed</td>
</tr>
<tr>
<td>FS3A1</td>
<td>199.7 (28,970)</td>
<td>load</td>
<td>30.00</td>
<td>0.150%</td>
<td>0.000%</td>
<td>0.000%</td>
<td>302.3 (43.8)</td>
<td>0.0</td>
<td>-</td>
<td>-</td>
<td>&gt;10^7</td>
<td>not failed</td>
</tr>
</tbody>
</table>

[a] \( \Delta \varepsilon_{p}/2 \) = \( \Delta \varepsilon/2 - \Delta \sigma/2E \).
[b] \( N_{50\%} \) is defined as the midlife cycle.
[c] \( (N_f)_{10\%} \) is defined as 10% load drop.
[d] \( (N_f)_{50\%} \) is defined as 50% load drop.
[e] IGL = inside gage length; OGIT = outside gage length but inside test section.
[f] Invalid test due to inclusion detected at failure surface. Data from this test is not included in data fittings.
[g] Data from these tests are not included in data fittings.
Table 3-4 Summary of constant amplitude completely reversed fatigue test results for forged steel 11V37- Direction B.

<table>
<thead>
<tr>
<th>Specimen ID</th>
<th>1st Cycle E Value (ksi)</th>
<th>Test control mode</th>
<th>Test freq. Hz</th>
<th>Δε/2 %</th>
<th>Δε/2 (calculated) %</th>
<th>Δε/2 (measured) %</th>
<th>Δσ/2 (ksi)</th>
<th>σm (ksi)</th>
<th>2N_{50%} reversals</th>
<th>2(N_f)_{10%} reversals</th>
<th>2(N_f)_{50%} reversals</th>
<th>Failure location</th>
</tr>
</thead>
<tbody>
<tr>
<td>FS1B1</td>
<td>193.3 (28,040)</td>
<td>strain</td>
<td>0.10</td>
<td>0.699%</td>
<td>0.410%</td>
<td>0.379%</td>
<td>580.6 (84.2)</td>
<td>1.4</td>
<td>1,024</td>
<td>2,404</td>
<td>2,702</td>
<td>IGL</td>
</tr>
<tr>
<td>FS2B2</td>
<td>196.9 (28,550)</td>
<td>strain</td>
<td>0.50</td>
<td>0.500%</td>
<td>0.222%</td>
<td>0.208%</td>
<td>559.5 (81.1)</td>
<td>7.8</td>
<td>4,096</td>
<td>10,238</td>
<td>12,476</td>
<td>IGL</td>
</tr>
<tr>
<td>FS1B3</td>
<td>189.9 (27,542)</td>
<td>strain</td>
<td>1.20</td>
<td>0.300%</td>
<td>0.063%</td>
<td>0.067%</td>
<td>477.0 (69.2)</td>
<td>12.5</td>
<td>8,192</td>
<td>12,110</td>
<td>12,484</td>
<td>IGL</td>
</tr>
<tr>
<td>FS2B4</td>
<td>208.6 (30,252)</td>
<td>strain</td>
<td>1.20</td>
<td>0.300%</td>
<td>0.061%</td>
<td>0.056%</td>
<td>481.6 (69.9)</td>
<td>11.2</td>
<td>16,384</td>
<td>32,768</td>
<td>36,326</td>
<td>IGL</td>
</tr>
<tr>
<td>FS2B6</td>
<td>189.7 (27,515)</td>
<td>load</td>
<td>1.80</td>
<td>0.199%</td>
<td>0.005%</td>
<td>0.008%</td>
<td>391.6 (56.8)</td>
<td>-1.8</td>
<td>8,192</td>
<td>1,089,354</td>
<td>1,089,370</td>
<td>IGL</td>
</tr>
<tr>
<td>FS1B4</td>
<td>193.6 (28,073)</td>
<td>strain</td>
<td>2.00</td>
<td>0.175%</td>
<td>0.007%</td>
<td>0.003%</td>
<td>337.7 (49.0)</td>
<td>-1.1</td>
<td>92,942</td>
<td>702,898</td>
<td>702,904</td>
<td>IGL</td>
</tr>
<tr>
<td>FS2B1</td>
<td>195.6 (28,362)</td>
<td>load</td>
<td>30.00</td>
<td>0.175%</td>
<td>0.000%</td>
<td>0.000%</td>
<td>352.7 (51.2)</td>
<td>0.0</td>
<td>-</td>
<td>-</td>
<td>&gt;10^7</td>
<td>not failed</td>
</tr>
<tr>
<td>FS1B2</td>
<td>206.2 (29,901)</td>
<td>load</td>
<td>30.00</td>
<td>0.150%</td>
<td>0.000%</td>
<td>0.000%</td>
<td>302.3 (43.8)</td>
<td>0.0</td>
<td>-</td>
<td>-</td>
<td>&gt;10^7</td>
<td>not failed</td>
</tr>
</tbody>
</table>

[a] Δε_p/2_{calculated} = Δε/2 - Δσ/2E.
[b] N_{50%} is defined as the midlife cycle.
[c] (N_f)_{10%} is defined as 10% load drop.
[d] (N_f)_{50%} is defined as 50% load drop.
[e] IGL = inside gage length; OGIT = outside gage length but inside test section.
[f] Data from these tests are not included in data fittings.
Table 3-5  Summary of constant amplitude completely reversed fatigue test results for forged steel 11V37- Direction C.

| Specimen ID | 1st Cycle E Value (GPa) (ksi) | Test control mode | Test freq. Hz | Δε/2 (%) | Δε/2 (calculated) (%) | Δε/2 (measured) (%) | Δσ/2 MPa (ksi) | σm MPa (ksi) | 2N50% reversals | 2(Nf)10% reversals | 2(Nf)50% reversals | Failure location(\textsuperscript{e}) |
|-------------|-------------------------------|-------------------|--------------|----------|-----------------------|-------------------|----------------|------------|----------------|----------------|----------------|----------------|----------------|
| FS2C1       | 193.3 (28,040)                | strain            | 0.10         | 0.697%   | 0.415%                | 0.377%            | 567.4 (82.3)   | 1.2 (0.2)  | 1,024          | 2,338          | 2,354          | IGL            |
| FS2C3       | 196.9 (28,550)                | strain            | 0.10         | 0.698%   | 0.411%                | 0.389%            | 579.0 (84.0)   | -2.6 (-0.4) | 1,024          | 2,080          | 2,166          | IGL            |
| FS1C5       | 189.9 (27,542)                | strain            | 0.50         | 0.499%   | 0.237%                | 0.198%            | 528.2 (76.6)   | -2.8 (-0.4) | 2,048          | 5,454          | 6,252          | IGL            |
| FS1C6       | 208.6 (30,252)                | strain            | 0.50         | 0.499%   | 0.236%                | 0.205%            | 529.2 (76.8)   | -3.5 (-0.5) | 4,096          | 7,276          | 8,080          | IGL            |
| FS2C2       | 189.7 (27,515)                | strain            | 1.20         | 0.299%   | 0.057%                | 0.066%            | 488.6 (70.9)   | -23.1 (-3.4) | 8,192          | 24,544         | 24,870         | IGL            |
| FS1C4       | 193.6 (28,073)                | strain            | 1.20         | 0.300%   | 0.070%                | 0.054%            | 462.0 (67.0)   | 7.2 (1.0)   | 16,384         | 31,560         | 35,416         | IGL            |
| FS1C3       | 195.6 (28,362)                | strain            | 2.00         | 0.199%   | 0.004%                | 0.010%            | 392.8 (57.0)   | -44.5 (-6.5) | 98,678         | 191,256        | 191,260        | IGL            |
| FS1C2       | 206.2 (29,901)                | strain            | 1.80         | 0.200%   | 0.016%                | 0.008%            | 370.1 (53.7)   | 81.3 (11.8)  | 127,166        | 171,280        | 171,290        | IGL            |
| FS1C1       | 220.5 (31,976)                | load              | 2.00         | 0.174%   | 0.011%                | 0.004%            | 329.1 (47.7)   | -3.2 (-0.5)  | 80,198         | 267,976        | 267,980        | IGL            |
| FS2C4       | 216.6 (31,417)                | load              | 30.00        | 0.175%   | 0.000%                | 0.000%            | 352.7 (51.2)   | 0.0 (0.0)   | -             | -              | 758,248        | OGIT           |

\[a\] \(\Delta \varepsilon /2\)\textsuperscript{(calculated)} = \(\Delta \varepsilon /2 - \Delta \sigma /2E.\)

[b] \(N_{50\%}\) is defined as the midlife cycle.
[c] \((Nf)_{10\%}\) is defined as 10% load drop.
[d] \((Nf)_{50\%}\) is defined as 50% load drop.
[e] IGL = inside gage length; OGIT = outside gage length but inside test section.
Table 3-6  Summary of constant amplitude completely reversed fatigue test results for cast aluminum A356-T6.

<table>
<thead>
<tr>
<th>Specimen ID</th>
<th>1st Cycle E Value (ksi)</th>
<th>Test control mode</th>
<th>Test freq. Hz</th>
<th>( \Delta \varepsilon /2 )%</th>
<th>( \Delta \varepsilon /2 ) (calculated) %</th>
<th>( \Delta \varepsilon /2 ) (measured) %</th>
<th>( \Delta \sigma /2 ) (ksi)</th>
<th>( \sigma _m ) (ksi)</th>
<th>( 2N_{50%} ) reversals</th>
<th>( 2(N_{f})_{10%} ) reversals</th>
<th>( 2(N_{f})_{50%} ) reversals</th>
<th>Failure location(^{[e]})</th>
</tr>
</thead>
<tbody>
<tr>
<td>CA1-1-4</td>
<td>76.9 (11,158)</td>
<td>strain</td>
<td>0.10</td>
<td>0.699%</td>
<td>0.322%</td>
<td>0.272%</td>
<td>294.4 (42.7)</td>
<td>-2.2</td>
<td>64</td>
<td>152</td>
<td>168</td>
<td>IGL</td>
</tr>
<tr>
<td>CA1-1-17</td>
<td>73.9 (10,715)</td>
<td>strain</td>
<td>0.10</td>
<td>0.699%</td>
<td>0.317%</td>
<td>0.273%</td>
<td>298.2 (43.3)</td>
<td>-5.3</td>
<td>256</td>
<td>464</td>
<td>488</td>
<td>IGL</td>
</tr>
<tr>
<td>CA1-1-1</td>
<td>76.1 (11,042)</td>
<td>strain</td>
<td>0.50</td>
<td>0.499%</td>
<td>0.125%</td>
<td>0.112%</td>
<td>291.3 (42.3)</td>
<td>-7.4</td>
<td>256</td>
<td>512</td>
<td>650</td>
<td>OGIT</td>
</tr>
<tr>
<td>CA1-1-12</td>
<td>70.7 (10,259)</td>
<td>strain</td>
<td>0.50</td>
<td>0.500%</td>
<td>0.131%</td>
<td>0.087%</td>
<td>287.4 (41.7)</td>
<td>-1.1</td>
<td>512</td>
<td>1,408</td>
<td>1,416</td>
<td>IGL</td>
</tr>
<tr>
<td>CA1-1-20</td>
<td>72.9 (10,574)</td>
<td>strain</td>
<td>1.10</td>
<td>0.374%</td>
<td>0.048%</td>
<td>0.022%</td>
<td>254.6 (36.9)</td>
<td>0.3</td>
<td>2,048</td>
<td>6,214</td>
<td>6,272</td>
<td>OGIT</td>
</tr>
<tr>
<td>CA1-2-9</td>
<td>75.8 (10,997)</td>
<td>strain</td>
<td>1.10</td>
<td>0.375%</td>
<td>0.035%</td>
<td>0.026%</td>
<td>265.1 (38.4)</td>
<td>8.8</td>
<td>4,096</td>
<td>7,138</td>
<td>10,584</td>
<td>IGL</td>
</tr>
<tr>
<td>CA1-1-2</td>
<td>76.7 (11,117)</td>
<td>strain</td>
<td>1.20</td>
<td>0.300%</td>
<td>0.009%</td>
<td>0.005%</td>
<td>227.4 (33.0)</td>
<td>-6.4</td>
<td>8,192</td>
<td>16,384</td>
<td>18,208</td>
<td>OGIT</td>
</tr>
<tr>
<td>CA1-1-11</td>
<td>72.5 (10,510)</td>
<td>strain</td>
<td>1.20</td>
<td>0.299%</td>
<td>0.018%</td>
<td>0.005%</td>
<td>219.6 (31.8)</td>
<td>33.9</td>
<td>8,192</td>
<td>16,166</td>
<td>19,958</td>
<td>IGL</td>
</tr>
<tr>
<td>CA1-1-22</td>
<td>74.7 (10,828)</td>
<td>load 20.00</td>
<td>1.50</td>
<td>0.225%</td>
<td>0.008%</td>
<td>0.002%</td>
<td>169.3 (24.6)</td>
<td>-1.9</td>
<td>8,192</td>
<td>463,482</td>
<td>463,486</td>
<td>IGL</td>
</tr>
<tr>
<td>CA1-1-14</td>
<td>72.8 (10,558)</td>
<td>load 20.00</td>
<td>1.80</td>
<td>0.199%</td>
<td>0.014%</td>
<td>0.002%</td>
<td>144.6 (21.0)</td>
<td>-7.1</td>
<td>16,384</td>
<td>862,032</td>
<td>862,044</td>
<td>OGIT</td>
</tr>
<tr>
<td>CA1-1-16</td>
<td>78.3 (11,348)</td>
<td>load 20.00</td>
<td>15.00</td>
<td>0.200%</td>
<td>0.000%</td>
<td>0.000%</td>
<td>156.6 (22.7)</td>
<td>0.0</td>
<td>-</td>
<td>-</td>
<td>308,330</td>
<td>IGL</td>
</tr>
<tr>
<td>CA1-1-7</td>
<td>77.0 (11,166)</td>
<td>load 20.00</td>
<td>20.00</td>
<td>0.149%</td>
<td>0.002%</td>
<td>0.000%</td>
<td>115.0 (16.7)</td>
<td>-0.1</td>
<td>-</td>
<td>-</td>
<td>846,826</td>
<td>OGIT</td>
</tr>
<tr>
<td>CA1-2-5</td>
<td>77.0 (11,166)</td>
<td>load 20.00</td>
<td>20.00</td>
<td>0.150%</td>
<td>0.002%</td>
<td>0.000%</td>
<td>115.5 (16.7)</td>
<td>-9.5</td>
<td>-</td>
<td>-</td>
<td>1,993,158</td>
<td>IGL</td>
</tr>
<tr>
<td>CA1-1-15</td>
<td>77.0 (11,166)</td>
<td>load 30.00</td>
<td>30.00</td>
<td>0.125%</td>
<td>0.002%</td>
<td>0.000%</td>
<td>96.3 (14.0)</td>
<td>0.0</td>
<td>-</td>
<td>-</td>
<td>&gt;10^7</td>
<td>not failed</td>
</tr>
</tbody>
</table>

[a] \( \Delta \varepsilon /2 \) (calculated) = \( \Delta \varepsilon /2 - \Delta \sigma /2 \varepsilon \).
[b] \( N_{50\%} \) is defined as the midlife cycle.
[c] \( (N_{f})_{10\%} \) is defined as 10% load drop.
[d] \( (N_{f})_{50\%} \) is defined as 50% load drop.
[e] IGL = inside gage length; OGIT = outside gage length but inside test section.
[f] Data from these tests are not used in data fittings.
Table 3-7 Summary of constant amplitude completely reversed fatigue test results for cast iron 65-45-12.

<table>
<thead>
<tr>
<th>Specimen ID</th>
<th>1st Cycle Value E Value (ksi)</th>
<th>Test control mode</th>
<th>Test freq. Hz</th>
<th>Δε/2 %</th>
<th>Δεp/2 % (calculated)</th>
<th>Δεp/2 % (measured)</th>
<th>Δσ/2 MPa (ksi)</th>
<th>σm MPa (ksi)</th>
<th>2N50% (reversals)</th>
<th>2(Nf)10% (reversals)</th>
<th>2(Nf)50% (reversals)</th>
<th>Failure location</th>
<th>2N50% (reversals)</th>
</tr>
</thead>
<tbody>
<tr>
<td>CI1-1</td>
<td>176.0 (25,524)</td>
<td>strain</td>
<td>0.50</td>
<td>0.500%</td>
<td>0.269%</td>
<td>0.247%</td>
<td>412.5 (59.8)</td>
<td>-10.9</td>
<td>512</td>
<td>1,032</td>
<td>1,122</td>
<td>IGL</td>
<td></td>
</tr>
<tr>
<td>CI1-8</td>
<td>179.5 (26,029)</td>
<td>strain</td>
<td>0.50</td>
<td>0.499%</td>
<td>0.267%</td>
<td>0.245%</td>
<td>413.4 (60.0)</td>
<td>-7.6</td>
<td>1,024</td>
<td>2,100</td>
<td>2,122</td>
<td>IGL</td>
<td></td>
</tr>
<tr>
<td>CI1-3</td>
<td>179.4 (26,018)</td>
<td>strain</td>
<td>1.20</td>
<td>0.299%</td>
<td>0.080%</td>
<td>0.068%</td>
<td>392.2 (56.9)</td>
<td>-20.0</td>
<td>8,192</td>
<td>12,348</td>
<td>13,816</td>
<td>IGL</td>
<td></td>
</tr>
<tr>
<td>CI1-9</td>
<td>166.4 (24,140)</td>
<td>strain</td>
<td>1.20</td>
<td>0.299%</td>
<td>0.086%</td>
<td>0.065%</td>
<td>381.5 (55.3)</td>
<td>-5.7</td>
<td>8,192</td>
<td>13,680</td>
<td>13,788</td>
<td>IGL</td>
<td></td>
</tr>
<tr>
<td>CI1-5</td>
<td>185.3 (26,880)</td>
<td>strain</td>
<td>1.80</td>
<td>0.197%</td>
<td>0.012%</td>
<td>0.016%</td>
<td>330.9 (48.0)</td>
<td>-40.3</td>
<td>8,192</td>
<td>87,888</td>
<td>87,894</td>
<td>IGL</td>
<td></td>
</tr>
<tr>
<td>CI1-10</td>
<td>184.2 (26,710)</td>
<td>strain</td>
<td>1.80</td>
<td>0.199%</td>
<td>0.015%</td>
<td>0.019%</td>
<td>328.9 (47.7)</td>
<td>19.5</td>
<td>8,192</td>
<td>93,352</td>
<td>93,366</td>
<td>IGL</td>
<td></td>
</tr>
<tr>
<td>CI1-6</td>
<td>180.6 (26,192)</td>
<td>load</td>
<td>30.00</td>
<td>0.150%</td>
<td>0.000%</td>
<td>0.000%</td>
<td>270.9 (39.3)</td>
<td>0.0</td>
<td>-</td>
<td>-</td>
<td>611,254</td>
<td>IGL</td>
<td></td>
</tr>
<tr>
<td>CI1-11</td>
<td>180.6 (26,192)</td>
<td>load</td>
<td>10.00</td>
<td>0.150%</td>
<td>0.000%</td>
<td>0.000%</td>
<td>270.9 (39.3)</td>
<td>0.0</td>
<td>-</td>
<td>-</td>
<td>794,946</td>
<td>IGL</td>
<td></td>
</tr>
<tr>
<td>CI1-13</td>
<td>180.6 (26,192)</td>
<td>load</td>
<td>15.00</td>
<td>0.130%</td>
<td>0.000%</td>
<td>0.000%</td>
<td>234.8 (34.0)</td>
<td>0.0</td>
<td>-</td>
<td>-</td>
<td>5,674,804</td>
<td>IGL</td>
<td></td>
</tr>
<tr>
<td>CI1-14</td>
<td>180.6 (26,192)</td>
<td>load</td>
<td>30.00</td>
<td>0.130%</td>
<td>0.000%</td>
<td>0.000%</td>
<td>234.8 (34.0)</td>
<td>0.0</td>
<td>-</td>
<td>-</td>
<td>4,491,966</td>
<td>IGL</td>
<td></td>
</tr>
</tbody>
</table>

[a] \( \Delta_{ep}/2^{(calculated)} = \Delta_{ep}/2 - \Delta_{e}/2E \).
[b] \( N_{50\%} \) is defined as the midlife cycle.
[c] \( (Nf)_{10\%} \) is defined as 10% load drop.
[d] \( (Nf)_{50\%} \) is defined as 50% load drop.
[e] IGL = inside gage length; OIT = outside gage length but inside test section.
[f] For plastic strain values, the calculated value is the average of E values from the first cycle of strain-controlled fatigue tests. (Range: 24140-26880 ksi, average value: 25884 ksi).
Table 3-8  Summary of fatigue constants for the three geometrical directions A, B and C of forged steel 11V37.

<table>
<thead>
<tr>
<th>Fatigue Properties</th>
<th>Direction A</th>
<th>Direction B</th>
<th>Direction C</th>
</tr>
</thead>
<tbody>
<tr>
<td>Fatigue strength coefficient, $\sigma'_f$, MPa (ksi)</td>
<td>1,156.8 (167.8)</td>
<td>1265.2 (183.5)</td>
<td>1316.3 (190.9)</td>
</tr>
<tr>
<td>Fatigue strength exponent, $b$</td>
<td>-0.082</td>
<td>-0.093</td>
<td>-0.103</td>
</tr>
<tr>
<td>Fatigue ductility coefficient, $\epsilon'_f$</td>
<td>3.032</td>
<td>26.893</td>
<td>1.814</td>
</tr>
<tr>
<td>Fatigue ductility exponent, $c$</td>
<td>-0.791</td>
<td>-1.062</td>
<td>-0.770</td>
</tr>
<tr>
<td>Fatigue strength, $S_f @ 10^6$ cycles, MPa (ksi)</td>
<td>352 (51.0)</td>
<td>326 (47.2)</td>
<td>295 (42.7)</td>
</tr>
<tr>
<td>Cyclic strength coefficient, $K'$, MPa (ksi)</td>
<td>1,269.5 (184.1)</td>
<td>1049.1 (152.2)</td>
<td>948.6 (137.6)</td>
</tr>
<tr>
<td>Cyclic strain hardening exponent, $n'$</td>
<td>0.137</td>
<td>0.106</td>
<td>0.094</td>
</tr>
</tbody>
</table>
Figure 3-1  Specimen configuration and dimensions (all dimensions in mm).

Figure 3-2  From left to right: forged steel, cast aluminum, and cast iron steering knuckles. The three perpendicular geometrical directions of the forged steel steering knuckle are; one along spindle, one along the arm perpendicular to the body, and one along the body.
Figure 3-3 True stress versus true plastic strain for (a) forged steel 11V37, (b) cast aluminum A356-T6, and (c) cast iron 65-45-12.
Figure 3-4 Monotonic stress-strain curves for (a) forged steel 11V37, (b) cast aluminum A356-T6, and (c) cast iron 65-45-12.
Figure 3-5  Superimposed monotonic stress-strain curves for the three materials; forged steel 11V37, cast aluminum A356-T6, and cast iron 65-45-12.
Figure 3-6  True stress amplitude versus normalized number of cycles for (a) forged steel 11V37, (b) cast aluminum A356-T6, and (c) cast iron 65-45-12.
Figure 3-7  True stress amplitude versus number of cycles for (a) forged steel 11V37, (b) cast aluminum A356-T6, and (c) cast iron 65-45-12.
Figure 3-8  True stress amplitude versus calculated true plastic strain amplitude for (a) forged steel 11V37 in three directions, (b) cast aluminum A356-T6, and (c) cast iron 65-45-12.
Figure 3-9  True stress amplitude versus true strain amplitude for (a) forged steel 11V37 in three directions, (b) cast aluminum A356-T6, and (c) cast iron 65-45-12.
Figure 3-10  Superimposed true stress amplitude versus true strain amplitude curves for the three materials; forged steel 11V37 in direction A, cast aluminum A356-T6, and cast iron 65-45-12.
Figure 3-11  Superimposed plots of cyclic and monotonic stress-strain curves for (a) forged steel 11V37 in direction A, (b) cast aluminum A356-T6, and (c) cast iron 65-45-12.
Figure 3-12  Superimposed plot of cyclic and monotonic stress-strain curves for the three materials; forged steel 11V37 in direction A, cast aluminum A356-T6, and cast iron 65-45-12.
Figure 3-13 Composite plot of midlife hysteresis loops for (a) forged steel 11V37 in direction A, (b) cast aluminum A356-T6, and (c) cast iron 65-45-12.
Figure 3-14 (a) True stress amplitude versus reversals to failure, (b) calculated true plastic strain amplitude versus reversals to failure, and (c) true strain amplitude versus reversals to failure for forged steel 11V37 in three directions.
Figure 3-15  True stress amplitude versus reversals to failure for (a) forged steel 11V37 in direction A, (b) cast aluminum A356-T6, and (c) cast iron 65-45-12.
Figure 3-16 Superimposed true stress amplitude versus reversals to failure for the three materials; forged steel 11V37 in direction A, cast aluminum A356-T6, and cast iron 65-45-12.
Figure 3-17  Calculated true plastic strain amplitude versus reversals to failure for (a) forged steel 11V37 in direction A, (b) cast aluminum A356-T6, and (c) cast iron 65-45-12.
Figure 3-18  Superimposed calculated true plastic strain amplitude versus reversals to failure for the three materials; forged steel 11V37 in direction A, cast aluminum A356-T6, and cast iron 65-45-12.
Figure 3-19  True strain amplitude versus reversals to failure for (a) forged steel 11V37 in direction A, (b) cast aluminum A356-T6, and (c) cast iron 65-45-12.
Figure 3-20  Superimposed true strain amplitude versus number of reversals to failure for the three materials; forged steel 11V37 in direction A, cast aluminum A356-T6, and cast iron 65-45-12.
Figure 3-21  Neuber fatigue life curves for forged steel 11V37, cast aluminum A356-T6, and cast iron 65-45-12.
Chapter Four

4 Finite Element Analysis

The objective of the stress analysis was to obtain the complete three dimensional stress and strain distributions at a potential failure site, facilitating fatigue life predictions. Depending on the method of fatigue life prediction, stress analysis can be linear or nonlinear. Linear elastic analysis is the most common type of stress analysis pursued in automotive design and analysis. The majority of automobile engine components like camshaft, crankshaft and connecting rod operate within the material elastic region, and therefore linear elastic analysis is feasible to model their behavior. On the other hand, most components in chassis and suspension undergo occasional overloads in service and cyclic plasticity becomes a major factor to define their stress-strain response, particularly at stress concentrations. In these cases linear analysis is not sufficient to analyze component's behavior and nonlinear elastic-plastic analysis should be performed.

Although in this study the components underwent unidirectional loading condition, the state of stress at many locations was found to be multiaxial. In addition, gross yielding existed for a number of experimental applied moment cases. Therefore, both linear finite element analysis (FEA) to be used along with a stress/strain correction method, and nonlinear FEA to directly estimate elastic-plastic distributions of stress and strain were performed. IDEAS FEA program and its help library bookshelf were employed for the analysis. Details of the FEA are discussed in this chapter.
4.1 Component Geometry Generation

The geometries of components were generated as solid parts using a coordinate measuring machine and the modeler tools of the software. The resulting solid models are presented in Figure 4-1. The weights of the generated models were 2.35 kg, 2.38 kg and 4.62 kg versus the actual components weight of 2.4 kg, 2.4 kg and 4.7 kg for the forged steel, cast aluminum and cast iron steering knuckles, respectively. These show differences of 2%, 1 % and 2 %, respectively, and verify the accuracy of the geometries generated for analysis. The fillets and chamfers of the components were removed (except at the critical locations) in the models used for the analysis in order to reduce the complexity of the models and the runtime.

4.2 Model Boundary Conditions

The boundary conditions and loading were selected to represent the assumed component testing conditions as shown in Figure 4-2. To verify the model with the specified boundary conditions other alternatives were analyzed by switching the loading and boundary conditions and also by releasing any one of the fixed points to ensure the critical locations remained the same.

For the cast aluminum steering knuckle in service while the loading is applied to the strut joints through struts, the four hub bolt holes are connected to the wheel assembly. Several trials for boundary conditions were analyzed including fixing the whole area of the four hub bolt holes, fixing the centerline of the hub bolt holes, only fixing the pair of bolt holes away from the load application point, and fixing two points at the middle area of the hub. It was found that except for the case of fixing the whole area of the bolt holes, all the other three cases provided approximately similar results at the critical location. This critical
location was the node at the area that the crack initiated during component testing (Section 5.5). Moreover, the strain readings from strain gages were reasonably close to the strain values for these three options as discussed in Section 5.3. The option to fix the whole area of the bolt hole resulted in lower value of stress at the node of this critical location and the highest stresses occurred at a different location. In addition, for this option the strain value at the location where strain gage was installed was different from the strain gage reading. Based on these two observations, the choice of fixing the hub bolt hole-centerlines was selected as the base boundary condition case for the cast aluminum steering knuckle. For the cast iron steering knuckle, where the geometry and service conditions were close to the cast aluminum steering knuckle, similar loading and boundary conditions were applied.

