DESIGN AND ANALYSIS OF FIXTURING IN ASSEMBLY OF SHEET METAL COMPONENTS OF HELICOPTERS

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FATİH MEHMET BAYAR

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Approval of the Graduate School of Natural and Applied Sciences

____

Prof. Dr. Canan Özgen
Director

I certify that this thesis satisfies all the requirements as a thesis for the degree of Master of Science.

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Prof. Dr. S. Kemal İder
Head of Department

This is to certify that we have read this thesis and that in our opinion it is fully adequate, in scope and quality, as a thesis for the degree of Master of Science.

____

Prof. Dr. Mustafa İlhan Gökler
Supervisor

Examining Committee Members

Prof. Dr. R. Orhan Yıldırım	(METU, ME)	________________
Prof. Dr. Mustafa İlhan Gökler	(METU, ME)	________________
Prof. Dr. Haluk Darendeliler	(METU, ME)	________________
Asst. Prof. Dr. İlhan Konukseven	(METU, ME)	________________
Cpt. H. Erdem Akın (M.S.)	(Turkish Land Forces)	________________

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Fatih Mehmet Bayar

ABSTRACT

DESIGN AND ANALYSIS OF FIXTURING IN ASSEMBLY OF SHEET METAL COMPONENTS OF HELICOPTERS

Bayar, Fatih Mehmet
MS., Department of Mechanical Engineering
Supervisor : Prof. Dr. Mustafa İlhan Gökler

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Assembling of the compliant parts used in aviation industry is a challenging process. Assembly fixtures are quite important tools in this effort and widely used in industry. In fixturing of easily deformable sheet metal parts, besides restraining the rigid body motion of the parts, the possible deformations that may occur during the assembly process and the spring-back effect on the final product need to be taken in to consideration. In order to guarantee a successful assembling, in other words, to obtain the final product within specified tolerances, a systematic approach to the fixture design problem is required. The designer should predict the correlation between the input variations and the final assembly variation, especially, for the complex assemblies.

This study proposes a design and analysis approach in fixturing of sheet metal assemblies for helicopter components. The design of an assembly fixture for a particular tail cone has been completed convenient to the existing locating principles. Finite Element Analysis (FEA) has been realized in simulating the assembling process in order to predict the possible variation of the interested feature on a complex assembly due to deformations.

Keywords: Sheet Metal Assembly, Fixture Design, Helicopter Components, Finite Element Analysis

ÖZ

HELİKOPTER SAC PARÇALARININ MONTAJINA YÖNELİK BAĞLAMA APARATI TASARIM VE ANALİZİ

Bayar, Fatih Mehmet
Yüksek Lisans, Makine Mühendisliği Bölümü
Tez Danışmanı : Prof. Dr. Mustafa İlhan Gökler

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Havacılık endüstrisinde kullanılan esnek sac parçaların montajı zorlu bir işlemdir. Bu çabada bağlama aparatları önemli bir yere sahiptir ve endüstride yaygın olarak kullanılmaktadır. Kolayca deforme olabilen sac parçalar için bağlama aparatı tasarımında, parçaların katı cisim hareketlerinin önlenmesinin yanı sıra, montaj işlemi esnasında oluşabilecek deformasyonların ve son üründeki esnemenin de dikkate alınması gereklidir. Montajın başarı ile yapılabilmesi, diğer bir deyişle, belirtilen toleranslara uygun nihai ürün elde edebilmesi için tasarım problemine sistematik bir yaklaşım uygulanmalıdır. Tasarımcı, özellikle karmaşık yapılarda, girdi boyutsal sapmalar ile son üründeki sapmaları ilişkilendirebilmelidir.

Bu çalışmada, helikopter sac malzemelerinin montajına yönelik bağlama aparatı tasarım ve analizlerinde kullanılacak bir yaklaşım açıklanmıştır. Mevcut yerleştirme ilkelerine uygun olarak belirlenen bir kuyruk konisi için

bir bağlama aparatı tasarımı yapılmıştır. Deformasyona bağlı olarak nihai ürün üzerindeki önemli geometrik unsurlarda oluşabilecek sapmaların belirlenebilmesi için “Sonlu Eleman Analizi” kullanımı gösterilmiştir.

