RESOLUTION OF INDETERMINATE LANDING GEAR STRUCTURE DESIGN

by

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ABSTRACT

Aircraft landing gear structural designs involves a balance of weight, cost and robustness while not compromising on safety. On some large commercial aircraft, the introduction of a second main supporting brace has led to an indeterminate structure in that there is redundancy in the load paths. This introduces two major challenges for structural design. The first challenge involves the introduction of a multiple load path. Understanding load path in the landing gear is critical in order to optimize the structure for weight. This report focuses on analysis techniques geared to resolving this indeterminate load path in order to mitigate this risk and optimize the design. The second major challenge is introduced by a compressive load in one of the braces during an in flight airload condition which impedes the ability for the landing gear to freefall, which is a requirement in aircraft design. Solving this problem involves introducing a pre-tension in the brace by force shortening the geometry. An indeterminate design introduces increased complexity and requires more simulation and analysis than that of a determinant design in order to accomplish the optimization demanded by the aerospace industry.

AKNOWLEDGEMENTS

The author wishes to acknowledge Messier-Dowty Inc.

Messier-Dowty is the global leader in the design and manufacturing of landing gear systems with a stake of over 50% of the landing gear market. Based in France, Messier-Dowty has sites in France, England, Canada, and the United States.

The author has worked as a Stress Analyst at Messier-Dowty for 4 years prior to the writing of this report, working to design and solve the problems of Indeterminate Landing Gears. Most of the analysis and modeling presented in this report was done by the author for Messier-Dowty and is the property of Messier-Dowty Inc.

The authors experience as Stress Lead on the Sukhoi Superjet-100 has led to a first hand knowledge of the design of Indeterminate Landing Gears thanks to the opportunities presented by Messier-Dowty.

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TABLE OF CONTENTS

LIST OF FIGURES

LIST OF APPENDICES

NOMENCLATURE

MLG - Main Landing Gear

NLG – Nose Landing Gear

RR – Tire Rolling Radius

VAT – Vertical Axle Travel

Kt – Stress Concentration Factor

FTU – Ultimate Tensile Strength Allowable

FTY – Yield Strength Allowable

FBRU – Ultimate Bending Strength Allowable

FSU – Ultimate Shear Strength Allowable

FOU – Trapezoidal Intercept Stress

L – Socket Length

A1, A2 – Socket Bending Moment Arm

B1, B2 – Socket Bearing Length

W1, W2 – Socket Load per Unit Length

R1, R2 – Socket Reaction Forces

M – Moment

F – Force

Q – First Moment of Inertia

I – Second Moment of Inertia

ε_f' - fatigue ductility coefficient

$2N_f$ – reversals to failure

σ_f' – fatigue strength coefficient

b – fatigue strength exponent

c – fatigue ductility exponent

1 INTRODUCTION

The landing gear is a primary structure of the aircraft which has the function of absorbing the energy of the aircraft landing. The landing gear also provides many other functions such as steering, retracting and extending, as well as braking. Most civil landing gears are designed for safe life design as opposed to a damage tolerant design seen on much of the aircraft. A safe life design means that the landing gear is designed and qualified to meet the required life of the aircraft.

Modern day design of landing gear structures uses multiple analysis tools to ensure the design meets three prime objective:

1) Safety, 2) Weight, and 3)Cost

In landing gear design with regards to the first objective, Safety, there is no compromise and the success in meeting this objective is done through analysis and testing. The second and third objectives are more commercial objectives where priority can vary from one project to another. Cost versus weight tradeoffs are common to the design iteration process as the removal of weight from components typically leads to increased manufacturing costs while the failure to meet a weight targets is typically met with penalties agreed upon in the contract with the airframer.

In the structural analysis of landing gear, a significant effort is made to get it right the first time. Components do not go into production until there is confidence based on the analysis that the components will meet their strength and fatigue requirements. Qualification of landing gear to be used on an aircraft is based on tests for strength, fatigue, endurance, and environmental. The analysis done in the design stage is to ensure that the landing gear will pass these tests on the first attempt. The failure to qualify components in the first attempt leads to added cost in re-design and re-testing of the component as well as potential life limitation and replacement of early production units in the case of failures on fatigue test. For this reason, advances analysis techniques are used to simulate the required testing in the design phase.

There are many different variations of landing gear geometries. Landing gear geometries typically need to meet the following three objectives, 1. Transfer loads into the aircraft efficiently, 2. Fit into the allotted space in the bay, and 3. Be retractable. For a given aircraft geometry, those three criteria limit the possible landing gear geometries.

On some of the larger aircraft, the need to efficiently transfer loads from the ground into the wing, drives the design of an indeterminate landing gear structure. Indeterminance in a structure means that there are multiple load paths. This is to say that static equilibrium cannot be determined based on geometry alone. The stiffness in the system becomes required in order to solve equilibrium.

On a landing gear structure, the indeterminate structure comes through the introduction of a second brace. Braces on landing gears are used to support the main leg of the landing gear under lateral loading conditions. This dual brace landing gear design was originally introduced by Boeing Aircraft Company. It is rumored that the a second brace was introduced as a required fix on one of the early Boeing aircraft and has been grandfathered into all new Boeing designs ever since.

The introduction of the second brace introduces more design challenges then could be imagined at first conception. Obviously, the redundancy of the structure requires additional analysis and design iteration in order to properly understand the load paths and optimize the structure for weight. Another significant design concern which was not initially known, is that there is a condition in flight during extension of the landing gear, where the airloads acting on the landing gear may put compressive loading into the aft brace such that full extension in a freefall condition is not possible. In order to resolve this issue, the aft brace is designed with a length shorter than the non-deflected geometry in order to preload the brace with a tensile load.

Boeing is the only Aircraft Manufacturer with the dual brace landing gear in service today. Some recent aircraft currently in development have adopted this Boeing design in order to optimize the design of their wing. The Sukhoi Superjet-100 and the Airbus A350 are two development aircraft which have adopt the dual brace design.

This report investigates the analysis and simulation approach to modern day landing gear design based on the methods of an industry leading company in landing gear design. The methods and analysis approach presented is geared to the resolution of the design problem of an indeterminate landing gear.

2 STUCTURAL DESIGN APPROACH

The design of a landing gear structure goes through many milestones throughout its development. It is important that the landing gear design integrates with the airframe structure, which is more than simply geometry. When designing an indeterminate load path system into the landing gear, the flexibility of the wing attachments become a requirement to proceed with the landing gear design.

In the early stages of design once a contract has been awarded, the JDP (Joint Development Phase) portion of the development in underway. During this time, the landing gear designers work closely with the airframe designers to come up with a design concept that meet the objectives. This phase ends with the Preliminary Design Review (PDR) before which geometry and basic design approach needs to be agreed between the landing gear design and airframe design. After PDR, the detailed design phase involves issuing detailed drawings. In order to reach the next milestone of CDR, a typical CDR deliverable is 80% of detailed drawing released. With the completion of CDR, manufacturing begins to cut metal chips and manufacturer the flight test and qualification test hardware. Figure 1 shows these design milestones in the order in which they occur.

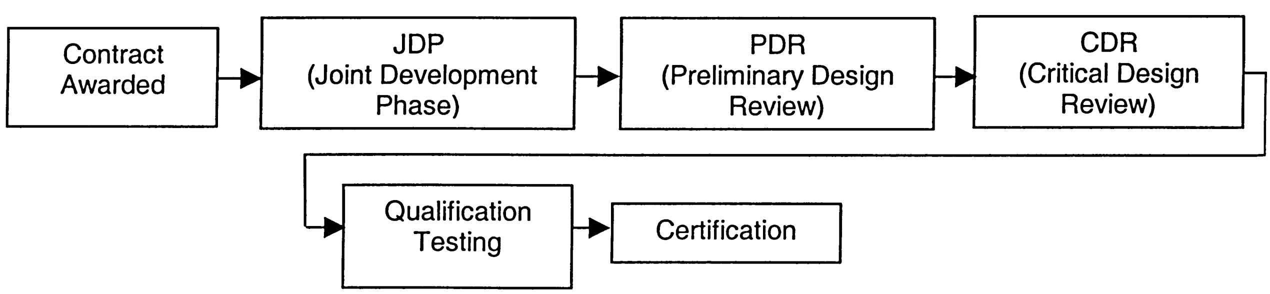


Figure 1: Landing Gear Structural Design Milestones

The design approach is iterative as changes throughout an iteration can affect the initial inputs of that iteration. This is even more predominant when designing a dual brace indeterminate landing gear structure. Initially, a design concept is developed to meet the requirement of fitting into the aircraft and meeting the airframers requirements for gear location, height, and tire selection. With this basic definition of the landing gear, as well as the center of gravity (c.g.) box defined for the aircraft, definition of the Ground load conditions can begin. The ground loads are required to begin the structural analysis of the landing gear. With the geometry of the landing gear defined, an internal load finite element model can be constructed. With an indeterminate landing gear design, it is necessary to have the aircraft flexibility modeled into this finite element model at this time. The internal loads allow for understanding the load paths throughout the landing gear and to do all the basic sizing of all major landing gear structural components. This basic sizing would include selection of optimal cross sectional shapes for the nature of the loading, as well as appropriate materials. Both cross sectional shape and material selection may be limited by or selected based on form and function of the design. The internal loads also provide a flexibility of the landing gear required as an input for simulating the landing loads which are also required for structural analysis.

Figure 2 outlines the flow of information involved in the iteration of the landing gear design process. For structural design, three main disciplines are interconnected in the information flow in order to progress the design. Design, Dynamics and Stress departments work as a team to flow the required information and specialties in order to iterate through a design.

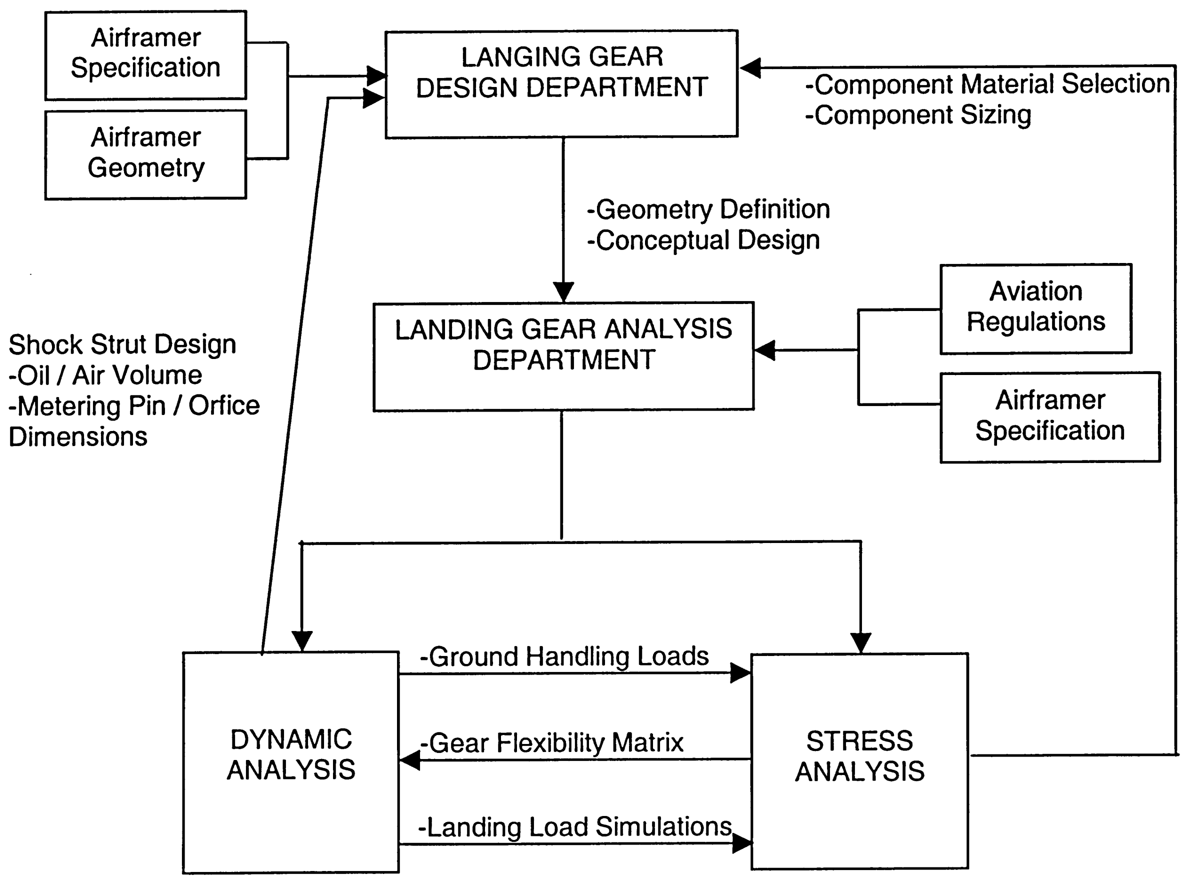
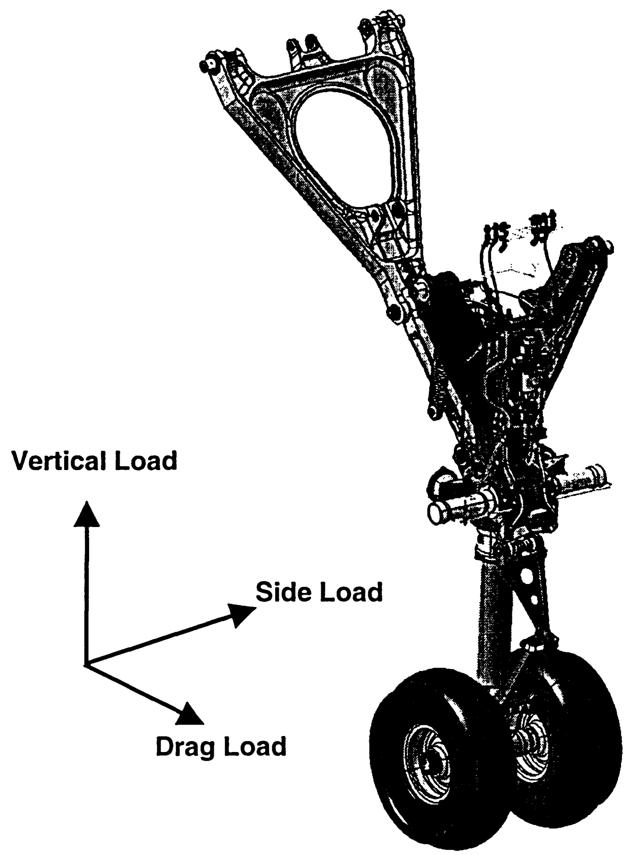


Figure 2: Design Iteration Flow

3 GROUND LOADS

The term 'ground loads' refers to the loads acting on the landing gear tires which loads the axle. On a typical tripod arrangement common to most civil and military aircraft, there is a single Nose Landing Gear (NLG) and two Main Landing Gears (MLG) attached to either the wing or the fuselage in the area of the wings. The share of the aircraft weight on the NLG or the MLGs is based on the center of gravity of the aircraft. During ground maneuvering events such as braking, the NLG can see added loading due to pitch over which is a function of mass, vertical c.g. location, and deceleration. The load across the MLGs can be distributed during maneuvering events such as a turning event where a centrifugal force at the aircraft c.g. forcing more load on the MLG furthest from the turning center.