4.3 Meshing of the Finite Element Models

To generate the mesh for the components, free meshing feature of the software was employed since it has no geometry restrictions and it could be defined on complicated volumes. The free mesh generator uses an algorithm that minimizes element distortion. Three-dimensional linear tetrahedron solid elements with three translational degrees of freedom were used. Although parabolic element offers more accurate results, the analysis time, especially for nonlinear analysis, is immensely more than that of linear elements. Nonlinear analysis at the highest moment level with parabolic elements showed negligible difference in results with those using linear tetrahedral element and, therefore, the linear element was selected for modeling. For the complex three-dimensional geometry of the steering knuckles, solid element offers more accurate results and IDEAS-10 software employs tetrahedron solid elements. In spite of a global mesh size for each component, free
local meshing feature was used to increase the number of elements at the vicinity of the critical points.

Convergence of stress and strain energy was considered as the criteria to select the mesh size. Too much refinement at the critical points would result in extremely lengthy analysis time and was, therefore, avoided. Figure 4-3 shows the variation of stress at the critical points of each steering knuckle. Global mesh size of 5.1 mm for the forged steel and cast aluminum steering knuckles and 3.8 mm for the cast iron steering knuckle and local mesh size of 0.1 mm for the forged steel and 0.64 mm for the cast aluminum and cast iron steering knuckles were specified. The critically-stressed locations were spindle fillets for the forged steel, and hub bolt-holes for the cast aluminum and cast iron steering knuckles. Due to higher stress concentration at the spindle fillet of the forged steel steering knuckle, large stress gradient existed at this location, and therefore, more refined mesh size was implemented. As another verification of convergence of the mesh, the averaged and unaveraged stress values at the critical nodes were confirmed to be approximately equal. Figure 4-4 shows the three meshed components with the darker areas representing those with higher mesh density.

Adaptive meshing module as a tool to improve a mesh by moving nodes, splitting elements, or remeshing the model, was examined. The software derives the new mesh by analyzing the data variation along the boundaries and within the interior regions of the faces. Adaptive meshing could be used to reduce elemental distortion or refine a mesh in areas where error estimates are highest. Energy error norm is used as basis for the adaptive meshing method. The energy error norm (EEN) is based on the strain energy error. It is computed from the elemental strain energy errors and the model’s total strain energy from Zienkiewicz-Zhu parameter (IDEAS-10 Help Library Bookshelf, 2002):
\[
\text{Energy error norm} = \sqrt{\frac{\sum \text{elemental strain energy errors}}{\text{total strain energy}}} \times 100 \quad (4-1)
\]

where the summation is over the entire volume of the component.

The Convergence is determined by comparing EEN in the model to the EEN specified. For the forged steel steering knuckle with an initial mesh size of 5.08 mm, EEN was calculated to be 10.7%. Remeshing option of adaptive meshing that increases the number of elements at the areas with high strain energy error was applied and the first adaptive trial resulted in EEN equal to 7.9%. No more adaptive meshing was possible due to high distortion of the newly generated elements. This is attributed to the geometric complexity of the component. The component’s appropriate loading and boundary conditions were applied to the refined model and the results were compared to the results from the converged mesh using free local meshing, where they were found not to be equal. In addition, the averaged and unaveraged stress and strain results for the adaptive-meshed model were compared and they were not equal either. These show that the mesh did not converge. As a result, the final meshed models used for solution were those with free local meshing and the adaptive meshing tool was found not to be suitable for the components.

### 4.4 Material Nonlinear Behavior Models

To define the limit of elastic behavior in nonlinear analysis, von Mises yield criterion was assumed, using an associated flow rule in which the plastic potential function is the same as the yield function and the components of the plastic strain increment are given by a Prandtl-Reuss type equation. Ziegler-Prager kinematic hardening rule was selected, for which a bilinear stress-strain curve of the material is assumed. The appropriate portion of the material’s cyclic stress-strain curve was used to define material properties for each cyclic
moment level; i.e. for each applied cyclic moment level, a slope for the second line of the bilinear model was assumed and the stress and strain results were obtained from FEA. This slope was found by trail and error in an iterative process until the resultant stress-strain point matched the experimental material cyclic stress-strain curve. Figure 4-5 shows the material model for the two higher moment levels.

Figure 4-6 illustrates the actual cyclic stress-strain curves at different cyclic moment levels including the values of stress and strain at the critical points of failure at each level obtained from FEA also superimposed. It was observed that the assumed model provides stress/strain values in the component with a reasonable error of less than 10% with respect to the actual material stress-strain curve. Note that even at the lower moment level, which can be considered as an indication of long-life service of the components, the material undergoes local plastic deformation. This is evidence that mere use of linear elastic FEA is not sufficient for reliable fatigue life predictions.

4.5 Model Solutions

The model solution equation solver for linear or nonlinear analysis creates a set of simultaneous linear equations that must be solved. For instance, the number of equations created for the highest load level of the forged steel steering knuckle model with 4245 nodes and 19098 elements was 12570 equations. The direct method is represented by Gaussian elimination along with Choleski decomposition or factorization method. With Choleski decomposition, the matrix of equations is factorized into a lower triangular, diagonal, and upper triangular matrix; the upper triangular matrix is the transpose of the lower triangular matrix. After factorization, the equations are solved for the unknown values by performing a
forward and backward substitution upon the load vector. Following decomposition of the stiffness matrix, the displacements are calculated and strain and stresses are obtained.

A nonlinear solution proceeds incrementally with equilibrium iterations performed at every solution point. The stiffness matrix in the nonlinear solution doesn’t participate in the equilibrium statement as is the case in linear analysis. The stiffness relates incremental displacements to incremental forces. The software computes tangent stiffness based on current geometry, stress, and plastic strain. Then it computes applied loads based on current configuration, and computes internal force based on current displacements and plastic strains. In the next step, it computes force residual, computes displacement increment, updates displacements, computes plastic strain increments at element integration points, and checks for convergence. Energy convergence ratio is the convergence criterion that uses both displacements and forces. The amount of work done by the residual forces in the $i$th iteration is compared with the first iteration (IDEAS-10 Help Library Bookshelf, 2002). As an example, the nonlinear solution for the highest load level of the forged steel steering knuckle took four iterations to converge.

4.6 **Analysis of the Results**

The finite element models as shown in Figure 4-2 were solved for loading and unloading under the assumed component testing conditions with the assumed moment levels as listed in Table 5-2. The procedure used to obtain maximum and minimum stresses and strains in a loading cycle is as follows; first, loading was applied to the FEA model at the maximum moment assumed to be used in the test and von Mises stress at the critical location was obtained ($\sigma_{VM,max}$). Then the moment range assumed to be used in the experiment was applied to the FEA model and von Mises stress range at the same critical
location was obtained ($\Delta \sigma_{VM}$). It should be noted that when using the maximum moment, the stress-strain relation used was the material cyclic curve based on amplitudes, whereas when using the moment range, the cyclic stress-strain curve was based on ranges (i.e. equation of the hysteresis loop). Having $\sigma_{VM,max}$ and $\Delta \sigma_{VM}$, the range and mean values of stresses and strains were then calculated.

Figure 4-7 shows contours of von Mises stress for the maximum moment level of 1515 N.m for forged steel and 2230 N.m for cast aluminum and cast iron steering knuckles. For the forged steel and cast aluminum steering knuckles at the higher moment levels, yielding occurred both gross (at the spindle and the hub, respectively) and locally (at the fillet and hub bolt-holes, respectively), whereas for the cast iron steering knuckle only local yielding occurred at the critical points (hub bolt-hole). The contours of stress and strain at the lower moment levels were different, showing only local plastic deformation, as shown in Figure 4-8. In this case, the spindle and the hub area for the forged steel and cast aluminum steering knuckles undergo elastic deformation.

Table 4-1 lists the $x$, $y$ and $z$ components, principal and equivalent von Mises values of stress and strain for the three components at the critical locations for the highest moment level applied in the analysis. The $z$ direction aligns with the spindle axis for the forged steel steering knuckle, and the $y$ and $x$ directions align with the strut arm for the cast aluminum and cast iron steering knuckles, respectively. The spindle 1st step fillet area for the forged steel and hub bolt hole for the cast aluminum and cast iron steering knuckles were found to be the high-stressed locations with high stress gradient. The effect of stress concentration can be seen in Figure 4-9 that shows the distribution of local stress at the spindle 2nd step fillet and nominal stress at a cross section remote from the fillet for the maximum moment level of 1515 N.m. For spindle radius smaller than 10 mm the material behavior is elastic,
while for larger radii it becomes inelastic. Figure 4-10 shows the variation of stress at different locations of the forged steel steering knuckle with respect to moment steps to provide a comparison between the level of stress at the critical location and other locations. The stress concentration produced by a notch depends on the mode of loading applied to the component. Not only the stress concentration at the fillet of the forged steel steering knuckle increases the stress level, but also the stress gradient is higher than, for instance, tension loading due to the loading mode (bending).

Examining the data in Table 4-1 clearly demonstrates the multiaxial nature of stress and strain at the critical locations, although the primary loading on the components are unidirectional. The ratio of $\sigma_y/\sigma_z$ in the case of the forged steel steering knuckle is 0.36, $\sigma_x/\sigma_y$ for the cast aluminum steering knuckle is 0.69, and $\tau_{yz}/\tau_{xy}$ is -0.36 for the case of the cast iron steering knuckle. Therefore, equivalent values of stress and strain should be calculated to account for multiaxiality. Since the stresses and strains obtained form these analyses were used to predict fatigue life, the proportionality of stresses is also an important issue for selection of an appropriate fatigue life prediction model. Only one source of load exists for the primary loading that these components undergo, therefore the stresses are proportional throughout the components, i.e. the stresses increase and decrease in-phase as the primary loads increase and decrease, respectively. For proportional stressing, von Mises stress and strain have been found effective in calculating the equivalent values as a result of multiaxiality (Stephens et al., 2000), and were used for subsequent fatigue life analyses.

Another observation that could be made from these data is the stress state at the critical locations. The values of $\sigma_2/\sigma_1$ for the critical locations are 0.36, 0.12 and 0.06 while the values of $\varepsilon_2/\varepsilon_1$ are -0.09, -0.35 and -0.13 for the forged steel, cast aluminum and cast iron steering knuckles. This indicates that at the critical location, the state of nearly plane strain
prevails for the forged steel steering knuckle, while the state of stress at the critical locations of the cast aluminum and cast iron steering knuckles is closer to plane stress. Slightly different results were obtained for lower moment levels due to limited plastic deformation, but the same conclusion about the state of stress could be drawn for them too.

An important point for the fatigue analysis to be mentioned here is the effect of the notch on local stresses and strains due to cycling. Cyclic loading generates residual stress at the notch and therefore, makes the local deformation behavior different from the nominal behavior. Table 4-2 lists the nominal moment ratios and local stress ratios as results of loading and unloading simulations of the components. As the nominal R-ratio remains almost constant (close to zero), significant negative R-ratio is observed for most of the simulations as a result of the residual stress generated at the stress concentrations (i.e. critical locations). This phenomenon is more pronounced at the higher moment levels, as expected, because of increased plastic deformation and nonlinear material behavior.
Table 4-1 Components of stress and strain for the critical locations of the forged steel ($M = 1515$ N.m), cast aluminum ($M = 2230$ N.m) and cast iron ($M = 2230$ N.m) steering knuckles.

<table>
<thead>
<tr>
<th></th>
<th>Forged Steel (2nd step fillet)</th>
<th>Cast Aluminum (hub bolt hole)</th>
<th>Cast Iron (hub bolt hole)</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Stress</strong></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>$\sigma_x$ (MPa)</td>
<td>44</td>
<td>145</td>
<td>-8</td>
</tr>
<tr>
<td>$\sigma_y$ (MPa)</td>
<td>258</td>
<td>210</td>
<td>200</td>
</tr>
<tr>
<td>$\sigma_z$ (MPa)</td>
<td>710</td>
<td>40</td>
<td>-6</td>
</tr>
<tr>
<td>$\tau_{xy}$ (MPa)</td>
<td>3</td>
<td>146</td>
<td>207</td>
</tr>
<tr>
<td>$\tau_{xz}$ (MPa)</td>
<td>43</td>
<td>-16</td>
<td>36</td>
</tr>
<tr>
<td>$\tau_{yz}$ (MPa)</td>
<td>-3</td>
<td>-32</td>
<td>-74</td>
</tr>
<tr>
<td>$\sigma_1$ (MPa)</td>
<td>713</td>
<td>332</td>
<td>334</td>
</tr>
<tr>
<td>$\sigma_2$ (MPa)</td>
<td>259</td>
<td>41</td>
<td>19</td>
</tr>
<tr>
<td>$\sigma_3$ (MPa)</td>
<td>42</td>
<td>23</td>
<td>-167</td>
</tr>
<tr>
<td>$\tau_{max}$ (MPa)</td>
<td>336</td>
<td>154</td>
<td>250</td>
</tr>
<tr>
<td>$\sigma_{VM}$ (MPa)</td>
<td>603</td>
<td>296</td>
<td>431</td>
</tr>
<tr>
<td><strong>Strain</strong></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>$\varepsilon_x$</td>
<td>-0.00430</td>
<td>0.00110</td>
<td>-0.00160</td>
</tr>
<tr>
<td>$\varepsilon_y$</td>
<td>-0.00060</td>
<td>0.00340</td>
<td>0.00360</td>
</tr>
<tr>
<td>$\varepsilon_z$</td>
<td>0.00700</td>
<td>-0.00270</td>
<td>-0.00160</td>
</tr>
<tr>
<td>$\varepsilon_{xy}$</td>
<td>0.00010</td>
<td>0.01000</td>
<td>0.01100</td>
</tr>
<tr>
<td>$\varepsilon_{xz}$</td>
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<td>0.00180</td>
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<td>$\varepsilon_{yz}$</td>
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<td>-0.00230</td>
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<tr>
<td>$\varepsilon_1$</td>
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<td>0.00770</td>
<td>0.00710</td>
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<td>$\varepsilon_2$</td>
<td>-0.00064</td>
<td>-0.00270</td>
<td>-0.00096</td>
</tr>
<tr>
<td>$\varepsilon_3$</td>
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<td>-0.00330</td>
<td>-0.00570</td>
</tr>
<tr>
<td>$\gamma_{max}$</td>
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<td>0.01100</td>
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</tr>
<tr>
<td>$\varepsilon_{VM}$</td>
<td>0.00670</td>
<td>0.00710</td>
<td>0.00740</td>
</tr>
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</table>
Table 4-2  Nominal moment ratios \((R_m)\) and local stress ratios \((R_\sigma)\) indicating variation of \(R_\sigma\) resulting from different levels of notch residual stress generated during cycling as a function of \(M_{max}\).

<table>
<thead>
<tr>
<th></th>
<th>(M_{max}) (N.m)</th>
<th>(M_{min}) (N.m)</th>
<th>Nominal Moment Ratio</th>
<th>(\sigma_{VM, max}) (MPa)</th>
<th>(\sigma_{VM, min}) (MPa)</th>
<th>Local Stress Ratio</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Forged Steel</strong></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>1515</td>
<td>75</td>
<td>0.05</td>
<td>603</td>
<td>-264</td>
<td>-0.44</td>
</tr>
<tr>
<td></td>
<td>1240</td>
<td>75</td>
<td>0.06</td>
<td>565</td>
<td>-214</td>
<td>-0.38</td>
</tr>
<tr>
<td></td>
<td>965</td>
<td>75</td>
<td>0.08</td>
<td>514</td>
<td>-123</td>
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</tr>
<tr>
<td></td>
<td>825</td>
<td>75</td>
<td>0.09</td>
<td>478</td>
<td>-61</td>
<td>-0.13</td>
</tr>
<tr>
<td><strong>Cast Aluminum</strong></td>
<td></td>
<td></td>
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<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>2230</td>
<td>75</td>
<td>0.03</td>
<td>296</td>
<td>-134</td>
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<td></td>
<td>1595</td>
<td>75</td>
<td>0.05</td>
<td>279</td>
<td>-24</td>
<td>-0.09</td>
</tr>
<tr>
<td></td>
<td>1305</td>
<td>75</td>
<td>0.06</td>
<td>251</td>
<td>5</td>
<td>0.02</td>
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<tr>
<td></td>
<td>1195</td>
<td>75</td>
<td>0.06</td>
<td>237</td>
<td>14</td>
<td>0.06</td>
</tr>
<tr>
<td><strong>Cast Iron</strong></td>
<td></td>
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<td></td>
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<tr>
<td></td>
<td>2230</td>
<td>75</td>
<td>0.03</td>
<td>431</td>
<td>-276</td>
<td>-0.64</td>
</tr>
<tr>
<td></td>
<td>1595</td>
<td>75</td>
<td>0.05</td>
<td>412</td>
<td>-229</td>
<td>-0.56</td>
</tr>
<tr>
<td></td>
<td>1305</td>
<td>75</td>
<td>0.06</td>
<td>405</td>
<td>-114</td>
<td>-0.28</td>
</tr>
<tr>
<td></td>
<td>1195</td>
<td>75</td>
<td>0.06</td>
<td>388</td>
<td>-85</td>
<td>-0.22</td>
</tr>
</tbody>
</table>
Figure 4-1  From left to right the solid models of the forged steel, cast aluminum and cast iron steering knuckles.

Figure 4-2  From top to bottom boundary conditions for the forged steel, cast aluminum and cast iron steering knuckles.
Figure 4.3 von Mises stress versus mesh size at critical locations of (a) forged steel, (b) cast aluminum and (c) cast iron steering knuckles to check mesh convergence.
Figure 4-4  From top to bottom, the generated mesh for the forged steel, cast aluminum and cast iron steering knuckles.
Figure 4-5 Superimposed cyclic stress-strain curves and the bilinear models used in the nonlinear FEA for the forged steel, cast iron and cast aluminum steering knuckle materials. The data points are from the material cyclic tests.
Figure 4-6  Actual material cyclic stress-strain behavior and the corresponding local von Mises stress and strain from FEA (shown with plus signs) for (a) forged steel, (b) cast aluminum and (c) cast iron steering knuckles.
Figure 4-7  From top to bottom, contours of von Mises stress at highest moment levels for forged steel ($M_{max} = 1515$ N.m), cast aluminum ($M_{max} = 2230$ N.m) and cast iron ($M_{max} = 2230$ N.m) steering knuckles. The stress values of the color bar are in MPa.
From top to bottom, contours of von Mises stress at lowest moment levels for forged steel ($M_{\text{max}} = 825 \text{ N.m}$), cast aluminum ($M_{\text{max}} = 1195 \text{ N.m}$) and cast iron ($M_{\text{max}} = 1195 \text{ N.m}$) steering knuckles. The stress values of the color bar are in MPa.
Figure 4-9 The distribution of stress at the spindle 2nd step fillet and at a section remote from the 2nd step fillet of the forged steel steering knuckle for a moment of 1515 N.m. The nominal stress profile was used to obtain the nominal values of stress.
Figure 4-10  Signed von Mises† stress history at different locations of the forged steel steering knuckle.

† Signed von Mises uses the sign of the absolute maximum principal stress to give either a positive or negative von Mises stress (This procedure is used in FEA/fatigue software codes such as nCode™ and MSC.Fatigue™).
Chapter Five

5 Component Fatigue Behavior and Comparisons

In fatigue design problems, material properties, loading, geometry, manufacturing aspects, and service environment are considered as five major aspects to be dealt with, and the effect of geometry has always been one of the challenging issues. The methodologies to transfer material properties obtained from specimen fatigue tests to fatigue behavior of real components where neither a nominal stress nor a notch factor could be defined, have been questionable. In addition, simulating manufacturing parameters like surface finish and residual stress, if not impossible, is complicated. Therefore direct component testing, though time consuming and expensive, has become a necessity in fatigue design. Contrary to specimen testing, for which numerous standards have been developed, component testing is more a matter of designer’s practice. Various parameters such as simulation of the real-life condition, collecting useful data, and relating test results to analytical predictions determine the applicability of the conducted tests.

To be able to compare the fatigue behavior of the steering knuckles, and to make it possible to verify analytical life predictions, constant-amplitude load-control fatigue tests were performed for forged steel and cast aluminum steering knuckles. The component testing was only conducted for forged steel and cast aluminum steering knuckles. In this chapter, the procedure to conduct the component tests is detailed, including considerations prior to and during the tests, and test results are presented and discussed.
5.1 Identifying the Components and their Service Loadings

The suspension systems of the vehicle, which the steering knuckles belong to, were identified and the loading and attachment conditions were investigated. The forged steel steering knuckle belongs to the rear suspension system of a four-cylinder sedan vehicle. It is a symmetric component with one plane of symmetry. The cast aluminum steering knuckle belongs to the front suspension of a six-cylinder minivan. The nomenclature for these components is shown in Figure 5-1.

Figure 5-2 and Figure 5-3 show the forged steel and cast aluminum steering knuckles as installed in the suspension system of the vehicles, respectively. For the case of the forged steel steering knuckle (Figure 5-2), the strut mounting holes are connected to the strut joints, the front and rear lateral links connect to the chassis and the tension strut joint is attached to the tension strut that is fixed to the chassis bracket. The hub and bearing assembly mount on the spindle. The hub bearing sits on the spindle middle step. The inner part of the hub attaches to the mounting holes, while the outer part connects to the wheel and is free to rotate. For the case of the cast aluminum steering knuckle (Figure 5-3), the strut mounts on the steering knuckle vertical arm, and the control arm and the stabilizer bar attach to the horizontal arm and lower body. The body also attaches to the caliper from its two outer bolt holes and to the wheel hub from the four mounting holes.

The primary loading condition for the forged steel and cast aluminum steering knuckles was simulated as shown in Figure 5-4 and Figure 5-5, respectively. For the forged steel steering knuckle (Figure 5-4), because of the symmetrical round geometry of the spindle, the forces and moment in the \( y \) or \( z \) directions result in a uniaxial stress along the spindle direction and a failure location at the second step of the spindle. Therefore, the
loading could be simplified to a single moment applied to the spindle in the $y$ or $z$ direction. The torsional moment and axial force in the $x$ direction are minor due to the presence of bearings on the spindle. In addition, attachment of the hub assembly to the steering knuckle mounting holes prevents the transfer of these loads to the body. For the cast aluminum steering knuckle (Figure 5-5) the primary loading is in the form of a moment applied in the $y$ direction to the arm, while the body is fixed through its four bolt hole.

5.2 Fixture Assembly Design and Preparation

Prior to preparing the fixture parts, finite element models of the simulated loading conditions were analyzed in order to determine and verify the critical locations and the loading and restraint arrangements. The critical points of highest stress in the components for the primary loading conditions discussed in Sections 5.1 were detected from the stress analysis, as described in Chapter 4. Accordingly, specific test fixtures for each one of the two steering knuckles were designed and machined.

The exploded view of the forged steel steering knuckle fixturing is shown in Figure 5-6 and the assembled model and the real fixture assembly is shown in Figure 5-7. In this arrangement the spindle was fixed by a 2-piece block where threaded rods tightened the block to the spindle. A pair of L-shaped moment arms transferred the load from the testing machine loading actuator to the spindle blocks in the form of bending load. The strut and suspension connections on the steering knuckle body were fixed to the bench using round and square blocks.

The exploded view of the cast aluminum steering knuckle fixturing is shown in Figure 5-8. For this component a two-strut-attachment test was conducted, as shown in the assembled model and the real fixture assembly of Figure 5-9. In this arrangement, the strut
attachment of the arm was connected from both sides to a pair of moment arms. The moment arms transferred the bending load from the loading actuator to the steering knuckle.

The four hub bolt holes were fixed to the bench.

5.3 Pre-Test Analysis and Considerations

A number of considerations were made to ensure correctness of the tests. Strain gages were positioned on the components and the strain readings were compared to analytical values and finite element analysis (FEA) results. To validate the test setups, values of strains as measured by strain gages in component testing and as predicted using FEA were compared, and are listed in Table 5-1. The strain gages for the forged steel steering knuckle were positioned at the vicinity of the spindle root and the first step fillets, and for the cast aluminum steering knuckle two gages were positioned at the goose neck of the strut arm and two at the hub bolt holes where crack initiation was observed during component testing. These locations are identified in Table 5-1. Depending on the location of the gage, the proper component of the strain obtained from the FEA was selected for comparison.

The differences between measured and predicted strains obtained for the two steering knuckles were less than 18% and were considered reasonable for the complex steering knuckle geometries. The error can be attributed to: 1) difficulty in matching the exact corresponding node on the FEA model to the strain gage location, 2) difference in boundary conditions between FEA and actual test setup, 3) the unwanted friction force that exists between the point of load application and the moment arms in test (although this was reduced significantly by using roller bearings), and 4) measurement errors related to strain gage averaging particularly for cast aluminum steering knuckle due to high strain gradient at
the strain gage locations. The measured strains also confirmed the symmetry of and linear variations with elastic loading.

For the forged steel steering knuckle, which has a relatively simpler geometry, results of strain calculations from analytical equations of mechanics of materials are also listed in Table 5-1. These results are mostly in between the measured and FEA-predicted strains. It should be noted that the position of the strain gages and the magnitudes of the applied loads were such that all measured strains were in the elastic range. In addition, it was confirmed that change of the gripped spindle length for the forged steel steering knuckle does not affect the strain reading at the locations close to the spindle step.

For the cast aluminum steering knuckle, the bolt pretension and the accompanied compressive stress on the engaged area of the component was estimated using theory of machine design (Shigley and Mischke, 1989) for a bolt torque of 13.6 N.m (applied to the bolt during the component tests) and a bolt having major thread diameter of 12.7 mm to be 5.4 kN and 14 MPa (2 ksi), respectively. The stress value was negligible compared to the stresses generated due to the experiment loading and were not considered in subsequent stress analysis and predictions.

A closed-loop servo-controlled hydraulic 100 kN axial load frame was used to conduct the tests. The calibration of the system was verified prior to the beginning of the tests. As shown in Figure 5-6 to Figure 5-9, a rod end bearing joint was used to apply the load from the actuator to the moment arms, in order to avoid any out of plane bending. Due to relative rigidity of the fixtures, the effect of horizontal friction force was found to be significant at the joint-fixture contact point. Therefore, a needle roller bearing was installed on each side of the pin of the bearing, allowing the moment arm to roll horizontally to minimize friction force. Figure 5-10 shows the details of attachment of the rod-end and
needle roller bearings to the load cell and moment arms for the forged steel steering knuckle testing. Care was taken to ensure symmetry of the bending load transferred from the two moment arms. All fixture bolts and nuts were tightened with identical torque values to maintain consistency.

Table 5-2 lists the maximum moment, moment amplitude, ratio of maximum to minimum moments, and test frequencies. The moment levels were determined based on stress analysis results, the true stress-true strain curve of the materials, and standard load cases. The actual load cases for these specific components were not available, but two standard load cases, Lotus load condition that is used in Daewoo Motors (Lee and Lee, 2003) and ULSAS standard load cases (AISI, 2001) were examined. The primary load of $3g$ acting on the wheel is the vertical bump for both of the load cases, where $g$ is the acceleration of gravity. Based on component weights of 1195 kg (2635 lb) and 1983 kg (4372 lb), and the wheel hub moment arms (from the center of the wheel bearings to the critical location of the components) of 21.9 mm and 25.4 mm, maximum moments of 1000 N.m and 2000 N.m were obtained for the forged steel and cast aluminum steering knuckles, respectively. The highest maximum moment levels in the component test were specified to be 1515 N.m and 2230 N.m for the forged steel and cast aluminum steering knuckles, respectively.

A minimum moment of 75 N.m was used in all tests, corresponding to an $R$-ratio $(M_{\text{min}}/M_{\text{max}})$ of less than 0.08. A total of seven component tests at four moment levels with amplitudes between 380 N.m and 720 N.m for the forged steel steering knuckle, and a total of six tests at four moment levels with amplitudes between 560 N.m and 1075 N.m for the cast aluminum steering knuckle were conducted. The frequency of the tests ranged from 0.5
Hz for higher moment levels, to 5 Hz for lower moment levels. The moment levels chosen resulted in fatigue lives between $10^4$ and $2 \times 10^6$ cycles.

### 5.4 Collecting Experimental Data

Displacement amplitude versus cycle data of the components during each test were monitored in order to record macro-crack nucleation (i.e. a crack on the order of several millimeters), growth, and fracture stages. Due to the nature of the loading and restraints on both steering knuckles, the locations of crack initiation could not be reached to enable detecting crack nucleation. Therefore, a marked displacement amplitude increase during the test was considered as the crack nucleation point, and a sudden increase as the final fracture.