Anahtar Kelimeler: Sac Malzeme Montajı, Bağlama Aparatı Tasarımı, Helikopter Parçaları, Sonlu Eleman Analizi

To my wife

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INTRODUCTION

1.1 INTRODUCTION

Fixtures and jigs are unavoidable for some of the manufacturing operations. Today, advanced automated techniques are providing high production rates but still mostly depending on fixtures to satisfy the interchangeability.

For "bulk" workpieces, once every single part of an assembly is manufactured regarding the interchangeability requirements, building the assembly is straightforward since the counterparts provide necessary mating features. However, for sheet metal products, this is rarely the case because the mating features may easily deform under the joining forces. For this reason, assembly fixtures have quite important roles in manufacturing of complex sheet metal assemblies.

Nowadays, aircraft sheet metal work has very similar aspects when compared with the techniques used in early 1900's. One may still utilize the book, "How to Do Aircraft Sheet Metal Work" [1], published in 1942, as a reference book in today's applications. There are still many applications depending on the experience of the labor. This is partly because production rates and batch sizes do not always let it be feasible to automate the production line, partly because tolerance accumulation and interchangeability problem is a serious challenge for lightweight sheet metal applications.

A conventional air vehicle body comprises a fuselage, wings, a tail, and several control surfaces. It should also provide required strength for aerodynamic and inertial forces without excessively contributing to the overall weight. The body accommodates subsystems like power plant (engine), power transmission system, flight control system, avionic system, fuel, and hydraulic systems, etc. Some of the major components of a helicopter body are shown in Figure1.1. Most of the members of these subsystems are interchangeable elements. For this reason, the related surfaces of the body, where these subsystem members are fastened, are designed and manufactured to satisfy interchangeability. In addition to spare part interchangeability, some dynamic components need to be aligned to prevent excessive vibration. On the body of the air vehicle, the geometric counterparts of such dynamic components should also be assembled by means of fixtures to satisfy the alignment requirements. Except for the mating details, for the rest of the sheet metal members, interchangeability is not essential; otherwise, extra tooling is required.

To succeed in assembling the components within the specified tolerances is one of the leading considerations as it is directly related to the assembly quality and functionality, i.e., obtaining the interchangeability of the components. In order to control the degree of dimensional and geometrical variation in assembling, some assembly tools are needed. In aviation industry, it is a common methodology to fabricate the components in various manufacturing facilities at different locations and then to assemble the final air vehicle. Either in the final assembling or during the fabrication of lower degree subassemblies, assembly fixtures are used. Sheet metal parts to be assembled in these tools have highly compliant nature, which means that they are easily deformable. For this reason sheet metal fixturing needs some different considerations and principles from fixturing of “bulk” workpieces. Although a sheet metal assembly becomes reinforced and has an enhanced stiffness as the order of the assembly increases, subassemblies and single components are quite compliant. The challenging effort of keeping the variation of the assembly within specified ranges, is still depending on practical experience, i.e., the craftsman approach, which depends on “trial and error” methods. However, this approach contributes a lot to the cost per product especially for the small volumes, as it is too much time consuming.