For all ground and landing conditions considered, the loads are calculated for each landing gear in terms of an orthogonal coordinate system in either ground or aircraft reference frames. The three dimensions of the load coordinate frame are defined as vertical, drag, and side where vertical is up (Z), drag is positive aft (X), and side is positive inboard (Y). As a standard for MLG analysis, the port gear is typically analyzed. Figure 3 shows the right handed coordinate system used in landing gear structural analysis.



Figue 3: Right Handed Landing Gear Coordinate Frame

3.1 Landing Loads

Landing loads simulations will typically consider a dozen or more landing events with varying weight, forward velocity and decent velocity. Within each of these conditions, there are various load events to be considered as the energy is absorbed and the gear oscillates in the forward aft directions (Figure 4).

PROPRIETARY INFORMATION REMOVED

Figure 4 : Landing Simulation Loads Plot (Ref. 13)

During a landing, three major loading events occur which are significant to the design of the landing gear. The first loading event that occurs during a landing is called the Spin Up event in which the positive drag load (FX) acting aft peaks. This loading event occurs as the wheels are spun up from an angular velocity of zero, to one that matches the speed of the landing aircraft. Since the tires are free to spin (i.e., there are no brakes applied at this time), the inertial loading of spinning up the tire translates into a peak in aft acting drag load on the axle.

The reaction to this Spin Up load is called the spring back which is a peak in the forward acting drag load (-FX) on the landing gear. This peak is always less than the Spin Up peak as there is damping in the system. The final loading even during a landing to consider is the Maximum Vertical Reaction (MVR) which occurs after the Spin Up and Spring Back oscillations. This is where most of the energy of the landing is absorbed. The significance of the loading during the MVR is the magnitude of the vertical load (FX).

Another significant consideration with each of the three landing loading events is the VAT at which each event occurs. Vertical Axle Travel (VAT) is a measure of the shock strut stroke in the vertical axis of the aircraft coordinate frame. There is a direct relationship between VAT and shock strut stroke. For a non-articulated landing gear (cantilever type), the relationship may be based on a simple rake angle of the shock strut relative to the vertical axis. For a articulated landing gear (trailing arm type), the relationship between stroke and VAT is based on a more complicated calculation of mechanical advantage.

The importance of VAT for a given loading condition is due to the fact that it defines the geometry of the structure under the loading condition. For example, under a Spin-Up loading condition with aft acting drag loads, a smaller VAT translates into a further 'boomed out' strut resulting in larger moment arms thereby more critical to the landing gear and aircraft structure.

The nose landing gear design is particularly designed by landing load conditions, as it does not see significant lateral loading from braking and turning described in Section 7.3. The lateral load conditions that design the piston and braces on a nose gear are due to landing 'Spin Up' and Lateral Drift Landing conditions. Typically Piston and Axle diameters, as well as braces on a nose landing gear, are sized by the 'Spin Up' and 'Spring Back' landing conditions, where at the trunnion arms and aircraft attachments are designed by the Lateral Drift Landing condition.

3.2 Ground Handling Loads

Ground handling loads are very significant to the landing gear design. The calculation of these loads are particular to the static weight distribution of a particular aircraft and a particular landing gear on the aircraft, but acceleration and friction requirements are defined in the Federal Aviation Regulations. The most common ground handling loads seen by an aircraft are Turning events and Braking events. These two loads are particularly important in the design of the main landing gears as they typically represent the highest side load condition and aft acting drag load condition.

The Turning condition requires that the lateral load be 0.5 of the vertical reaction [Ref. 14, FAR 25.495]. This is a significant load condition which typically sizes the axle and piston diameters of a main landing gear.

The Braked Roll condition requires the consideration of a coefficient of friction of 0.8 with the associated vertical reaction [Ref. 14, FAR 25.493]. This condition typically sizes braces and Main Fittings on the main landing gear of an aircraft.

The ground handling loads are significant design conditions as unlike the landing load conditions, there is no means of modifying parameters to reduce the loads as the regulation are set based on the static weight distribution of the aircraft. Landing loads can be optimized for the structure by profiling the metering pin and orifice, thereby, controlling the oil/air mixture, and thus the energy absorption characteristics of the strut. A good design of a main landing gear aims at designing the landing loads to be slightly less critical than the ground handling loads, thus not designing any components. The tradeoff is that by profiling the metering pin such that the 'Spin Up' loads are less critical than the Braked Roll condition, the consequence is that the Maximum Vertical Reaction (MVR) increases which can be critical for the aircraft attachments.

3.3 Take Off Loads

Little in the way of takeoff loads is critical to the landing gear, with exception to rejected takeoff conditions, which are considered in Ground Handling Loads. In recent years, however, flight test data has revealed that, during a takeoff event, there is a loading internal to the shock strut which is of some significance. As the aircraft takes off, the lift off can occur quite rapidly, leaving little time for the shock strut to extend. The result of such a rapid extension result in the strut free extending into the fully extended state. The significance of the Free Extension is based on the stroke from which it is free to extend from. The further the strut is stroked in, the more significant the loading for the strut and it has further distance to extend. The weight of the unsprung mass, which includes the rolling stock such as the wheels, tires and brakes, as well as the piston, axle and lower torque link, increases the velocity of which strut is extending, and all the energy associated with this is taken out internally through the strut as it bottoms.

The consideration of this is a design condition for some internal components for both strength considerations as well as fatigue.

4 INTERNAL LOADS

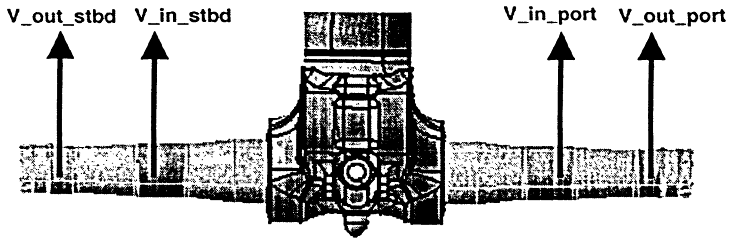
Significant derivation and simulation are required in defining the Ground Loads. The Ground Loads define the load acting at the axle and tires in the Vertical, Forward/Aft, and Side directions and the stroke positions at which each of the load conditions occur. With this information, a more detailed analysis of the load path through the landing gear structure, into the airframe structure is required for sizing of both the landing gear and the airframe. The ground loads are applied to a Finite Element beam model through loading at the axle and reacted at the airframe in order to derive the internal loads of the landing gear structure. This section describes the details and complexities of this analysis.

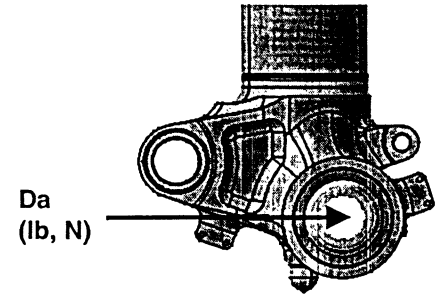
4.1 Wheel Load Distribution

With the ground loads for each design condition defined, the consideration of how this load is introduced into the wheel/tire is next considered. Most typical medium size aircraft have landing gear with two tires per landing gear which will be considered in this example. When there is more than one tire per landing gear, the distribution across the two wheels in this case needs to be considered. For purposes of strength analysis, each of the load cases is typically considered with the following five wheel load distributions; 50:50, 60:40, 40:60, 100:0, 0:100. The wheel load distribution of 100:0 and 0:100 are flat tire conditions. FAR requirements for these flat tire conditions typically require the flat tire ground load to be 60% of that of both tires inflated.

4.2 Axle Loading

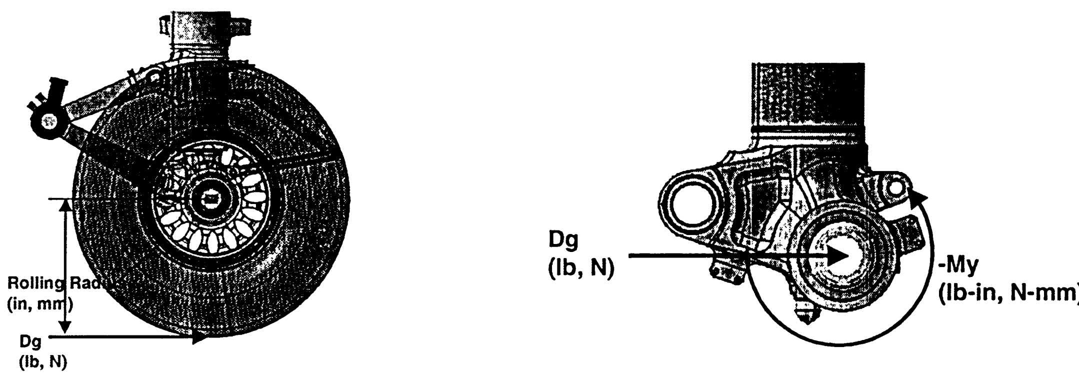
Each of the ground load components, vertical, drag, and side will be considered for how these loads transmit from the wheel/tire to the axle, and ultimately the rest of the landing gear structure. Vertical loading is the most simple load to conceptualize as the weight of the aircraft act on the tires which is transmitted into the axle through the wheel bearing (Figure 5a). A drag load in a load that acts forward/aft relative to the aircraft and can act either the axle (Figure 5b) or at the ground contact with the tire (Figure 5c). The difference in the loading location is due to the nature of the loading condition. For a drag load to act at the axle, this is due to the inertial loading when the aircraft first touches down and the 'Spin Up' and 'Spring Back' loading events occur. The touchdown of the aircraft 'Spins Up' the tires creating an inertial loading in the aft direction with an oscillating 'Spring Back' affect creating a forward acting load. The drag load will act at the tire interface with the ground during a braking event where an aft acting drag load will be generated. A reverse braking event will create a forward acting drag load at the ground. The side load condition (Figure 5d) occurs when during a turning event or a drift landing event. Since the method of this load transfer is via friction between the tire and ground, the side load always acts at the ground. The significance of understanding whether the load acts at the ground or at the axle is to properly understand any additional moment arms introduced into the system.



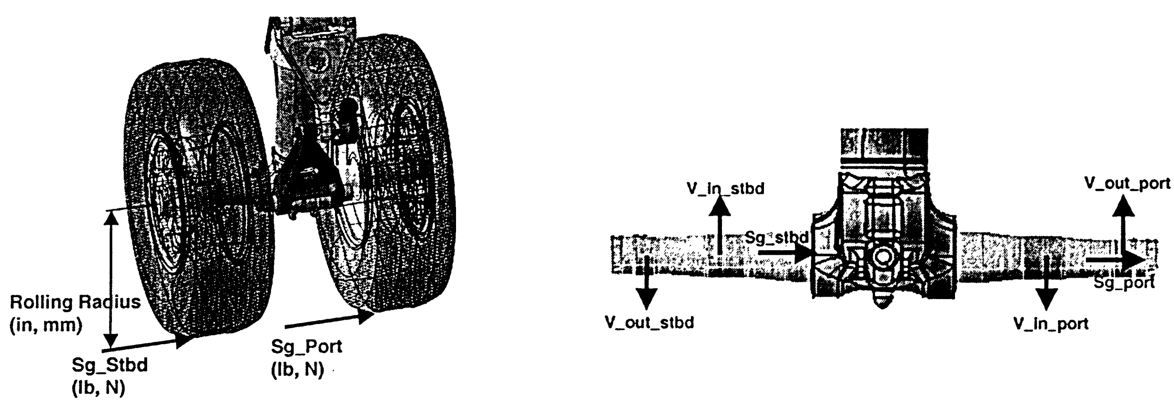


5a) Vertical Loading (FZ)

5b) Drag Loading at the axle
(FX)



5c) Drag Loading at the Ground due to Braking (FX)



5d) Side Loading (FY)

Figure 5: Axle Loading

4.3 Strength Loads Set Reduction

For strength analysis of the landing gear, all landing, ground handling, and other load condition requirements are examined to determine the critical cases for different regions of the landing gear mechanism. There is not one condition that is critical for strength analysis of the landing gear, as different load conditions are critical for different regions of the structure. Practically speaking there are typically a half dozen to a dozen load conditions that will be critical for different regions of the structure. Determining these critical load conditions cannot be done at the axle load level, however, and requires internal loads analysis to determine all axle loading conditions and their effect to the internal loads.