Variations of displacement amplitude versus cycles for two typical tests of the forged steel and cast aluminum steering knuckles are shown in Figure 5-11. As can be observed from this figure, for the forged steel steering knuckle the displacement amplitude was nearly constant until about the end of the test. This indicates that the time lag between macro-crack nucleation and fracture was a small fraction of the total life. On the other hand, for the cast aluminum steering knuckle the crack growth portion of the life was significant. The crack lengths of the cast aluminum steering knuckles were also visually observed and recorded. For the typical cast aluminum steering knuckle data in Figure 5-11, the crack lengths were 8 mm, 13 mm, 20 mm and 27 mm at $N/N_f$ equal to 0.3, 0.5, 0.7 and 0.9, respectively, where crack grew with an approximately linear trend versus number of cycles. The lives to failure used in latter comparisons for the cast aluminum steering knuckle were considered to be those of macro-crack nucleation.
5.5 Experimental Results, Analysis, and Comparisons

Table 5-2 presents the component test data and Figure 5-12 shows the applied moment amplitude versus fatigue life curves for the two components. Although a plateau was observed for the forged steel steering knuckle as could be seen in Figure 5-12, the cast aluminum steering knuckle did not exhibit this behavior at the selected lower moment amplitude levels. The stress amplitude versus fatigue life curves of the two steering knuckles are superimposed in Figure 5-13. The stresses in this chart are the local von Mises stresses at the critical locations of the components obtained from nonlinear FEA (as discussed in Chapter 4). For the cast aluminum steering knuckle S-N lines based on failure defined as either macro-crack nucleation or fracture are shown. On the average, about 50% of the cast aluminum steering knuckle life is spent on macro-crack growth. This figure also shows that the forged steel steering knuckle results in about two orders of magnitude longer life than the cast aluminum steering knuckle, for the same stress amplitude level. This occurs at both short as well as long lives. Note that the difference can be even larger at long lives, due to the run-out data points for the forged steel steering knuckle. It could also be seen from this figure that the highest load levels provided life in the range of $10^4$ to $5 \times 10^4$ cycles. Moment levels corresponding to this life range could be considered as representative of overload conditions for suspension components, such as a steering knuckle, in service.

Different fracture surface characteristics were observed for the forged steel and cast aluminum steering knuckles. As could be seen in Figure 5-14 for a typical steering knuckle, the failed forged steel steering knuckles had a typical ductile material fatigue failure surface including crack initiation, smooth crack growth and rough fracture sections. The failed cast aluminum in Figure 5-15 could be seen with a relatively longer crack growth portion (as
observed in the displacement monitoring curve), as compared the crack growth portion of the forged steel steering knuckle.
Table 5-1  Measured and predicted strain values at 680 N.m static moment. Locations of the gages are also shown.

| Gage Number | Measured Strain (μstrain) | Predicted Strain \( \frac{P \, Mc}{A \, T} \) (μstrain) | Diff. Meas. and FEA (%)
<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>Predicted Strain from FEA (μstrain)</td>
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</tr>
<tr>
<td></td>
<td></td>
<td>Diff. Meas. and FEA (%)</td>
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</tr>
<tr>
<td><strong>Forged Steel</strong></td>
<td></td>
<td></td>
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</tr>
<tr>
<td>1</td>
<td>542</td>
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<td>583</td>
</tr>
<tr>
<td>2</td>
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</tr>
<tr>
<td>2</td>
<td>534</td>
<td>-</td>
<td>470</td>
</tr>
<tr>
<td>3</td>
<td>228</td>
<td>-</td>
<td>268</td>
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<tr>
<td>4</td>
<td>289</td>
<td>-</td>
<td>320</td>
</tr>
</tbody>
</table>
Table 5-2 Component test data of forged steel and cast aluminum steering knuckles investigated.

<table>
<thead>
<tr>
<th>$M_{\text{max}}$ (Nm)</th>
<th>$M_a$ (Nm)</th>
<th>R-Ratio</th>
<th>$N_f$ Crack Nucleation (cycle)</th>
<th>$N_f$ Fracture (cycle)</th>
<th>Test Freq. (Hz)</th>
<th>Remarks</th>
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</thead>
<tbody>
<tr>
<td>Forged Steel</td>
<td></td>
<td></td>
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</tr>
<tr>
<td>1515</td>
<td>720</td>
<td>0.05</td>
<td>57,200</td>
<td></td>
<td>2</td>
<td>1</td>
</tr>
<tr>
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<td>720</td>
<td>0.05</td>
<td>48,800</td>
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<td>0.06</td>
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<td>3</td>
<td>1</td>
</tr>
<tr>
<td>965</td>
<td>445</td>
<td>0.07</td>
<td>&gt;323,800</td>
<td></td>
<td>3</td>
<td>1,2</td>
</tr>
<tr>
<td>965</td>
<td>445</td>
<td>0.07</td>
<td>&gt;1.5E6</td>
<td></td>
<td>3</td>
<td>1,2</td>
</tr>
<tr>
<td>965</td>
<td>445</td>
<td>0.07</td>
<td>&gt;560,700</td>
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<td>1,2</td>
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<tr>
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<td>&gt;1.4E6</td>
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<td>5</td>
<td>1,2</td>
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<tr>
<td>Cast Aluminum</td>
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<tr>
<td>2230</td>
<td>1075</td>
<td>0.04</td>
<td>15,000</td>
<td>29,500</td>
<td>1</td>
<td></td>
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<tr>
<td>2230</td>
<td>1075</td>
<td>0.04</td>
<td>35,000</td>
<td>58,500</td>
<td>1</td>
<td></td>
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<tr>
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<td>755</td>
<td>0.05</td>
<td>292,000</td>
<td>409,000</td>
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<tr>
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<td>0.05</td>
<td>190,000</td>
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<tr>
<td>1195</td>
<td>560</td>
<td>0.07</td>
<td>278,000</td>
<td>869,900</td>
<td>4</td>
<td></td>
</tr>
</tbody>
</table>

1. There was no marked difference between crack nucleation and fracture lives of the forged steel steering knuckle.
2. These parts broke at a location other than the critical location due to fixture part breaking, therefore they were considered as run-out tests.
Figure 5-1 Nomenclature used for forged steel (top) and cast aluminum steering knuckles in this study.
Figure 5-2  Forged steel steering knuckle within the rear suspension system of a 4-cylinder sedan.
Figure 5-3  Cast aluminum steering knuckle within the front suspension system of a 6-cylinder minivan.
Figure 5-4 Simulated primary loading and restraints on the forged steel steering knuckle for component testing.

Figure 5-5 Simulated primary loading and restraints on the cast aluminum steering knuckle for component testing.
Figure 5-6 Exploded view of the forged steel steering knuckle test fixture: 1) steering knuckle, 2) spindle grip, 3) moment arm, 4) side blocks, 5) end bar, 6) test bench, 7) load actuator, 8) load cell, 9) rod-end bearing and pin, 10) needle roller bearings.

Figure 5-7 Schematic drawing (left) and the fixture for forged steel steering knuckle test arrangement.
Figure 5-8  Exploded view of the cast aluminum steering knuckle test fixture: 1) steering knuckle, 2) body fixing blocks, 3) arm grip pins, 4) moment arm adjusting pins, 5) moment arm, 6) test bench, 7) load actuator, 8) load cell, 9) rod-end bearing and pin, 10) needle roller bearings.

Figure 5-9  Cast aluminum steering knuckle test arrangement showing fixturing schematic (left), and arm fixturing close up.
Figure 5-10  Load application through rod-end bearing and a pair of needle roller bearings to avoid out-of-plane bending and horizontal friction force.

Figure 5-11  Displacement amplitude versus normalized cycles for typical forged steel and cast aluminum steering knuckles during tests.
Figure 5-12  Applied moment amplitude versus fatigue life curves for (a) forged steel and (b) cast aluminum steering knuckles.
Figure 5-13  Superimposed stress amplitude versus fatigue life curves for forged steel and cast aluminum steering knuckles.

Figure 5-14  Typical fractured (left) and fracture surface of the forged steel steering knuckle.
Figure 5-15  Typical fractured (top) and fracture surface of the cast aluminum steering knuckle.
Chapter Six

6 Fatigue Life Predictions and Comparisons with Experimental Results

In general, four fatigue life prediction models are commonly used. These include the stress-life model, the strain-life model, the fatigue crack growth model, and a combination of fatigue crack growth and strain-life models. The first two models consider the macro-crack nucleation phase as failure. The stress-based approach is typically used in terms of nominal stresses. Therefore, it does not directly account for the plastic strain at the notch root, which can have significant effect on fatigue behavior of components under cyclic plasticity conditions. The predictions of this approach for notched members, such as the fillet at the spindle step of the forged steel steering knuckle and the hub bolt holes of the cast aluminum and cast iron steering knuckles, are usually conservative. Local values of stress could also be used in stress-life predictions (here called local stress-life or $\sigma$-N method).

Strain-based approaches are very common in life prediction of notched components. In these cases, the behavior of the material at the root of the notch is best described in terms of strain. The approaches pursued here are nominal stress approach, local strain approach using nominal stresses, and local stress and strain approaches using finite element analysis (FEA) results. The fatigue crack growth model was not pursued in this study, since damage tolerance design, which assumes preexisting cracks, is not yet as practical for ground vehicle parts due to costly and, in most cases, infeasible inspection of cracks. The overall methodology followed for life prediction in this study is shown in Figure 6-1.
In the following sections the life prediction paths pursued are implemented and discussed. The details of fatigue life calculations for each method at the highest moment level for the forged steel and cast aluminum steering knuckles are provided in Appendix B.

### 6.1 Nominal Stress Approach

One of the approaches used for the forged steel steering knuckle is the nominal stress approach (path 1 in Figure 6-1). This method could not be used for the cast aluminum steering knuckle since nominal stress at the vicinity of the notch could not be explicitly defined for this component. In nominal S-N approach, the nominal values of maximum, mean and alternating stresses were calculated from analytical equations (i.e. for nominal elastic behavior, $S = P/A + Mc/I$). Nominal yielding occurred at higher maximum moment levels of 1515 N.m and 1240 N.m. The maximum and minimum nominal stress distributions at a section remote from the fillet, i.e. where the stress distribution becomes uniform, were obtained by applying maximum and minimum moments to the FEA model of Figure 4-2 for the forged steel steering knuckle and using a nominal stress distribution on a cross section remote from the spindle 2nd step fillet as in Figure 4-9. Two mean stress correction methods were used to account for the effect of mean stress. These include the Gerber parabola:

$$\frac{S_u}{S_{sy}} + \left(\frac{S_m}{S_u}\right)^2 = 1$$  \hspace{1cm} (6-1)

and the modified Goodman equation:

$$\frac{S_u}{S_{sy}} + \frac{S_m}{S_u} = 1$$  \hspace{1cm} (6-2)

The fatigue life was then obtained from a Basquin-type equation:

$$S_{sy} = \sigma'_f \left(2N_f\right)^a$$  \hspace{1cm} (6-3)
To find the exponent \( B \), the S-N line was drawn considering the effect of the notch. The stress concentration factor, \( K_t = 2.2 \), at the fillet was obtained from stepped shaft stress concentration chart. Peterson’s equation, \( K_f = 1 + \frac{K_t - 1}{1 + a/r} \), was used to calculate the fatigue notch factor, \( K_f = 2.1 \), where \( a \) is a material characteristic length obtained from an empirical relationship between \( S_u \) and \( a \), i.e. \( a = 0.0254 \left( \frac{2070}{S_u} \right)^{1.8} \) with \( S_u \) in MPa and \( a \) in mm (Stephens et al., 2000). Then the fatigue limit (at \( 10^6 \) cycles), \( S_f \), was reduced to \( S_f / K_f \), and \( B \) was found to be -0.133. No surface finish effect was considered due to highly polished surface of the spindle at the fillet. Table 6-1 lists nominal stress amplitude and mean stress, equivalent nominal stress amplitude for \( R = -1 \) (\( S_{Nf} \)), and predicted and experimental fatigue lives. Figure 6-2 shows stress amplitude versus predicted and experimental lives using the nominal S-N approach. For the Gerber model, a life difference of less than a factor of 6 compared to the experimental results could be observed. The modified Goodman predictions were more conservative, with predicted lives shorter by about a factor of 50.

6.2 Local Strain Approach Using Nominal Stresses

The nominal values of stress were also used as input to the local strain approach (path 2 in Figure 6-1). To use this method, a notch rule such as Neuber’s rule or strain energy density rule is required to obtain the local stresses and strains. Neuber’s rule relates nominal stress and strain with local stress and strain behavior in such a way that the geometric mean value of the stress and strain concentration factors is equal to the Hookian (theoretical) stress concentration factor (Neuber, 1961). The strain energy density rule is based on the assumption that the strain energy density at the notch root is nearly the same for linear elastic notch behavior and elastic-plastic notch behavior, as long as the plastic
deformation zone at the notch is surrounded by an elastic stress field (Molski and Glinka, 1981). Because of this requirement and considering the fact that general yielding exists at the two higher moment levels, the latter method is not best suited for the case of forged steel steering knuckle.

Neuber’s assumption has shown good predictions for plane stress conditions and is usually conservative by over-predicting strain in most other cases (Stephens et al., 2000), even though this is not always the case (Härkegård and Mann, 2003; Zeng and Fatemi, 2001). In order to investigate the applicability of the notch formula used, the state of stress and strain at the critical sites of this steering knuckle was investigated. As discussed in Section 4.6, the results of stress analysis for the forged steel steering knuckle at the transition point between the fillet and the spindle second step (the highest stressed point of the component) showed that the state of plane strain could be considered for the critical location and Neuber’s rule is expected to provide conservative results.

To deal with nominal yielding at the two higher moment levels (1515 N.m and 1240 N.m) in local strain approach, two methods were followed. The first method is based on considering the fact that Neuber’s rule in its original form is an approximation method that also applies to the general case of large scale yielding (Neuber, 1961; Härkegård and Mann, 2003), the local stresses and strains were obtained from this general form:

\[ \sigma \varepsilon = S \varepsilon K_t^2 \]  

(6-4)

where the relation between nominal stress \( S \) and nominal strain \( \varepsilon \) is based on Ramberg-Osgood equation, rather than the linear relation \( S = \varepsilon E \). The state of stress at the root of the notch is multiaxial (as discussed in Section 4.6) and to account for this, a method proposed by Hoffmann and Seeger (1985) was implemented and the equivalent stress concentration factor, \( K_{eq} \), was used in Equation (6-4) instead of \( K_t \), where:
In this equation \( a_e \) and \( b_e \) denote the elastic stress ratios, \( a_e = \frac{\sigma_{e2}}{\sigma_{e1}} \) and \( b_e = \frac{\sigma_{e3}}{\sigma_{e1}} \). The second method applied is based on a generalized application of Neuber’s rule for nonlinear net cross section behavior (i.e. general yielding) proposed by Seeger and Heuler (1980). In this method, by defining modified nominal stress and strain incorporating the plastic limit load, Neuber’s rule is generalized for the case of inelastic net section behavior. They proposed this generalization for Neuber’s rule as:

\[
\frac{K_{eq}^2 S^2}{E} \frac{E e^*}{S^*} = \varepsilon = \frac{\sigma^2}{E} + \sigma \left( \frac{\sigma}{K''} \right)^{\gamma''}
\]

(6-6)

The modified nominal stress \( S^* \) is calculated from:

\[
S^* = \frac{P}{P_p} S_y
\]

(6-7)

\( P \) is the maximum load of the test and \( P_p \) is the plastic limit load defined for elastic-perfectly plastic material and characterizes the onset of general yielding of net section area. Here, the loads were replaced by moments due to specific loading condition for the components. The modified nominal strain, \( \varepsilon^* \), is obtained from the material stress-strain law.

Once the local amplitudes of stress, \( \sigma_a \), and strain, \( \varepsilon_a \), were calculated, fatigue life was obtained from the Smith-Watson-Topper (SWT) life equation that incorporates the effect of mean stress:

\[
\sigma_{\max} e_a E = (\sigma'_f)^2 (2N_f)^{b} + \sigma'_f e'_f E (2N_f)^{b+c}
\]

(6-8)

Table 6-2 lists the stress results from FEA and predicted stresses based on Neuber’s rule, and Seeger-Heuler’s generalized application of Neuber’s rule. When the estimated stresses of these two stress estimation approaches are compared, it can be found that for the
maximum applied moment levels where large-scale yielding existed, the latter approach estimates 14% and 5% higher local stress amplitudes, but 30% and 25% lower mean stress compared to nonlinear FEA results than Neuber’s rule without generalization. In this regard, the stress and strain predictions by direct application of Neuber’s rule, as compared with the Seeger-Heuler’s generalized application of Neuber’s rule, were less conservative and closer to nonlinear FEA results. In this case the results of Neuber’s rule (without generalization) were considered as the base in life predictions.

It should be emphasized that these two methods compared here (i.e. Neuber’s rule, and Seeger-Heuler’s generalized application of Neuber’s rule) are notch deformation rules and are used to estimate stress and strain at the notch. Fatigue life prediction for the two higher moment levels that follows the stress and strain estimations resulted in 2 and 1.3 times more conservative lives for Neuber’s rule with Seeger-Heuler’s generalized application of Neuber’s rule.

Table 6-1 lists the local alternating and mean stresses and predicted and experimental lives, and Figure 6-3 illustrates SWT parameter \(\sqrt{\sigma_{\text{max}} \varepsilon_{\sigma} E}\) versus predicted and experimental lives, where it could be seen that the predictions are conservative by more than a factor of 7. This could be explained by the suggestion that Neuber’s rule is shown to be more applicable to plane stress states.

6.3 Local Stress and Strain Approaches Using FEA Results

The crack nucleation models could also be applied using local stress and strain values obtained directly from FEA at the failure location (paths 3 and 4 in Figure 6-1). The complex geometry of the cast aluminum steering knuckle is an example for which no notch
factor could be defined and according to Sonsino, Kaufmann and Grubišić (1997), for such conditions transferability of material test data could be performed only through local equivalent stresses or strains in the critical failure areas. Here, the local elastic-plastic stresses and strains were obtained from FEA using two approaches; first, by full-scale elastic-plastic FEA of the steering knuckles (path 3 in Figure 6-1), and second, a combination of elastic FEA and a Neuber-type stress correction (path 4 in Figure 6-1). The stress-life and strain-life methods were then implemented to predict fatigue life. To account for the multiaxial stress state, equivalent von Mises values were used for both components. Since the maximum principal stress theory is more commonly used for brittle materials such as cast aluminum, life predictions using maximum principal stresses and strains were also performed for the cast aluminum steering knuckle.

Using the nonlinear FEA results (path 3 in Figure 6-1), the local equivalent stresses and strains corresponding to the experimental loading conditions were obtained by applying equal moments to the simulated finite element models and performing nonlinear FEA. The local R-ratios for stress are different from the applied moment R-ratio due to local plasticity, as indicated in Section 4.6. In stress-life approach, the effect of mean stress was accounted for using three mean stress models. These include Gerber’s equation (Equation (6-1), but for local stresses), the stress-life version of the SWT model:

\[ \sigma_{\text{max}} \sigma_u = \sigma'_f^2 \left(2N_f\right)^{2b} \]  

and the commonly used modified Goodman equation (Equation (6-2), but for local stresses). The Basquin equation (Equation (6-3), but for local stress) was then used to obtain the fatigue life, where \( B = b \). For the stress-life version of SWT model, fatigue life was found directly from Equation (6-9).
Superimposed local stress amplitude versus predicted and experimental lives for different mean stress correction models based on stress-life approach is presented in Figure 6-4. Comparing the predictions with test results for the forged steel steering knuckle in Figure 6-4a, the Gerber’s model could be seen to offer close predictions. The modified Goodman’s prediction is the most conservative, by an order of magnitude in fatigue life. For the cast aluminum steering knuckle in Figure 6-4b, Gerber’s line is relatively close to the experimental life line by factors of 5 in shorter lives and a factor of 2 in longer lives, and offers relatively conservative predictions. The SWT and modified Goodman predictions are very conservative. In addition, comparing Figure 6-3 and Figure 6-4a, i.e. the nominal and local approaches for the forged steel steering knuckle, it could be seen that predictions of the local approach are closer to the experimental lives. This is partly due to the fact that the local approach directly accounts for residual stresses generated because of local plasticity, whereas this effect is not considered in the nominal stress approach.

In the strain-life method using the nonlinear FEA results, to take the effect of mean stress into account in fatigue life predictions two models were considered; the Morrow’s mean stress model:

\[
\varepsilon_a = \frac{\varepsilon_f' - \varepsilon_m'}{E} \left(2N_f\right)^b + \varepsilon_f' \left(2N_f\right)^c
\]  \hspace{1cm} (6-10)

and the strain-life version of SWT model, Equation (6-8). The predictions for the cast aluminum steering knuckle were also performed using maximum principal stress, in addition to von Mises stress. Superimposed local strain amplitude versus predicted and experimental lives for different mean stress correction models based on strain-life approach are presented in Figure 6-5. Morrow’s mean stress correction model provides the closest prediction, within a factor 2 in life for the forged steel, and within a factor of 3 in life for the cast aluminum
steering knuckle. The predictions for the cast aluminum steering knuckle based on the maximum principal stress were about twice more conservative than the von Mises predictions, and are not shown.

In the second category of local stress and strain approach using FEA stresses and strains, elastic FEA results were used in conjunction with a Neuber-type stress correction (path 4 in Figure 6-1) to account for inelastic deformation (Chu, 1997). To estimate the local elastic-plastic stresses and strains, Neuber’s rule was applied in terms of equivalent quantities of multiaxial stresses, and the equivalent stresses were computed from von Mises flow criterion (Socie and Marquis, 2000), which is a reasonable assumption for multiaxial proportional stresses. If \( \sigma^e \) and \( \varepsilon^e \) are defined as the elastically calculated notch stress and strain, and \( \sigma^p \) and \( \varepsilon^p \) are the elastic-plastic notch stress and strain, Neuber’s rule becomes:

\[
\sigma^p \varepsilon^p = \sigma^e \varepsilon^e
\]  

(6-11)

The second equation to obtain elastic-plastic stress and strain is the cyclic stress-strain curve of the material:

\[
\varepsilon^e = \frac{\sigma^e}{E} + \left( \frac{\sigma^e}{K^e} \right)^{\frac{1}{n^e}}
\]  

(6-12)

To better observe the differences between the nonlinear FEA and linear FEA combined with Neuber-correction, Table 6-3 compares the stress amplitude, mean stress and fatigue life of these two approaches. In this table, the fatigue lives using SWT mean stress correction models for \( \sigma \)-N and \( \varepsilon \)-N approaches are listed. It could be seen that the lives obtained using the linear FEA and Neuber-correction are close to the nonlinear FEA predictions. In particular, for the case of cast aluminum steering knuckle, the nonlinear FEA results in conservative lives, while the linear FEA with Neuber-correction predicts closer lives to experimental results. It could be concluded that the linear FEA plus Neuber-
correction method is a capable approach to life prediction of components, especially considering the fact that it does not require the more complicated and time-consuming nonlinear FEA.

6.4 Fatigue Performance Comparison of Components

Superimposed stress amplitude versus life curves based on stress-life approach, and SWT parameter versus life based on the strain-life approach for the three steering knuckles are presented in Figure 6-6 and Figure 6-7, respectively. To obtain stress unit in Figure 6-7, square root of the left side of Equation (6-8) is plotted as the SWT parameter. Comparison of the forged steel, cast aluminum and cast iron steering knuckle prediction curves in Figure 6-7 demonstrates that the forged steel steering knuckle offers more than an order of magnitude longer life than the cast iron steering knuckle, at both short as well as long lives. As compared with the cast aluminum steering knuckle, the predicted lives for the forged steel steering knuckle are longer by about three orders of magnitude. The fatigue performance comparison results of the components of this study are also presented in Zoroufi and Fatemi (2004).

6.5 Other Considerations in Durability Assessment of Steering Knuckle

Vehicle steering knuckle is a critical mechanical component in terms of its fatigue behavior. Forging, casting, and machining induce residual stresses to the component. Residual stresses can be determined analytically or computationally having detailed knowledge of the local mechanical response during the induction of residual stress (Savaidis et al., 2002), or experimentally using, for instance, X-ray diffraction measurements (Stephens et al., 2000). Residual stresses at the critical locations of the component generated during the
manufacturing process could be a significant source of strengthening (if compressive) or weakening (if tensile), in terms of fatigue life. For the case of the forged steel steering knuckle under investigation, the main location suitable for inducing local compressive residual stress is the lower section of the spindle 2nd step fillets. This could be achieved by a number of ways including shot peening. Due to one-sided nature of loading that the component undergoes, this is a very effective method to increase fatigue strength.

Automotive chassis components are often used without corrosion protection. Therefore, fatigue performance under corrosive environments has to be assessed as part of the design process. Corrosion in conjunction with the effect of time, frequency and variable amplitude loading may play a prominent role in fatigue design, especially for aluminum alloys that have lower fatigue limits than steels. For instance, a study on durability assessment of forged and cast aluminum steering knuckles showed that fatigue resistance drastically drops under cyclic loading in corrosive environments (Heuler and Birk, 2002).

The current study was based on the simplified constant amplitude experiments and analysis, while the component in service undergoes the more complex variable amplitude loading including overloads and underloads. For a steering knuckle, this type of loading can be generated from panic brake, severe cornering and pot holes. As stated in (Bonnen and Topper, 1999), experience with long cracks and low-stress-levels have shown that tensile overloads can increase fatigue life while compressive overloads can decrease it. A tensile overload can cause a substantial crack tip plastic zone and, as a result of elastic constraint around the plastic zone, it leaves compressive residual stresses which result in crack growth retardation. The converse can occur for compressive overloads which decrease compressive residual stresses and, consequently, fatigue life.
As an example of the effects of variable amplitude loading on fatigue behavior of automotive parts, Bonnen and Topper (1999) conducted variable amplitude bending-torsion fatigue experiments on normalized SAE 1045 steel axle-shafts to determine the effects of overloads on fatigue life. Either periodic bending overloads or static bending loads were applied to these shafts to determine their effect on torsional fatigue. It was determined that these yield-stress-level bending excursions both decrease the torsional fatigue limit and shorten the torsional fatigue life at medium and long lifetimes. In the case of multiaxial loading, the overloads resulted in substantial reductions in fatigue strength at all lives such that at the fatigue limit this reduction was nearly a factor of three. Therefore, apart from the overload itself, other parameters such as stress level, sequence effect, cumulative damage and the type of loading may have substantial influence on fatigue behavior.

From another viewpoint, constant amplitude tests, as compared with realistic variable amplitude tests, may give insufficient data about the relative fatigue strength of components. Nevertheless, constant amplitude loading is specified in all the acceptance fatigue tests for components by taking commercial considerations into account. From a commercial viewpoint, an acceptance fatigue test needs to be simple, quick, and cheap. The risk of a fatigue failure in service may be kept low by making a simple test relatively severe. In many situations this will minimize the overall product cost, despite possible over-design (Pook, 1997). In situations where minimum weight design is important, such as the aircraft industry and more recently the automotive industry with the trend toward lighter and more fuel efficient vehicles, more expensive and time consuming variable amplitude tests are economically justified.

A number of references in the literature conducted real-time vehicle wheel load and moment measurements to obtain the applied loads and moments on steering knuckles (for
instance see Lee et al., 1995). It was found that loads and moments in all three directions are applied to a typical steering knuckle installed in a vehicle. Experiments and analyses based on multiaxial loading, rather than unidirectional, express the real loading condition. But due to complexity of such kind of testing for this component, unidirectional tests are typically conducted. For instance Lee et al. (1995), conducted separate unidirectional tests in two primary loading directions on the steering knuckle. Although unidirectional testing in the primary loading direction, as performed in this work, does not reflect the precise effect of real life history, it could be regarded as a simplified and practical means of estimating fatigue tolerance of the component.

Fretting fatigue is another mode of failure. Fretting, which is a surface wear phenomenon occurring between two contacting surfaces having oscillating relative motion of small amplitude, may exist between the steering knuckle and hub assembly, strut joints, suspension links, and chassis connections in service.

Surface finish effect could be very influential in fatigue evaluation and, normally, a surface finish reduction factor is applied to the fatigue strength of a component. However, in this study the fillet of the forged steel steering knuckle was machined and polished and, therefore, no surface finish factor was applied to life predictions. For the cast aluminum steering knuckle, due to the nature of the casting materials and the fact that the defects of a casting material are uniform internally and externally, no surface finish factor was implemented either.

In forged components, grain flow produces directional characteristics in properties such as strength, ductility, and resistance to impact and fatigue. The effect of anisotropy and directionality of the forged steel steering knuckle material fatigue behavior was investigated in this study, where it was found that the primary stressing direction offers about twice more
life than the other two directions. In castings, flow lines, porosity and several other casting
defects can show up, depending on casting practice. Solidification is not always uniform in
castings. Directional solidification is another process that might bring variability of behavior
in different directions.