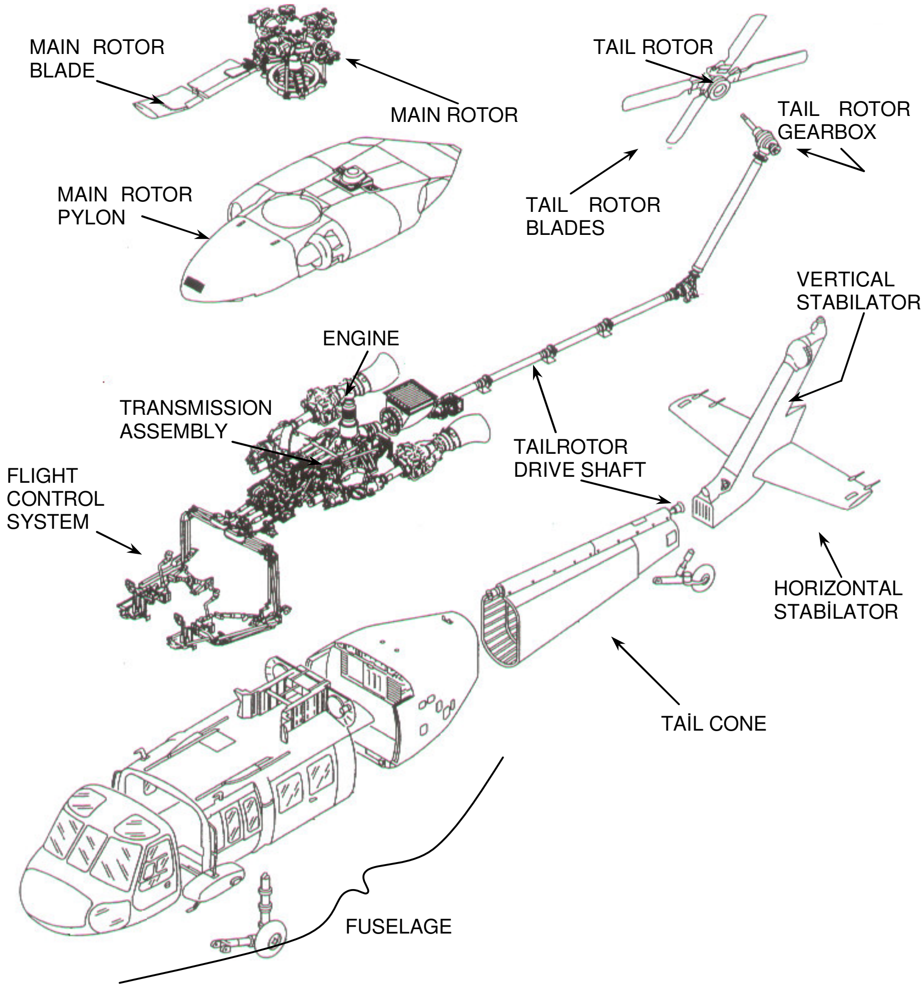


FIGURE 1.1 Major Components of a Helicopter Body [2]

This study focuses on a systematic approach to the fixturing of sheet metal components of helicopters. It is expected that understanding of fixturing principles and effects on assembly variation will contribute to the design of sheet metal assembly tools.

Throughout the study; the theoretical state of a geometrical feature specified by the given dimensions is defined as “the nominal state” of that feature. The difference between the actual state and the nominal state of a feature is defined as “the deviation” of that feature and the deviations of the same feature on a sample group of discrete parts are called as “the variation” of that feature. The allowable variation limit for a feature is “the tolerance range” and any deviation exceeding these limits is defined as “error”.

1.2 HELICOPTER STRUCTURE

Modern air vehicle structures are constructed primarily from sheet metal. The thin metal sheets have proven to be very efficient in resisting the shear or tensional loads, when the weight is the prior consideration. On the other hand, sheet metal parts should be stiffened to resist compression loads and normal-to-surface loads [3]. For that reason, they are generally stiffened with some typical members. Those members are classified under some common names and each group of stiffening parts have similar geometries within the group. In general, the stiffening members are either formed from sheet metal or manufactured directly from extruded stock materials or by machining.

When there is no stiffening members, in other words, the skin or shell is designed to resist all loads, this construction is called *monocoque*. The term comes from the French word meaning “shell only”.

However, for the cases where thick sheets should be used to resist the loading, monocoque structures are not feasible. Instead, the stiffening members are applied to form *semimonocoque* structures.

The group of common structural members in helicopter structure are given in the following subsections.

1.2.1 Skin

Skin (or “web”) of the helicopter structure is mostly made of aluminum alloy sheet metal and primarily resists shear and tension loads. Skin also provides a coverage for the rest of the structure. Figure 1.2 illustrates a portion of a skin surface and other elements of the structure.

In general, skin is the last assembled member of the structure during assembly process. Pre-formed skin parts are designed in such a way that after assembling process, the remaining portions are trimmed to fit to the geometry of the structure.