The complete loads set contains all loading conditions to consider and can consist of a couple of hundred load cases. This complete load set can be reduces by inspection by eliminating less critical load conditions based on load magnitude and landing gear stroke position. This is best explained by example. In the following Example, case 1 and 2 are inspected and it can be seen that the drag load is higher in case 1. Also, the Vertical Axle Travel (VAT) in case 1 in lower, meaning that the shock strut piston is further stoked out (i.e. gear is longer overall), which makes case 1 even more critical than case 2. For this reason, Case 2 can be eliminated from the complete loads set without any further analysis. When comparing case 3 with case 1, the load magnitudes are lower, but the landing gear stroke is fully extended in case 3. Therefore, it is not obvious whether Case 1, or Case 3 is more critical without considering more analysis. For this reason, both Case 1 and Case 3 would be run through the finite element beam model to determine criticality.

Example:

Case 1 : V = 200 kips, D = 75 kips @ VAT = 3 in.

Case 2 : V = 200 kips, D = 70 kips @ VAT = 10 in.

Case 3 : V = 190 kips, D = 70 kips @ VAT = 0 in.

This reduction by inspection is done for all cases in the complete load set.

4.4 Finite Element Beam Model

The internal-load finite element model is a relatively simple Finite Element model in that it consists of a relatively small number of degrees of freedom. Typically, a model for a landing gear would consist of a few hundred elements and nodes. These models are simple in the sense that the only output required from these models are internal loads, reactions and deflections. There is no stress output used from these models, and they are used as a load path understanding of the mechanism.

In order to design components on a landing gear, the internal loads throughout the landing gear structure need to be known. This is done through the generation of a finite element beam model. The finite element beam model is modeled a series of connected beams each modeled with representative stiffness. The connections between components are modeled with proper releases of degrees of freedom (DOF) to ensure realistic load transfer across joints.

Typical Finite Element software used to solve landing gear beam models are Nastran, Abaqus, and internally developed codes. In the aircraft industry, Nastran is typically the choice of airframers for the aircraft loads model, and typically the landing gear model is made to be compatible with the airframers model.

The first input for the finite element beam model is the geometry. Geometry is defined and agreed upon with the customer in the early stages of program. This includes exact interface connections with the aircraft, as well as shock absorber stroke and wheel and tire interface. With the geometry defined, one can construct and finite element beam model with nodes and elements in space.

The second input for the finite element beam model is the component stiffnesses. In order to include proper stiffness in the first iteration of the beam model, engineering judgment is often used. This judgment includes initial material selection of components as well general section types and sizes. Future updates of the beam model will incorporate updated stiffness based on the sizing performed from the internal loads developed in the first iteration of the beam model.

The importance of component stiffness is secondary for most landing gears. Landing gear beam models are run non-linear such that secondary deflections are considered in the final solution. These secondary deflections are therefore dependent on the component stiffness. For most landing gears, these secondary effects typically can affect the result by 5 to 10%. For certain types of landing gear configurations, the structure is indeterminate meaning that there are redundant load paths. In this situation, gear stiffness and aircraft stiffness have a primary effect on the resulting internal loads.

The third input for the finite element beam model is the loads which includes the axle loads, as well as any other loading such as brake torque, and actuator loading. The axles loads are applied to the axle bearing locations represented by nodes on the finite element beam model. The method of developing the axle loads was described earlier. Figure 6 illustrates an example of an indeterminant landing gear beam model displayed in 3-d to show the representative stiffness of the element properties.

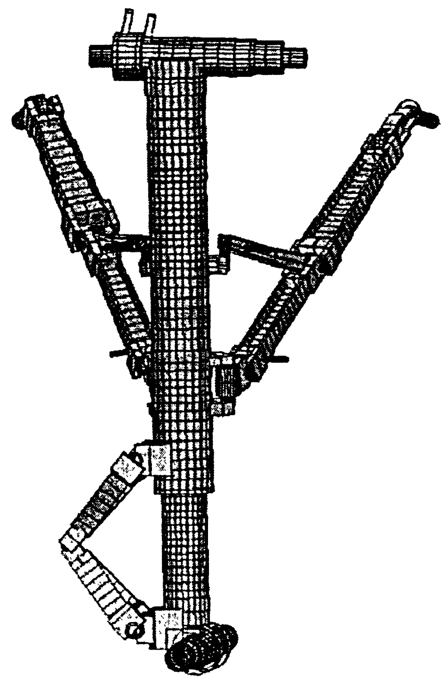


Figure 6: Finite Element Beam Model

Brake torque is reacted differently on different aircraft. In all configurations, the moments from the brake housing need to be transferred into the landing gear structure by some means. Typically the brake housing will be bolted on to a flange on the landing gear structure.

The load from the retraction actuator is also considered. Depending on the hydraulic schematic for a particular aircraft, the system pressure in the retraction actuator may be on or off during landing and ground handling events. If the retraction actuator pressure is left on, the addition of this load must be considered with the ground loads. Since this addition of load can be either additive or subtractive in different regions of the landing gear, both retraction actuator on and off are typically considered in the analysis.

The three main outputs from the finite element landing gear beam model are internal loads, aircraft reaction, and deflections. All these outputs are inputs into other levels of analysis of the landing gear.

Internal loads are required for strength analysis of the landing gear components. For classical analysis of components, the critical load conditions for the element and node of interest is used to calculate critical stresses for the section. For Finite Element analysis of components, the internal loads from the beam model are used to determine critical loads at the boundary conditions of the part to be analyzed.

The reactions of the finite element beam model of the landing gear are at the connections to the aircraft structure. The aircraft reactions for all critical load conditions considered are used for both calculation of the aircraft attachments on the landing gear side and the aircraft side. The airframer uses these reactions to size the structure in the wings/fuselage where the landing gears are installed.

The deflections from the finite element beam model are typically used as a check the analysis against the test article deflections. In the design phase, certain clearances within the landing gear structure are checked to ensure that there is no interferences such as tires with other structure.

5 STRENGTH ANALYSIS

The strength analysis of the landing gear involves ensuring that components do not yield and rupture for critical load conditions. The methods of strength analysis are separated into two streams: Classical Strength Analysis, and Finite Element Methods. Both methodologies can be used but both have advantages and disadvantages and typically complement each other when done in parallel.

Appendix 1 summarizes common basic stress formulas used in strength analysis of landing gear components. These formulas are based on solid mechanics theories.

5.1 Limit, Ultimate, and Design Conditions

Landing simulation and book case requirement dictate the limit conditions for design of the landing gear. The term limit implies that the condition is the worst of its kind that that particular aircraft can expect to see in service. For example, limit landing conditions consider high descent velocities for the simulation. For braking conditions, 0.5-0.6 G deceleration is considered. For turning conditions, high friction coefficients are considered to produce high side loads acting on the landing gear. When designing for limit load conditions, the pass criteria is no yielding.

The ultimate condition simply applies a safety factor of 1.5 to the limit load condition. The pass criteria for this condition allows gross section yielding but no rupture under this load condition.

For US military programs, requirements for Design conditions are sometimes specified. These Design conditions are considered much like the limit conditions but there is no safety factor to be applied to a design condition. MIL standards dictate the pass criteria for these conditions which generally requires no yielding. Some local yielding is acceptable however, provided that it does not affect the functionality of the landing gear.

5.2 Classical Strength Analysis

For many years before the introduction of finite elements, these classical methods were used for the entirety of the landing gear analysis. Analysis in this form requires preparing input files for Fortran programs to perform typical solid mechanics analysis such as buckling analysis, basic section analysis, lug analysis, and socket analysis to name a few. The inputs typically require information about the loading, the geometry, and material properties. The output from these Fortran analyse provides a detailed description of the stress calculation, as well as a calculation of a margin of safety.

5.2.1 Analysis by Sections

Analysis by sections is a very practical method for sizing components. This method involves analyzing the cross section of the component based on the loading in the component. On a landing gear, all major structural components have multiple section cuts and using the internal loads from the beam model, the critical cases for each section can be determined and the appropriate cross sectional sizes determined based on an acceptance criteria for the material allowables. The method is performed in the following steps.

Step 1 : Internal Loads from the Beam Model

The first step is to select the section location in the beam model and an appropriate element and node associated with the section of interest. Figure 7 shows the node and element of a particular section of interest from which the internal beam model loads are extracted.

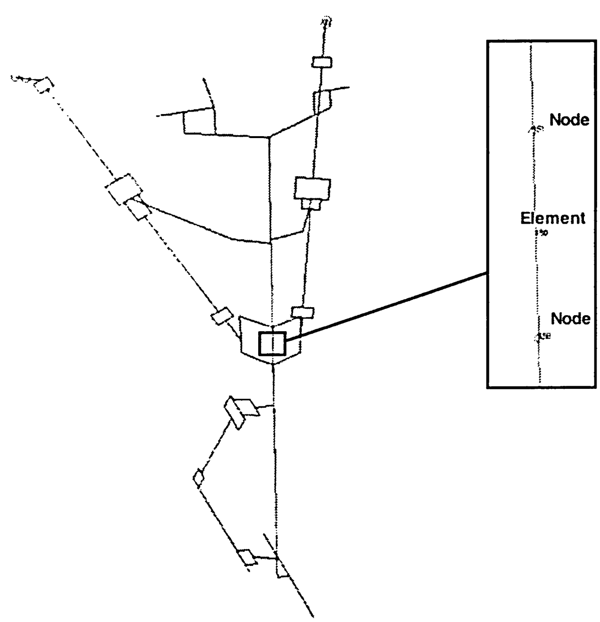


Figure 7 : Beam Model Internal Loads

From the strength loads deck, for every load condition considered, there is a set of internal loads at the element and node of interest. This will consist of three orthogonal forces, and three orthogonal moments. The criticality of a certain loadcase will be based on the magnitude and direction of the load. A smaller magnitude in a different direction could be more critical if the cross section is not symmetrical (i.e. I-Beam).

Step 2 : Cross Section Geometry

The cross section of the geometry has to be defined for the analysis. In the early design phase, it may be necessary to try different types of cross sectional shapes to determine what is most efficient, but an understanding of the nature of the internal loading typically leads to the selection of the appropriate shape. The cross sectional shape may not be based on what structural efficient but also based on what's required for the kinematics, envelope, and manufacturability.

For the given cross section, the section properties are calculated such as area, moments of inertia, etc... Figure 8 shows the cross sectional properties of the area of interest from the Catia model.

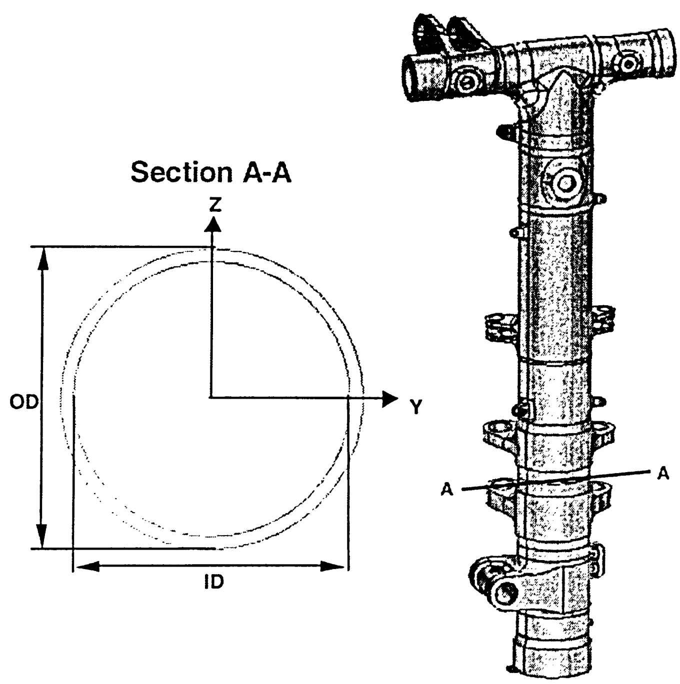


Figure 8 : Cross Section Geometry

Step 3 : Stress Calculation

With the loading and geometry defined, the calculation of the nominal stress is calculated based on solid mechanicals fundamentals for axial, bending, shear, and torsional stress. Where appropriate, hoop stresses are calculated. For most regions on the landing gear, the nominal stress alone is not sufficient to describe the stress level at the sections as geometric features lead to stress concentrations known as a Kt's.

A stress concentration exists when a geometric feature drives a directional shift in the load path. The more severe the geometric feature (i.e. smaller radius), the higher the stress concentration.

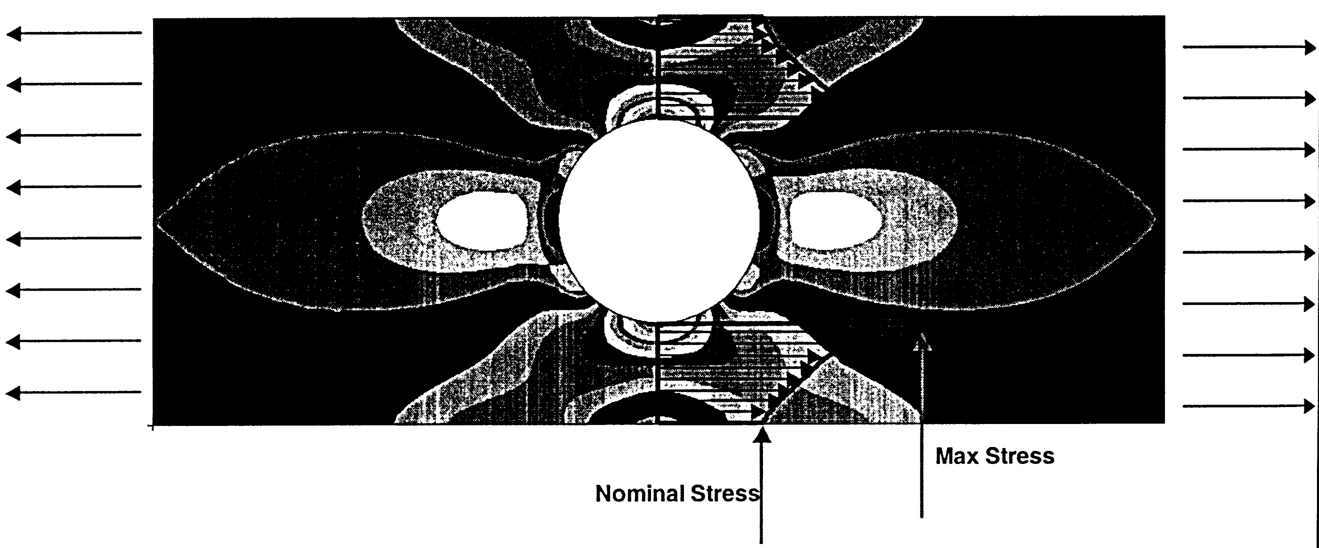


Figure 9 : Stress Concentration

Figure 9 shows a typical example of a stress concentration in a flat plate in tension with a hole. A typical stress concentration for this example is 3 meaning that the maximum stress in three times the calculated nominal stress.