It should be noted that the influencing parameters on fatigue behavior of a steering
knuckle or similar components are not limited to those mentioned, nor are the approaches
to address such problem only confined to those implemented. One of the main purposes of
this chapter was to provide a feasible methodology to approach the durability assessment of
a fatigue-critical automotive component.
Table 6-1 Nominal and local stresses and predicted and experimental fatigue lives for the forged steel steering knuckle using nominal stress and local strain approaches.

<table>
<thead>
<tr>
<th>Applied Moment Amplitude (N.m)</th>
<th>$S_y$ (MPa)</th>
<th>$S_m$ (MPa)</th>
<th>$S_N$</th>
<th>Fatigue Life (cycles)</th>
<th>$\epsilon_N$</th>
<th>$\sqrt{\sigma_{max} \epsilon_a E}$ (MPa)</th>
<th>Predicted Fatigue Life (cycles)</th>
<th>Experiment Fatigue Life (cycles)</th>
</tr>
</thead>
<tbody>
<tr>
<td>720</td>
<td>268</td>
<td>301</td>
<td>422</td>
<td>309</td>
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<td>10,400</td>
<td>701</td>
<td>57,280</td>
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<td></td>
<td>0.29</td>
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<td>8,800</td>
<td>48,879</td>
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<td>388</td>
<td>288</td>
<td>1,900</td>
<td>17,800</td>
<td>674</td>
<td>12,900</td>
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<td></td>
<td></td>
<td>0.26</td>
<td>594</td>
<td>594</td>
<td>105,197</td>
</tr>
<tr>
<td>450</td>
<td>219</td>
<td>252</td>
<td>317</td>
<td>242</td>
<td>8,700</td>
<td>65,100</td>
<td>620</td>
<td>27,500</td>
</tr>
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<td>0.22</td>
<td>524</td>
<td>524</td>
<td>323,849*</td>
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<td></td>
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<td>1,565,630*</td>
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<td>380</td>
<td>185</td>
<td>219</td>
<td>253</td>
<td>200</td>
<td>47,100</td>
<td>278,600</td>
<td>570</td>
<td>81,800</td>
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<tr>
<td></td>
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<td></td>
<td></td>
<td></td>
<td>0.18</td>
<td>455</td>
<td>570</td>
<td>250</td>
</tr>
</tbody>
</table>

* The component broke at a location different from the predicted failure location due to fixture part breaking, therefore considered as run-out test.
Table 6-2  Comparison of FEA results and predicted stresses and strains obtained from Neuber’s rule, and Seeger Seeger-Heuler’s generalized application of Neuber’s rule for the forged steel steering knuckle. The predicted fatigue lives are calculated from strain-life method.

<table>
<thead>
<tr>
<th>Applied Moment Amplitude (N.m)</th>
<th>$S_a$ (MPa)</th>
<th>$S_m$ (MPa)</th>
<th>$\sigma_a$ (MPa)</th>
<th>$\sigma_m$ (MPa)</th>
<th>$\varepsilon_a$ (%)</th>
<th>Predicted Fatigue Life (cycles)</th>
<th>Neuber's Rule</th>
<th>Predicted Fatigue Life (cycles)</th>
<th>Seeger-Heuler's Generalized Application of Neuber's Rule</th>
<th>Predicted Fatigue Life (cycles)</th>
</tr>
</thead>
<tbody>
<tr>
<td>720</td>
<td>268</td>
<td>301</td>
<td>456</td>
<td>245</td>
<td>0.29</td>
<td>8,800</td>
<td></td>
<td></td>
<td>517</td>
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<tr>
<td>580</td>
<td>253</td>
<td>286</td>
<td>440</td>
<td>235</td>
<td>0.26</td>
<td>12,900</td>
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<td></td>
<td>397</td>
<td>223</td>
</tr>
<tr>
<td>380</td>
<td>185</td>
<td>219</td>
<td>345</td>
<td>225</td>
<td>0.18</td>
<td>81,800</td>
<td></td>
<td></td>
<td>345</td>
<td>225</td>
</tr>
</tbody>
</table>
Table 6-3  Comparison of alternating and mean stress values, SWT parameter and fatigue life for two stress calculation methods; nonlinear FEA, and elastic FEA plus Neuber-correction using $\sigma$-N and $\varepsilon$-N methods. Fatigue lives for $\sigma$-N and $\varepsilon$-N approaches were both obtained from the SWT model. Nonlinear FEA results were taken as the base for comparison.

<table>
<thead>
<tr>
<th>Applied Moment Amplitude (N.m)</th>
<th>$\sigma_a$ (MPa)</th>
<th>$\sigma_m$ (MPa)</th>
<th>Fatigue Life (cycles)</th>
<th>$\sqrt{\sigma_{max}^2 + \varepsilon_E^2}$ (MPa)</th>
<th>Fatigue Life (cycles)</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Forged Steel Steering knuckle</strong></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>720</td>
<td>434</td>
<td>437</td>
<td>-1</td>
<td>170</td>
<td>177</td>
</tr>
<tr>
<td>580</td>
<td>390</td>
<td>374</td>
<td>4</td>
<td>176</td>
<td>198</td>
</tr>
<tr>
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<td>294</td>
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<td>195</td>
<td>220</td>
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<tr>
<td>380</td>
<td>270</td>
<td>250</td>
<td>8</td>
<td>209</td>
<td>225</td>
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<tr>
<td><strong>Cast Aluminum Steering knuckle</strong></td>
<td></td>
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<td></td>
</tr>
<tr>
<td>1080</td>
<td>215</td>
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<td>6</td>
<td>81</td>
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<td>560</td>
<td>112</td>
<td>104</td>
<td>6</td>
<td>125</td>
<td>119</td>
</tr>
</tbody>
</table>
Figure 6-1  Durability assessment methodology pursued in this study.
Figure 6-2  Superimposed experimental life and stress-life prediction using the nominal stress approach for the forged steel steering knuckle.

Figure 6-3  Superimposed experimental life and local strain-life predictions using nominal stresses as input for the forged steel steering knuckle.
Figure 6-4  Superimposed local stress amplitude versus experimental life and predictions of stress-life model for (a) forged steel and (b) cast aluminum steering knuckle using nonlinear FEA results.
Figure 6-5 Superimposed local strain amplitude versus experimental life and predictions of strain-life model for (a) forged steel and (b) cast aluminum steering knuckle using nonlinear FEA results.
Figure 6-6  Stress amplitude versus life curves based on the stress-life approach for the forged steel, cast aluminum and cast iron steering knuckles.

Figure 6-7  SWT parameter versus life curves based on the strain-life approach for the forged steel, cast aluminum and cast iron steering knuckles.
Chapter Seven

7 Optimization Including Material, Manufacturing and Cost Considerations

The objective of the optimization study was to reduce weight and manufacturing cost of the forged steel steering knuckle while maintaining or improving its fatigue strength. It was attempted to have a more general look at the optimization. Therefore reducing the mass of the component, reducing manufacturing costs, and improving fatigue performance were focused simultaneously.

Manufacturing costs and fatigue strength of a steering knuckle depend on service conditions, geometry, material and manufacturing processes. Service conditions are typically dictated to the designer. Therefore, geometry, material and manufacturing parameters were attempted in this study as design variables. The modifications were approached in two stages; first without changing the component’s attachment geometry and focusing on steering knuckle’s body; and second, with limited change in attachment geometry and focusing on the spindle as well as the body. The material alternatives search in this study considered replacing the current material with materials of superior fatigue performance, and subsequently, reducing dimensions and weight. Manufacturing parameter modifications to improve fatigue performance or reduce manufacturing costs included precision forging instead of conventional forging, warm forging instead of hot forging, reducing manufacturing steps, and surface enhancement.
In this chapter, first component specifications including material, manufacturing process details and production cost parameters are discussed. This follows with the definition of optimization problem that introduces the objective function, constraints and design variables. Then, the details of optimization analysis and results are discussed.

### 7.1 Component Specifications

Considering the optimization task globally, makes it necessary to have knowledge about the component, its service conditions, material of construction, manufacturing processes and parameters that affect its cost. The nomenclature used in the optimization study is shown in Figure 5-1, and Figure 5-2 shows the component as installed in the suspension system of the vehicle. The exact service loading conditions for this specific component were not available, but a general case of Figure 5-4 was considered as a valid simulation of the primary loading condition for the optimization study.

The forged steel steering knuckle’s material, AISI/SAE 11V37, is a resulphurized (free machining) high-strength low-alloy (or microalloyed) steel that contains 0.37% carbon. High strength low-alloy steels, as defined in ASM Machining Handbook (ASM, 1989), are classified as a separate steel category than alloy steels, and are similar to as-rolled mild-carbon steels but with enhanced mechanical properties obtained by the addition of small amounts of alloys. In some cases, special processing techniques such as controlled rolling and accelerated cooling methods are also used. Due to these additions, low alloy steels have mechanical properties that are superior to those of the carbon steels for designated applications. Superior properties usually mean higher strength, hardness, wear resistance, toughness, and more desirable combinations of these properties. Microalloyed forging steels reduce manufacturing costs by means of a simplified thermo-mechanical treatment (i.e., a
controlled cooling following hot forging) that achieves the desired properties without additional heat treatments (i.e. quenching and tempering) required by conventional carbon and alloy steels. Addition of vanadium enhances strength and toughness of steel. It also forms carbides that increase wear resistance. The increase in fatigue strength from vanadium averages between 5 and 15 MPa per 0.01 wt% vanadium, depending on carbon content and rate of cooling from hot rolling (Nakamura et al., 1993). Vanadium can have negative impact on machinability (Bayer, 2003).

The steering knuckle’s main manufacturing processes are hot forging and machining. Figure 7-1 shows the manufacturing process flow chart. The description of each step is as follows, where the information about the forging and the machining processes were obtained from the OEM and the part’s machining company, respectively.

1. SAE 11V37 is in the form of hot rolled cold drawn bar. The raw material samples are sent to the inspection laboratory to be chemically checked.

2. In the cutting department, the bars are cut or sheared to the dimensions of the steering knuckle.

3. The bars are induction-heated or heated in furnace between 1100°C to 1260°C.

4. The forging sequence begins with bar stock. When the stock has reached the proper forging temperature, it is delivered to the impression die from the furnace.

5. The fullering and edging operations improve grain structure, reduce the cross-sectional area of the stock where needed, and gather the metal for other sections‡.

‡ Fuller is a portion of the die that is used in hammer-forging primarily to reduce the cross section and lengthen a portion of the forging stock. The fullering impression is often used in conjunction with an edger, which is the portion of the die impression that distributed metal, during forging, into areas where it is most needed to facilitate filling the cavities of subsequent impressions to be used in the forging sequence.
6. The blocking operation forms the steering knuckle into its first definite shape. This involves hot working of the metal in several successive blows of the hammer or press, thus compelling the workpiece to flow into and fill the blocking impression in the dies.

7. Flash is produced and appears as flat, unformed metal around the edge of the part.

8. The exact shape of each steering knuckle is obtained by the impact of several additional blows in the hammer that force the stock to completely fill every part of the finishing impression.

9. The flash is removed with trimming in a mechanical press. A set of trim dies consists of a sharp shearing edge, produced to the exact contour of the forging at the flash line, and a punch, also contoured to fit the forging.

10. The trimmed steering knuckles are now ready for shot cleaning where the scales and surface impurities remained from the process are removed.

11. The machining process is categorized into four steps: (1) turning the spindle, (2) conformity checking, (3) milling the holes and making the threads, and (4) inspection and packing.

12. In the first machining step (turning the spindle) the steering knuckle is placed and located on the V-block area of the fixture (Figure 7-2). The turning process is conducted in two steps by rougher and finisher tools. The turning process takes 80 seconds.

13. In the 1st inspection step, the spindle portion of the part is verified to have the required quality by visual and eddy current inspections.

14. In the milling step, the unmachined forged steering knuckle is loaded on the fixtures (Figure 7-3). This step takes 160 seconds. The following operations are performed:
   a. The strut joint holes are milled.
b. The strut joint chamfers are made.

c. The mounting hole chamfers are made.

d. The tension strut steps are milled.

e. The hub mounting holes are drilled.

f. The spindle threads are cut.

g. The hub mounting hole threads are cut.

h. The lateral link hole is made.

15. In the 2\textsuperscript{nd} inspection step the part is gaged and visually checked to make sure the holes are tapped, the threads are on the spindle, and the chamfers are in the mounting holes and flange holes.

16. The finished steering knuckles are cleaned and dispatched.

A realistic optimization process necessitates a good understanding of the production cost attributes and engineering parameters that influence them in manufacturing of the component. These are shown in Table 7-1 for the component of this study. In manufacturing of the steering knuckle, the cost attributes of a unit steering knuckle consist of the raw material, forging process and machining process costs. The cost of each process consists of variable and fixed costs.

The engineering parameters related to manufacturing were defined in four main categories in Table 7-1; the general production parameters, the forging process parameters, the machining process parameters and the machining lubrication parameters. The parameters that affect the costs were identified. For instance, a forging parameter like temperature affects tools and initial investment costs of the forging process, while a machining parameter like feed rate influences coolant and tooling costs of the machining process. The overhead costs include energy consumption during the processes. The inspection costs depend on the
design and customer requirements, but are not affected by the process parameters. It is known through the machining company that the price of the forged part (including the raw material) is 50% of the finished part price and the remaining 50% is the machining process’s share.

7.2 Optimization Problem Definition

Mathematically, a general constrained optimization problem could be defined by an objective function, design variables and a set of constraints (Vanderplaats, 1999):

Minimize $F(X)$

Subject to:

$g_j(X) \leq 0 \quad j = 1, m$ inequality constraints

$h_k(X) = 0 \quad k = 1, l$ equality constraints

$X_i^l \leq X_i \leq X_i^u \quad i = 1, n$ side constraints

where $X = \begin{bmatrix} X_1 \\ X_2 \\ \vdots \\ X_n \end{bmatrix}$

design variables

In terms of a structural design like the current case of steering knuckle redesign, $F(X)$ represents a quality to be minimized, such as the mass of the component, maximum stress at critical locations or cost. $g(X)$ is a bounded constraint like allowable stress at certain geometry locations, and $h_k(X)$ is an equality constraint like fixed dimensions and geometry limitations. The design variables such as size and shape of the component are restricted by the upper and lower limits. Material and manufacturing parameters are also considered as side variables in order to obtain an improved design.
There are several types of optimization approaches applied to automotive structural design (Ferreira et al., 2003). *Size Optimization* defines the design variables in terms of discrete parameters of the system in study. Generally these parameters do not change the overall shape of the component and only the size is modified. Parameters such as geometrical properties such as thickness, diameter, and area are used as design variables in size optimization.

On the other hand, *Shape Optimization* and *Topological Optimization* methods should change appearance of the geometrical domain in the study. In shape optimization the boundaries of the domain are mapped by a set of control variables that defines the coordinates of the domain borders. These coordinates are changed in order to minimize the objective function. Therefore the final shape of the system is changed to find the optimal design to satisfy the requirements. Topological optimization generally changes the shape of the system and is more applicable to the conceptual design stage. The initial geometrical domain is roughly defined and the optimization algorithms create several voids in the whole system in order to achieve the best material distribution to obtain the optimum design.

The overall optimization approach in this work was a combination of quantitative and qualitative methods (i.e. mass reduction, cost reduction, and improving fatigue performance using alternative materials and considering manufacturing aspects). The optimization work was performed in two stages. Figure 7-4 shows a schematic of the alternatives considered; in Stage I with no change in attachment geometry, the geometry variables were considered to be the thicknesses at different locations of the body considering manufacturing limitations. Additional manufacturing modifications to reduce cost were also investigated. In Stage II with limited changes in the attachment geometry, spindle was
redesigned and alternative materials and additional manufacturing operations were investigated to reduce weight and cost.

For each optimization stage of Figure 7-4, a localized shape optimization procedure was conducted on the steering knuckle considering manufacturing limitations. The material and manufacturing processes as design variables were more used as means of design modification rather than optimization. Figure 7-5 shows the algorithm of the optimization process. The general flow of the solution for geometric optimization started with identifying pre-process design data such as design variables and constraints. Then, finite element analysis (FEA) on the original component enabled investigating the distribution of parameters such as stress, stress gradient, and displacement. Design constraints were evaluated and the important ones were retained. A parametric study was performed to observe how a defined change in the model affected structural response. Following that, the optimization problem was created and solved, where the analysis data were updated after each one of the iterations. Referring to the primary load calculations in Chapter 5, the nominal stresses generated from the primary loading on the component are essentially elastic, so linear elastic analysis was found sufficient for the optimization study. The iterations continued as necessary till the convergence criteria were met (Vanderplaats and Miura, 1986; Lee and Lee, 2003). For the material alternatives and manufacturing processes a trial and error approach combined with qualitative investigation was followed by selecting replacement materials, proposing alternative manufacturing processes, and adding or eliminating some of the manufacturing steps. The constituents of the analysis are detailed here after.
7.2.1 Objective Function

The objective function is the function or parameter that is optimized. In this optimization of the steering knuckle, weight and manufacturing cost of the component were considered as the objective functions.

7.2.2 Constraints

Stress and geometry constraints were defined based on the selected optimization stage. In order to maintain fatigue performance of the component, equivalent local von Mises stress amplitude at the original model’s critical location (spindle 2\textsuperscript{nd} step fillet) under primary loading conditions (fixed steering knuckle body and moment applied to the spindle) was obtained at a certain moment amplitude. This moment amplitude could change during the optimization stage, but the optimization was carried out in such a way that the equivalent local stress amplitude at any location of the optimized model did not exceed the equivalent stress amplitude at the critical location of the original model under equal moment amplitude:

\[(\Delta \sigma_{\text{equivalent}} \text{ @ any geometry location}) - (\Delta \sigma_{\text{equivalent}} \text{ @ spindle second step fillet}) \leq 0\]

This criterion is reasonable considering the stress history at different locations of the component (Figure 4-10). According to this figure, the stresses generated in the component at different locations vary proportionally. It could also be noticed that for the root of the 2\textsuperscript{nd} step of the spindle (the critical location), stress is higher than at other locations.

Referring to the component’s nomenclature of Figure 5-1, the following constraints were considered for each optimization stage.

**Stage I:** The optimized steering knuckle was expected to be interchangeable with the existing one. Therefore, the following dimensions were kept unchanged:

- strut joint center-points, and inner diameters;
- front and rear lateral links center-points and inner diameters;
- tension strut joint center-point and inner diameters;
- center-points and size of the bolt holes of the hub mounting plate;
- spindle geometry.

**Stage II:** The first four constraints of Stage I remained the same but the spindle geometry could be changed.

### 7.2.3 Design Variables

Design variables specify what could change in the optimization process. Depending on each specific optimization stage of Figure 7-4, the geometry variables were selected as follows.

**Stage I:**
- body thickness and the geometry of the portion to be removed;
- strut joints outer diameters and thicknesses;
- front and rear lateral links outer diameter and thickness;
- tension strut joint outer diameter and thickness;
- hub mounting plate geometry and thickness.

**Stage II:** The variables of Stage I plus spindle shape and diameter; spindle was redesigned to a two step shaft with conic transition to reduce the stress concentration and subsequently weight.

Considering the material as a design variable, alternative steels that have better fatigue behavior could reduce component’s weight. In this regard, a number of high strength steels, high strength low-alloy steels (microalloyed steels), and advanced high strength steels were considered as possible substitutes. Taking the results of the fatigue performance
comparison of Chapter 6 into account, substitution of a lighter metal like aluminum or cheaper metal like cast iron was not considered as an option since one of the goals of the optimization of this component was to maintain the overall shape, strength, stiffness and fatigue performance of the component. A lighter or less strong material like aluminum or cast iron lends to an inevitably bigger component to acquire the required strength and stiffness, and a completely altered geometry.

Manufacturing processes as design variables were investigated from two points of view; first to reduce manufacturing costs, and second to improve fatigue strength of the component. The manufacturing processes and process steps of Figure 7-1 were reevaluated to investigate the possibility of replacing the current process with an improved one, and limiting or eliminating any post-forging process without affecting the fatigue strength, in order to reduce manufacturing costs. On the other hand, the options considered to improve fatigue strength of the component were inducing compressive residual stress by surface rolling at the spindle fillet, carburizing, nitriding, and surface hardening.

7.3 Optimization Analysis, Results and Discussions

The optimization analysis was based on the guideline of Figure 7-4, where the 2nd optimization stage compliments the 1st stage. Depending on the requirements of the designer, the proper stage could be selected as detailed here after.

7.3.1 Stage I – No Change in Attachment Geometry

Stage I of the optimization guideline (Figure 7-4) included geometry optimization without changing the attachment geometry, and cost saving in the manufacturing process. Since the objective of this stage was to maintain the current geometry with no attachment
changes, improving fatigue performance by substituting superior materials or additional manufacturing processes, which would allow having smaller spindle dimensions, was not considered for this stage.

7.3.1.1 Optimization Using Geometry Variables

A baseline FEA of the original model showed that some parts of the steering knuckle experience lower stresses for the primary load case considered. The procedure of Figure 7-5 was implemented. The shape optimization techniques were applied to lower the weight of the component. The I-DEAS Optimization software was used to perform automatic redesign of the component. Redesign allows varying the design parameters (Section 7.2.3), set limits on the design (stress), and set goals for the design (minimize stress by varying body thickness). The software then tries to find the optimal structure that satisfies these criteria.

Following the optimization, redesign histories (design changes at each optimization step) were examined including design goal, design limit, and design parameter for all iterations to find whether the design is feasible and whether it is the best feasible design possible. In a feasible design, the component falls within its stress limit while its economical manufacturing feasibility is verified. The upper bound of the design goal is the actual value of the design goal scaled up by the maximum stress limit violation. The lower bound is based on limit violation and on the predicted change in design goal value obtained from the optimization algorithm. It is always less than or equal to the actual value.

In this optimization stage the attachment geometry of the steering knuckle was kept unchanged and the geometry variable was considered to be the thickness at different locations of the body considering manufacturing limitations. Reducing the thickness of the
body close to the tension strut connection hole will likely result in distortion of the component during the forging process and was avoided. The thickness in the area surrounded by lateral link joint and the lower strut joint was not reduced due to presence of stress gradient and manufacturing limitations. Therefore, the area surrounded by strut mounting holes and hub mounting section was targeted to reduce the thickness.

To find the appropriate perimeter of the section to be removed, the rate of change of stress at that portion of the component was determined while the component was under primary loading condition. The distribution of stress at that portion for the original model is shown in Figure 7-6. The red areas in this figure that are basically the stress concentration points were found to have the highest stress and stress gradient sensitivity to thickness, thus a pattern was sought to keep the reduced thickness area away enough from these sensitive points. This pattern could be seen in Figure 7-7.

The depth of the removed section was selected as the design variable, while the constraints as mentioned in Section 7.2.2 were applied to the model. The design goal was to minimize the mass of the component. Figure 7-8 shows the variation of actual component mass, mass upper bound and mass lower bound with the optimization iterations. It should be noted that the mass value in this figure reflects the whole assembly including the moment arms. Figure 7-9 illustrates the stress limit variation in the optimized area versus number of iterations. After 15 optimization iterations, the stress limit in the optimized area was still only 40% of the stress at the critical location of the component (spindle 2nd step fillet). In this case, the possible detrimental effect of the forged surface in the optimization region is accounted for and the design is feasible, i.e. the stresses generated in the optimized area do

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5 This was a software limitation. Moment could not be applied directly to the spindle but should be in the form of load since in IDEAS a moment can not be applied to a geometrical point and it should be applied to a node. But the math programming redesign optimization task only accepts geometry based FEA models (that can not include any boundary condition on nodes or elements).
not exceed the limit. Figure 7-10 shows the depth of the removed area versus number of iterations.

The highest weight saving happens by removing the above mentioned section of the body from the component. Investigating the forging process applicable to this modification, it was found that the deeper a hole on a surface, the harder the forging process becomes. The web can also be punched out resulting in additional weight saving, however forging cost may increase since punching out would require additional process**. Therefore, a depth of 4 mm was selected following the Forger’s recommendation. The final optimized thickness of the body in the mass-reduced area is 8 mm, versus the original thickness of 16 mm. Figure 7-11 shows the filleted schematic of the optimized portion and Figure 7-12 provides the dimensions of the redesigned portion. Figure 7-13 shows the distribution of von Mises stress in the component before and after optimization for this part of Stage I for an arbitrary bending moment of 690 N.m. The stress distribution in the material-removed area becomes more uniform for the optimized component, indicating reduction of stress gradient in the section. It could also be seen that the level of stress in the optimized section increased after optimization, while the maximum stress and the critical locations of the component remained the same, showing the fact that in the optimized model the load is distributed more evenly in the component.

For other optimization potentials at this stage, FEA showed that the strut joints outer diameters and thicknesses were subjected to lower stress levels, and therefore could be reduced. However, this reduction’s contribution to weight loss was seen to be minor; therefore these dimensions were kept unchanged. The tension strut joint outer diameter and thickness were kept unchanged due to the likelihood of distortion of that portion of the

** Private communication with Joe Grosso of Citation Corporation.
component in manufacturing in addition to the attachment requirements of the tension strut to the steering knuckle.

The analysis showed that reduction of lateral link joint’s outer diameter and thickness, and the hub mounting plate thickness was possible. In addition, such changes would not affect the integrity of the component and are manufacturable. Following the mathematical redesign optimization of IDEAS, the outer diameter of the joint and the thickness of the mounting plate were selected as design parameters, separately, and the maximum stress at the critical location of the component was chosen as the stress limit. The optimized lateral link joint outer diameter and the thickness of the mounting plate were found to be 20.8 mm and 11.4 mm, versus the original 24 mm and 13 mm, respectively. Figure 7-14 shows the modified dimensions of the optimized part. In this redesign, the component’s weight became 2.13 kg earning 9.4% weight saving compared to the original model. Figure 7-15 shows the distribution of von Mises stress in the component before and after optimization for the Stage I optimization (including body, hub mounting and lateral link joint optimization) for an arbitrary bending moment of 690 N.m. The stress level, though higher in some areas within the material-removed sections, is still lower than that of the critical point. This optimization option does not result in additional forging cost††.

Another factor to be considered following the optimization is the stiffness of the optimized model compared to the original model. Comparing the displacement of the model before and after optimization, it was found that a 15% overall displacement increase (measured at the point of application of the load) occurs after optimization. But an important factor to be considered along with this comparison is the absolute value of the displacement. Taking standard load cases into account (e.g. ULSAS standard load case, AISI,

†† Private communication with Joe Grosso of Citation Corporation.
and assuming a primary load to be the vertical bump (see Section 5.3 for details), the
maximum amount of displacement of the spindle as a result of 1 kN.m moment in the
primary loading direction (applied in the form of concentrated force to the moment arms)
equals to less than 0.1 mm. Therefore, the stiffness does not affect the functionality of the
optimized component in service.

7.3.1.2 Modifications to Manufacturing Processes

In the framework of this study and for optimization Stage I of Figure 7-4 with no
change in attachment geometry, the approaches considered to reduce manufacturing costs
were precision forging instead of conventional forging, warm forging instead of hot forging,
and reducing machining steps.

Precision Forging versus Conventional Forging

The current practice in the forging of steering knuckle results in a considerable
percentage of the material being wasted as flash, in addition to generating non-precise parts
that require machining to get to the required tolerances at the interfaces with other
suspension parts. The weight of the as-forged steering knuckle is 3.1 kg, while the weight of
the finished steering knuckle is 2.4 kg. Therefore 0.7 kg (about 22%) of the material is
removed during post forging processes, mainly machining. Reducing the amount of
machining will also cut the component’s machining cost significantly.

Despite these advantages of precision forging, it should be noted that a major
advantage of closed-die forging with flash is that the volume of the preform can vary within
a wider range than for flashless forging. This makes it possible to continuously manufacture
products with the same quality. Precision forging requires a higher accuracy and a
significantly improved process monitoring and control. Moreover, the tools needed for
precision forging and the process conditions have to be monitored carefully because of the stricter product tolerances that need to be achieved. Thus a couple of process parameters such as forces, temperature, and process-times need to be monitored (Reinsch et al., 2003).

Considering these facts, a research program could be incited for the steering knuckle based on the studies on precision-flashless forging by Reinsch et al. (2003) and Vazquez and Altan (2000). Although these studies were conducted on forged connecting rods, due to similarities with steering knuckle from manufacturing point of view, they could be modified and extended for the steering knuckle of this study with the following objectives:

1. Design a tooling concept that can save material by allowing the formation of only a small amount of flash;
2. The volume of the initial preform and the volume of the cavity at the end of the process must be the same. The mass distribution and positioning of the preform must be very exact;
3. Optimize the tooling design by simulations. Find a geometry that requires the smallest load to fill the cavity without causing defects;
4. Establish guidelines and procedures to design blockers and preforms in order to accelerate the development of the production process of the forged part.