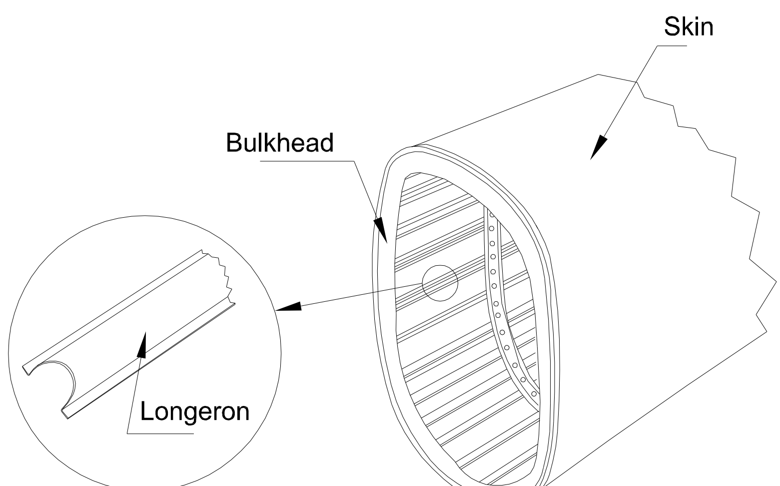


Figure 1.2 Elements of Body Structure

1.2.2 Bulkheads

The structural element which transfers concentrated loads in transverse directions to the shell of a structure, is called bulkhead (synonymously “ring” or “frame”). When used for wing, bulkheads are generally called as ribs. As shown in Figure 1.3, bulkheads have typical geometrical features, like holes to both lighten and stiffen the bulkhead, and flanges to attach the bulkhead to the skin surface continuously around their perimeters.

Bulkheads are most often the primary members to locate and clamp during the assembly process of an structure.

For some heavy lift or military purposes the bulkheads may be fabricated by machining from forged aluminum alloys.

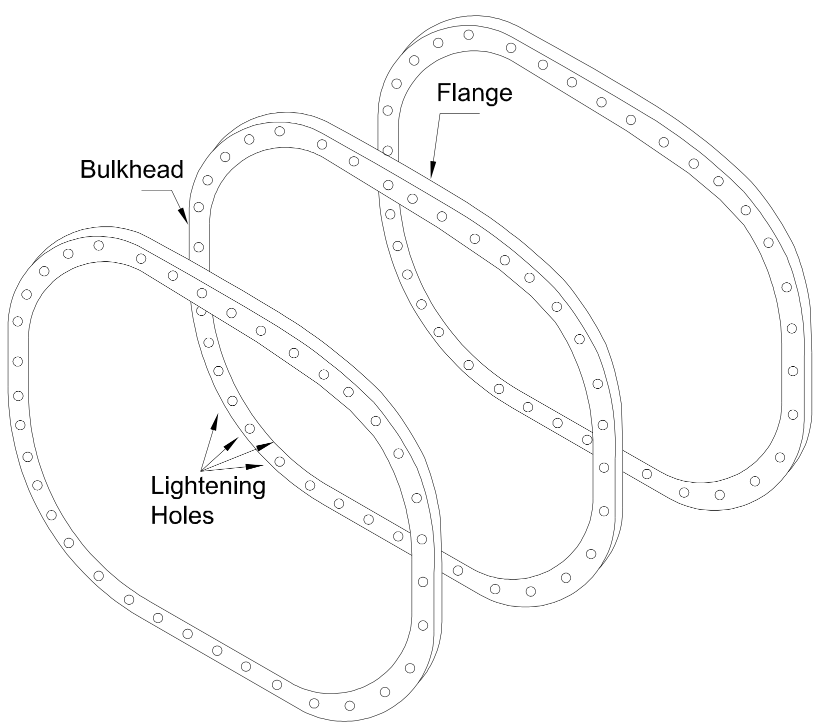


Figure 1.3 Typical Fuselage Bulkheads

1.2.3 Longerons

Longerons or stringers are the members to resist compressive loads. They generally lie along the longitudunal axis, as shown in Figure 1.2. Longerons are sensitive to buckling, and they are usually connected to the bulkheads to provide column support. They are also joined to the skin along their length. In wing structures, the equivalent of the longerons are called “spars”.

Longerons may be in different cross-sections. However, they have typical features to increase the stiffness and to apply joining. Generally, in assembling sequence, longerons are assembled after the placement of bulkheads.

1.2.4 Functionality Considerations for Helicopter Body Structure

Understanding the expected functionality of a product is important in order to construct a successful method to analyze the dimensional and geometric variation of that product. This section is devoted to the investigation of the functions of a helicopter structure.

Tolerance values of a geometric feature give the margin of dimensions and form of a feature where that feature successfully fits to the corresponding features in the assembly and performs its function. There are many subsystems mounted on the helicopter body. These subsystems have considerable number of parts and every part has its own goemetrical features. Tolerance values appointed to a feature depend on the function of that feature and may be quite wide or close. This is also valid for the corresponding features of assembled components of the body structure.

The successful assembly of a helicopter body is required;

a) to satisfy the “principal dimensions” of the helicopter, like the length, width, height, rotor diameter of the helicopter or main rotor tail rotor clereance, etc,

b) to provide the counterpart mating surfaces for moving components like windows, doors, cowlings,
c) to provide the base for the subsystem components or their fittings and supports.