From a practical approach, the nominal stress level is relatively simple to determine as described above. By multiplying this nominal stress by a factor greater than one based on the geometric feature, allows a more practical method for predicting the actual stress level. Empirical results for stress concentration factors exist in charts based on the geometry and type of loading. Figure 10 shows an example of such a reference for a tube with a cross hole under an axial load. Determining the Kt is a simple matter of describing the geometry of the tube and the cross hole. This Kt is used to multiply the nominal stress and determine the predicted stress level for that section.

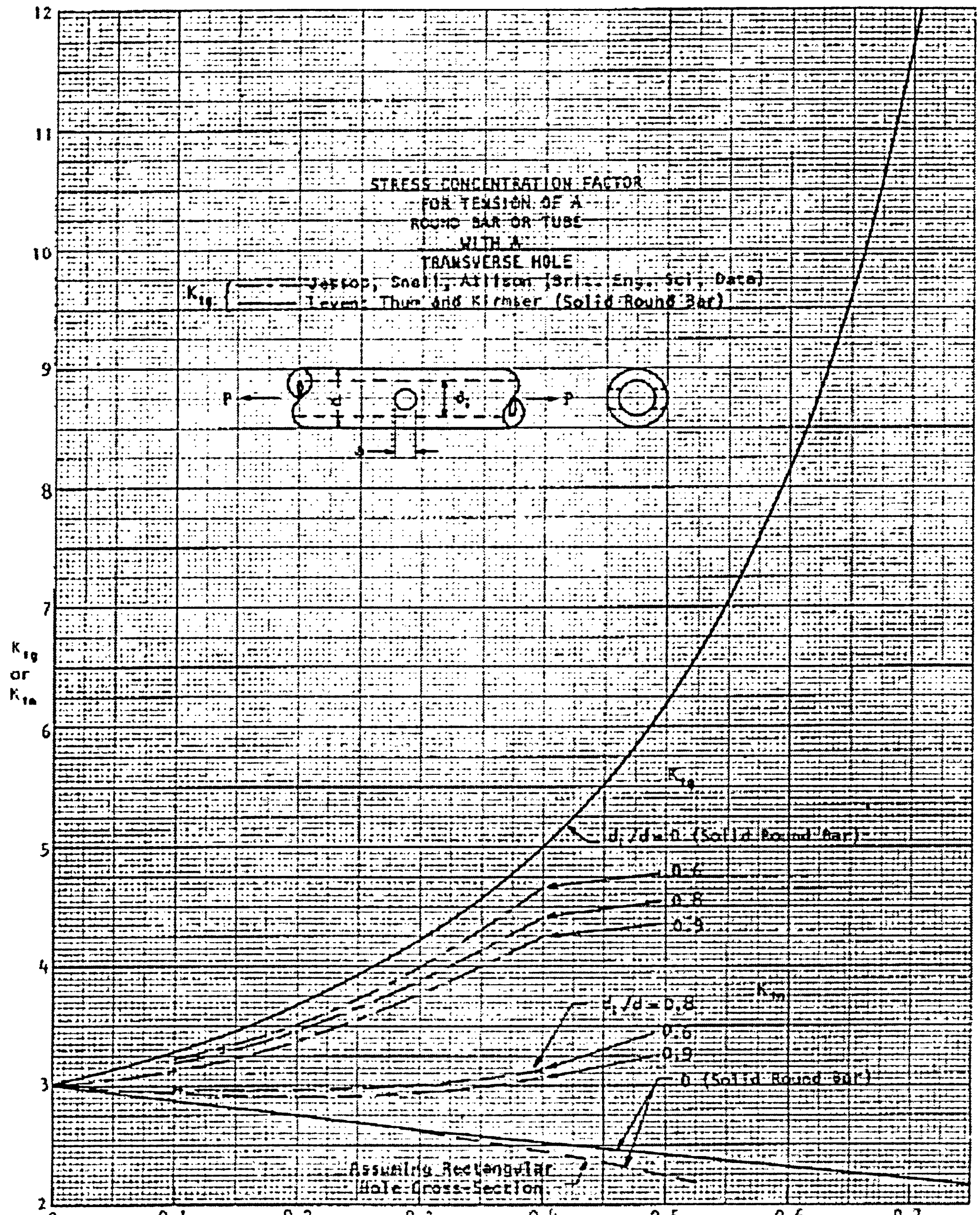


Figure 10 : Stress Concentration Chart [Ref. 8]

Step 4 : Material Properties

The material properties of the component being analyzed defined the criteria for acceptance of the predicted stress level. Material properties are calculated based on coupon testing of the material. Allowable for tension, compression, and shear, for both yield and rupture, are required in determining acceptance of the analysis. An allowable is a statistically reduced value where the fidelity of which will depend on the number of coupons tested and from the number of heat lots of material tested. Many strength allowables for metals used in the aerospace industry can be found in MIL-HDBK [REF. 3], which is a government document which has derived allowables for many steels, aluminums and titanium's. In certain instances, there are no published allowables for certain material allowables such as FCY or FSU. It is possible to derive an allowable based on only a few data points from test coupons by using the higher fidelity A-Basis allowables such as FTU and FTY an (REF. APPENDIX 7).

The stress-strain curve of a material is derived based on testing of coupons for a given material. Stress-strain curves can be derived for tensile coupons as well as compressive coupons. This provides information on the material such as elastic modulus, yield and rupture points as well as the fictitious stress (reference stress) which is useful in deriving bending allowable for a material. The stress-strain curve itself can be approximated as an exponential relationship up to the yield point in an equation called the Ramberg-Osgood equation. Appendix 6 describes the method of determining the stress-strain curve and deriving the Ramberg-Osgood equation from the stress-strain curve. Expressing the curve as an equation becomes convenient in deriving other allowables such as bending allowables described in Appendix 3.

The bending allowable is significant as it was found that under a bending stress, the material shows capability of exceeding yield and rupture stress for tensile coupons in a bending loading situation. The method for deriving this capability is described by the Cozzone method in Appendix 3. The significance to landing gear design is that this increased allowable allows components primarily sized by bending stresses to be smaller such as the shock strut piston and cylinder diameters. This has a significant impact on the final weight of the landing gear.

Step 5 : Margin of Safety

The basis for acceptance of an analysis is to show a positive margin of safety. The margin of safety is calculated based on the ratio of the allowable stress to the calculated stress. The allowable stress is based on material allowables for the material being analyzed.

$$M.S. = \frac{\sigma_{Allowable}}{\sigma_{Calculated}} - 1$$

A margin of safety equal to zero means that the calculated stress level is at the allowable stress level for the material.

For limit load calculations, the material is not allowed to yield. The allowables used for this condition are yield allowables with a positive margin of safety, meaning that no yielding should occur.

For ultimate load calculation, the limit load is multiplied by a factor of safety of 1.5 and the allowable for the material rupture is considered. A positive margin of safety means that the material will not rupture under 1.5 times the limit load condition.

A more detailed description of the calculation of Margin of Safety is described in Appendix 5.

5.2.2 Socket Analysis

Socket analysis is a very common analysis technique used in landing gear design as there are many sockets and clevis arrangements in the mechanism. Socket analysis is used to determine the internal loading within the pin of the socket of clevis as well as the load distribution within the socket. There are typically two variation of a socket analysis that are considered. In one case, the joint is a clevis arrangement and it is considered as discontinuous as the bushings do not extend throughout the entire socket. In another variation, the pin is cantilevered out from the socket and the busing are continuous. The load distribution throughout the socket is considered as a sinusoidal distribution. From a design standpoint, we are interested in determining the peak load distribution for design of the socket and bushing. Also, the loading of the pin is of interest in order to understand the peak bending moments in the pin.

Appendix 2 show the derivation and formula used in a typical socket analysis. Appendix 1 shows a sample output of a socket analysis showing the loading along the length of the pin, as well as the peak load per unit length values.

5.3 Finite Element Analysis

Finite Element Analysis is a vastly used tool in landing gear structural analysis. The major advantage over the other classical approaches mentioned previously is that it removes most idealization that is made in the analysis. From a practical standpoint, the classical methods are much quicker for basic sizing of components. When it comes to detailed analysis, particularly in regions of transitions and stress concentrations, the finite element method is more robust in reducing risk and predicting actual stress levels.

The primary use of finite element analysis in the design of landing gear is to provide a better means for predicting stress levels in geometries which are too complicated to idealize by other means. Also, in multiple component assemblies, the interaction of the load flow through the components can be simulated using contact boundary conditions which allows for understanding of secondary load paths which may not be evident in the static balance of the assembly.

In short, finite elements provide risk reduction for the design and allow the design to be more aggressive for weight by removing more material based on the prediction of the stress levels.

In Figure 11, an assembly of a partial piston with the axle installed is modeled. Contact boundary conditions are modeled between the piston and the axle as well as at the pin where the torque links connect.

PROPRIETARY INFORMATION REMOVED

Figure 11: Finite Element Assembly model

In the Figure 12, the deformed shape of the assembly model in the loaded state shows the nature of loading condition. The exaggerated deformation is useful to understand the effect of the load condition on the structure and can be used as a tool to ensuring the boundary conditions are modeled correctly, as well as to check that the deformations do not cause any interference with other components in the assembly.

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Figure 12: Finite Element Assembly Model Deformation

The primary result of interest from the finite element results is typically the stress or strain. Stress and strain results can be considered either only in the elastic range or can be considered for the effects of plasticity. Typically in a fatigue or limit strength load condition, the stress and strain are kept in the elastic region of the stress-strain curve, as to avoid any yielding in these load conditions. Figure 13 shows the stress fringe plot for the assembly finite element model. The overall fringe plot gives indications of load path, as well as stress concentration regions.

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Figure 13: Finite Element Assembly Model Stress Fringe

5.3.1 Finite Element Model Validation

The most important aspect of Finite Element Modeling is the validation of the results. Validation of finite element results are typically done by some or a combination of all of the following means :

1. Test Results
2. Reactions Forces & Beam Model Comparison
3. Deformed Shape
4. Free Body Diagram
5. Energy Balance

Test Results

Depending on the availability of test results, it may be possible to check the finite element results against proven test results. Strain gauge results can be used to compare stress results at different locations on the component. Photoelastic results are typically used to compare locations of stress concentrations. It can be very useful in comparing fringe patterns in order to provide confidence in the boundary conditions of the finite element results. Realistically, however, test results are not available for initial design. There are some situations such as a redesign or a non-conformance of a part, where a finite element model would be created with existing test results. Correlating to test results yields the highest fidelity of finite element model.

Reaction Forces & Beam Model Comparison

One of the fundamental checks to do when validating finite element results is check the reaction forces against what is expected. If analyzing a component, this may involve comparing the reactions of the component finite element model to internal loads from a full landing gear beam model.

Deformed Shape

The deformed shape (Reference Figure 9) provides a visual on what the loads and boundary conditions are doing to the component. Mistakes in loading or boundary conditions can be caught in this visual check.

Free Body Diagram

This is often a check that is forgotten with the ever increasing reliance on computer simulation. A free body diagram of a component is the best means of ensuring static equilibrium and understanding load path.

Energy Balance

In complex finite element models where contact boundary conditions are defined between multiple components, convergence issues may lead to the use of a stabilization tool in the software which adds damping to locally stabilize the contact constraint. It is important to track the energy used to stabilize these local instabilities and ensure that the magnitude of this energy loss is insignificant relative the to total strain energy of the model.

6 FATIGUE ANALYSIS

Fatigue analysis of landing gears is as important as the strength analysis. Many landing gear component failures are as a result of fatigue. More than half of the structural analysis time in new design involves the fatigue analysis. The fatigue theory used in the design process is that of strain-life.

6.1 Strain Life

The methodology employed in the fatigue analysis of landing gear structure is a strain-life based fatigue approach. The difference of this method compared to a stress-life approach is that the critical locations of components (notches) response are strain or deformation dependent. In low stress levels in high cycle fatigue where stress and strain are linearly related, the load control method (stress life) and the strain controlled level (strain life) are considered to be equivalent. At high stress levels beyond yield in the low cycle fatigue region, the strain controlled approach best models the material behavior.

In order to apply this methodology in a means which minimizes a company's risk of fatigue failure while still producing a low weight design, the company has undertaken a material characterization program for all of the typical materials used in design. The following is a list of the most common materials which are used in design.

- 300M Steel
- 4340 Steel
- 4330V Steel
- Titanium 5553
- Titanium 10-2-3
- 15-5 Stainless
- 7075-T73 Aluminum
- 7010-T73 Aluminum

For each of these materials, strain controlled coupon testing is performed typically from 4 or more heat lots with three samples per 5 strain levels tested for each heat lot. This provides the data required to produce a strain life reference curve for each material. This typically involves approximately 60 polished coupons for each material type. These coupons are run at an R=-1 stress ratio in order to produce an R=-1 strain life curve.