In order to fill the cavity and produce components, Vazquez and Altan (2000) concluded that the formation of a reasonable amount of flash should be allowed. A guideline to achieve this goal could be planned as:

1. Define a preform for the flashless forging of the steering knuckle by physical modeling experiments. An option would be plasticine billets and aluminum tooling;
2. Obtain the volume distribution in the steering knuckle;
3. Perform FE simulations of the component production to obtain the material flow and tool-billet contact conditions;

4. Determine the shape parameters to optimize the preform geometry;

5. Perform FE simulation with the optimized geometry;

6. Design the tool for forging of the component with controlled amount of flash by performing several simulations with different geometrical parameters (flash location and thickness) to optimize the geometry of the tooling.

   It should be noted that, in spite of the advantage of reduced machining, precision forging requires more complex dies that need to be replaced more often, and higher forging loads that may increase the production cost of the component.

**Warm Forging versus Hot Forging**

The current forging process of the steering knuckle of this study is performed hot. The idea of replacing hot forging with warm forging is based on the study by Aloi et al. (1997), where an experimental study of warm and hot forged microalloyed-steel-transmission hubs showed that the optimal combination of strength and toughness is achieved in a warm forged, fan-cooled condition.

Warm forming is an energy efficient process that allows for a part to be manufactured to a near net shape, in fewer operations, and with less material waste than in hot forming. Warm forging, as used in the study of Aloi et al. (1997), includes heating a blank to a controlled temperature between 760°C (1400°F) and 1040°C (1900°F), low temperatures in the austenite single-phase region, and forging into shape. The technology allows for near net shape forging of parts with intricate detail and much closer tolerances, as compared to hot forging. By nature, warm forging also results in a significant refinement of the austenite grain size.
Warm forging processes result in a more efficient utilization of material by reducing the amounts of both flash and scale which are formed. In one type of warm forging process, closed-die flashless forging, the tooling cavity completely surrounds the material, thereby eliminating all flash. The blank weight must be precisely controlled so that, in combination with warm forming, no flash trim is formed on the exterior of the forging envelope. Since no flash is generated, a trim press is not required in processing.

Associated reduced scale formation, inherent after forging at lower temperatures, not only reduces material losses, it also contributes to the near net shape of the final part. Additionally, lightly scaled parts would not require scale removal in the forge press or final shot cleaning operation. The lower forming temperature requires lower electrical consumption not only reducing the operational cost but also the capital cost of the induction heating power supply. The smaller power supply requires less floor space and a more compact forging line. Die life can actually be improved in lower temperature forging operations due to decreased thermal loading and reduced amounts of abrasive forging scale. In warm forging of microalloyed steels, improved toughness can be accomplished through refinement of the austenite grain size.

Warm forging does however, have limitations and requires specialized equipment due to the fact that the resultant forming pressures in this process are extremely high. For example, the forging pressure increases when decreasing the forging temperature. Since the press load capacities are higher than that required for hot forging a similar size part, the tooling must be able to withstand the higher stress levels imparted during forging. This places an overall limitation on the size of forging that can be formed at lower temperatures.

According to the study by Aloi et al. (1997), in forging of 1037 and 1040 microalloyed steels the room temperature toughness values were 32 to 42 Joules (24 to 31 ft-
lbs) higher for the two microalloyed steels after forging at 1010°C (1850°F) versus 1230°C (2250°F), while 0.2% offset yield strength was reduced by only 4 to 23 MPa (1 to 3 ksi). The hubs passed a fatigue test of the induction hardened splines, a push-out weld test, and a wide-open throttle durability test of the hub parts. Successful implementation of both hubs into production was also achieved, and included acceptable weldability, induction hardening response, and good productivity through comparable machinability.

The results obtained in the study of Aloi et al. (1997) demonstrate the advantages with regard to mechanical properties, processability, and overall cost of warm forged microalloyed steels for any number of applications. These benefits can be fully recognized in applications requiring a good combination of strength and toughness, including the steering knuckle of this study. On the other hand, since the material is forged at higher yield stress in warm forging, the process requires higher loads that reduce die life. Therefore, while substituting warm forging is considered as one of the options in manufacturing process modification of the steering knuckle, it is necessary to evaluate the cost impact of the option in a more detailed study.

**Eliminating or Limiting Some of the Manufacturing Steps**

Figure 7-16 shows the forged steering knuckle prior to shipping to the machining plant. Here as a modification, the strut and hub mounting holes are proposed to be extrusion-pierced during the forging process. In the modified process, starting with a slug or billet from a steel bar, the steering knuckle is produced in the first stage by pressing or hammering. Then at the second stage the hub and strut mounting holes are extrusion-pierced. The final stage for this part finish pierces the holes. Therefore, the forging process is modified and the machining work of the mounting holes could be reduced.
The changes in cost due to this process modification could be evaluated both qualitatively and quantitatively. Referring to the cost attributes (Table 7-1), by implementing these changes, the forging process costs attributed to tools and dies, and initial investment and setup increase. According to Jung (2002), the machining cost is mainly based on machining time that is composed of set-up time, operation time, and non-operation time. The current modification has minor influence on the tool and machine set-up time. The operation time is the duration laps from feed engagement to feed disengagement and is composed of the rough cutting time, the finish cutting time, and tool approach time. The main contribution of the current modification will be on the rough cutting time. The non-operation time, which is the lapse time after a workpiece is mounted except the operation time, has also minor effect within the current modification.

The rough cutting time is proportional to the machined volume while the finish cutting time is proportional to the area being machined. For the majority of machining operations, the main portion of machining time is consumed by rough machining (e.g. for the example part investigated by Jung (2002) the finish machining was estimated to be 15% of the rough machining time). Based on this concept, the reduction in machining time and subsequently machining cost could be related to the reduction in removed volume. In this modification of the steering knuckle, 8.9 cm³ less volume will be removed during the machining process (about 10% of the total material currently being removed) saving about 10% of machining and 5% of total component production time and, subsequently, cost. It should be noted that the changes in forging process need to be implemented only once, imposing a one-time investment, while the saving in machining costs is attributed to every component being manufactured. Therefore the modification proposed is justified.
7.3.2 Stage II – Limited Change in Attachment Geometry

Stage II of the optimization guideline (Figure 7-4) included geometry optimization with limited change in attachment geometry (i.e. modifying spindle geometry), substituting alternative materials with superior performance, and improving fatigue performance by additional manufacturing operations. This stage intended to obtain a smaller and, therefore, lighter spindle that reduces material and manufacturing costs.

7.3.2.1 Optimization Using Geometry Variables

For this optimization stage, the spindle of the optimized component from Stage I was redesigned to obtain higher fatigue performance and reduced weight. A schematic of this redesign compared to the original design is shown in Figure 7-17. In this model, the diameters of the first and third steps of the spindle were maintained and the second step, on which the bearing is installed in the current design, was reshaped to a tapered design. Instead of the current cylindrical-bore roller bearing, a spherical roller bearing with tapered bore (Figure 7-18) was selected. Due to removal of the second step fillet, which was the failure location in component tests, the stress concentration at this location decreased. In the original design, the maximum stress generated by the primary loading condition on the component occurred at the 2nd step fillet. The diameter of the spindle’s first step in this redesigned model was optimized in a way that the maximum stress generated by the primary loading in the component did not exceed the maximum stress generated in the original model with the same magnitude of load. In addition, the influence of spindle 1st step fillet radius was investigated.

Figure 7-19 shows the stress distribution caused by the primary loading and a typical moment magnitude of 690 N.m. In the redesigns with the fillet radii of 1.3 mm (the original
1st step radius) and 2.6 mm, the component’s weight became 2.07 kg and 2.03 kg, earning 2.8% and 4.7% weight savings compared to Stage I optimization, and 11.9% and 13.6% weight savings compared to the original model, respectively. Due to the reduction of the effect of stress concentration with increasing the fillet radius, the weight of the spindle is reduced. But this increase of the radius depends on the design requirements of the mounting hub and should be made compatible with the suspension system design. The dimensions of the spindle portion after the redesign with 1.3 mm spindle 1st step fillet radius (the original size) are shown in Figure 7-20. The forging cost difference between a tapered and a straight stem is minimal. There would be added cost in machining a tapered stem and in induction hardening the tapered stem after machining. Typically a tapered stem would be rough machined, hardened, then finish machined. Even though this could result in significant cost impact to the knuckle, it can still yield a net wheel end system cost save‡‡. All these factors should be taken into account at the detailed design stage of this plan. Figure 7-21 shows the model of Stage II optimization.

7.3.2.2  Modifications Using Alternative Materials

Alternative materials with superior fatigue properties were considered as another group of design variables within the Stage II optimization (Figure 7-4) and general optimization procedure of Figure 7-5. In this respect, the monotonic and cyclic properties of a number of steels were obtained (AISI bar steel fatigue database, undated; Johnson et al., 2000). In addition, material selection guidelines recommended by Bayer (2003) for automotive applications were considered. Since the objective of this material search was to find a material with superior fatigue properties, fatigue strengths of alternative materials were

‡‡ Private communication with Joe Grosso of Citation Corporation.
compared to that of the current SAE 11V37 used to manufacture the steering knuckle. Those materials superior in fatigue strength (Table 7-2 lists their mechanical properties) were investigated individually.

The effect of alloying elements on mechanical properties, service conditions and manufacturability (considering Table 7-3), strength, ductility, static strength at overloads, and excess deformation were evaluated and the proposed alternatives are listed in Table 7-4. It should be noted that microalloyed steels had higher priority since no heat treatment is required to achieve the desired properties such as hardness and strength. The non-microalloyed steels bring additional cost for heat treatment and unwanted distortion during this process may occur, resulting in the need for additional machining and are, therefore, not recommended. Materials with lower machinability are also not desirable due to cost impact on machining. For the particular case of 1117 steel, a microalloyed version of this material (i.e. 11V17), if produced and tested for mechanical properties, can be a potential alternative.

A key manufacturing parameter of interest is machinability of the selected materials. The machinability of carbon or alloy steels is influenced by their carbon or alloy content, microstructure, amount of soft or free machining particles, oxide inclusions, work-hardening rate and hardness. The AISI Bar Machinability Sub-committee is in the process of generating a database of automotive bar steel machinability under single point carbide turning conditions, including more than 30 steel grades (Joseph and Stout, 2004). Since this database and other machinability rating sources (e.g. ASM Handbook, Vol. 16) do not include most of the potential materials being studied in this work, hardness of the materials was considered as the criterion to evaluate and compare machinability of the alternative materials. It is shown that hardness has adverse effect on cutting speed and feed rate in machining (Figure 7-22); hence it could be considered as a reasonable criterion for machinability comparison.
The following microstructural and manufacturability considerations should also be noted. All of the candidate steels rely on two fundamental mechanisms to develop a proper microstructure and the desired properties. The first is to achieve a uniform fine grained austenite microstructure during forging, which transforms to a uniform fine grained room temperature microstructure during cooling after forging. The second is to precipitate a fine dispersion of vanadium particles (vanadium nitride, vanadium carbide, vanadium carbonitride) during the transformation of austenite to the room temperature microstructure. The forging and cooling practices required for each are functions of the composition. While all of the grades are forgeable in that parts can be formed, the required reheating temperatures, forging temperatures and cooling practices will be composition dependent. The MoVTi group of steels offers some advantages in that molybdenum suppresses unwanted high temperature precipitation of vanadium particles, and lowers austenite transformation temperatures. The net effect is a finer dispersion of vanadium particles in a finer room temperature microstructure resulting in better properties.

The properties of two of the steels, 1522MoVTi and 1522MoVTiS, are as hot-rolled properties. Caution should be exercised in comparing these properties to the other grades where the properties are given after a forging and cooling practice. The hot deformation and cooling conditions achieved during hot rolling in a steel mill may not be the same as those used during forging. As a result the properties could change. Also, the properties of the 15R30V grade are given after hot rolling followed by a high temperature heat treatment. If this heat treatment is required for a forged part, the cost of the part will increase significantly because of the additional heat treating step.§§

§§ Private communications with Dr. Tom Oakwood, formerly of Inland Steel Co.
Based on the results of the investigation presented in Table 7-4 and the aforementioned discussion, the candidate materials, i.e. SAE 15V24, 15R30V, 1522 MoVTi, 1522 MoVTiS, 1534 MoVTi and 1534 MoVTiSi, are proposed as alternatives. Limited weight saving can be achieved by replacing the potential alternative materials (0.4% to 2.1%), mainly due to geometrical constraints. If comprehensive changes to the geometry are allowed or for other components with fewer constraints, the weight saving will be more significant. Among these alternatives, 15V24 is the lowest cost material in terms of raw material cost, comparable in cost to the current 11V37. The group of MoVTi steels has significantly higher raw material cost. In particular, molybdenum is the high cost alloy element. More specific selection requires availability of cost data for the specific metal and performing manufacturability tests of the metals, both of which were out of the scope of this study.

7.3.2.3 Modifications to Manufacturing Processes

Within the framework of Stage II optimization of Figure 7-4 and optimization procedure of Figure 7-5, fatigue strength of the spindle could be improved, allowing reduced spindle size, and therefore weight and cost. Additional manufacturing operations such as surface hardening and shot peening to induce compressive residual stress are considered to achieve this purpose as discussed here.

Surface carburizing, nitriding and hardening could be applied to the fatigue-critical area (i.e. spindle fillet) of the subject steering knuckle for improvement of fatigue performance, and also improved wear and fretting resistance, since the steering knuckle is in contact with other chassis components. Carburizing process could be applied to steel to increase the carbon content of the surface layer. It leads to a higher fatigue resistance and a much improved wear resistance. Nitriding of low-alloy steels can significantly increase the
hardness of the surface layer by some precipitation phenomena which also improve the fatigue resistance. Nitriding also gives a volume increase and as a result residual compressive stresses in the surface layer.

Hardening of a surface layer is also done by induction hardening and thus without introducing extra carbon or nitrogen. Results in Figure 7-23 show improvements obtained for the spindle of a truck, which in service caused some fatigue problems at the smaller radius (3 mm). After induction hardening the S-N curve was raised considerably, and fatigue problems did not occur anymore. The etched cross section of the axle in Figure 7-23 reveals the depth of the induction hardened layer, which in this case is on the order of a few millimeters.

Compressive residual stresses improve fatigue resistance by acting as negative mean stress. For the component of this study, no shot peening is performed, but there is a possibility of compressive residual stress generation due to shot cleaning that is performed after forging. In addition, residual stresses may exist due to the forging process. Since measuring the residual stress generated due to processing on the component was not within the scope of this study, no residual stress effect due to processing is considered.

To achieve higher fatigue performance, the major location that deserves inducing local compressive residual stress is the spindle fillets, as shown in Figure 7-24. Residual stresses can be induced by mechanical methods, thermal methods, and machining. Among the most widely used mechanical processes for producing beneficial compressive surface residual stresses for enhancing long and intermediate fatigue lives are surface rolling, shot peening and shot blasting. Shot peening and shot blasting are not applicable to the case of forged steel steering knuckle since the spindle surface is machined after forging. On the other hand, surface rolling could be applied to the spindle steps. In Figure 7-25 a hardened
rolling wheel is pressed in the notch of a rotating-bending specimen. As a result, the plastic deformation occurs in the surface layer of the notch root. As shown by the graph, the fatigue limit under rotating-bending was significantly increased, up to more than twice the original value.

The modifications proposed in this section should be looked into at the detailed design stage, considering the issues of manufacturability and cost. The weight saving due to each modification should be compared with the manufacturing limitations and cost impact of the technique. In addition, the dimensions of the wheel end system should be taken into consideration, where design constraints may limit dimensional changes.

### 7.3.3 Discussions

To summarize the results from the optimization analysis, Table 7-5 provides a comparative list of component’s original weight and the weight reductions of each optimization stage. Overall weight and cost reductions of at least 12% and 5%, respectively, are estimated for the manufacturing process. The cost of the saved material is additional reduction, though not very considerable due to small portion of material cost within the total production cost.

The optimization results showed somewhat limited changes for this particular component. In this regard, a few points should be clarified. First, this component is relatively small, compared to steering knuckles with similar or relatively similar service conditions. Second, it has many attachment compatibility constraints. A more comprehensive change on constraints can result in drastic and unwanted design alterations in suspension system. For instance, an optimization plan could be proposed as an interference-fit assembly to make the component in two parts namely body and spindle, in which the
spindle fits in the body. This plan requires a number of major changes in suspension system design including the geometry of strut, the location of lateral link joint, the position of hub mounting holes, and in general the overall geometry of the component. Furthermore, some manufacturing limitations make the optimization options limited. For instance, reducing the thickness of the body close to the tension strut connection hole results in distortion of the component during the forging process. Also, the forging process becomes more complicated as the depth of the removed part from a surface increases. The web can ultimately be punched out resulting in additional weight saving, however forging cost may increase since punching out would require additional process. In addition, material can not be removed from the area surrounded by lateral link joint and the lower strut joint, again due to limitations in the manufacturing process. All these confine the optimization options to what was discussed for Stage I (Section 7.3.1). Modifications to spindle, as considered in Stage II (Section 7.3.2), are also restricted because of numerous changes that will be imposed on the wheel end system. The techniques proposed in this stage to improve fatigue strength are also pending a detailed design study to investigate weight saving versus cost impact of the particular technique and the limitations of the wheel end system design.

It should be emphasized that, following the general goal of the present study as mentioned in Chapter 1, this steering knuckle was used as an example part in the optimization study to investigate a methodology for optimizing a forged automotive component with durability constraints for weight and cost reductions. Therefore, the emphasis in this study was more on the optimization process, rather than optimization of the particular component used. In spite of the limited optimization achieved for this particular component, the approach that was followed is applicable to other forged
components. Components with fewer geometrical restrictions have much higher potential for weight reduction and cost savings.
Table 7-1  Cost attributes for manufacturing the steering knuckle.

<table>
<thead>
<tr>
<th>Cost Attributes</th>
<th>Engineering Parameters</th>
<th>General Production Parameters</th>
<th>Forging Process Parameters</th>
<th>Machining Process Parameters</th>
<th>Lubrication Parameters</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>Program Life</td>
<td>Annual Prod. Volume</td>
<td>Material Choice</td>
<td>Physical Volume</td>
</tr>
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<td>Forging Process</td>
<td>Variable</td>
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<td>-</td>
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<td>Raw Material Costs</td>
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<td>Labor</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td></td>
<td>Tools and Dies</td>
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<td>-</td>
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<td>-</td>
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<td>Fixed</td>
<td>Initial Investment</td>
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<td>✓</td>
<td>✓</td>
<td>✓</td>
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<td></td>
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<td>Setup</td>
<td>-</td>
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<tr>
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<td>Overhead</td>
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<td>✓</td>
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<td></td>
<td>Inspection*</td>
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<tr>
<td>Machining</td>
<td>Variable</td>
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</tr>
<tr>
<td>Process</td>
<td>Labor</td>
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<td>Coolant</td>
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<td>Tools</td>
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<tr>
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<td>Initial Investment</td>
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<tr>
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<td>Overhead</td>
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<tr>
<td></td>
<td>Inspection*</td>
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</tbody>
</table>

✓ : Engineering parameters affecting cost attributes.
* The inspection costs depend on customer requirements and component design criteria, not the production parameters.
Table 7-2  Preliminary list of alternative materials with superior fatigue strength, as compared to the currently used alloy steel (SAE 11V37).

<table>
<thead>
<tr>
<th>SAE Grade</th>
<th>Process</th>
<th>$S_f$ @10$^6$ Cycles (MPa)</th>
<th>$S_y$ (MPa)</th>
<th>$S_u$ (MPa)</th>
<th>$%R/A$</th>
<th>$%EL$</th>
<th>BHN</th>
<th>Ref.</th>
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<tbody>
<tr>
<td>11V37</td>
<td>Specimen from forged steering knuckle</td>
<td>352</td>
<td>556</td>
<td>821</td>
<td>37</td>
<td>21</td>
<td>240</td>
<td>This study</td>
</tr>
<tr>
<td>10B21</td>
<td>Quenched and Tempered</td>
<td>537</td>
<td>1062</td>
<td>1105</td>
<td>70.5</td>
<td>33.4</td>
<td>322</td>
<td>AISI Database</td>
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<tr>
<td>1117</td>
<td>Quenched and Tempered Simulated Carburized Core</td>
<td>386</td>
<td>655</td>
<td>777</td>
<td>55.0</td>
<td>37.0</td>
<td>193</td>
<td>AISI Database</td>
</tr>
<tr>
<td>15B35</td>
<td>Quenched and Tempered Simulated Carburized Core</td>
<td>468</td>
<td>866</td>
<td>940</td>
<td>64.5</td>
<td>37.3</td>
<td>286</td>
<td>AISI Database</td>
</tr>
<tr>
<td>15V24</td>
<td>Hot Formed, Cont. Cooled Specimens From Control Arm</td>
<td>379</td>
<td>646</td>
<td>878</td>
<td>61.1</td>
<td>36.0</td>
<td>243</td>
<td>AISI Database</td>
</tr>
<tr>
<td>15R30V</td>
<td>Hot rolled at 1260°C, Austenitized at 1150°C for 1 hr, Air cool</td>
<td>404</td>
<td>593</td>
<td>861</td>
<td>-</td>
<td>17.7</td>
<td>255</td>
<td>Johnson et al. (2000)</td>
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<tr>
<td>1522MoVTi</td>
<td>As hot rolled</td>
<td>428</td>
<td>600</td>
<td>852</td>
<td>-</td>
<td>21.5</td>
<td>267</td>
<td>Johnson et al. (2000)</td>
</tr>
<tr>
<td>1522MoVTiS</td>
<td>As hot rolled</td>
<td>438</td>
<td>598</td>
<td>950</td>
<td>-</td>
<td>23</td>
<td>248</td>
<td>Johnson et al. (2000)</td>
</tr>
<tr>
<td>1534 MoVTi</td>
<td>Forced air cooled after forging</td>
<td>443</td>
<td>624</td>
<td>904</td>
<td>-</td>
<td>17.6</td>
<td>272</td>
<td>Johnson et al. (2000)</td>
</tr>
<tr>
<td>1534MoVTiS</td>
<td>Forced air cooled after forging</td>
<td>483</td>
<td>558</td>
<td>1021</td>
<td>-</td>
<td>24.6</td>
<td>272</td>
<td>Johnson et al. (2000)</td>
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<tr>
<td>4130 Al</td>
<td>Quenched and Tempered</td>
<td>528</td>
<td>1284</td>
<td>1483</td>
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<td>28.0</td>
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<tr>
<td>4140</td>
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<td>4320</td>
<td>Quenched and Tempered Simulated Carburized Core</td>
<td>547</td>
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<td>4620</td>
<td>Thru Carburized Simulated Case</td>
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<td>5120</td>
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<td>472</td>
<td>780</td>
<td>1008</td>
<td>58.0</td>
<td>46.0</td>
<td>252</td>
<td>AISI Database</td>
</tr>
<tr>
<td>5140</td>
<td>Quenched and Tempered</td>
<td>497</td>
<td>957</td>
<td>1039</td>
<td>52.7</td>
<td>28.3</td>
<td>305</td>
<td>AISI Database</td>
</tr>
<tr>
<td>51860</td>
<td>Quenched and Tempered</td>
<td>771</td>
<td>1830</td>
<td>1970</td>
<td>21.6</td>
<td>36.2</td>
<td>450</td>
<td>AISI Database</td>
</tr>
<tr>
<td>8620</td>
<td>Thru Carburized Simulated Case</td>
<td>458</td>
<td>920</td>
<td>1202</td>
<td>1.0</td>
<td>1.0</td>
<td>583</td>
<td>AISI Database</td>
</tr>
<tr>
<td>8695 LAB HEAT</td>
<td>Lab. Melted Simulated Carburized Case w/o IGO</td>
<td>401</td>
<td>780</td>
<td>1496</td>
<td>1.3</td>
<td>1.3</td>
<td>539</td>
<td>AISI Database</td>
</tr>
<tr>
<td>8822</td>
<td>Quenched and Tempered Simulated Carburized Core</td>
<td>442</td>
<td>918</td>
<td>1131</td>
<td>56.0</td>
<td>40.0</td>
<td>252</td>
<td>AISI Database</td>
</tr>
<tr>
<td>9254 Al</td>
<td>Quenched and Tempered</td>
<td>845</td>
<td>2270</td>
<td>2950</td>
<td>4.0</td>
<td>3.9</td>
<td>584</td>
<td>AISI Database</td>
</tr>
<tr>
<td>9254V (MA)</td>
<td>Quenched and Tempered</td>
<td>710</td>
<td>1870</td>
<td>2050</td>
<td>35.1</td>
<td>16.2</td>
<td>536</td>
<td>AISI Database</td>
</tr>
<tr>
<td>9310</td>
<td>Quenched and Tempered Simulated Carburized Core</td>
<td>479</td>
<td>804</td>
<td>902</td>
<td>70.8</td>
<td>31.0</td>
<td>240</td>
<td>AISI Database</td>
</tr>
</tbody>
</table>
Table 7-3 Effect of common alloying elements in automotive steels (Bayer, 2003).

<table>
<thead>
<tr>
<th>Element</th>
<th>Effects</th>
</tr>
</thead>
<tbody>
<tr>
<td>Carbon</td>
<td>The principal strengthening element in steel. It can have a great effect on numerous metallurgical properties. Increased carbon levels can provide increased hardness and strength. Lower carbon (less than 0.25%) improves weldability, ductility, and toughness at the expense of strength.</td>
</tr>
<tr>
<td>Manganese</td>
<td>Increases strength and toughness. Manganese has one of the strongest elemental effects on steel’s hardenability (the ability of steel to harden at a depth from the surface through quenching). Higher levels have a negative effect on weldability.</td>
</tr>
<tr>
<td>Sulfur</td>
<td>Considered an impurity, except when intentionally added to improve machinability. It combines with manganese to produce manganese sulfide (MnS) inclusions, which assist as “chip breakers” in machining steels. Higher sulfur levels have a detrimental effect on impact resistance.</td>
</tr>
<tr>
<td>Silicon</td>
<td>A deoxidizer, it is added to steel to tie up free oxygen. The term “killed steel” is used when it is deoxidized, thus providing improved internal soundness and surface quality. Higher levels slightly increase hardenability; however, silicon can have a negative impact on machinability.</td>
</tr>
<tr>
<td>Nickel</td>
<td>When combined with other alloying elements, it produces steels with excellent strength and low-temperature toughness in the quenched and tempered condition.</td>
</tr>
<tr>
<td>Chromium</td>
<td>Provides wear resistance, hardenability, and low temperature toughness. At high levels, it provides corrosion and oxidation resistance, and assists in maintaining strength levels at elevated temperatures.</td>
</tr>
<tr>
<td>Molybdenum</td>
<td>Has a strong effect on hardenability (similar to manganese). Molybdenum also increases strength at elevated temperatures.</td>
</tr>
<tr>
<td>Aluminum</td>
<td>Acts as a deoxidizer and helps control grain size. It can have a negative impact on machinability.</td>
</tr>
<tr>
<td>Niobium</td>
<td>Helps produce fine grain steel, and improves the strength of microalloyed steels.</td>
</tr>
<tr>
<td>Vanadium</td>
<td>Also helps produce fine grain steel. Additionally, it can be used to increase strength, impact toughness, and hardenability. Vanadium can have a negative effect on machinability.</td>
</tr>
<tr>
<td>Titanium</td>
<td>Primarily a deoxidizer and nitrogen scavenger in the making of boron steels. Also acts as a grain refiner. Titanium can have a detrimental effect on machinability.</td>
</tr>
<tr>
<td>Boron</td>
<td>Increases hardenability in steel with less than 0.8% carbon, replacing other alloying elements.</td>
</tr>
</tbody>
</table>
Table 7-4  Potential alternative materials to replace the current 11V37 steel. All the steels are in the form of bars.