The considerations stated in an order above requires wider to the closer tolerances, respectively. The general dimensions of an helicopter body may have variations in the order of several centimeters, where variations in dimensions of subsystem components supports are within a few milimeter (or may even be under a milimeter).

In general, among the subsystems of a helicopter, the power transmission system is most critical. Supports of the dynamic components of power transmission system, like shafts, bearings, gearboxes, etc., require an additional alignment procedure to satisfy the close assembly tolerance of dynamic components. The structures on which such supports lie need to satisfy closer tolerances than the adjustment limits of the alignment procedure in order to guarantee a successful alignment. The components of a usual power transmission system are shown in Figure 1.4.

The alignment of bearings and gearboxes of relatively long tail rotor drive shaft is an example of the required close tolerance assembly applications. The tail cone should maintain its straightness and the dimensions between its related features should remain within the tolerances after the assembling of the structure.

Similarly, for the moving parts like, doors, windows and sliding cowlings tolerances are important in order to provide sufficient isolation (of wind, noise, water leakeage, etc).

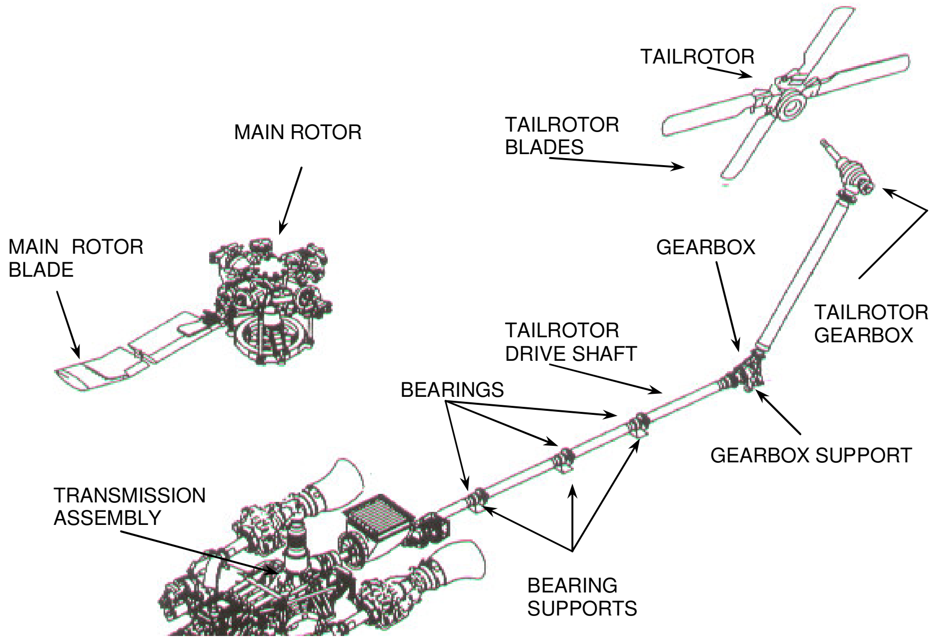


Figure 1.4 Components of Power Transmission System

1.3 INDUSTRIAL PRACTICE

1.3.1 Sheet Metal Joining Methods

Aeronautical structures widely depend on sheet metal products. Once the sheet metal parts are fabricated by means of selected forming methods, they need to be assembled properly to achieve the desired final product.

In aviation industry, necessity to light weight makes the considerations challenging. A very high percentage of the parts, even structural or not, constituting an air vehicle, is made of sheet metal. In this study “sheet metal” refers to the stock materials in sheet form, thickness between 0.006 inches (0.15 mm) up to ¼ inches (6.35 mm) [4]. A very high percentage of these sheet metal parts are made of aluminum alloys where magnesium alloys, titanium alloys and steel alloys are other common materials.

Aluminum alloys have superb specific strength and corrosion resistance. Beside these advantages, from joining point of view, the major disadvantage of most of the high-strength aluminum alloys used in aviation industry is poor weld ability [5, 6]. Due to this fact, main joining method for structural assemblies of aluminum alloy sheet metal components is riveting. Although Resistance Spot Welding and Gas Tungsten Arc Welding (GTAW) are also used for joining operations, they are rare compared to riveting, and beside some exceptions, applications are commonly for non-structural components [7].