With the strain life coupon testing results for 4 heat lots at 5 different strain levels representing R=-1, the Coffin-Manson curve is used to fit the data. A sample Coffin-Manson strain-life curve is plotted in Figure 14.

$$\frac{\Delta\varepsilon}{2} = \underbrace{\frac{\sigma_f^{'}}{E}\left(2N_f\right)^b}_{\text{elastic}} + \underbrace{\varepsilon^{'}\left(2N_f\right)^c}_{\text{plastic}} \quad \text{[Ref. 4]}$$

where:

ε_f' - fatigue ductility coefficient

$2N_f$ – reversals to failure

σ_f' – fatigue strength coefficient

b – fatigue strength exponent

c – fatigue ductility exponent

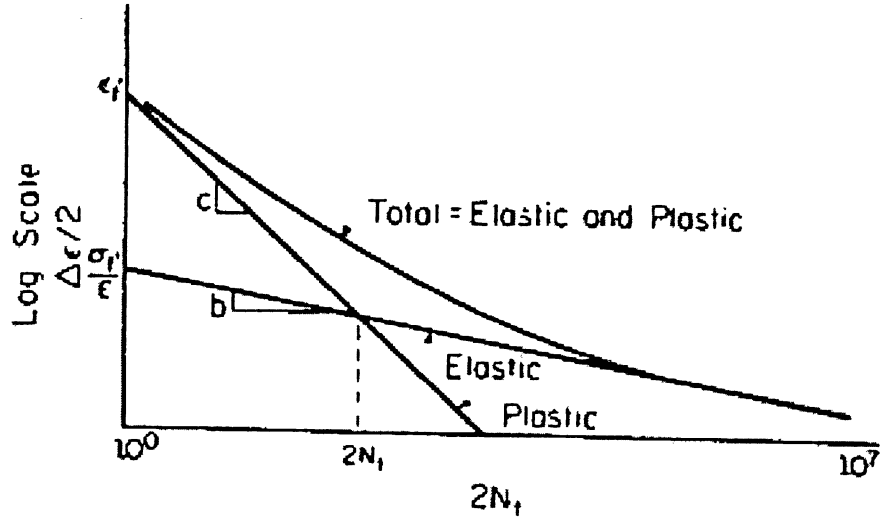

Figure 14 : Coffin-Manson Strain Life Curve [Ref. 4]

Since the reference curve is generated at an R=-1 stress ratio, for practical design purposes, a means of adjusting the curve to different mean stresses is required. Mean stress correction can be performed using a relationship described by Smith-Watson-Topper as follows:

$$\sigma_{\max}\frac{\Delta\varepsilon}{2}=\frac{\left(\sigma_f'\right)^2}{E}\left(2N_f\right)^{2b}+\sigma_f'\varepsilon_f'\left(2N_f\right)^{(b+c)} \quad \text{[Ref. 4]}$$

The actual landing gear components do not reflect the polished coupons upon which the fatigue data is derived, and therefore, surface conditions must be considered in order to see their detriment to the fatigue life.

In addition to surface roughness, surface treatments can be detriment or enhancing effect to the materials fatigue life. Surface treatments such as anodizing aluminum, CAD, Chrome and HVOF, have detrimental effects on the fatigue life of a component, whereas, shot peening can improve the components fatigue performance. The company has undertaken testing of all materials under all applicable surface treatment conditions in order to fully understand the effect on the strain life curve.

6.2 Landing Gear Fatigue Spectrum

The landing gear fatigue spectrum is a flight-by-flight listing of load conditions which the aircraft will typically see. This will include fewer cycles of high load conditions such at 0.6 G turns or 0.5 μ braking events or high descent velocity landings. Less severe, more typical load conditions will have many more cycles. The basis for determining the severity of these loads and the frequency of such events is based on flight test data, as well as experience. Fatigue loads are organized in a flight by flight basis, in that the order of the loading is representative of reality.

A realistic fatigue spectrum is extremely important as it can drive the design. The fatigue spectrum which you derive is aimed to represent real every day operational loading which will predict the every day loading on the landing gear. Typically load conditions used in fatigue spectrums are derived from flight test data in order to understand the real loads acting on the aircraft in service. It is important to have the spectrum representative, in that it is not overly conservative, adding weight, and not under conservative, compromising safety.

6.3 Finite Element Fatigue Analysis

In order to apply the theory of strain-life to a practical design process, there are a number of tools available. One of the more recent additions at the disposal of the analyst is FE-Fatigue software. Since most detailed analysis involves finite elements, a tool has been created to use these finite element strain results for fatigue analysis. Figure 15 outlines the procedure.

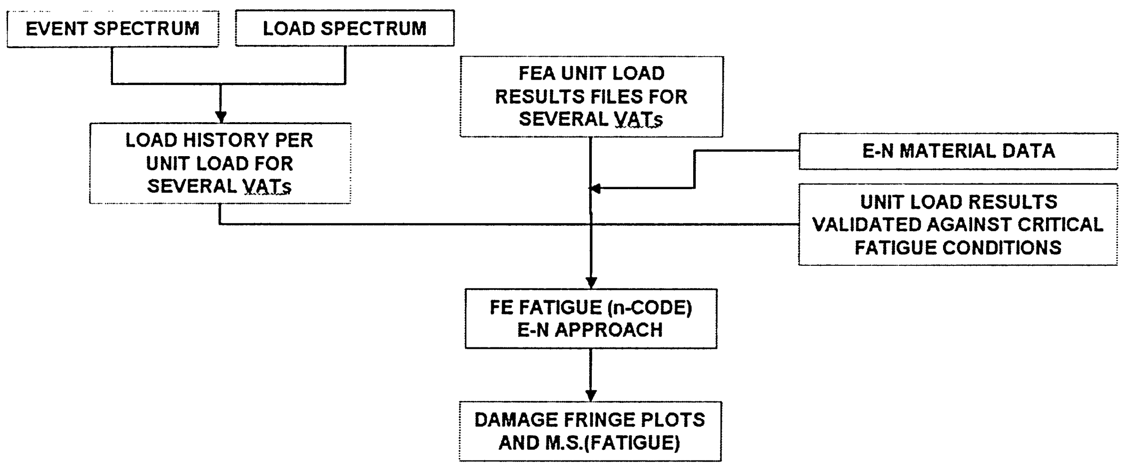

Figure 15 : FE-Fatigue Analysis Flow Chart

Essentially, the fatigue analysis using finite elements involves three types of inputs, the spectrum inputs, the finite element results inputs, and the material inputs.

The spectrum inputs contain the detailed fight by flight events list in the events spectrum, and the list of fatigue loads in the load spectrum. The material data is a database of strain life curves for different material and applicable surface treatments. The finite elements results represent unit load results.

6.3.1 Unit Load Superposition

Unit load superposition is a means of generating results for many different load conditions by superimposition unit load conditions. This assumes a linear relationship between results and unit loads. For the application of landing gear fatigue analysis, this method works well as the loading on the landing gear can be simplified to loads at a the axle in components of vertical, drag at the axle, drag at the ground, and side. By analyzing the results at the unit level, the results of the finite elements can be superimposed for the many load conditions that make up the fatigue spectrum.

For landing gear analysis, the stroke of the shock strut or Vertical Axle Travel (VAT), will vary for different load conditions. Since VATs affect the overall geometry of the system, the load transfer through the system is different for different VATs. For this reason, the unit loads have to be run a different VAT models which encapsulates the loads spectrum.

When using a unit load superposition approach to analysis, it is important to remember that the deformation within the structure is non-linear, and that the linear assumption is an approximation. For this reason, it is a good practice to run critical fatigue cases and compare the results to the superimposed results for the same case. If significant discrepancies exist, the difference should be included as a separate analysis parameter in order to avoid inaccuracies. Previous methods used have been to apply factors to the unit loads results, such that the superimposed results equal the actual results for fatigue critical load conditions.

7 INDETERMINANCE ISSUES

The design challenges associated with dual brace indeterminate landing gear design involve the structural analysis, as well as the freefall analysis.

7.1 Indeterminate Structures

Analysis of an indeterminate structure is more challenging than that of a determinate structure for the reason that the stiffness of the structure is of primary importance in achieving static equilibrium. The dual brace arrangement allows the load to be shared across the structure on the aft spar and the gear beam, which can be an advantage depending on the design of the wing.

From a structural design point of view, understanding the load path through the indeterminate structure is crucial in order to have an optimized design. Modeling the stiffness of both the landing gear, as well as the aircraft structure is needed to properly understand how the load is distributed. With four attachment points in the aircraft, the flexibility of the aircraft and the landing gear will determine how the load is distributed across the four points.

Figure 16 shows a finite element model of an indeterminate landing gear design. The aircraft flexibility is a flexibility matrix input into the model, based on information from the airframer.

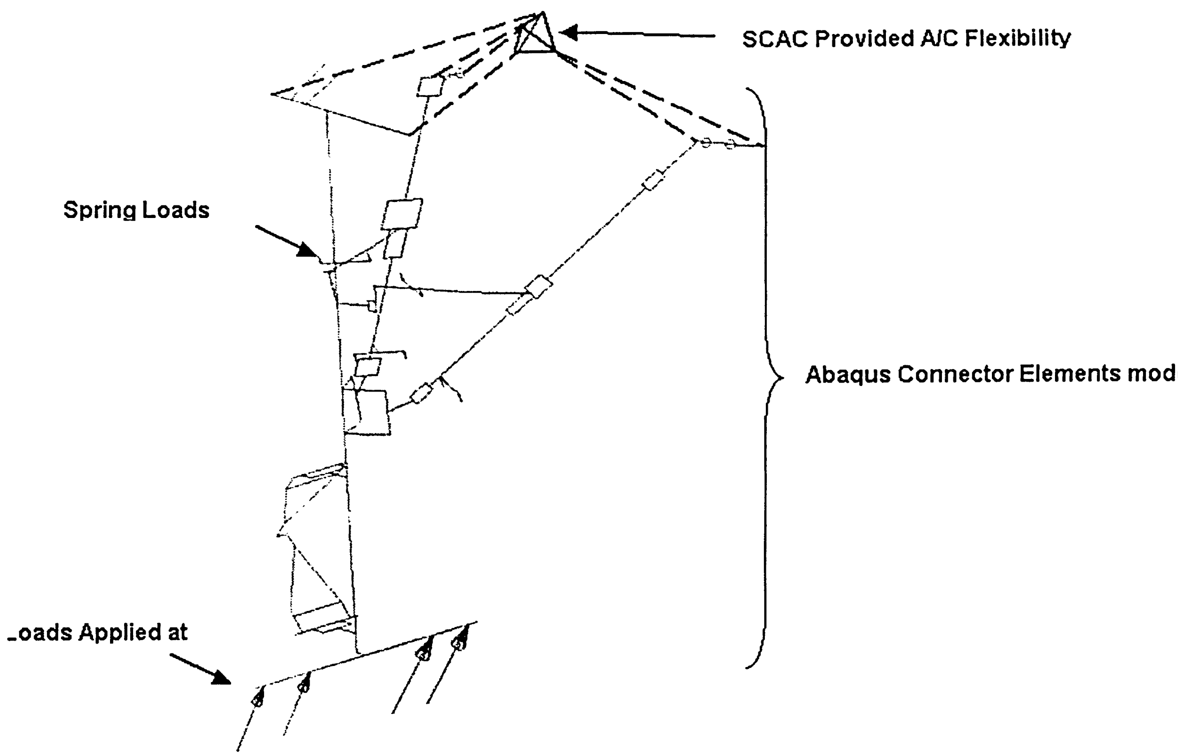

Figure 16 : Indeterminent Landing Gear Design

The importance of the correct stiffness to the load distribution requires more iteration than determinant designs in order to keep the loads up to date with the design. On determinant gears, the stiffness only has a secondary effect of the loads in that the loading in the deflected shape and change the loads from the undeflected shape by up to 10%.

Figure 17 shows a 3D finite element model of the dual brace design. In this model, the aircraft attachment points are tied in to the flexibility matrix representing the airframe. All joints are modeled with contact boundary conditions, and load is applied at the upper and lower bearings of the piston.

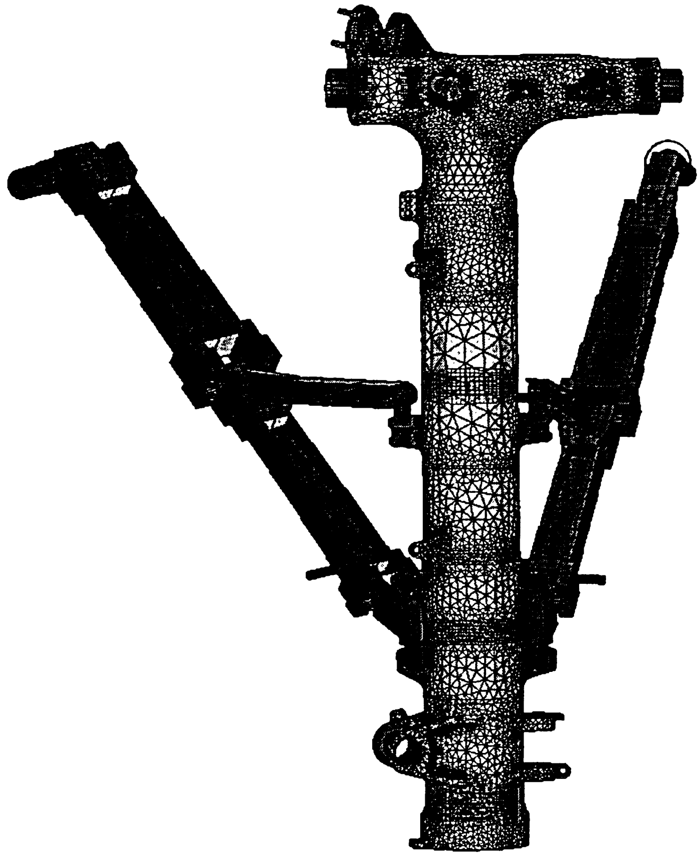

Figure 17 : Indeterminent Landing Gear Finite Element Model

The finite element modeling of this indeterminate structure is of vital importance in understanding the load paths which is required to produce a design which is optimized for weight. The stiffness information from the airframer and for the landing gear itself is essential in order to produce a model with the accuracy required.

7.2 Freefall Analysis

The freefall condition is a requirement for all landing gear designs where the gear must be able to go down and locked without hydraulic aid. The weight of the landing gear, and the springs on the braces help to bring the landing gear down and locked once the uplock hook has released (Reference Appendix 8). The dual brace indeterminate structure design introduces a problem into the ability of the gears to come down and locked in a freefall condition.