<table>
<thead>
<tr>
<th>Material</th>
<th>Process</th>
<th>Microstructure</th>
<th>Application</th>
<th>Advantage(s)</th>
<th>Disadvantage(s)</th>
</tr>
</thead>
<tbody>
<tr>
<td>15V24</td>
<td>hot forged and control-cooled vanadium-</td>
<td>ferrite-pearlite</td>
<td>control arms</td>
<td>• It has superior strength, ductility and fatigue properties</td>
<td></td>
</tr>
<tr>
<td></td>
<td>strengthened carbon steel</td>
<td></td>
<td></td>
<td>• Its machinability is comparable to the current material</td>
<td></td>
</tr>
<tr>
<td>15R30V</td>
<td>hot forged and control-cooled</td>
<td>coarse ferrite-pearlite</td>
<td>automotive/aerospace</td>
<td>• It has superior strength and fatigue properties</td>
<td>Its ductility is slightly lower</td>
</tr>
<tr>
<td></td>
<td>vanadium-strengthened steel with</td>
<td></td>
<td></td>
<td>• Its hardness is close to the current material</td>
<td></td>
</tr>
<tr>
<td></td>
<td>hardening characteristics.</td>
<td></td>
<td></td>
<td>• It is a microalloyed steel so the produced part will not need heat treatment</td>
<td></td>
</tr>
<tr>
<td>1522</td>
<td>hot forged and control-cooled</td>
<td>acicular ferrite and bainitic along with</td>
<td>automotive/aerospace</td>
<td>• It has superior strength and fatigue properties</td>
<td>Its hardness is slightly higher (less machinable)</td>
</tr>
<tr>
<td>MoVTi</td>
<td>nontraditional steel with softening</td>
<td>conventional ferrite and pearlite</td>
<td></td>
<td>• It is a microalloyed steel so the produced part will not need heat treatment</td>
<td>Titanium generates forgeability issues</td>
</tr>
<tr>
<td></td>
<td>characteristics</td>
<td></td>
<td></td>
<td>• Titanium particles aid in refining austenite grain size during forging</td>
<td>It has higher raw material cost</td>
</tr>
<tr>
<td>1522</td>
<td>hot forged and control-cooled</td>
<td>acicular ferrite and bainitic along with</td>
<td>automotive/aerospace</td>
<td>• It has superior strength, ductility and fatigue properties</td>
<td></td>
</tr>
<tr>
<td>MoVTiS</td>
<td>high sulfur (0.05 weight percent) steel with</td>
<td>conventional ferrite and pearlite</td>
<td></td>
<td>• Its hardness is almost equal to the current material (equivalent machinability)</td>
<td></td>
</tr>
<tr>
<td></td>
<td>softening characteristic</td>
<td></td>
<td></td>
<td>• It is a microalloyed steel so the produced part will not need heat treatment</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td>• Titanium particles aid in refining austenite grain size during forging</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td>• Titanium generates forgeability issues</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td>• It has higher raw material cost</td>
<td></td>
</tr>
<tr>
<td>1534</td>
<td>hot forged and control-cooled</td>
<td>acicular ferrite and bainitic along with</td>
<td>automotive/aerospace</td>
<td>• It has superior strength and fatigue properties</td>
<td></td>
</tr>
<tr>
<td>MoVTi</td>
<td>low silicon/0.35 weight percent carbon steel</td>
<td>conventional ferrite and pearlite</td>
<td></td>
<td>• It is a microalloyed steel so the produced part will not need heat treatment</td>
<td></td>
</tr>
<tr>
<td></td>
<td>with softening characteristic</td>
<td></td>
<td></td>
<td>• Titanium particles aid in refining austenite grain size during forging</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td>• Its hardness is slightly higher (less machinable) and ductility slightly lower</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td>• Titanium generates forgeability issues</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td>• It has higher raw material cost</td>
<td></td>
</tr>
<tr>
<td>1534</td>
<td>hot forged and control-cooled</td>
<td>acicular ferrite and bainitic along with</td>
<td>automotive/aerospace</td>
<td>• It has superior strength, ductility and fatigue properties</td>
<td></td>
</tr>
<tr>
<td>MoVTiSi</td>
<td>high silicon/0.35 weight percent carbon steel</td>
<td>conventional ferrite and pearlite</td>
<td></td>
<td>• It is a microalloyed steel so the produced part will not need heat treatment</td>
<td></td>
</tr>
<tr>
<td></td>
<td>with softening characteristic</td>
<td></td>
<td></td>
<td>• Titanium particles aid in refining austenite grain size during forging</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td>• It has softening characteristics</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td>• Its hardness is slightly higher (less machinable)</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td>• Titanium generates forgeability issues</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td>• It has higher raw material cost</td>
<td></td>
</tr>
</tbody>
</table>
### Table 7-5 Summary of the results of optimization stages.

<table>
<thead>
<tr>
<th>Geometry Change</th>
<th>Process Change</th>
<th>Component Weight (kg)</th>
<th>Weight Reduction (%)</th>
<th>Cost Reduction (%)</th>
<th>Remarks</th>
</tr>
</thead>
<tbody>
<tr>
<td>Original</td>
<td></td>
<td>2.35</td>
<td>-</td>
<td></td>
<td></td>
</tr>
<tr>
<td><strong>Stage I</strong></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>Material removed from body</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>Hub mounting thickness optimized</td>
<td>2.13</td>
<td>9.4</td>
<td>5% + material saving</td>
<td></td>
</tr>
<tr>
<td></td>
<td>Lateral link joint thickness optimized</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td><strong>Stage II</strong></td>
<td>Spindle redesigned</td>
<td>2.07 or less***</td>
<td>11.9 and up</td>
<td>5% + material saving</td>
<td></td>
</tr>
</tbody>
</table>

1. Precision forging and warm forging are other proposed process changes for Stage I. These options should be more extensively evaluated in a detailed design stage for their cost impact.

2. Surface enhancement (carburizing, nitriding, hardening) and inducing compressive residual stress at surface by surface rolling are proposed additional processes that will reduce component weight by increasing fatigue strength in Stage II. These options should be more extensively evaluated in a detailed design stage for their cost impact.

*** Depending on spindle fillet radius.
Figure 7-1   Manufacturing process flow chart for the steering knuckle.
Figure 7-2  Locating the forged steering knuckle on the V-block to turn spindle.

Figure 7-3  Unmachined forging loaded on the fixtures to be milled. The steering knuckle shown in this picture is slightly different from the one being studied here.
Figure 7-4  Steering knuckle optimization stages followed in this study.
Figure 7-5  Steering knuckle optimization procedure.
Figure 7-6  Stress distribution under primary loading at the vicinity of top strut attachment (top left), bottom strut attachment (top right), hub mounting part (bottom left) and the upper curve of steering knuckle body.

Figure 7-7  Pattern of the section to reduce the thickness in Stage I optimization.
Figure 7-8  Actual design goal (solid line), the design goals upper bound (dashed line) and the design goals lower bound (dotted line) versus number of iterations. The mass includes the moment arms and the unit is kg.

Figure 7-9  Stress limit in the optimized area versus number of iterations. The stress unit is MPa.
Figure 7-10  Design variable (depth of the removed area) versus number of iterations. The design parameter unit is mm.

Figure 7-11  Optimized section in Stage I optimization.
Figure 7-12  Dimensions of the redesigned portion of the steering knuckle’s body in Stage I optimization. All dimensions are in mm.

Figure 7-13  von Mises stress distribution for an arbitrary moment of 690 N.m before (top) and after optimizing for the removed section from the body (first part of Section I optimization). The stress unit is MPa.
Figure 7-14  The original (left) and optimized dimensions of hub mounting and lateral link in Stage I optimization. The dimensions are in mm.

Figure 7-15  von Mises stress distribution for an arbitrary moment of 690 N.m before (top) and after optimizing for Stage I for the optimized area. The stress unit is MPa.
Figure 7-16  The forged part showing the strut and hub mounting holes.

Figure 7-17  Stage I (left) and Stage II models with redesigned spindle.

Figure 7-18  Spherical roller bearing with tapered bore to be installed on the tapered section of the redesigned spindle (SKF Catalog).
Figure 7-19  von Mises stress distribution on the spindle-redesigned steering knuckle due to an arbitrary moment of 690 N.m. The stress unit is MPa.

Figure 7-20  Dimensions of the original (top) and redesigned-spindle (bottom) steering knuckles. The dimensions are in mm.
Figure 7-21  Front (left) and isometric views of Stage II optimization.

Figure 7-22  Cutting speed and feed rate (both indexes of machinability) versus hardness of some steels (Trent and Wright, 2000).
Figure 7-23  Results of fatigue tests ($R = -0.6$) for induction hardened and non-hardened truck stub axle (top) and depth of induction hardened zone revealed by etching (bottom) (Schijve, 2001).
Figure 7-24  Spindle fillets as the highest stressed areas of the steering knuckle are recommended for inducing compressive residual stress.

Figure 7-25  Effect of rolling of the notch root on fatigue behavior under rotating-bending fatigue of 37CrS4 steel (Kloos et al., 1987).
Chapter Eight

8 Summary and Conclusions

The effects of manufacturing process on fatigue design and optimization of automotive components using experimental, numerical and analytical tools were investigated. Even though the methodologies developed apply to a wide range of automotive and other components, vehicle steering knuckles made of forged steel, cast aluminum, and cast iron were selected as example parts for this study.

An extensive literature review was performed to compare manufacturing processes of automotive components focusing on mechanical behavior, identify methods used in durability assessment and optimization aspects. Material monotonic and fatigue properties were obtained and compared through strain-controlled tests on forged steel, cast aluminum, and cast iron specimens. Linear and nonlinear finite element analyses were employed to obtain three dimensional stress and strain distributions in the components to facilitate fatigue life predictions and optimization. Component tests were conducted to compare fatigue behavior of the components made from different manufacturing processes and to verify analytical evaluations and life predictions.

A general procedure for durability design of automotive components was developed and pursued for the example parts. The strengths and shortages of various approaches including nominal stress, local stress and local strain approaches were investigated. An optimization procedure for a forged automotive component was also developed considering manufacturing, material, and cost aspects, using the forged steel steering knuckle as the
example part. The manufacturing process details and production cost parameters for the example part were discussed, the optimization problem was defined and the results were analyzed. The findings of this study are summarized below.

**Material Fatigue Behavior and Comparisons**

1. From tensile tests and monotonic deformation curves it is concluded that forged steel is considerably stronger and more ductile than cast aluminum and cast iron. Cast aluminum and cast iron reached 37% and 57% of forged steel ultimate tensile strength, respectively. The yield strength of cast aluminum and cast iron is also lower, 42% and 54% of the forged steel, respectively. The percent elongation, as a measure of ductility, of cast aluminum and cast iron were found to be 24% and 48% of the forged steel, respectively.

2. From strain-controlled cyclic tests it is concluded that the cyclic deformation curve of the forged steel is independent of the geometrical direction (i.e. isotropic behavior). For the fatigue behavior, however, some degree of anisotropy was observed. Both the long-life as well as the short-life fatigue of forged steel were observed to be longer (by about a factor of two) in the direction coinciding with the primary stressing direction of the forged steering knuckle.

3. The cyclic yield strength of cast aluminum and cast iron were found to be 54% and 75% of forged steel, respectively. The cyclic strain hardening exponent of cast aluminum and cast iron was 46% and 55% of the forged steel, respectively. These indicate the higher cyclic strength of forged steel against yielding, and its higher resistance to plastic deformation.

4. Significantly better S-N fatigue resistance of the forged steel was observed, as compared with the two cast materials. Comparison of long-life fatigue strength (defined as the
fatigue strength at $10^6$ cycles) shows that the fatigue limit of cast aluminum and cast iron are only 35% and 72% of the forged steel, respectively. In addition, while the fatigue strength of forged steel at $10^6$ cycles is expected to remain about constant at longer lives, fatigue strength of the two cast materials is expected to continuously drop with longer lives.

5. Forged steel was found to be superior to cast aluminum and cast iron with respect to low cyclic fatigue (i.e. cyclic ductility). In automotive design, cyclic ductility can be a major concern when designing components subjected to occasional overloads, particularly for notched components, where significant local plastic deformation can occur.

6. Comparisons of strain-life fatigue behavior of the three materials demonstrate the superiority of the forged steel over cast aluminum and cast iron. The forged steel provides about a factor of 5 longer lives in the short-life regime, compared to the cast aluminum and cast iron. In the high-cycle regime, forged steel results in about an order of magnitude longer life than the cast iron, and about a factor of 3 longer life, compared to the cast aluminum.

7. Neuber stress versus life plot, which considers the combined effects of stress and strain amplitudes, shows forged steel to have about two orders of magnitude longer life than cast iron and four orders of magnitude longer life than cast aluminum.

**Finite Element Analysis**

8. In order to avoid a complex meshed model that increases the FEA run-time, a relatively coarse global mesh size, and a finer mesh at the vicinity of the critical points using free local meshing feature was selected for each component. This procedure increased the
computational efficiency of the model significantly, particularly for nonlinear models where material deformation was elastic-plastic.

9. Even at the lower loading level, which can be considered as an indication of long-life service of the components, the material undergoes local plastic deformation. This is evidence that mere use of linear elastic FEA is not sufficient for reliable fatigue life predictions.

10. The spindle 1st step fillet area for the forged steel and hub bolt hole for the cast aluminum and cast iron steering knuckles were found to be high-stressed locations with high stress gradient. Both stress concentration as well as stress gradient due to the mode of loading applied (i.e. bending in this case) are major factors in making an area fatigue-critical location.

11. Although the primary loading on the components is unidirectional, it is shown that the stress and strain at the critical locations are multiaxial. The type of primary loading that the components undergo generates proportional stresses throughout the components. For proportional stressing, von Mises stress and strain have been found effective in calculating the equivalent values as a result of multiaxiality, and were used for fatigue life analyses.

12. At the critical location the state of plane strain prevails for the forged steel steering knuckle, while the state of stress at the critical location of the cast aluminum and cast iron steering knuckles is closer to plane stress. Knowledge of the state of stress and strain at the critical location of the components helps in choosing the appropriate deformation model, leading to more accurate fatigue life predictions.

13. FEA simulation for cyclic loading is important for fatigue analysis since cyclic deformation material response can be vastly different from monotonic deformation
response. In addition, the local and nominal behaviors are generally different under various loading conditions. For example, as the nominal stress R-ratio remains almost constant (close to zero), significant negative stress R-ratio is observed for most of the simulations as a result of the residual stress generated at the stress concentrations due to local plastic deformation.

**Component Fatigue Behavior and Comparisons**

14. Strain gages were used to validate the readings with analytical calculations. The differences between experimentally measured and FEA-predicted strains obtained for the forged steel and cast aluminum steering knuckles were found to be reasonable for the complex geometries considered.

15. Based on the component testing observations, crack growth life was found to be a significant portion of the cast aluminum steering knuckle fatigue life (on the average, about 50% of the cast aluminum steering knuckle life is spent on macro-crack growth), while crack growth life was an insignificant portion of the forged steel steering knuckle fatigue life.

16. Component testing results showed the forged steel steering knuckle to have about two orders of magnitude longer life than the cast aluminum steering knuckle, for the same stress amplitude level. This occurred at both short as well as long lives. Comparison of the strain-life prediction curves of the components demonstrated that the forged steel steering knuckle offers more than an order of magnitude longer life than the cast iron steering knuckle.

17. The failed forged steel steering knuckle had a typical ductile material fatigue failure surface including crack initiation, smooth crack growth and rough fracture sections. The failed cast aluminum had a relatively longer crack growth portion as compared to the
crack growth portion of the forged steel steering knuckle. The failure locations in the component tests agreed with FEA predictions.

**Fatigue Life Predictions**

18. The nominal stress approach cannot be used for complex component geometries, such as the cast aluminum steering knuckle in this study due to the fact that for complex geometries, nominal stress can not be defined explicitly. For the forged steel steering knuckle, the predictions of the nominal S-N approach were conservative, by about a factor of seven on fatigue life, as compared to the experimental results.

19. The local stress or strain approaches in conjunction with the FEA results were found to provide better life predictions, as compared with the commonly used nominal S-N approach. This is partly due to the fact that the local approaches directly account for the residual stresses from local plastic deformation.

20. The local strain approach using nominal stresses for the forged steel knuckle in conjunction with Neuber’s rule predicted conservative lives, by about an order of magnitude, as compared with experimental results. This confirms the suggestion that Neuber’s rule is more applicable to plane stress states, since plane strain state existed at the fatigue-critical location of the forged steel knuckle.

21. Life predictions based on local approaches using linear elastic FEA results in conjunction with Neuber-corrected stresses were found to be close to those obtained based on nonlinear elastic-plastic FEA results. Therefore, the simpler and less time consuming linear elastic FEA, when modified to correct for plastic deformation, is an effective and capable approach for life prediction of components with complex geometries and/or loadings.
22. For the local stress approach, Gerber’s mean stress parameter provides better predicted fatigue lives, as compared with the experimental lives, than the commonly used modified Goodman equation. For the local strain approach, Morrow’s mean stress parameter provides better predicted fatigue lives than the Smith-Watson-Topper mean stress parameter.

**Optimization**

23. Manufacturing process considerations, material and cost parameters are major constituents of a general optimization procedure with durability constraints for automotive component. A geometrical optimization without these considerations is not a practical approach for such high volume components.

24. The proposed material alternatives provide higher fatigue strength for the component. Manufacturability and cost are two other main issues that are critical to the final selection of the replacing material(s). Limited weight saving is achieved by replacing the potential alternative materials, mainly due to geometrical constraints. If comprehensive changes to the geometry are allowed or for other components with fewer constraints, the weight saving will be more significant.

25. Additional manufacturing operations such as surface hardening and surface rolling to induce compressive residual stress can be considered to improve fatigue strength of the forged steel steering knuckle at the spindle fillet area.

26. Overall weight and cost reductions of at least 12% and 5%, respectively, are estimated for the example part following the optimization task. The cost of the saved material is additional reduction, though not very considerable due to small portion of material cost within the total production cost. Due to the small size of the forged steel steering knuckle and many attachment compatibility constraints, limited changes could be
implemented during the optimization process. More comprehensive changes require a more detailed design of the component and the suspension system.

27. The approach that was followed is applicable to other forged components. Components with fewer geometrical restrictions than the steering knuckle considered have much higher potential for weight reduction and cost savings.
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Houshito, S., Watanabe, Y., Goka, M., and Ishihara, Y., 1989, “Feasibility Study on the 
Application of High Strength Ductile Iron to Automotive Chassis Parts, *International Journal of 


Substitution Process by Powder Metallurgy in Automobile Parts,” *Journal of Materials Processing 


Appendix A - Summary of Literature Review for Fatigue Life Prediction of Automotive Components

This appendix summarizes a number of relatively recent studies related to fatigue life prediction of automotive components. The procedure followed in each study is briefly discussed and significant points are mentioned. The articles are arranged in the order of years published in order to have a sense of the progress in the field.
Author(s): Blarasin and Farsetti (1989)

Article Title: A procedure for the rational choice of microalloyed steels for automotive hot-forged components subjected to fatigue loads

Component(s): Steering knuckle of the front suspension of a commercial vehicle

Stress-Strain Calculation: Nominal stress history (corrugated proving ground) + Neuber’s rule

Life Prediction Method(s): Strain-life

Experiment Verification: Variable amplitude uniaxial component test

Author’s Remarks:
1. Overestimated predictions for Q&T-steel components; attributed to underestimate of the stress concentration as well as to differences between the microstructure of the component and the fatigue specimens.
2. Good agreement between calculated and experimental lives for microalloyed-steel components.
<table>
<thead>
<tr>
<th>Author(s):</th>
<th>Conle and Mousseau (1991)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Article Title:</td>
<td>Using vehicle dynamics simulations and finite-element results to generate fatigue life contours for chassis components</td>
</tr>
<tr>
<td>Component(s):</td>
<td>Upper control arm of an automotive front suspension</td>
</tr>
<tr>
<td>Stress-Strain Calculation:</td>
<td>Elastic FEA + Neuber plasticity correction</td>
</tr>
<tr>
<td>Life Prediction Method(s):</td>
<td>Critical plane</td>
</tr>
<tr>
<td>Experiment Verification:</td>
<td>None</td>
</tr>
</tbody>
</table>
| Author’s Remarks: | 1. The method finds the critical angle for fatigue calculations of each element for all load cases, not just the critical ones.  
2. The vehicle subsystem models (e.g. bushings) need to be correlated with improved laboratory tests and real service load behavior.  
3. Other methods for multiaxial to uniaxial equivalencing should be investigated.  
4. Verification with actual experimental results at each stage of analysis needs to be done. |
| Prediction Procedure: | 1. Forces generated on the wheels are recorded on proving ground.  
2. DADS\textsuperscript{10} model computes the upper control arm forces.  
3. Elastic FEA calculates loads on each element.  
4. Critical fatigue activity plane is determined by an elastic multiaxial superposition routine.  
5. Stresses normal to this maximum principal stress plane are rainflow counted, converted to local stress and strain with Neuber’s rule, and summed for fatigue damage. |

\textsuperscript{10} Dynamic Analysis and Design System
Author(s): Heyes, Milsted and Dakin (1994)

Article Title: Multiaxial fatigue assessment of automotive chassis components on the basis of finite element models

Component(s): General – steering knuckle as example

Stress-Strain Calculation: Elastic FEA (+ Neuber plasticity correction for strain-life predictions)

Life Prediction Method(s):
- Strain-life: abs max principal – normal strain
- abs max principal – SWT
- abs max principal – Morrow
- signed Tresca criterion
- signed von Mises criterion

Critical plane:
- normal strain, \( \varepsilon_n \)
- shear strain, \( \gamma \)
- Bannantine normal, \( \varepsilon_n \sigma_{max} \)
- Fatemi and Socie, \( \gamma(1+n\sigma_{max}/\sigma_y) \)

Experiment Verification: None

Author’s Remarks:
A two stage assessment procedure is proposed. First, critical events and locations are identified through preliminary life estimates and examination of strain ranges, and summary statistics are obtained concerning the behavior of the elastic stress tensor. A more detailed analysis of the critical events and locations is then carried out, with the simplest credible method appropriate to the prevailing stress-strain state being used to assess life.
Prediction Procedure

Computer Aided Engineering in Fatigue (CEA-FATIGUE) methodology was applied:

1. **Reference Elastic Stresses:** The reference stresses corresponding to each load source was first determined by performing static-elastic finite element analyses, and was then multiplied by the load factor histories for the entire elastic stress histories.

2. **Elastic Stress History Synthesis:** The elastic stress components due to numerous static loads were combined to derive the time history for each stress components. That is

   \[ \sigma_{i}(t) = \sum \sigma_{ik} \cdot L_{i}(t) \]

3. **Failure Mode Computation:** The critical plane approach was adopted. The plane where the fatigue crack initiates, the critical plane, was defined as the plane on which major elastic principal stresses were maximized during the entire multiaxial cyclic loading history. The search for the critical plane was found necessary, because the principal stress/strain axes vary with time for non-proportional loading problems. The maximum normal stress and normal strain amplitude on the critical plane were the damage parameters that characterized the fatigue life through the SWT criterion. It was noted that the SWT parameters are applicable to the failure modes characterized by crack initiation on planes of maximum principal stress, and are especially satisfactory for automotive suspension and engine components with cast materials.

4. **Rainflow Cycle Counting:** The objective of cycle counting is to reduce all irregular stress history to a table of number of cycles versus stress/strain levels (including mean and amplitude) so that a stress/strain life curve obtained with fully reversed loading cycle may be used for estimating the fatigue life of the component. A cycle is defined as a closed loop on a cyclic stress-strain curve. The one-pass rainflow algorithm was adopted for performing cycle counting.

5. **Mean Stress Effect and Plasticity Justification:** The SWT equation was used for mean stress effect. When the elastic stresses obtained from the elastic finite element analysis exceeded the cyclic yield stress range, the stresses were adjusted. Glinka’s energy density method was adopted for the plasticity justification.

6. **Damage Accumulation:** The fatigue life was calculated by accumulating the damage due to the stress/strain cycles using Miner’s (linear damage) rule.
**Author(s):** Taylor (1996 and 1997)
Taylor, Zhou, Ciepalowicz and Devlukia (1999)

**Article Title:**
1. Crack modeling: A novel technique for the prediction of fatigue failure in the presence of stress concentrations
2. Mixed-mode fatigue from stress concentrations: an approach based on equivalent stress intensity

**Component(s):** General - Crankshaft

**Stress-Strain Calculation:** Elastic FEA

**Life Prediction Method(s):** Crack modeling method (CMM)

**Experiment Verification:** Constant amplitude uniaxial (bending and torsion) component test

**Author’s Remarks:**
1. The method can be successfully applied to components which are analyzed using FEA. Predictions for the fatigue limit of an automotive crankshaft were correct to within 3% of the experimental value.
2. The method is simple to use, requiring only elastic FE data and basic material properties. It is an improvement over existing methods for the analysis of sharp notches, which require elastic/plastic analyses combined with empirical corrections for the effect of notch size and stressed volume.
3. Predictions were not significantly affected by refinement of the FE mesh, even though this had a strong effect on the value of the hot-spot stress. This is very advantageous for the practical use of the technique.

**Prediction Procedure**

1. The elastic stresses along a line moving away from the point of maximum stress are examined and compared to the stress distribution for a crack of standard geometry. The geometry chosen was that of a through-crack in an infinite plate under uniform tension for which the variation of stress with distance from the crack tip was shown by Westergaard.
2. Results from the FE analysis of the component, in the form of a stress/distance curve, are compared with a curve derived from the infinite plate crack equation. The constants of the equation are varied until a best possible fit is obtained.
<table>
<thead>
<tr>
<th>Author(s):</th>
<th>Prediction Procedure</th>
</tr>
</thead>
<tbody>
<tr>
<td>Taylor (1996 and 1997)</td>
<td></td>
</tr>
<tr>
<td>Taylor, Zhou, Ciepalowicz and Devlukia (1999)</td>
<td></td>
</tr>
</tbody>
</table>

4. The CMM suffers from problems when faced with a notch or crack which is physically short. It is expected that the CMM prediction will give a fatigue limit that is too high in these cases.

3. The value of $K$, and thus $\Delta K$ is then known because for the Westergaard geometry the solution is given by $\Delta K = \Delta \sigma (\pi D)^{1/2}$ where $\Delta \sigma$ and $D$ are the constants of the Westergaard equation.

4. The fatigue limit for the component is predicted by finding the applied loads on the FE model which are just sufficient to make $\Delta K = \Delta K_{th}$. Because the FE model is linear-elastic this is usually just a matter of scaling.

5. In line with the approach of Smith and Miller, a second prediction of fatigue limit for the component must be made by finding the loads for which the local stress at the stress-concentration is equal to the unnotched fatigue limit.

6. The second prediction will be the correct one if we are in the blunt-notch regime. In any event the higher of the two predictions of fatigue limit is used.
Author(s): Chu (1997)

Article Title: Multiaxial fatigue life prediction method in the ground vehicle industry

Component(s): General

Stress-Strain Calculation:
- Strain-gaged local history
- Elastic FEA + Neuber's plasticity correction
- Elastic-plastic FEA

Life Prediction Method(s):
- SWT ($\sigma_{max}$, $\epsilon_a$) and biaxial version of it ($\sigma_{max}$, $\epsilon_a$ + $|\tau_{max}|$, $\gamma_a$)

Experiment Verification: Not applicable

Author's Remarks:
1. The multiaxial fatigue analysis program is separated into two parts: the stress analysis part and the fatigue damage analysis part.
2. The objective of the stress analysis is to obtain the complete 3D stress and strain history of a potential failure site.
3. The fatigue damage analysis involves rainflow counting of the fatigue events and the summing up of their damage. The method is similar to that used for uniaxial problems, with the additional complexity of the biaxial damage criterion and the critical plane approach: using Brown and Miller's biaxial damage criterion as an example, for every potential failure plane, damage events are determined by rainflowing the shear strain component ($\gamma$). During each shear strain defined damage event, the variation of normal strain ($\epsilon$) is also recorded. The damage caused by each event is then assessed by the biaxial damage parameter ($\gamma_a + K\epsilon_a$). The (shortest) fatigue life (on the most critical plane) can be determined only after the fatigue analysis is performed on each and every potential failure plane.
**Author(s):** Sonsino, Kaufmann, Foth and Jauch (1997)

**Article Title:** Fatigue strength of driving shafts of automatic transmission gearboxes under operational torques

**Component(s):** Transmission shaft (induction-hardened surface)

**Stress-Strain Calculation:** Elastic FEA (the measured strains showed minor plastic strains)

Normal stress hypothesis was used for the hardened surface because of its brittleness and distortion energy hypothesis was used for the tougher non-hardened core.

**Life Prediction Method(s):** Nominal and local stress-life

**Experiment Verification:** Constant and variable amplitude component tests

**Author’s Remarks:**

1. The nominal stress concept (case A) is not recommended because the nominal stresses do not always correlate with the local stresses.

2. Concepts based on local stresses and strains (Cases B and C) describe the material behavior better than the methods based on nominal loads (Cases A and D).

3. The $P_j$ method sometimes results in overestimation of fatigue life.

4. The following information must be available and the following procedures must be followed in order to make a reliable prediction of fatigue life:
   - S-N curve of the component on the basis of the local stresses/strains, including the influences of the geometry, material, surface condition and residual stresses on fatigue behavior.
   - Actual local material states at the locations that are critical for failure.
   - Component and material related Haigh diagram for the determination of the reference S-N curve for damage calculation:

Five fatigue life calculation procedures were followed:

- **Case A:** The nominal stress concept: the calculation starts from the external load that corresponds to the nominal shear stress, as well as to the local strain, because of the approximate linearity observed between load and local strain.

- **Case B:** SWT ($P_{SWT}$-N curve derived from the S-N curve of the shaft is used instead of the S-N curve of the unnotched specimen).