Bonding methods are also applied for sheet metal joining and they are becoming popular. However, bonding methods still can not be taken as an exact alternative for riveting.

In this study, the joining method for the assemblies of sheet metal should be considered as riveting.

1.3.2 Sheet Metal Assembling

As a manufacturing process, assembling means joining at least two parts to each other to obtain a specific geometry or function. If manufacturing a product in one piece is not possible due to economical or fuctionality reasons, the remaining way is to manufacture as discrete parts and assemble these parts to reach the final product.

Sheet metal parts are widely used in automative and aviation industries. These parts are manufactured quite economically by using metal forming processes. However, considerable effort is needed to ensure the proper assembly of these discrete parts.

A sheet metal part generally requires more than one assembly stage to reach the final product. Mostly, the part is suited in an assembly fixture to locate properly according to its assembly-mates. On this assembly tool, the part is joined to the subassembly and extra operations may be performed to prepare the part for the following assembly steps (for example: drilling a guiding hole or a fastener hole on the part, etc). After transferring the subassembly from one assembling tool station to another and completing the steps, the final product comes into existence. It should be noted that the term “final product” used here may refer to a complete air vehicle or just a very basic subassembly. This depends on the facility’s product theme.

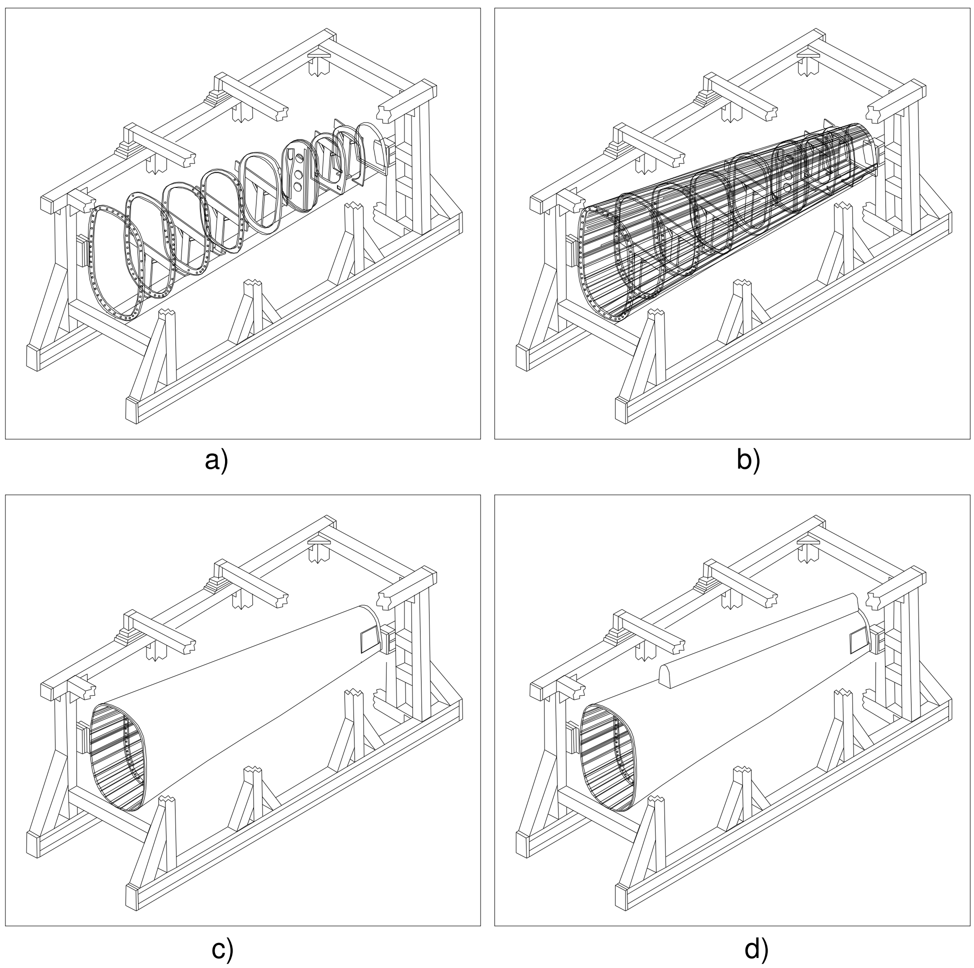


Figure 1.5 Stages of Assembling

The stages of assembling process are peculiar to the product itself but always performed in a hierarchical manner. Figure 1.5 shows the basic sheet metal parts of an illustrative tail cone of a helicopter and the assembling stages for that tail cone on an illustrative assembly fixture. Here at (a) bulkheads are located to the assembly fixture and clamped. The following step, (b), demonstrates locating and fastening of the longerons. Next step is the application of the pre-formed skin, which is the surface sheet metal. Finally, the complementary parts and accessories are fastened onto the tail cone, and the critical fastening holes of fittings and supports are drilled and reamed.