Airload on the landing gear as it is in freefall will introduce tension into the forward brace and compression into the aft brace. Though the tension in the forward brace helps put it into lock, the compression in the aft brace has the opposite effect and keeps the brace from going into lock. To overcome this problem, without introducing very large downlock springs, a method of brace shortening has been introduced. This method involves producing the aft brace to be shorter than the geometry of its installation. This introduces a pre-tension into the brace in the unloaded state. If this pre-tension is set correctly, it will act against the compressive load driven by airloading and allow the gear to come down and locked.

Introducing this shortening condition into the design is challenging. The first step in determining this shortening requirement is to simulate the airload conditions which generate high compressive loads in the aft brace. Once the design condition is found for the highest airload condition, this compressive load is used to determine the required shortening to generate an equivalent but opposite pre-tension in the brace. Simulation aids in determining the range of required shortening, but in the end, a series of tests on the actual landing gear with a range of different length braces will determine the appropriate brace shortening.

8 SUMMARY

Landing gear structural design involves ensuring structural integrity for strength and fatigue. The aerospace industry designs for light weight in order to carry more payload, which leads to designs with exotic materials, and extensive analysis and simulation in order to push the design to maximum efficiency. Most landing gears are determinant structures where load paths are not extremely difficult to define. In component design, the load is the basis for all design and analysis of the component, and the pedigree and fidelity of that load must be low risk in order to ensure that the design will be appropriate for the conditions that will be seen in service. The introduction of indeterminance into the landing gear structure adds a risk in the load determination for components. To mitigate this risk, there are two options. One option is to add risk factors to the analysis to account for uncertainties, which is not appealing to the profitability of a project from a weight point of view. The other option is to increase the amount of analysis and simulation in order to mitigate the risk.

The amount of simulation in landing gear structural analysis has increased exponentially, even over the past ten years, with most of the beneficial contributions coming from increase computational capabilities in software and hardware. The indeterminate problem, however good the simulation, still involves risk, and the introduction of risk factors is still necessary, but kept to a minimum.

The two main problems introduced through an indeterminate, dual brace design landing gear are redundant load paths, and freefall issues. Determining load path on all landing gear designs is accomplished through landing and ground load simulations providing loads to be analyzed in a finite element beam model. With indeterminate landing gears, the stiffness of the landing gear and the aircraft structure become that much more critical in correctly predicting load paths, and as a result, more iteration is required throughout the design life, as well as better transfer of information between the airframer and the landing gear supplier. Additionally, the use of 3-D finite element modeling with full contact boundary conditions provides a means of modeling detailed components accurately. The freefall issue involves the introduction of a compressive load under airload conditions which impedes the locking of the brace in a freefall under certain airload conditions. The resolution of this problem involves either introducing very large springs which can force the brace into lock, or by introducing a pre-tension into the brace by making the length geometrically smaller than the installation length, thereby introducing a pre-tension. The pre-tension in the brace alleviates the compressive load during the airload condition, thereby, not inhibiting the locking of the brace.

Landing gear analysis involves simulation for both the loads and the structural analysis. Simulation is made more realistic by modeling 3-d components with representative stiffnesses important in predicting load paths. Increased computational capabilities allows for more complex meshing in finite elements, as well as more complicated contact boundary condition and multi-part assembly models. All these efforts are aimed at better prediction and, thereby, a better product on the first time through.

The engineering of landing gear is an ongoing tradeoff of weights, costs, and robustness, which can go on and on, but at some point, to get the product out the door, the compromises are made and a design is established. Within the industry itself, there are many debates over which design practices are more efficient, and some major airframers stick to their beliefs of which approaches are better than others. For example, the dual brace landing gear design is used extensively on the Boeing product but never before on the Airbus product. Recently, Airbus has unveiled its plans for the A350 as a reaction to the Boeing 787. It is interesting to note that this design is to be the dual brace arrangement.

APPENDIX 1 – Sample Analysis Outputs [Ref. 2]

Section Analysis

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PROPRIETARY INFORMATION REMOVED

Socket Analysis

PROPRIETARY INFORMATION REMOVED

APPENDIX 2 – Socket Analysis Theory [Ref. 7]

Combined Cosine–Uniform Socket Distribution (Discontinuous)

This section presents the combined cosine-uniform discontinuous socket load distribution. Under the combined cosine-uniform distribution, a uniform-distributed load and a cosine-distributed load are applied across the length of the socket. The socket loading inside each bushing length is a combination of both distributed loadings.

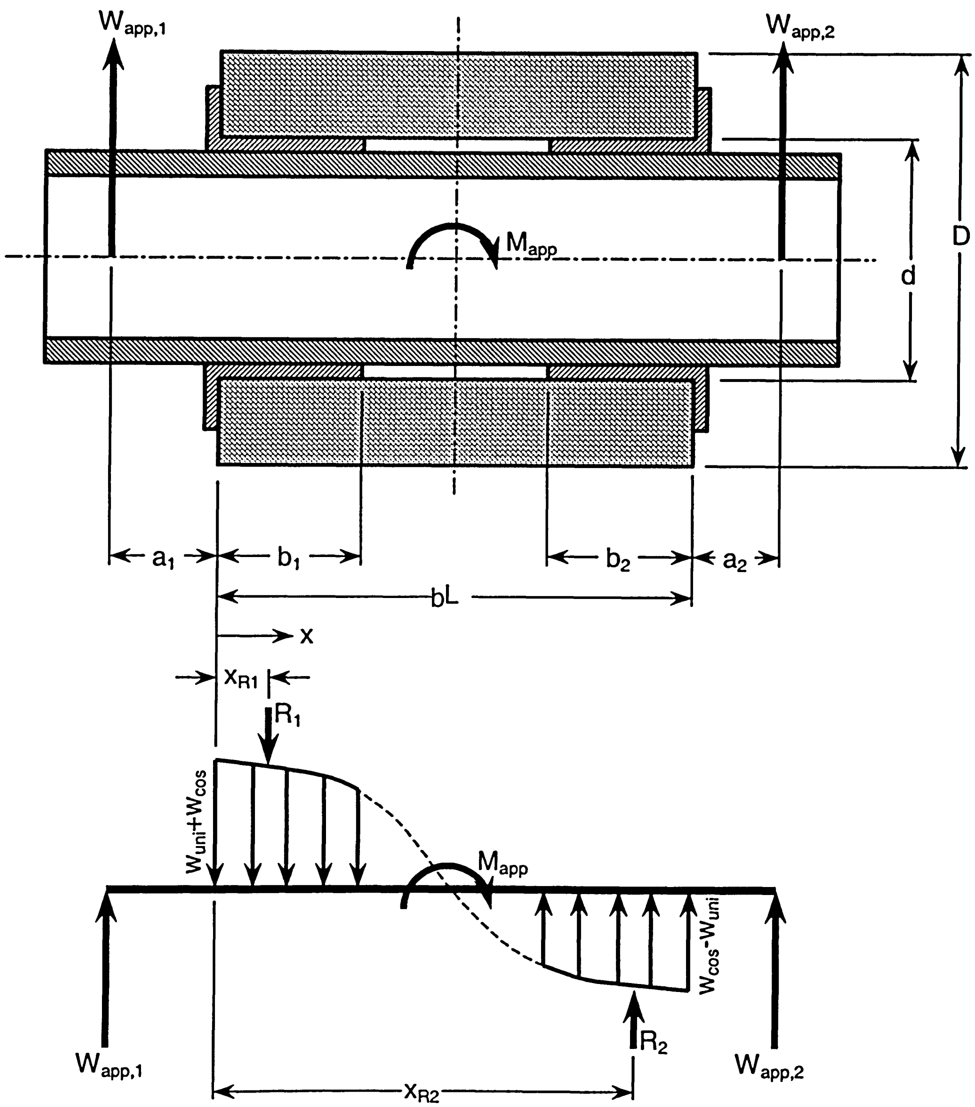


Figure 18: Socket Geometry and Load Distribution for Cosine-Uniform Distribution

Note that the arguments of all sine and cosine functions are in radians.

Reaction Positions (Approximate)

$$x_{R1} = \frac{b_1}{2} \cdot \left[1 - \frac{\left(\frac{\pi \cdot b_1}{L}\right)^2}{12 \cdot \left(1 + \frac{w_{uni}}{w_{cos}}\right) - 2 \cdot \left(\frac{\pi \cdot b_1}{L}\right)^2} \right]$$

$$x_{R2} = L - \frac{b_2}{2} \cdot \left[1 - \frac{\left(\frac{\pi \cdot b_2}{L}\right)^2}{12 \cdot \left(1 - \frac{w_{uni}}{w_{cos}}\right) - 2 \cdot \left(\frac{\pi \cdot b_2}{L}\right)^2} \right]$$

Reaction Forces

$$R_1 = \frac{w_{cos} \cdot L}{\pi} \cdot \sin\left(\frac{\pi \cdot b_1}{L}\right) + w_{uni} \cdot b_{1_1}$$

$$R_2 = \frac{w_{cos} \cdot L}{\pi} \cdot \sin\left(\frac{\pi \cdot (L - b_2)}{L}\right) - w_{uni} \cdot b_{2_{22}}$$

Use the constant H to simplify the subsequent equations, defined as:

$$H = \frac{b_1^{\,2} + 2 \cdot L \cdot b_2 - b_2^{\,2}}{2 \cdot (b_1 + b_2)}$$

Cosine load distribution magnitude:

$$w_{cos} = \frac{\pi^2}{L^2} \cdot \left[\frac{M_{app} + W_{app,1} \cdot (a_1 + H) - W_{app,2} \cdot (L + a_2 - H)}{2 - \cos\left(\frac{\pi \cdot b_1}{L}\right) - \cos\left(\frac{\pi \cdot b_2}{L}\right) - \frac{\pi}{L} \cdot (b_1 - H) \cdot \sin\left(\frac{\pi \cdot b_1}{L}\right) + \frac{\pi}{L} \cdot (L - b_2 - H) \cdot \sin\left(\frac{\pi \cdot b_2}{L}\right)} \right]$$

Uniform load distribution magnitude:

$$w_{uni} = \frac{W_{app,1} + W_{app,2} - \frac{w_{cos} \cdot L}{\pi} \cdot \left[\sin\left(\frac{\pi \cdot b_1}{L}\right) - \sin\left(\frac{\pi \cdot b_2}{L}\right) \right]}{b_1 + b_2}$$

The distributed load on the pin at any point inside the length of the socket is described by:

$$w(x)=\begin{cases} w_{cos}\cdot\cos\left(\dfrac{\pi\cdot x}{L}\right)+w_{uni} & \text{if} \quad 0\le x\le b_1 \\ 0 \quad \text{if} \quad b_1<x<L-b_2 & \\ w_{cos}\cdot\cos\left(\dfrac{\pi\cdot x}{L}\right)+w_{uni} & \text{if} \quad L-b_2\le x\le L \end{cases}$$

Shear Load on the Pin Inside the Socket

$$v(x)=\int_0^x w(x)\cdot dx - W_{app,1}$$

Bending Moment on the Pin Inside the Socket

$$m(x)=\int_0^x w(t)\cdot(x-t)\cdot dx - W_{app,1}\cdot(a_1+x)-M_{app}$$

Modified Combined Cosine - Uniform Socket Distribution (CONTINUOUS)

The socket geometry and load distribution illustrated below is a subset of the above case. In this situation there is one applied load and one applied moment with contact throughout the length of the pin within the socket. However since the load distribution is a cosine contour it is governed by the previous equations.

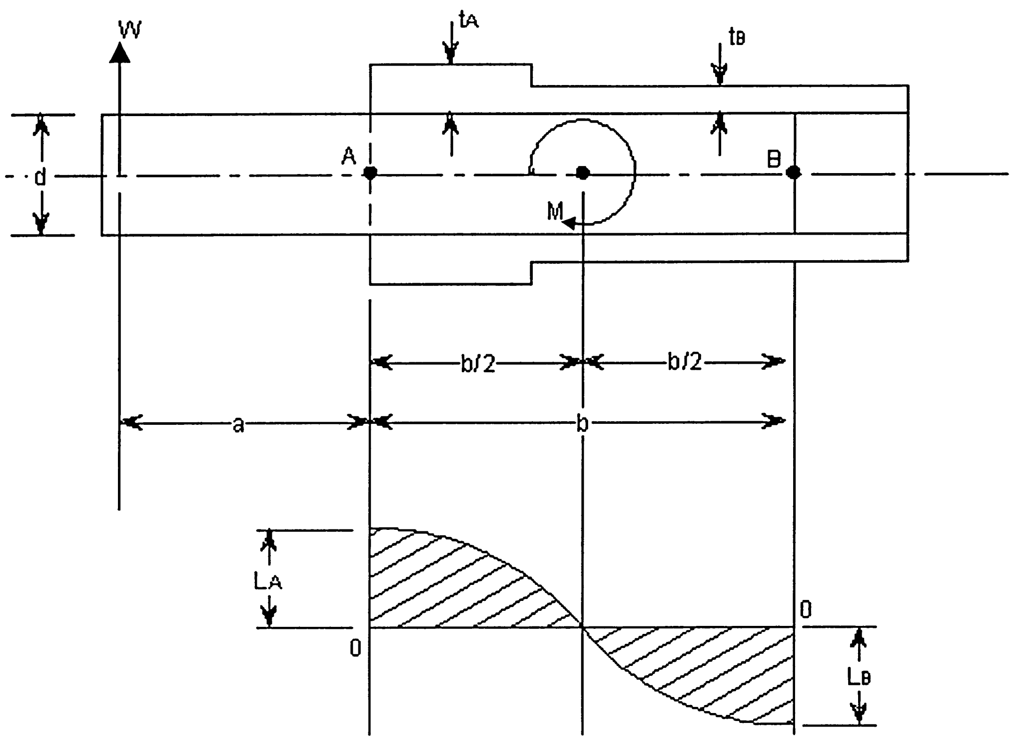


Figure 19 : Socket Geometry and Load Distribution for Modified Cosine-Uniform Distribution