- **Case C:** Component related Haigh diagram (the local rainflow matrix of strains ($\varepsilon_a$, $\varepsilon_m$, $n$) is initially calculated from the rainflow matrix of loads ($M_{T,a}$, $M_{T,m}$, $n$), the cyclic stress-strain curve determined using non-hardened, unnotched specimens subjected to axial loading was used to find the local rainflow matrix of stresses ($\sigma_a$, $\sigma_m$, $n$), and the influence of mean stress is incorporated using a Haigh diagram).

- **Case D:** Modification of S-N curve using Liu-Zenner procedure.

- **Case E:** Using $P_j$ parameter based on fracture mechanical considerations.
Cyclic stress-strain curves for the particular material states at the critical failure locations.

- Local (equivalent) strains or stresses in the critical area.
- Loads and local parameters in the form of a rainflow matrix.
- Real damage sums applicable for the type of spectrum, geometry, stress and material state (surface, residual stresses) at the locations that are critical for failures. The real damage sum should refer to the method of fatigue life prediction that was used. Also, a comparison must be made between the service spectrum that was expected and the spectrum for which the real damage sums were derived.

that assume a micro-crack from the very beginning of the fatigue life. The local strain at the crack tip is calculated as the amplitude of the effective cyclic J-integral.
**Author(s):** Sonsino, Kaufmann, and Grubišić (1997)

<table>
<thead>
<tr>
<th><strong>Article Title:</strong></th>
<th>Transferability of material data for the example of a randomly loaded forged truck stub axle</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Component(s):</strong></td>
<td>Truck stub axle</td>
</tr>
<tr>
<td><strong>Stress-Strain Calculation:</strong></td>
<td>linear elastic FEA</td>
</tr>
<tr>
<td><strong>Life Prediction Method(s):</strong></td>
<td>Local stress-life</td>
</tr>
<tr>
<td><strong>Experiment Verification:</strong></td>
<td>Constant and variable amplitude component tests</td>
</tr>
</tbody>
</table>

**Prediction Procedure**

**CASE A:** Method based on rain-flow matrices; Amplitude transformation via the damage parameter $P_{SWT}$ and a component and material related compensation of the mean values using a Haigh diagram are performed for the strain-time histories on the basis of their respective rain-flow matrices.

- A1: Amplitude transformation with the damage parameter $P_{SWT}$
- A2: Amplitude transformation with component related Haigh diagram.

**CASE B:** Method based on peak value sequences; Besides introducing the rain-flow matrices, the peak value sequences are introduced so that the sequence of the loading can numerically be taken into account cycle by cycle.

- B1: Amplitude transformation with the damage parameter $P_{SWT}$
- B2: Amplitude transformation with the damage parameter $P_i$.

**CASE C:** Method based on level crossing and range pair counting;

- C1: Mean value formation between the counts of level crossings and range pairs;
- C2: The original Miner damage calculation is retained, but the reference S-N curve has the mean value of the slopes of the S-N curve for the failure criteria total rupture and the S-N curve for crack propagation.

**Author's Remarks:**

1. The problem that arises before the fatigue life calculation is the transferability of data predominantly determined for specimens on the basis of nominal stress to components where neither a nominal stress nor a notch factor can be defined in most cases and where the dimensions often deviate from those of the specimens investigated.

2. The preconditions for the transferability of material data are fulfilled only after consideration of:
   - the same failure criterion (first detectable crack),
   - the local equivalent stresses or strains and,
   - the maximum stressed/strained material volume.

3. In addition, differences between the materials, states of surfaces, surface layers and residual stress must also be accounted for.

4. The local concept in its extended form is applicable, namely after taking into account local stresses/strains, stress/strain gradients and the maximum stressed/strained material volume.

5. If the maximum stressed/strained material volume cannot be calculated in a relatively simple manner, it can be derived by approximation via FE calculations using strain/stress gradients, and its influence on fatigue strength can be evaluated.

6. The real damage sum, which is a function of material, component, load
Prediction Procedure

...type and spectrum, is not known in most cases. This is a decisive limitation to the transferability of real damage sums determined on specimens. Available experience with real damage sums can therefore only be utilized for a preliminary dimensioning within the framework of a defined method (for particular methods, different empirical damage sums are used).

7. When over-dimensioning is not acceptable, verification tests prior to release are indispensable, in particular for vital components like stub axles, which must never fail.
<table>
<thead>
<tr>
<th><strong>Author(s):</strong></th>
<th>Béranger, Berard, and Vittori (1997)</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Article Title:</strong></td>
<td>A fatigue life assessment methodology for automotive components</td>
</tr>
<tr>
<td><strong>Component(s):</strong></td>
<td>Renault Safrane suspension arm</td>
</tr>
<tr>
<td><strong>Stress-Strain Calculation:</strong></td>
<td>Elastic FEA</td>
</tr>
<tr>
<td><strong>Life Prediction Method(s):</strong></td>
<td>Critical plane and hyper-sphere procedures of Dang Van</td>
</tr>
<tr>
<td><strong>Experiment Verification:</strong></td>
<td>Constant-amplitude load-control component tests</td>
</tr>
<tr>
<td><strong>Author’s Remarks:</strong></td>
<td>1. The minimum value of the safety factor on the part is located in the area where the failure occurred on the test rig. Its value is 1.02. This result exhibits a very good correlation between the experiment and the fatigue life prediction since the theoretical value of 1 and the actual safety factor are very close. 2. It was shown that the safety factors obtained are highly dependent on the material data. The best results were derived from material data representative of the exact surface finish of the part, namely shot-peened.</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th><strong>Prediction Procedure</strong></th>
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<tbody>
<tr>
<td>1. The result of the static linear elastic FE calculation is used to generate the load cycle.</td>
</tr>
<tr>
<td>2. The Dang Van fatigue model ( (\tau(t)+\alpha p(t)-\tau_o &gt; 0) ) was used with a material damage line, whose coefficients are ( \tau_o = 217 ) MPa and ( \alpha = -1.38 ). They correspond to the shot-peened material, for a life of ( 10^7 ) cycles, with a probability of failure of 50%. Several stress ratios are used to identify these coefficients. Therefore any loading cycle, with various stress ratios, can be predicted with these coefficients using the Dang Van model.</td>
</tr>
<tr>
<td>3. The result of the calculation is a value of a so called safety factor ( S_f ). This factor can be defined as the normalized smallest distance of the loading path in the ((\tau,\rho)) Dang Van diagram to the damage line. When ( S_f ) is greater than 1 the component is supposed to be safe. On the contrary when ( S_f ) of some elements becomes less than unity, failure may occur.</td>
</tr>
<tr>
<td>4. Several damage lines representative of the various surface finishes have been tested for the prediction of the safety factors associated to the fatigue life.</td>
</tr>
</tbody>
</table>
### Author(s): Devlukia and Bargmann (1997)

<table>
<thead>
<tr>
<th>Component(s):</th>
<th>Suspension arm</th>
</tr>
</thead>
<tbody>
<tr>
<td>Stress-Strain Calculation:</td>
<td>Strain measurement during tests</td>
</tr>
<tr>
<td>Life Prediction Method(s):</td>
<td>Strain-life (deterministic and probabilistic)</td>
</tr>
<tr>
<td>Experiment Verification:</td>
<td>Constant and variable amplitude component tests</td>
</tr>
</tbody>
</table>

#### Deterministic:
- The surface roughness parameters for the as-forged surface were found. These values, together with the tensile strength of the material were used to predict the surface finish factor, $K_{sf}$.
- This factor was used in Neuber’s rule to predict the reduction in fatigue lives. There, in place of the fatigue notch factor $K_f$, the term $K_t/K_{sf}$ was used, $K_t$ being the geometric stress concentration factor.
- The surface factor and the residual stress measured value were used in Morrow’s equation to predict their effects on the fatigue performance.
- Miner’s rule was used for damage summation.

#### Probabilistic:
- In this analysis, both the surface finish factor and the residual stress were assumed to be independent random variables. Experimental measurements carried out on a batch of components established the normal density, mean value and standard deviation of the surface finish factor and the residual stress.
- The above probabilistic inputs were used to calculate the probability of failure based on the Coffin-Manson equation, the hysteresis loop equation and Neuber’s rule.
- The residual stress effects were predicted using the Goodman correction, and the surface roughness effects were predicted substituting $K_t/K_{sf}$ for $K_f$ in Neuber’s rule.
- The “CPFI-Complete Probability Fast Integration” method was used to predict the probability of failure.
- In the CPFI approach, the fatigue-life distribution of a component, and hence the reliability, i.e. the probability that the random lifetime $N$ does not fall below a given value, is always expressed in a multiple-integral closed form.
Article Title: Fatigue analysis and the local stress–strain approach in complex vehicular structures

Component(s): General

Stress-Strain Calculation: Analytical, elastic FEA + Neuber plasticity correction, elastic-plastic FEA

Life Prediction Method(s):
1. Total strain vs life;
2. SWT parameter $\sigma_{\text{max}} \varepsilon$ vs life;
3. Morrow-Landgraf: replace $\sigma^'$ by $(\sigma^'+\sigma)$;
4. Von Mises' equivalent stress vs life;
5. Brown-Miller $\gamma + K\varepsilon$ vs life;
6. Fatemi-Socie-Kurath $\gamma(1+\frac{\sigma_{\text{max}}}{\sigma})$ vs life;
7. Chu’s modified SWT $\sigma_{\text{max}} \varepsilon + |\tau|_{\text{max}} \gamma$ vs life.

Local stress-strain analysis options:

Strength of materials

Finite elements

Elastic

Plastic

Component calibration

Local $\varepsilon$-history

Material properties derived from standard strain life test results.

A three-dimensional stress–strain model, adapted from that of Mroz.

The model must be able to simulate reversed multiaxial stress/strain paths.

Damage assessment must be flexible enough to handle several multiaxial fatigue damage criteria. A critical plane search for the most damaging direction is necessary.

A multiaxial Neuber type of plasticity correction method must be used to translate the elastic local stress estimates into approximations that are corrected for plastic stress–strain behavior.

Large finite element models of vehicular components tend to have many load input points, and thus must utilize an elastic unit load analysis combined with a superposition procedure of each load points service history, to produce each element’s stress history. It is critical that the

Author(s): Conle and Chu (1997)
Prediction Procedure

<table>
<thead>
<tr>
<th>Author(s):</th>
<th>Conle and Chu (1997)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Problem areas that require further research:</td>
<td></td>
</tr>
<tr>
<td>1. The simulation of the non-linear stress–strain or load deflection behavior of elements such as tires and bushings.</td>
<td></td>
</tr>
<tr>
<td>2. Accounting for ‘small cycle’ effects caused by periodic overload conditions in the loading.</td>
<td></td>
</tr>
</tbody>
</table>
**Prediction Procedure**

1. It is proposed that in order for fatigue failure to occur, the stress level must be high enough not only at the maximum stress point (the hot spot) but also for some distance around the hot spot (the stress at one point, a given distance from the hot spot or to the stress averaged over a line of given length). These distances are material-dependent.

2. Consider a central crack in an infinite plate subjected to a remote uniaxial load \( \Delta \sigma \). The stress ahead of the crack tip can be expressed, when \( r < a \), as:

   \[
   \Delta \sigma(r, \theta) = \Delta \sigma \sqrt{\frac{r}{2 \pi}} \cos \left( \frac{\theta}{2} \right) \left[ 1 + \sin \left( \frac{\theta}{2} \right) \sin \left( \frac{3\theta}{2} \right) \right]
   \]

3. If this crack is loaded at its fatigue limit, then \( \Delta K = \Delta K_{th} \); therefore the stress range at the point \( r = L/2, \theta = 0 \) on the stress-distance curve is equal to the plain fatigue limit:

   \[
   \Delta \sigma \left( r = \frac{L}{2}, \theta = 0 \right) = \Delta \sigma_0
   \]

4. When \( \Delta K = \Delta K_{th} \), the average stress along a straight line of length \( 2L \) is equal to the plain fatigue limit:

   \[
   \frac{1}{2L} \int_0^{2L} \Delta \sigma(r, \theta = 0) \, dr = \Delta \sigma_0
   \]

5. Also the stress averaged over a semicircle of radius \( r = L, \theta = \pm \pi/2 \) to \( -\pi/2 \), is approximately equal to the plain fatigue limit (in fact it is larger by about 10
| **Author(s):** | Taylor, Bologna and Bel Knani (2000)  
| Susmel and Taylor (2003) |
|---|---|
| **Prediction Procedure** |  

gradient, the stress at the critical distance may be lower than that for the blunt notch.

4. The approach was able to predict correctly the location of fatigue failure, and also predicted the fatigue life with an error of a factor of 1.85, equivalent to an error of less than 6% on stress, which is within acceptable tolerances.

6. The last 3 equations provide the basis for three critical distance methods for estimating the fatigue limit of the cracked body: using the stress at a single point, averaged along a line or averaged over an area.

\[
\frac{4}{\pi l} \int_{-\pi/2}^{\pi/2} \int_0^{\pi/2} \Delta \sigma(r, \theta) \, dr \, d\theta \approx \Delta \sigma_0
\]
**Author(s):** Tang, Ogarevic and Tsai (2001)

**Article Title:** An integrated CAE environment for simulation-based durability and reliability design

**Component(s):** General – lower control arm

**Stress-Strain Calculation:** quasi-static FEA

**Life Prediction Method(s):**
- von Mises equivalent strain approach
- ASME Boiler and Pressure Vessel Code strain approach
- tensile critical plane method
- shear critical plane method

**Experiment Verification:** None

**Author's Remarks:**
1. The development of a simulation-based environment for durability and reliability analysis of a mechanical component is implemented.
2. This environment carries out the life prediction and reliability assessment of mechanical components based on duty cycle loads or dynamic stress time histories. It involves multidiscipline, such as geometry generation, structural analysis, structural dynamics, fatigue life prediction, and reliability analysis.
3. The software structure of this environment consists of three subworkspaces, computer servers, database and data server, and a communication channel. Each subworkspace collects several tools to perform an engineering activity, namely, dynamic stress computation, fatigue life prediction, and reliability analysis.

### Prediction Procedure

**Software component structure:**

<table>
<thead>
<tr>
<th>Component</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>PATRAN/HyperMesh</td>
<td>Geometry Modeler</td>
</tr>
<tr>
<td>ANSYS/NASTRAN/ABAQUS</td>
<td>FE Computation Server</td>
</tr>
</tbody>
</table>

**The program runstream:**

1. **Quasi-static Loading Generation** → **FE Analysis** → **Dynamic Stress** → **Stress/Strain Estimation**

**The dataflow and dependency of life prediction subworkspace:**

1. **Stress Spectrum** → **Interface Tool** → **Dynamic Stress History** → **Conversion Tool** → **Interface Tool**
2. **Elamin or Elastic/Plastic Strain File** → **Material Property Database** → ** damage & Initiation Life Prediction Tool**
Author(s): Kocabicak and Firat (2001) and Firat and Kocabicak (2004)

Article Title: 1. Numerical analysis of wheel cornering fatigue test
2. Analytical durability modeling and evaluation-complementary techniques for physical testing of automotive components

Component(s): Passenger car wheel

Stress-Strain Calculation: Elastic and elasto-plastic FEA using a linear or non-linear kinematic hardening material law with isotropic von Mises yield criterion

Life Prediction Method(s): Strain-life:
Strain amplitudes based on effective strain and stress measures
Critical plane concepts based on SWT and Fatemi-Socie parameters

Experiment Verification: Constant and variable amplitude bi-axial load-controlled cornering fatigue test

Author's Remarks:
1. Best correlations are obtained with critical plane parameters, while effective parameters performed relatively poor.
2. Fatigue test cycles predicted using Fatemi–Socie damage parameter based on critical plane concepts are considerably close to test cycles for all wheel loads, and falls in to a band within a factor of 3 to 5.
3. In order to investigate the trends predicted with increasing number of cycles, the numerical simulation for the wheel failure location is conducted using Fatemi–Socie parameter, and the variation of damage per cycle is determined to be monotonically decreasing function, indicating that the estimation of fatigue life using the damage predicted with the first cycle after monotonic loading is an appropriate strategy resulting in an conservative results for this particular case. The reduction in damage per cycle is determined to be approximately 13%.
4. The critical plane parameters involving mean stress terms perform equally similar predictions under non-proportional unbalanced loading case considered here, and Fatemi–Socie and Smith–Watson–Topper models based on critical plane concept constitute a pair of damage parameters applicable in both cases within the margin of acceptable accuracy from engineering point of view in the design of wheels with conformance to anti-fatigue requirements in biaxial cornering tests.
Author(s): Savaidis (2001)

| Article Title: | Analysis of fatigue behavior of a vehicle axle steering arm based on local stresses and strains |
| Component(s): | Vehicle steering arm |
| Stress-Strain Calculation: | Strain-time sequence measured at the failure critical location |
| Life Prediction Method(s): | local stress-strain: $P_{SWT}$ damage parameter $P_J$ damage parameter |
| Experiment Verification: | Variable amplitude uniaxial (bending) component test |
| Author’s Remarks: | The Local Strain Approach allows an analysis of fatigue behavior of engineering components under operational loading from the point of view of material mechanics. In conjunction with the two parameters, $P_{SWT}$ and $P_J$, it is able to display and estimate influencing factors such as mean stresses, load sequence effects above and below the endurance limit and technological effects such as the roughness and residual stresses explicitly. |

**Damage Parameters**

The SWT damage parameter: $P_{SWT} = \sqrt{(\sigma_s + \sigma_w)e/E}$

and the one developed by means of elastic-plastic fracture mechanics on a semi-elliptical microcrack ($P_J$):

$$P_J = \frac{\Delta J_{eff}}{a} = \frac{1.24 \cdot \Delta \sigma_{eff}^2 + 1.02 \cdot \Delta \sigma_{eff} \cdot \Delta \varepsilon_{eff} - \Delta \sigma_{eff}}{E \cdot a}$$

have been used. Since the $P_J$-$N$ curve is closely related to the crack growth law, its analytical description occurs according to a power-law function: $P_J^* \cdot N = Q$ for $P_J > P_J, E$, where the constant $Q$ denotes the fatigue life at $P_J = 1$ N/mm². The slope $m$ is evaluated using regression analysis throughout the experimental results. The subscript $E$ indicated the endurance limit.

**Technological Effects**

Surface roughness: Factor $\kappa$ describes the decrease of the stress endurance limit due to surface roughness:

$$\kappa = \frac{\sigma_{E, rough}}{\sigma_{E, pol}} = 1 - 0.22 \cdot (\log R_z)^{0.64} \cdot \log \sigma_{UTS} + 0.45 \cdot (\log R_z)^{-0.53}$$

The influence of the residual stress-strain state on the fatigue life prediction is considered by adding the value of the residual strain $\varepsilon_R$ to the local strain-time sequence $\varepsilon_{cycle}$ measured:

$$\varepsilon(t) = \varepsilon_{cycle}(t) + \varepsilon_R$$
Local Stress-Life Approach

Reference: S-N curve of a "similar" component

Prediction:

1. The influence of heat treatment on fatigue resistance of case-hardened components is taken into account by using S-N curves obtained from "similar" components as reference. The reference S-N curves are transformed to a $\sigma_{e,a}$-$N$ curve for the critical location using the stress concentration...
factor $K_n$ (or a load influence factor $i$). To the $\sigma_e$-$N$ curve a highly stressed volume $I_V$ and a Weibull exponent $k_W$ are assigned.

2. Failure criterion is crack initiation of technical size in the case.

3. “Similarity” means that the components are equal in stress state at the critical locations for crack initiation, base material, case hardening (type, case depth, etc.), residual stresses, and surface condition.

4. The weakest link model is recommended to consider the size effect on the crack initiation life.

5. The parameter $P_{SWT,e}$ proved to be suitable to take the mean stress influence into account. Cyclic loading may be limited by the static strength of a component, e.g. with fully tensile loading (load ratio $0 < R < 1$). If the Weibull exponent for monotonic loading $k_{W,m}$ is known, the size effect on static strength can also be estimated with the weakest link model.
Author(s): Haiba, Barton, Brooks and Levesley (2002)

Article Title: Review of life assessment techniques applied to dynamically loaded automotive components

Component(s): Lower control arm of a double wishbone suspension system

Stress-Strain Calculation:
- The standard time domain approach that involves the application of the quasi-static stress analysis method.
- The more sophisticated time domain approach that involves the application of the transient dynamics analysis method.
- The frequency domain approach that involves the application of the harmonic stress analysis method.

Life Prediction Method(s):
- The standard time domain approach that involves stress or strain cycle counting, damage prediction, and finally life estimation. The stress histories required for the above processes can be obtained from either a quasi-static or transient stress analysis.
- The frequency domain approach that involves using the power spectral density of the component stress/strain histories, estimating the expected number of cycles at several pre-specified stress ranges, and finally predicting the life. The stress histories required prior to the application of this approach can be obtained from a harmonic stress analysis.

Experiment Verification: None

Author’s Remarks:
1. The reference life assessment strategy that uses transient stress analysis and time domain life assessment is very expensive. Consequently, it cannot be used as a part of an optimization algorithm based on fatigue life.
2. The strategy that involves quasi-static stress analysis and time domain life assessment is very efficient. The minimum life predicted using this strategy is very accurate when compared with that obtained using the reference strategy. Additionally, this strategy has the capability to predict nodal life distribution that is identical to the distribution obtained using the reference strategy when considering just the design regions of the component under consideration. Consequently, this strategy can be used as a part of an optimization algorithm based on fatigue life.
3. The life assessment strategy that uses harmonic stress analysis and frequency domain life assessment cannot be used to predict reliable values of minimum life and the results obtained using this strategy cannot be used within an optimization algorithm based on fatigue life.
Appendix B - Life Prediction Calculation Details

The details of life prediction calculations for the forged steel (FS) and cast aluminum (CA) steering knuckles for each life prediction model used and for the highest load level of the component testing are provided here.

I Nominal S-N Approach for FS Steering knuckle

Applied Loads and moments are:

\[
\begin{align*}
P_{\text{max}} &= 1100 \text{lbf} = 4893 \text{N} \\
M_{\text{max}} &= 1100 \times 12.18 = 4893 \times 0.309 \text{lbf.in} = 1513 \text{N.m} \\
P_a &= 525 \text{lbf} = 2335 \text{N} \\
M_a &= 525 \times 12.18 = 2335 \times 0.309 \text{lbf.in} = 722 \text{N.m} \\
P_{\text{min}} &= 50 \text{lbf} = 222 \text{N} \\
M_{\text{min}} &= 50 \times 12.18 = 222 \times 0.309 \text{lbf.in} = 69 \text{N.m} \\
P_m &= 575 \text{lbf} = 2558 \text{N} \\
M_m &= 575 \times 12.18 = 2558 \times 0.309 \text{lbf.in} = 791 \text{N.m}
\end{align*}
\]

Perform nonlinear finite element analysis (FEA) at a cross section 0.5 in above spindle second step fillet:

Apply \(P_{\text{max}}\) → find \(S_{\text{max}} = 568.1 \text{ MPa}\) & apply \(P_{\text{min}}\) → find \(S_{\text{min}} = 33.0 \text{ MPa}\)

Then \(S_a = 267.5 \text{ MPa}\) & \(S_m = 300.6 \text{ MPa}\)

\(K_t = 2.2\) from stress concentration charts (e.g. see Stephens et al., 2000) for step shaft under bending.

Find \(K_f\):

\[
K_f = 1 + \frac{K_t - 1}{1 + a/r} \quad \text{where} \quad a = 0.0254 \left( \frac{2070}{S_a} \right)^{1.18} \quad \text{in mm with} \ S_a \text{ in MPa and} \ r = 0.05 \text{ in} = 1.27 \text{ mm},
\]

\(\rightarrow K_f = 2.1\)

Find S-N line including \(K_f\) effect at \(10^6\) cycles:

\[
\begin{array}{c|c}
\text{2N} & S_N \\
\hline
1 & \sigma' f = 1156.8 \text{ MPa} \\
2.00E+06 & S_f/K_f = 352/2.1 = 168.4 \text{ MPa}
\end{array}
\]
→ \( B = -0.133 \rightarrow S_{nf} = 1156.8 \left(2N_f\right)^{0.133} \)

Use modified Goodman equation to account for mean stress and find life:

\[
\frac{S_e}{S_{nf}} + \frac{S_m}{S_u} = 1 \rightarrow \frac{267.5 + 300.6}{821.1} = 1 \rightarrow S_{nf} = 422.0 \text{ MPa}
\]

\[
S_{nf} = \sigma_f' \left(2N_f\right)^B \rightarrow S_{nf} = 1156.8 \left(2N_f\right)^{0.133} \rightarrow N_f = 1000 \text{ cycles}
\]

Use Gerber equation to account for mean stress and find life:

\[
\frac{S_e}{S_{nf}} + \left(\frac{S_m}{S_u}\right)^2 = 1 \rightarrow \frac{267.5 + \left(\frac{300.6}{821.1}\right)^2}{1} = 1 \rightarrow S_{nf} = 308.9 \text{ MPa}
\]

\[
S_{nf} = \sigma_f' \left(2N_f\right)^B \rightarrow S_{nf} = 1156.8 \left(2N_f\right)^{0.133} \rightarrow N_f = 10400 \text{ cycles}
\]

II ε-N Approach using Nominal Stresses for FS Steering knuckle

Using Neuber's Rule without Generalization

Neuber’s Rule in its original form applies to small-scale as well as large-scale yielding:

\[
\sigma \varepsilon = S e K_{nq}^2
\]

where \( S \) & \( \varepsilon \) are nonlinear nominal stress and strain and \( K_{nq} \) is based on Hoffmann-Seeger’s multiaxial version of Neuber’s Rule for local multiaxial state of stress where elastic FEA results are available. To find \( K_{nq} \):

\[
K_{nq} = K_f \left[ \frac{1}{2} \left[ \left(1-a_e\right)^2 + \left(1-b_e\right)^2 + \left(a_e-b_e\right)^2 \right] \right]
\]

\[
a_e = \frac{\sigma_{e2}}{\sigma_{e1}} = 0.24 \text{ and } b_e = \frac{\sigma_{e3}}{\sigma_{e1}} = 0.06
\]

\( \sigma_{e1} \), \( \sigma_{e2} \) and \( \sigma_{e3} \) are the principal stresses from elastic FEA solution.

\( \rightarrow K_{nq} = 1.9 \)

The 2nd equation to be solved with the Neuber’s rule is the material law:
\[ \varepsilon = \frac{\sigma}{E} + \left( \frac{\sigma}{K'} \right)^{\frac{1}{n'}} \]

For maximum load:

\[ S_{\text{max}} = 568.1 \text{ MPa} \rightarrow e_{\text{max}} = \frac{S_{\text{max}}}{E} + \left( \frac{S_{\text{max}}}{K'} \right)^{\frac{1}{n'}} = \frac{568.1}{201500} + \left( \frac{568.1}{1269} \right)^{\frac{1}{0.137}} = 0.00565 \]

Then the equations are:

\[ \sigma_{\text{max}} e_{\text{max}} = S_{\text{max}} e_{\text{max}} K_{iy}^2 = 568.1 \times 0.00565 \times 1.9^2 = 11.6 \]

\[ e_{\text{max}} = \frac{\sigma_{\text{max}}}{201500} + \left( \frac{\sigma_{\text{max}}}{1269} \right)^{\frac{1}{0.137}} \]

Solving the above two equations simultaneously with a numerical method (MATLAB here):

\[ \sigma_{\text{max}} = 700.6 \text{ MPa} \]

\[ e_{\text{max}} = 0.0166 \]

For unloading:

\[ S_a = 267.5 \text{ MPa} \rightarrow \Delta S = 535 \text{ MPa} \rightarrow \]

\[ \Delta e = \frac{\Delta S}{E} + 2 \left( \frac{\Delta S}{2K'} \right)^{\frac{1}{n'}} = \frac{535.0}{201500} + 2 \left( \frac{535.0}{2 \times 1269} \right)^{\frac{1}{0.137}} = 0.00268 \]

Then the equations are:

\[ \Delta \sigma \Delta e = \Delta S \Delta e K_{iy}^2 = 535.0 \times 0.00268 \times 1.9^2 = 5.2 \]

\[ \Delta e = \frac{\Delta \sigma}{201500} + 2 \left( \frac{\Delta \sigma}{2 \times 1269} \right)^{\frac{1}{0.137}} \]

Solving the above two equations simultaneously with a numerical method (MATLAB here):

\[ \Delta \sigma = 912.0 \text{ MPa} \]

\[ \Delta e = 0.0057 \]

The strain amplitude is equal to \( \varepsilon_a = 0.00285 \)

Use Smith-Watson-Topper (SWT) equation to calculate life:
\[ \sigma_{\text{max}} e_a E = (\sigma'_y)^2 (2N_f)^{2b} + \sigma'_y e'_y E (2N_f)^{bne} \]

\[ 700.6 \times 0.00285 \times 201500 = 1156.8 (2N_f)^{2e-0.082} + 1156.8 \times 3.0315 \times 201500 (2N_f)^{-0.082-0.7912} \]

\[ \rightarrow N_f = 8800 \text{ cycles} \]

**Using Seeger-Heuler's Generalized Application of Neuber's Rule**

\[ \frac{K_e^2}{E} \frac{S^2}{E} \frac{E e'}{S'} = \frac{\sigma^2}{E} + \sigma \left( \frac{\sigma}{K'} \right)^{\gamma'} \]

Here \( S \) is the elastically calculated nominal stress:

\[ S_a = \frac{P_a}{A} + \frac{M_a e}{I} = \frac{2335 \times 309.4 \times 28}{\pi 28^2} + \frac{\pi 28^4}{64} = 340.3 \text{ MPa} \]

Similarly for \( P_a = 2558 \text{ N} \rightarrow S_a = 372.7 \text{ MPa} \)

Therefore: \( S_{\text{max}} = S_a + S_a = 713.0 \text{ MPa} \)

\( S^* \) is the modified nominal stress:

\[ S^* = \frac{P}{P_p} S_y \]

where \( P_p \) is the plastic limit load for elastic-perfectly plastic material at the cross section remote from the notch. Here, the appropriate version would be:

\[ S^* = \frac{M}{M_p} S_y \]

and \( M_p = 2028 \text{ N.m} \) (equivalent to 1470 lbf load at the actuator).