The equations introduced in section A can describe this geometry by changing the following variables:

$$M_{app} = M$$

$$W_{app,1} = W$$

$$W_{app,2} = 0$$

$$b_1 = b_2 = \frac{b}{2}$$

$$a_1 = 0$$

$$a_2 = 0$$

$$L = b$$

The approximate locations of the reaction forces are:

$$x_{R1} = \frac{b_1}{2} \cdot \left[1 - \frac{\left(\frac{\pi \cdot b_1}{L}\right)^2}{12 \cdot \left(1 + \frac{w_{uni}}{w_{cos}}\right) - 2 \cdot \left(\frac{\pi \cdot b_1}{L}\right)^2} \right]$$

$$x_{R1} = \frac{b}{4} \cdot \left[1 - \frac{\left(\frac{\pi}{b} \frac{b}{2}\right)^2}{12 \cdot \left(1 + \frac{w_{uni}}{w_{cos}}\right) - 2 \cdot \left(\frac{\pi}{b} \frac{b}{2}\right)^2} \right]$$

$$x_{R1} = \frac{b}{4} \cdot \left[1 - \frac{\left(\frac{\pi}{2}\right)^2}{12 \cdot \left(1 + \frac{w_{uni}}{w_{cos}}\right) - 2 \cdot \left(\frac{\pi}{2}\right)^2} \right]$$

$$x_{R2} = L - \frac{b_2}{2} \cdot \left[1 - \frac{\left(\frac{\pi \cdot b_2}{L}\right)^2}{12 \cdot \left(1 - \frac{w_{uni}}{w_{cos}}\right) - 2 \cdot \left(\frac{\pi \cdot b_2}{L}\right)^2} \right]$$

$$x_{R2} = b - \frac{b}{4}\cdot\left[1 - \frac{\left(\frac{\pi}{b}\frac{b}{2}\right)^2}{12\cdot\left(1 - \frac{w_{uni}}{w_{cos}}\right) - 2\cdot\left(\frac{\pi}{b}\frac{b}{2}\right)^2}\right]$$

$$x_{R2} = b - \frac{b}{4}\cdot\left[1 - \frac{\left(\frac{\pi}{2}\right)^2}{12\cdot\left(1 - \frac{w_{uni}}{w_{cos}}\right) - 2\cdot\left(\frac{\pi}{2}\right)^2}\right]$$

The reaction forces are:

$$R_1 = \frac{w_{cos}\cdot L}{\pi}\cdot\sin\left(\frac{\pi}{L}b_1\right) + w_{uni}\cdot b_1$$

$$R_1 = \frac{w_{cos}\cdot b}{\pi}\cdot\sin\left(\frac{\pi}{b}\frac{b}{2}\right) + w_{uni}\cdot\frac{b}{2}$$

$$R_1 = \frac{w_{cos}\cdot b}{\pi} + w_{uni}\cdot\frac{b}{2}$$

$$R_2 = \frac{w_{cos}\cdot L}{\pi}\cdot\sin\left(\frac{\pi\cdot(L - b_2)}{L}\right) - w_{uni}\cdot b_2$$

$$R_2 = \frac{w_{cos}\cdot b}{\pi}\cdot\sin\left(\frac{\pi}{b}\left(b - \frac{b}{2}\right)\right) - w_{uni}\cdot\frac{b}{2}$$

$$R_2 = \frac{w_{cos}\cdot b}{\pi}\cdot\sin\left(\frac{\pi}{b}\frac{b}{2}\right) - w_{uni}\cdot\frac{b}{2}$$

$$R_2 = \frac{w_{cos}\cdot b}{\pi} - w_{uni}\cdot\frac{b}{2}$$

To assist in finding the equation for the distributed load, H is defined as:

$$H = \frac{b_1{}^2 + 2\cdot L\cdot b_2 - b_2{}^2}{2\cdot(b_1 + b_2)}$$

$$H = \frac{\left(\frac{b}{2}\right)^2 + 2\cdot b\cdot\left(\frac{b}{2}\right) - \left(\frac{b}{2}\right)^2}{2\cdot\left(\frac{b}{2} + \frac{b}{2}\right)}$$

$$H = \frac{b}{2}$$

The magnitude of the cosine distribution is:

$$w_{cos} = \frac{\pi^2}{b^2}\cdot\left[\frac{M_{app}+W_{app,1}\cdot(a_1+H)-W_{app,2}\cdot(L+a_2-H)}{2-\cos\left(\frac{\pi\cdot b_1}{L}\right)-\cos\left(\frac{\pi\cdot b_2}{L}\right)-\frac{\pi}{L}\cdot(b_1-H)\cdot\sin\left(\frac{\pi\cdot b_1}{L}\right)+\frac{\pi}{L}\cdot(L-b_2-H)\cdot\sin\left(\frac{\pi\cdot b_2}{L}\right)}\right]$$

$$w_{cos} = \frac{\pi^2}{b^2}\cdot\left[\frac{M+W\cdot(a+b/2)-0}{2-\cos\left(\frac{\pi}{b}\frac{b}{2}\right)-\cos\left(\frac{\pi}{b}\frac{b}{2}\right)-\frac{\pi}{b}\cdot\left(\frac{b}{2}-\frac{b}{2}\right)\cdot\sin\left(\frac{\pi}{b}\frac{b}{2}\right)+\frac{\pi}{b}\cdot\left(b-\frac{b}{2}-\frac{b}{2}\right)\cdot\sin\left(\frac{\pi}{b}\frac{b}{2}\right)}\right]$$

$$w_{cos} = \frac{\pi^2}{b^2}\cdot\left[\frac{M+W\cdot(a+b/2)}{2}\right]$$

$$w_{cos} = \frac{1}{\left(\frac{12}{\pi^2}\right)b^2}\cdot[6M+6W\cdot a+3W\cdot b]$$

The magnitude of the uniform distribution is:

$$w_{uni} = \frac{W_{app,1}+W_{app,2}-\frac{w_{cos}\cdot L}{\pi}\cdot\left[\sin\left(\frac{\pi\cdot b_1}{L}\right)-\sin\left(\frac{\pi\cdot b_2}{L}\right)\right]}{b_1+b_2}$$

$$w_{uni} = \frac{W+0-\frac{w_{cos}\cdot b}{\pi}\cdot\left[\sin\left(\frac{\pi}{b}\frac{b}{2}\right)-\sin\left(\frac{\pi}{b}\frac{b}{2}\right)\right]}{\frac{b}{2}+\frac{b}{2}}$$

$$w_{uni} = \frac{W}{b}$$

The load distribution along the length of the socket is:

$$w(x) = w_{cos}\cdot\cos\left(\frac{\pi\cdot x}{b}\right)+w_{uni} \qquad \text{for} \qquad 0\le x\le b$$

$$w(x) = \frac{1}{\left(\frac{12}{\pi^2}\right)b^2}\cdot[6M+6W\cdot a+3W\cdot b]\cdot\cos\left(\frac{\pi\cdot x}{b}\right)+\frac{W}{b}$$

At x = 0, the load distribution is:

$$w(0) = L_A = \frac{1}{\left(\frac{12}{\pi^2}\right)b^2}\cdot\left[6M+6W\cdot\left(a+\frac{b}{2}\right)\right]\cdot(1)+\frac{W}{b}$$

$$M=W\left(a+\frac{b}{2}\right)$$

$$L_A = \frac{1}{\left(\frac{12}{\pi^2}\right)b^2}\cdot[6M+6M]+\frac{W}{b}$$

$$L_A = \frac{\pi^2}{b^2}\cdot(M)+\frac{W}{b}$$

At x = b, the load distribution is:

$$w(b) = L_B = \frac{1}{\left(\frac{12}{\pi^2}\right)b^2}\cdot\left[6M+6W\cdot\left(a+\frac{b}{2}\right)\right]\cdot(-1)+\frac{W}{b}$$

$$L_B = -\frac{1}{\left(\frac{12}{\pi^2}\right)b^2}\cdot[6M+6M]+\frac{W}{b}$$

$$L_B = -\frac{\pi^2}{b^2}\cdot(M)+\frac{W}{b}$$

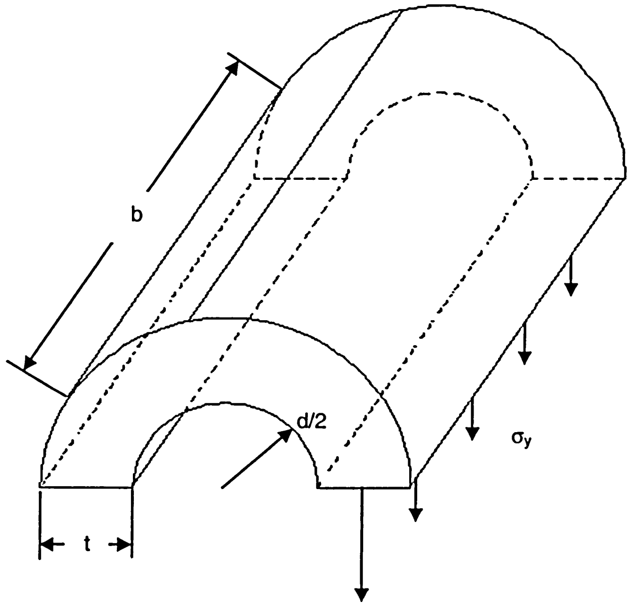
The shear load on the pin inside of the socket is:

$$v(x) = \int_0^x w(x)\cdot dx - W_{app,1}$$

The bending moment acting on the pin inside of the socket is:

$$m(x) = \int_0^x w(t)\cdot(x-t)\cdot dx - W_{app,1}\cdot(a_1+x)-M_{app}$$

Bursting Stress in Socket



The internal pressure force acting outwards must be in equilibrium with the cirumferential stress (σ_y) force.

$$\sigma_y = \frac{F_y}{2t \cdot b}$$

$$P_{int} = \frac{F_{int}}{d \cdot b}$$

$$L_i = \frac{F_{int}}{b}$$

thus

$$P_{int} = \frac{L_i}{d}$$

Equating F_y with F_{int} gives:

$$F_y = F_{int}$$

$$\sigma_y (2t \cdot b) = P_{int} (d \cdot b)$$

$$\sigma_y (2t) = \frac{L_i}{d} (d)$$

$$\sigma_y = \frac{L_i}{2t}$$

APPENDIX 3 – Calculation of Plastic Bending

The basis for plastic bending is that the modulus of rupture is usually greater than the actual material strength as shown in Figure 20 :

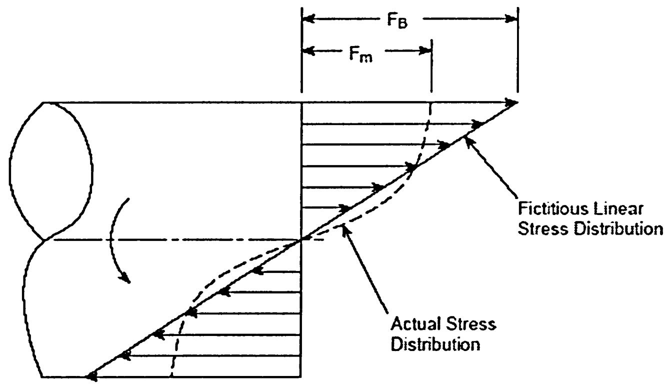


Figure 20: Simple Plastic Bending

Since it is desirable to use the classical beam formula in calculating bending stress, a fictitious failing stress, F_b, is calculated based on a method developed by F.P. Cozzone. This ficticious stress is referred to as the modulus of rupture.

The accuracy of the Cozzone method is as follows (Ref. 5, para. C3.6):

It is exact for rectangular sections under pure bending with moment vector parallel to a principal axis.

For double symmetric sections under pure bending and moment vector parallel to a principal axis, the accuracy would be within 5%.

Single symmetric sections will vary from practically exact to definitely unconservative (moment vector normal to axis of symmetry)

For sections subject to combined bending and axial load, the results will vary from practically exact to conservative.

For unsymmetrical bending, with and without axial load, the results will vary from practically exact to conservative.

The Cozzone method is based on the following assumptions and simplifications:

Failure is assumed to be predictable by an extreme fiber stress. Proper selection of an extreme fiber failure stress depends on the mode of failure of the cross-section. The mode of failure may be due to material rupture or local instabilities, i.e. buckling or crippling.

The cross-section is assumed to be fully effective.

The cross-section is assumed to not distort under bending. The Cozzone method should not be used for cross-sections that significantly change shape under bending loads, as they will give unconservative results. The Modulus of rupture for round tubes takes into account this distortion when publishing allowable bending stress data.

Strain distribution on the cross-section is assumed to be linear from the neutral axial to the extreme fibers.

Material tension and compression stress-strain properties are assumed to be identical. This assumption simplifies the analysis.

The actual non-linear stress distribution is replaced by a trapezoidal stress distribution, as shown in Figure 21. The trapezoidal intercept stress 'f_o' is to be determined so that the moment produced by the assumed trapezoidal stress distribution, about the neutral axis, is the same as the moment produced by the actual stress distribution for a rectangular cross-section. The actual stress distribution is assumed to be the stress-strain curve up to 'f_m' at the location of extreme fiber. The limiting value of 'f_m' is typically F_{ty} for limit analysis and F_{tu} for ultimate analysis.