Thus:

\[ S^* = \frac{M}{M_p} S_y = \frac{1515}{2028} \times 556.2 = 415.5 \text{ MPa} \]

\[ e^* = \frac{S^*}{E} + \left( \frac{S^*}{K'} \right)^{\gamma'} = \frac{415.5}{201500} + \left( \frac{415.5}{1269} \right)^{0.137} = 0.00235 \]

\[ \rightarrow \frac{E e^*}{S^*} = 1.14 > 1 \]
So the Seeger-Heuler model recommends the generalized application.

For maximum load:

\[ S_{\text{max}} = 713.0 \, \text{MPa} \]

Then the equations are:

\[
\frac{K^2 S_{\text{max}}^2}{E} \frac{E e^*}{S^*} = \sigma_{\text{max}}^2 + \sigma_m \left( \frac{\sigma_{\text{max}}}{K^*} \right)^{\frac{1}{6.137}} \\
\frac{1.9^2 713.0^2}{201500} 1.14 = \frac{\sigma_{\text{max}}^2}{201500} + \sigma_m \left( \frac{\sigma_{\text{max}}}{1269} \right)^{\frac{1}{6.137}} \\
\sigma_{\text{max}} = 689.2 \, \text{MPa}
\]

\[
\varepsilon_{\text{max}} = \frac{\sigma_{\text{max}}}{201500} \left( \frac{\sigma_{\text{max}}}{1269} \right)^{\frac{1}{6.137}} = 0.0151
\]

For unloading the plastic limit moment was found to be 2755 N.m (obtained from FEA for EPP unloading material at a cross section remote from the notch). Thus:

\[
\Delta S^* = \Delta M \frac{S_y}{M_p} = \frac{1445}{556.2} = 396.3 \, \text{MPa}
\]

\[
\Delta e^* = \frac{\Delta S^*}{201500} + 2 \left( \frac{\Delta S^*}{2 \times 1269} \right)^{\frac{1}{6.137}} = \frac{396.2}{201500} + 2 \left( \frac{396.2}{2 \times 1269} \right)^{\frac{1}{6.137}} = 0.00197
\]

\[
\rightarrow \frac{E \Delta e^*}{\Delta S^*} = 1
\]

Therefore Seeger-Heuler's model is not needed.

\[ \Delta S = 680.6 \, \text{MPa} \]

Then the equations are:

\[
\frac{K^2 \Delta S^2}{E} = \frac{\Delta \sigma^2}{E} + 2 \Delta \sigma \left( \frac{\Delta \sigma}{2K^*} \right)^{\frac{1}{6.137}} \\
\frac{1.9^2 680.6^2}{201500} = \frac{\Delta \sigma^2}{201500} + 2 \Delta \sigma \left( \frac{\Delta \sigma}{2 \times 1269} \right)^{\frac{1}{6.137}} \\
\rightarrow \Delta \sigma = 1034.9 \, \text{MPa}
\]

\[
\Delta e = \frac{\Delta \sigma}{201500} + 2 \left( \frac{\Delta \sigma}{2 \times 1269} \right)^{\frac{1}{6.137}} = 0.008
\]

The strain amplitude is equal to: \( \varepsilon_a = 0.004 \)
Use SWT equation to calculate life:

\[ \sigma_{\text{max}} \varepsilon_a E = (\sigma'_f)^2 (2N_f)^{2b} + \sigma'_f \varepsilon' E (2N_f)^{b+c} \]

\[ 689.2 \times 0.004 \times 201500 = 1156.8^2 (2N_f)^{2-0.082} + 1156.8 \times 3.0315 \times 201500 (2N_f)^{-0.082-0.7912} \]

\[ \rightarrow N_f = 4500 \text{ cycles} \]

III Local \( \sigma\)-N Approach using Nonlinear FEA Results for FS Steering knuckle

Perform nonlinear FEA and obtain results (von Mises) at the node located at the transition of fillet and spindle second step shaft:

Apply \( P_{\text{max}} = 4893 \text{ N} \) → find \( \sigma_{\text{max}} = 603.3 \text{ MPa} \) & \( \varepsilon_{\text{max}} = 0.0073 \)

Apply \( \Delta P = 4670 \text{ N} \) → find \( \Delta\sigma = 867.4 \text{ MPa} \) & \( \Delta\varepsilon = 0.0049 \)

Therefore:

\[ \sigma_a = 433.7 \text{ MPa} \quad \varepsilon_a = 0.00246 \]

\[ \sigma_m = 169.6 \text{ MPa} \quad \varepsilon_m = 0.0049 \]

\[ \sigma_{\text{min}} = -264.1 \text{ MPa} \quad \varepsilon_{\text{min}} = 0.00241 \]

Use modified Goodman equation to account for mean stress and calculate life:

\[ \frac{\sigma_a}{\sigma_{N_f}} + \frac{\sigma_m}{S_u} = 1 \rightarrow \frac{433.7}{821.1} + \frac{169.6}{1} = 1 \rightarrow \sigma_{N_f} = 546.6 \text{ MPa} \]

\[ \sigma_{N_f} = \sigma'_f \left(2N_f\right)^b \rightarrow 546.6 = 1156.8 \left(2N_f\right)^{-0.082} \rightarrow N_f = 4600 \text{ cycles} \]

Use Gerber equation to account for mean stress and calculate life:

\[ \frac{\sigma_a}{\sigma_{N_f}} + \left(\frac{\sigma_m}{S_u}\right)^2 = 1 \rightarrow \frac{433.7}{821.1} + \left(\frac{169.6}{821.1}\right)^2 = 1 \rightarrow \sigma_{N_f} = 453.0 \text{ MPa} \]

\[ \sigma_{N_f} = \sigma'_f \left(2N_f\right)^b \rightarrow 453.0 = 1156.8 \left(2N_f\right)^{-0.082} \rightarrow N_f = 4600 \text{ cycles} \]

Use the stress-life version of SWT model to account for mean stress and calculate life:

\[ \sigma_{\text{max}} \sigma_a = \sigma'_f^2 \left(2N_f\right)^{2b} \rightarrow 603.3 \times 433.7 = 1156.8^2 \left(2N_f\right)^{2-0.082} \rightarrow N_f = 10400 \text{ cycles} \]
IV Local σ-N Approach using Linear FEA Results and Neuber-Correction for FS Steering knuckle

Perform linear FEA and obtain result (von Mises) at the node located at the transition of fillet and spindle second step shaft:

Apply $P_{max} = 4893$ N → find $\sigma_{max} = 1079.8$ MPa & $\varepsilon_{max} = 0.0046$

Solve Neuber’s correction to obtain elastic-plastic stress and strain along with the material law:

$\sigma^\text{eq} \varepsilon^\text{eq} = \sigma^\text{eq} \varepsilon^\text{eq} \rightarrow \sigma^\text{eq} \varepsilon^\text{eq} = 4.97$

$\varepsilon^\text{eq} = \frac{\sigma^\text{eq}}{E} + \left(\frac{\sigma^\text{eq}}{K'}\right)^{\frac{1}{2}} \rightarrow \varepsilon^\text{eq} = \frac{\sigma^\text{eq}}{201500} + \left(\frac{\sigma^\text{eq}}{1269.5}\right)^{0.137}$

to find:

$\sigma_{max} = 614.5$ MPa & $\varepsilon_{max} = 0.0081$

Apply $\Delta P = 4670$ N→ find $\Delta \sigma = 1030.8$ MPa & $\Delta \varepsilon = 0.0044$

Solve Neuber’s correction to obtain elastic-plastic stress and strain along with the material law for unloading:

$\Delta \sigma^\text{eq} \Delta \varepsilon^\text{eq} = \Delta \sigma^\text{eq} \Delta \varepsilon^\text{eq} \rightarrow \Delta \sigma^\text{eq} \Delta \varepsilon^\text{eq} = 4.54$

$\Delta \varepsilon^\text{eq} = \frac{\Delta \sigma^\text{eq}}{E} + 2\left(\frac{\Delta \sigma^\text{eq}}{2K'}\right)^{\frac{1}{2}} \rightarrow \Delta \varepsilon^\text{eq} = \frac{\Delta \sigma^\text{eq}}{201500} + 2\left(\frac{\Delta \sigma^\text{eq}}{2 \times 1269.5}\right)^{0.137}$

to find:

$\Delta \sigma = 874.5$ MPa & $\Delta \varepsilon = 0.0052$

Therefore:

$\sigma_y = 437.3$ MPa $\varepsilon_y = 0.0026$

$\sigma_u = 177.3$ MPa $\varepsilon_u = 0.0055$

$\sigma_{min} = -260.0$ MPa $\varepsilon_{min} = 0.0029$

Use modified Goodman equation to account for mean stress and find life:
\[
\frac{\sigma_a}{\sigma_{Nf}} + \frac{\sigma_m}{S_u} = 1 \rightarrow \frac{437.3}{\sigma_{Nf}} + \frac{177.3}{821.1} = 1 \rightarrow \sigma_{Nf} = 557.6 \text{ MPa}
\]

\[
\sigma_{Nf} = \sigma'_f (2N_f)^b \rightarrow 557.6 = 1156.8(2N_f)^{0.082} \rightarrow N_f = 3600 \text{ cycles}
\]

Use Gerber equation to account for mean stress and find life:

\[
\frac{\sigma_a}{\sigma_{Nf}} + \left(\frac{\sigma_m}{S_u}\right)^2 = 1 \rightarrow \frac{437.3}{\sigma_{Nf}} + \left(\frac{177.3}{821.1}\right)^2 = 1 \rightarrow \sigma_{Nf} = 458.6 \text{ MPa}
\]

\[
\sigma_{Nf} = \sigma'_f (2N_f)^b \rightarrow 458.6 = 1156.8(2N_f)^{0.082} \rightarrow N_f = 39700 \text{ cycles}
\]

Use the stress-life version of SWT model to account for mean stress and calculate life:

\[
\sigma_{\text{max}} = \sigma'_f (2N_f)^{2b} \rightarrow 614.5 \times 437.3 = 1156.8^2(2N_f)^{2 \times 0.082} \rightarrow N_f = 8900 \text{ cycles}
\]

V Local σ-N Approach using Nonlinear FEA Results for CA Steering knuckle

With von Mises Stress and Strain

Perform nonlinear FEA and obtain results (von Mises) at the node located at 45º angle of hub bolt hole (the failure location):

Apply \( P_{\text{max}} = 6230 \text{ N} \rightarrow \sigma_{\text{max}} = 296.5 \text{ MPa} & \epsilon_{\text{max}} = 0.0067 \)

Apply \( \Delta P = 6005 \text{ N} \rightarrow \Delta \sigma = 430.9 \text{ MPa} & \Delta \epsilon = 0.00489 \)

Therefore:

\[
\sigma_s = 215.5 \text{ MPa} \quad \epsilon_s = 0.0024
\]

\[
\sigma_m = 81.0 \text{ MPa} \quad \epsilon_m = 0.0043
\]

\[
\sigma_{\text{min}} = -134.5 \text{ MPa} \quad \epsilon_{\text{min}} = 0.0018
\]

Use modified Goodman equation to account for mean stress and find life:

\[
\frac{\sigma_a}{\sigma_{Nf}} + \frac{\sigma_m}{S_u} = 1 \rightarrow \frac{215.5}{\sigma_{Nf}} + \frac{81.0}{302} = 1 \rightarrow \sigma_{Nf} = 294.2 \text{ MPa}
\]

\[
\sigma_{Nf} = \sigma'_f (2N_f)^b \rightarrow 294.2 = 666(2N_f)^{0.117} \rightarrow N_f = 550 \text{ cycles}
\]
Use Gerber equation to account for mean stress and find life:

\[
\frac{\sigma_a}{\sigma_{nf}} + \left( \frac{\sigma_m}{S_u} \right)^2 = 1 \rightarrow \frac{215.5}{\sigma_{nf}} + \left( \frac{81.0}{302} \right)^2 = 1 \rightarrow \sigma_{nf} = 232.1 \text{ MPa}
\]

\[
\sigma_{nf} = \sigma_f^2 \left( 2N_f \right)^b \rightarrow 232.1 = 666 \left( 2N_f \right)^{0.117} \rightarrow N_f = 4200 \text{ cycles}
\]

Use the stress-life version of SWT model to account for mean stress and calculate life:

\[
\sigma_{\text{max}} \sigma_a = \sigma_f^2 \left( 2N_f \right)^b \rightarrow 296.5 \times 215.5 = 666^2 \left( 2N_f \right)^{2c-0.117} \rightarrow N_f = 2000 \text{ cycles}
\]

**With Maximum Principal Stress and Strain**

Perform nonlinear FEA and obtain results (maximum principal) at the node located at 45° angle of hub bolt hole (the failure location):

Apply \( P_{\text{max}} = 6230 \text{ N} \) → find \( \sigma_{\text{max}} = 317.2 \text{ MPa} \) & \( \varepsilon_{\text{max}} = 0.0072 \)

Apply \( \Delta P = 6005 \text{ N} \) → find \( \Delta \sigma = 450.9 \text{ MPa} \) & \( \Delta \varepsilon = 0.0056 \)

Therefore:

\( \sigma_x = 225.5 \text{ MPa} \quad \varepsilon_x = 0.0028 \)

\( \sigma_y = 91.7 \text{ MPa} \quad \varepsilon_y = 0.0044 \)

\( \sigma_{\text{min}} = -133.8 \text{ MPa} \quad \varepsilon_{\text{min}} = 0.0016 \)

Use modified Goodman equation to account for mean stress and find life:

\[
\frac{\sigma_a}{\sigma_{nf}} + \frac{\sigma_m}{S_u} = 1 \rightarrow \frac{225.5}{\sigma_{nf}} + \frac{91.7}{302} = 1 \rightarrow \sigma_{nf} = 323.5 \text{ MPa}
\]

\[
\sigma_{nf} = \sigma_f^2 \left( 2N_f \right)^b \rightarrow 323.5 = 666 \left( 2N_f \right)^{0.117} \rightarrow N_f = 240 \text{ cycles}
\]

Use Gerber equation to account for mean stress and find life:

\[
\frac{\sigma_a}{\sigma_{nf}} + \left( \frac{\sigma_m}{S_u} \right)^2 = 1 \rightarrow \frac{225.5}{\sigma_{nf}} + \left( \frac{91.7}{302} \right)^2 = 1 \rightarrow \sigma_{nf} = 248.3 \text{ MPa}
\]

\[
\sigma_{nf} = \sigma_f^2 \left( 2N_f \right)^b \rightarrow 248.3 = 666 \left( 2N_f \right)^{0.117} \rightarrow N_f = 2350 \text{ cycles}
\]

Use the stress-life version of SWT model to account for mean stress and calculate life:
\[
\sigma_{\text{max}} \sigma_a = \sigma_f^2 (2N_f)^{2b} \rightarrow 317.2 \times 225.5 = 666^2 (2N_f)^{2c-0.117} \rightarrow N_f = 1250 \text{ cycles}
\]

VI Local \(\sigma\)-N Approach using Linear FEA Results and Neuber-Correction for CA Steering knuckle

Perform linear FEA and obtain results (von Mises) at the node located at 45° angle of hub bolt hole (the failure location):

Apply \(P_{\text{max}} = 6230 \text{ N} \rightarrow \sigma_{\text{max}} = 446.1 \text{ MPa} & \varepsilon_{\text{max}} = 0.0051\)

Solve Neuber’s correction to obtain elastic-plastic stress and strain along with the material law:

\[
\sigma_{\text{eq}} \varepsilon_{\text{eq}} = \sigma_{\text{eq}} \varepsilon_{\text{eq}} \rightarrow \sigma_{\text{eq}} \varepsilon_{\text{eq}} = 2.28
\]

\[
\varepsilon_{\text{eq}} = \frac{\sigma_{\text{eq}}}{E} + \left(\frac{\sigma_{\text{eq}}}{K'}\right)^\frac{V}{\alpha} \rightarrow \varepsilon_{\text{eq}} = \frac{\sigma_{\text{eq}}}{201500} + \left(\frac{\sigma_{\text{eq}}}{1269.5}\right)^{\frac{V}{0.137}}
\]

to find:

\(\sigma_{\text{max}} = 302.1 \text{ MPa} & \varepsilon_{\text{max}} = 0.0075\)

Apply \(\Delta P = 6005 \text{ N} \rightarrow \Delta \sigma = 430.9 \text{ MPa} & \Delta \varepsilon = 0.0049\)

Solve Neuber’s correction to obtain elastic-plastic stress and strain along with the material law for unloading:

\[
\Delta \sigma_{\text{eq}} \Delta \varepsilon_{\text{eq}} = \Delta \sigma_{\text{eq}} \Delta \varepsilon_{\text{eq}} \rightarrow \Delta \sigma_{\text{eq}} \Delta \varepsilon_{\text{eq}} = 2.11
\]

\[
\Delta \varepsilon_{\text{eq}} = \frac{\Delta \sigma_{\text{eq}}}{E} + 2 \left(\frac{\Delta \sigma_{\text{eq}}}{2K'}\right)^\frac{V}{\alpha} \rightarrow \Delta \varepsilon_{\text{eq}} = \frac{\Delta \sigma_{\text{eq}}}{201500} + 2 \left(\frac{\Delta \sigma_{\text{eq}}}{2 \times 1269.5}\right)^{\frac{V}{0.137}}
\]

to find:

\(\Delta \sigma = 405.6 \text{ MPa} & \Delta \varepsilon = 0.0052\)

Therefore:

\(\sigma_a = 202.8 \text{ MPa} \quad \varepsilon_a = 0.0026\)

\(\sigma_m = 99.3 \text{ MPa} \quad \varepsilon_m = 0.0049\)

\(\sigma_{\text{min}} = -103.5 \text{ MPa} \quad \varepsilon_{\text{min}} = 0.0023\)
Use modified Goodman equation to account for mean stress and find life:

\[
\sigma_u + \frac{\sigma_m}{S_u} = 1 \rightarrow \frac{202.8}{\sigma_{N_f}} + \frac{99.3}{302} = 1 \rightarrow \sigma_{N_f} = 301.8 \text{ MPa}
\]

\[
\sigma_{N_f} = \sigma'_f (2N_f)^b \rightarrow 301.8 = 666(2N_f)^{0.117} \rightarrow N_f = 450 \text{ cycles}
\]

Use Gerber equation to account for mean stress and find life:

\[
\frac{\sigma_u}{\sigma_{N_f}} + \left(\frac{\sigma_m}{S_{N_f}}\right)^2 = 1 \rightarrow \frac{202.8}{\sigma_{N_f}} + \left(\frac{99.3}{302}\right)^2 = 1 \rightarrow \sigma_{N_f} = 227.3 \text{ MPa}
\]

\[
\sigma_{N_f} = \sigma'_f (2N_f)^b \rightarrow 227.3 = 666(2N_f)^{0.117} \rightarrow N_f = 5000 \text{ cycles}
\]

Use the stress-life version of SWT model to account for mean stress and calculate life:

\[
\sigma_{max} \sigma_u = \sigma'_f (2N_f)^2 \rightarrow 302.1 \times 202.8 = 666^2 (2N_f)^{2 \times 0.117} \rightarrow N_f = 2400 \text{ cycles}
\]

**VII ε-N Approach using Nonlinear FEA Results for FS Steering knuckle**

See local stresses and strains in local σ-N approach using nonlinear FEA results for the FS steering knuckle.

Use Morrow equation to account for mean stress and calculate life:

\[
\varepsilon_a = \frac{\sigma'_f - \sigma_m}{E} (2N_f)^b + \varepsilon'_f (2N_f)^c
\]

\[
0.00246 = \frac{1156.8 - 169.6}{201500} (2N_f)^{-0.082} + 3.0315 (2N_f)^{-0.7912} \rightarrow N_f = 31700 \text{ cycles}
\]

Use Manson and Halford equation to account for mean stress and calculate life:

\[
\varepsilon_a = \frac{\sigma'_f - \sigma_m}{E} (2N_f)^b + \varepsilon'_f \left(\frac{\sigma'_f - \sigma_m}{\sigma'_f}\right)^\frac{b}{2} (2N_f)^c
\]

\[
0.00246 = \frac{1156.8 - 169.6}{201500} (2N_f)^{-0.082} + 3.0315 \left(\frac{1156.8 - 169.6}{1156.8}\right)^{-0.7912/0.082} (2N_f)^{-0.7912}
\]

\[\rightarrow N_f = 9400 \text{ cycles}\]
Use SWT equation to account for mean stress and calculate life:

$$
\sigma_{\text{max}} E = (\sigma'_f)^2 (2N_f)^{2b} + \sigma'_f \epsilon'_f E (2N_f)^{b+c}
$$

$$
603.3 \times 0.00246 \times 201500 = 1156.8^2 (2N_f)^{2c-0.082} + 1156.8 \times 3.0315 \times 201500 (2N_f)^{-0.082-0.7912}
$$

$$
\rightarrow N_f = 20900 \ \text{cycles}
$$

**VIII \ \epsilon-N \ Approach \ using \ Linear \ FEA \ Results \ and \ Neuber-Correction \ for \ FS \ Steering \ knuckle**

See local stresses and strains in local $\sigma-N$ approach using linear FEA results for the FS steering knuckle.

Use Morrow equation to account for mean stress and calculate life:

$$
\epsilon_a = \frac{\sigma'_f - \sigma_m}{E} (2N_f)^{b} + \epsilon'_f (2N_f)^{c}
$$

$$
0.0026 = \frac{1156.8 - 177.3}{201500} (2N_f)^{-0.082} + 3.0315 (2N_f)^{-0.7912} \rightarrow N_f = 24300 \ \text{cycles}
$$

Use Manson and Halford equation to account for mean stress and calculate life:

$$
\epsilon_a = \frac{\sigma'_f - \sigma_m}{E} (2N_f)^{b} + \epsilon'_f \left( \frac{\sigma'_f - \sigma_m}{\sigma'_f} \right)^{\frac{x}{2}} (2N_f)^{c}
$$

$$
0.0026 = \frac{1156.8 - 177.3}{201500} (2N_f)^{-0.082} + 3.0315 \left( \frac{1156.8 - 177.3}{1156.8} \right)^{-0.7912-0.082} (2N_f)^{-0.7912}
$$

$$
\rightarrow N_f = 6200 \ \text{cycles}
$$

Use SWT equation to account for mean stress and calculate life:

$$
\sigma_{\text{max}} E = (\sigma'_f)^2 (2N_f)^{2b} + \sigma'_f \epsilon'_f E (2N_f)^{b+c}
$$

$$
614.5 \times 0.0026 \times 201500 = 1156.8^2 (2N_f)^{2c-0.082} + 1156.8 \times 3.0315 \times 201500 (2N_f)^{-0.082-0.7912}
$$

$$
\rightarrow N_f = 16700 \ \text{cycles}
$$
IX ε-N Approach using Nonlinear FEA Results for CA Steering knuckle

With von Mises Stress and Strain

See local stresses and strains in local σ-N approach using nonlinear FEA with von Mises stress and strain for the CA steering knuckle.

Use Morrow equation to account for mean stress and calculate life:

\[ \varepsilon_a = \frac{\sigma'_f - \sigma_m}{E} (2N_f)^b + \varepsilon'_f (2N_f)^c \]

\[ 0.0024 = \frac{666.0 - 81.0}{78100} (2N_f)^{-0.117} + 0.0944 (2N_f)^{-0.6095} \rightarrow N_f = 16100 \text{ cycles} \]

Use Manson and Halford equation to account for mean stress and calculate life:

\[ \sigma'_f - \sigma_m = \left( \frac{\sigma'_f - \sigma_m}{\sigma'_f} \right)^{\gamma_b} (2N_f)^c \]

\[ 0.0024 = \frac{666.0 - 81.0}{78100} (2N_f)^{-0.117} + 0.0944 \left( \frac{666.0 - 81.0}{666.0} \right)^{-0.6095/-0.117} (2N_f)^{-0.7912} \]

\[ \rightarrow N_f = 12400 \text{ cycles} \]

Use SWT equation to account for mean stress and calculate life:

\[ \sigma_{max} \varepsilon_a E = (\sigma'_f)^2 (2N_f)^{2b} + \sigma'_f \varepsilon'_f E (2N_f)^{b+c} \]

\[ 296.5 \times 0.0024 \times 78100 = 666.0^2 (2N_f)^{2 \times -0.117} + 666.0 \times 0.0944 \times 78100 (2N_f)^{-0.117-0.6095} \]

\[ \rightarrow N_f = 5750 \text{ cycles} \]

With Maximum Principal Stress and Strain

See local stresses and strains in local σ-N approach using nonlinear FEA with maximum principal stress and strain for the CA steering knuckle.

Use Morrow equation to account for mean stress and calculate life:

\[ \varepsilon_a = \frac{\sigma'_f - \sigma_m}{E} (2N_f)^b + \varepsilon'_f (2N_f)^c \]
Use Manson and Halford equation to account for mean stress and calculate life:

\[
0.0028 = \frac{666.0 - 91.7}{78100} \left(2N_f\right)^{-0.117} + 0.0944 \left(2N_f\right)^{-0.6095} \rightarrow N_f = 5600 \text{ cycles}
\]

Use SWT equation to account for mean stress and calculate life:

\[
\sigma_{\text{max}} \varepsilon_a E = (\sigma_f')^2 (2N_f)^{2b} + \sigma_f' \varepsilon_f' E (2N_f)^{b+c}
\]

\[
317.2 \times 0.0028 \times 78100 = 666.0^2 (2N_f)^{-0.117} + 666.0 \times 0.0944 \times 78100 (2N_f)^{-0.6095}
\]

\[
\rightarrow N_f = 2700 \text{ cycles}
\]

X \ \varepsilon-N Approach using Linear FEA Results and Neuber-Correction for CA Steering knuckle

See local stresses and strains in local \(\sigma-N\) approach using linear FEA results with von Mises stress and strain for the CA steering knuckle.

Use Morrow equation to account for mean stress and calculate life:

\[
\varepsilon_a = \frac{\sigma_f' - \sigma_m}{E} \left(2N_f\right)^{b} + \varepsilon_f' \left(2N_f\right)^{c}
\]

\[
0.0026 = \frac{666.0 - 99.3}{78100} \left(2N_f\right)^{-0.117} + 0.0944 \left(2N_f\right)^{-0.6095} \rightarrow N_f = 8100 \text{ cycles}
\]

Use Manson and Halford equation to account for mean stress and calculate life:

\[
\varepsilon_a = \frac{\sigma_f' - \sigma_m}{E} \left(2N_f\right)^{b} + \varepsilon_f' \left(\frac{\sigma_f' - \sigma_m}{\sigma_f'}\right)^{\gamma_b} \left(2N_f\right)^{c}
\]

\[
0.0028 = \frac{666.0 - 91.7}{78100} \left(2N_f\right)^{-0.117} + 0.0944 \left(2N_f\right)^{-0.6095} \rightarrow N_f = 5600 \text{ cycles}
\]
Use SWT equation to account for mean stress and calculate life:

\[
\sigma_{\text{max}} \varepsilon_a E = (\sigma'_m)^2 (2N_f)^{2b} + \sigma'_m \varepsilon'_m E (2N_f)^{b+c}
\]

\[
302.1 \times 0.0024 \times 78100 = 666.0^2 (2N_f)^{2e-0.117} + 666.0 \times 0.0944 \times 78100 (2N_f)^{-0.117-0.6095}
\]

\[
\rightarrow N_f = 4050 \text{ cycles}
\]