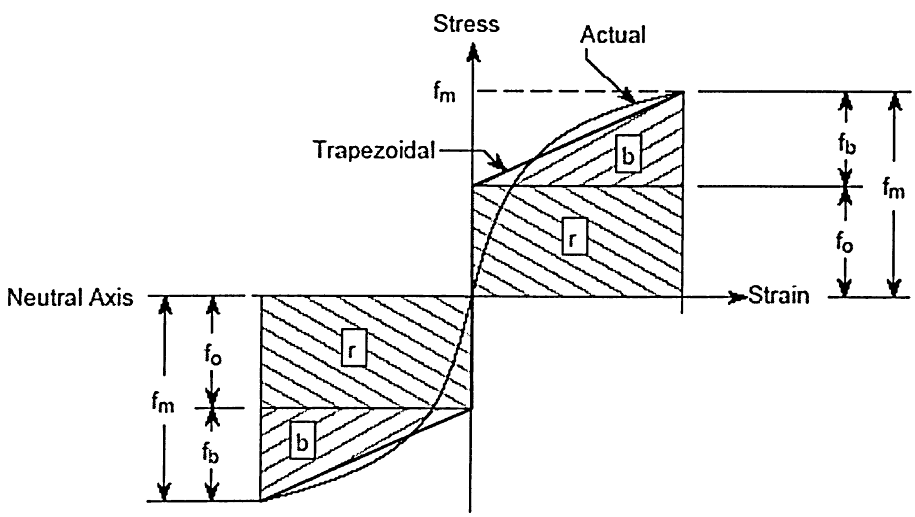


Figure 21: Cozzone Trapezoidal Stress Distribution

Assuming the rectangular and triangular portion of Figure 21 has resisting moments of m_r and m_b respectively, the total resisting moment may be expressed as follows :

$m_o = m_r + m_b$ Where m_o = total internal resisting moment

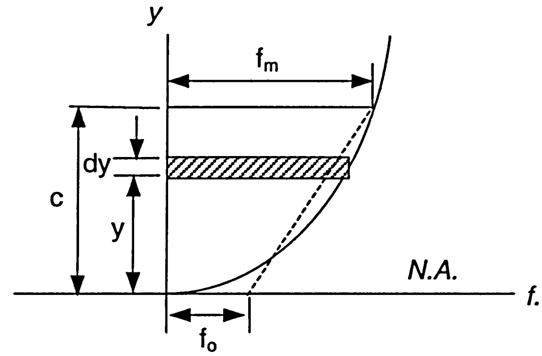
m_r = internal moment developed by portion (r)

m_b = internal moment developed by portion (b)

Since fb varies linearly from zero to f_b, the stress distribution is given in beam theory or $m_b = f_b I / c$ for the entire beach section. The stress variation on the portion m_r is described as rectangular, thus:

$$m_r = 2\int_0^c fydA$$

Given that region r from figure 4.4 is idealized as a rectangle with a constant force f_o:



$$m_r = 2f_o * \int_0^c ydA$$

Let $Q = \int_0^c ydA$

Then $m_r = 2f_o * Q$

For the outer fiber in Figure A2-2 , $f_b = f_m - f_o$

Therefore:

$$m_o = (f_m - f_o) * \left(\frac{I}{c}\right) + 2f_o * Q$$
$$= f_m + f_o * \left(\frac{2Qc}{I} - 1\right) \qquad (1)$$

Let $K = \dfrac{2Qc}{I}$ section shape factor (2)

and $F_b = \dfrac{m_o * c}{I}$. Then, from Equation 1:

$$F_b = f_m + f_o * (K - 1) \qquad (3)$$

The trapezoidal intercept stress, f_o, is a property of the material stress-strain curve. Since the calculation of this parameter requires integration of the stress-strain curve, it is convenient to have an equation to describe the curve. Walter Ramberg and William R. Osgood published a paper which decribes the stress-strain curve by three parameters. The Ramberg-Osgood result is an exponential equation relating stress and strain with an exponent specific to each curve.

$$e = \frac{f}{E} + 0.002 * \left(\frac{f}{F_{ty}}\right)^n \qquad (4)$$

Where : e = strain (in/in) , E = Modulus of Elasticity, f = stress, F_{ty} = Yield stress of material

n = Ramber-Osgood exponent

The Cozzone distribution from Figure 21 produces a moment about the neutral axis which is identical to the actual stress distribution. Therefore, the following must be true:

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APPENDIX 4 – Basic Stress Formulas

[Ref. 9]

Bending	$f_b = \frac{M \cdot c}{I} = \frac{M}{Z}$
Bearing pressure	$f_{br} = \frac{W}{0.85 \cdot D \cdot L}$ 0.85=lube groove factor
Direct tension	$f_t = \frac{W}{A}$
Direct compression	$f_c = \frac{W}{A}$
External pressure (not for pins, see below)	$f_{hc} = \frac{2 \cdot P_{hc} \cdot D^2}{D^2 - d^2}$ thick cylinder (D/t<20) $f_{hc} = \frac{P_{hc} \cdot D_{av}}{2 \cdot t}$ thin cylinder (D/t>20)
Internal pressure	$f_{ht} = \frac{P_{ht} \cdot (D^2 + d^2)}{D^2 - d^2}$ thick cylinder (D/t<20) $f_{hc} = \frac{P_{hc} \cdot D_{av}}{2 \cdot t}$ thin cylinder (D/t>20)
Direct shear	$f_s = \frac{V}{A}$
Bending induced shear (not for pins)	$f_s = \frac{W \cdot Q_m}{I \cdot b}$ non circular section $f_s = K_{bs} \frac{W}{A}$ with $K_{bs} = \frac{4}{3}\left[1 + \frac{D \cdot d}{D^2 + d^2}\right]$ circular section
Torsion	$f_{st} = \frac{T \cdot r}{J}$ circular section $f_{st} = \frac{T \cdot r}{k}$ non circular section
Hoop Compression for Pins	$f_{hc} = \frac{P_{hc} \cdot D_{av}}{3 \cdot t}$

APPENDIX 5 – Calculation of Margin of Safety

Interaction Equations (Octahedral Stress Equations)

The most general form of the Octahedral Shear-Stress Theory is for the triaxial stress state, or stress tensor. Other forms of the theory, such as for biaxial stress states, are simplifications of the equations presented in this section.

If the stress tensor has not been resolved into principal stresses, calculate the equivalent stress using:

$$f_{eq} = \sqrt{\frac{(f_x - f_y)^2 + (f_y - f_z)^2 + (f_z - f_x)^2 + 6 \cdot (f_{xy} + f_{yz} + f_{zx})^2}{2}} \quad \text{[Ref. 5]}$$

Most planar sections analysed are in a biaxial stress state, where one, or at most, two normal stresses are applied.

The Octahedral Shear-Stress Theory for the biaxial stress state is a simplification of the above equation. By assuming the stresses on one plane of the tensor to be zero (f_y, f_{yz}, f_{xy}), the general triaxial equation reduces to:

$$f_{eq} = \sqrt{\frac{f_x^{\ 2} + f_z^{\ 2} + (f_z - f_x)^2 + 6 \cdot f_{xz}^{\ 2}}{2}} \qquad (1)$$

Where for planar sections analysis :

$f_x = f_h$ (hoop stress component)

$f_z = f_a$ (axial stress component = bending stress [f_b] ± direct stress [f_t , f_c])

$f_{xz} = f_{ts}$ (total shear stress = direct shear [f_s] ± torsional shear [f_{st}])

A stress ratio is introduced for each stress components which ratios the stress component with a material allowable for that component :

$$R_x = \frac{f_x}{F_{Allow}}$$

Direct Axial Stress Ratio (Tensile) :

Limit : $R_t = \dfrac{f_t}{F_{ty}}$, Ultimate : $R_t = \dfrac{f_t}{F_{tu}}$

Direct Axial Stress Ratio (Compressive) :

Limit : $R_c = \frac{f_c}{F_{cy}}$, Ultimate : $R_c = \frac{f_c}{F_{cu}}$

Bending Axial Stress Ratio (Compressive) :

Limit : $R_b = \frac{f_b}{F_{by}}$, Ultimate : $R_b = \frac{f_b}{F_{bu}}$

Direct Shear Stress Ratio (Compressive) :

Limit : $R_s = \frac{f_s}{F_{sy}}$, Ultimate : $R_s = \frac{f_s}{F_{su}}$

Torsional Shear Stress Ratio (Compressive) :

Limit : $R_{st} = \frac{f_{st}}{F_{sy}}$, Ultimate : $R_{st} = \frac{f_{st}}{F_{su}}$

Total Axial Stress Ratio:

Section Under Tension :

Tensile Side: $R_n = R_t + R_b$, Compressive Side: $R_n = R_t - R_b$

Section Under Compression:

Tensile Side: $R_n = -R_c + R_b$, Compressive Side: $R_n = -R_c - R_b$

Total Shear Stress Ratio:

$R_{ts} = R_s \pm R_{st}$

The utilisation factor (U) is defined as the equivalent of the stress ratios calulated above. The definition for the utilization factor for a biaxial stress field is derived as follows :

A relationship between the shear and tensile allowable is given as follows:

$F_{tu} = \sqrt{3}F_{su}$ (Ref. MMPDS)

Given the $\sqrt{3}$ relationship between the tensile and shear allowable, the calculation of the utilization factor is related back the tensile allowable for the shear stress ratio, in combination with equation 1:

$$U = \sqrt{\frac{R_h{}^2 + R_n{}^2 + (R_n - R_h)^2 + 2 \cdot R_{ts}{}^2}{2}}$$

Which can be re-arranged to the following form:

$$U = \sqrt{R_h{}^2 + R_n{}^2 - R_n \cdot R_h + R_{ts}{}^2}$$

Margins of Safety

Margins of safety for both limit and ultimate conditions shall be calculated using factors of utilization derived from the appropriate interaction equation.

$$MS = \frac{1}{U \cdot K} - 1$$

Where: U is the utilization factor

K is any special factor required for design (e.g. fitting factor, casting factor, growth factor, etc.)

The basic static strength design criterion is to achieve margins of safety greater than zero for limit and ultimate loading conditions.

APPENDIX 6 – Stress Strain Curve

The are multiple methods for representing a stress strain curve. A convenient method for futur calculation is to describe the stress-strain curve through a mathematical equation.

The Ramberg and Osgood method assumes that an exponential relationship exits between stress and plastic strain by the following relationship.

$$e_p = 0.002\left(\frac{f}{f_{0.2ys}}\right)^n \quad \text{.......................................(1)}$$

This relationship is known to not be exact but sufficient for use **up to** the yield strength for the material, but **cannot** be used over the full-range stress-strain curve. Adding the linear elastic portion to equation 4.1, the Ramberg-Osgood equation becomes :

$$e_p = \frac{f}{E} + 0.002\left(\frac{f}{f_{0.2ys}}\right)^n \quad \text{.......................................(2)}$$

The exponent 'n' is determined using data points in the plastic region of the typical stress-strain curve.

In this method the plastic strains are plotted against stress on a logarithmic scale to reveal a linear relationship, where the slope of this curve represents the Ramberg-Osgood exponent 'n'.

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Figure 22 : Logarithmic Stress vs. Plastic Strain

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As a recommended check when using this method, it is important to reproduce the stress strain curve and ensure that stress/strain deviation between the two curves does not exceed 5%. Figure 23 shows the fit of the Ramber-Osgood curve to be acceptable up to the yeild point.

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Figure 23: Ramber-Osgood Curve Fit [Ref. 10]

APPENDIX 7 – Derived Material Properties

For materials where Messier-Dowty has perfomed testing for compression and shear, the allowables are derived using a method of inderect computation without regression through the use of reduced ratios. This methodology is described in MMPDS 9.5.7 (Ref. 3).

This method is an acceptable method for deriving Fcy and Fsu properties when sufficient data is not available for a more direct computation method. The derived mechanical property is determined by its relationship to an established tensile property (Ftu or Fty) for which an A-Basis has been established.

The proceedure involves pairing SUS measurements with TUS measurement for which Ftu(A-basis) is based, and pairing CYS measurements with TYS measurements for which Fty(A-basis) is based to establishe ratios. TUS and TYS are taken as the average values of the test data for which the A-basis is established.

The premise of this method is that the mean ratio of paired observations of the related properties (i.e. CYS/TYS) provides an estimate of the population means (i.e. Fcy/Fty).

For each of the test observations, the ratio is calculated as follows :

$$r = \frac{CYS}{TYS} \quad or \quad r = \frac{SUS}{TUS}$$

The population mean is calculated as follows :

$$R = \frac{F_{cy,su}}{F_{ty,tu}} = \bar{r} - t_{0.95} s / \sqrt{n}$$

where the t0.95 statistic is based on n-1 degrees of freedom [Ref. 3]

Once the population mean is establised, the derived allowable is calculated.

$$F_{cy,su} = R \cdot F_{ty,tu}$$

The basis of the derived property is assumed to be the basis for F_{ty} or F_{tu} tensile properties.

Example : Derived FSU for a material (Ref. 11)

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APPENDIX 8 – Uplock Design

The uplock is another important piece of structure to the landing gear mechanism is the uplock. The function of the uplock is to hold the landing gear or door up in the bay. When the landing gear is retracted, the latch of the uplock is released via hydraulic actuation which allows the hook to rotate through the spring load. As the hydraulics of the landing gear retraction actuator push the landing gear mating pin into the hook, the hook rotates and is locked into the latch. The hydraulic actuation which retracts the gear is then released and the landing gear hangs from the uplock. Figure 24 shows a typical landing gear uplock mechanism with dual hydraulic actuators, one which acts as an emergency release.

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Figure 24: Landing Gear Uplock Mechanism

The uplock structure see loading on the hook from combinations of hydraulic retraction, weight under g loading, and air loading on doors. The strength condition for designing uplock typically involves the worst combination of g loading, retraction loading and airloads.

For fatigue considerations, a typical fatigue spectrum involves various repeats of g loading at increasing frequency with decreasing g level for the in flight loading. It also involves the retraction and extension loading subject to every flight. The loading of such a uplock can be significant and the implications of an unwarranted release in flight could be severe so care in the structural analysis is essential. On the uplock in this example the design load is 70 metric tones.

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Figure 25 : Finite Element Analysis of Uplock Mechanism

A finite element analysis for a typical uplock hook involves modeling the proper contact between all components attached to the hook. The load goes through the uplock pin into the hook which pivots about a spigot. In the uplock pin is pushing up on the hook, the uplock will want to rotate one way but is stopped by a contact bolt. If the pin load is pulling the hook down, the load goes through the latch which is engaged on to the upper portion of the hook.

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