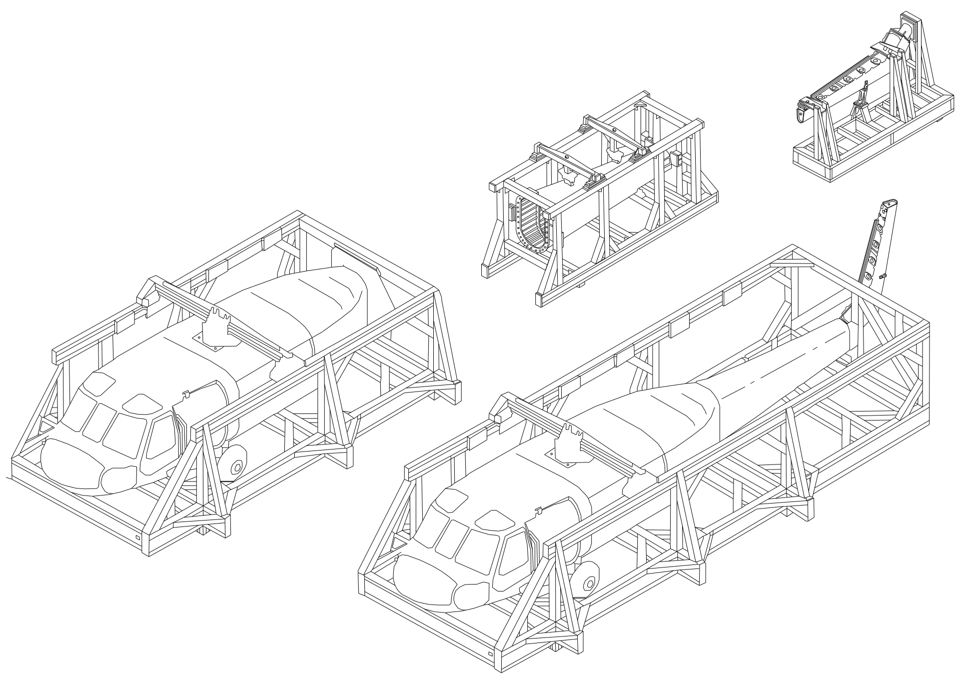


Figure 1.6 Several Fixtures Used in an Assembly Line [2]

The built-up procedure mentioned above is not necessarily to be completed using the same tooling base. Basically, there are two alternatives: first is to use a single fixture body with different sets of fixture details for different stages, subsequently. Second is using different sets of fixture details in different fixture bodies. The decision strongly depends on the production rate.

For high production rates, processes are divided into several assembly stations. In an ordinary assembly line, several stations exist with a determined number of assembly fixtures. During each stage, some of the components are added onto the semi-product assemblies, or some subassemblies joined to achieve higher order assemblies. Figure 1.6 shows some illustrative assembly fixtures for a sheet metal assembly line.

1.3.3 Design, Manufacture and Set-up of An Assembly Tool

As the assembling process for sheet metal parts has its own characteristics, the design of the tools required for this operation should concern several facts. Although assembly tools are commonly used in aviation industry, there is no rule of thumb for design procedure. In industry, several non-written principles are valid for designing of sheet metal assembly fixtures. Actually, these principles vary for each facility according to the experience gained and the available technology for measurement and manufacturing. A general flow diagram for design, manufacture and set-up of an assembly fixture is given in Figure 1.7.

A standard assembly fixture is composed of locators, clamps, extra supports and a base. Locators are used to place the workpiece into the fixture to provide deterministic loading. Located workpieces are secured by means of clamps. Locators and clamps of a fixture generally called as fixture elements. If required, extra supports are added to resist the forces occurring during the process. The base of the fixture provides the necessary surfaces to hold these components together. A detailed introduction about fixtures is given in section 2.1.

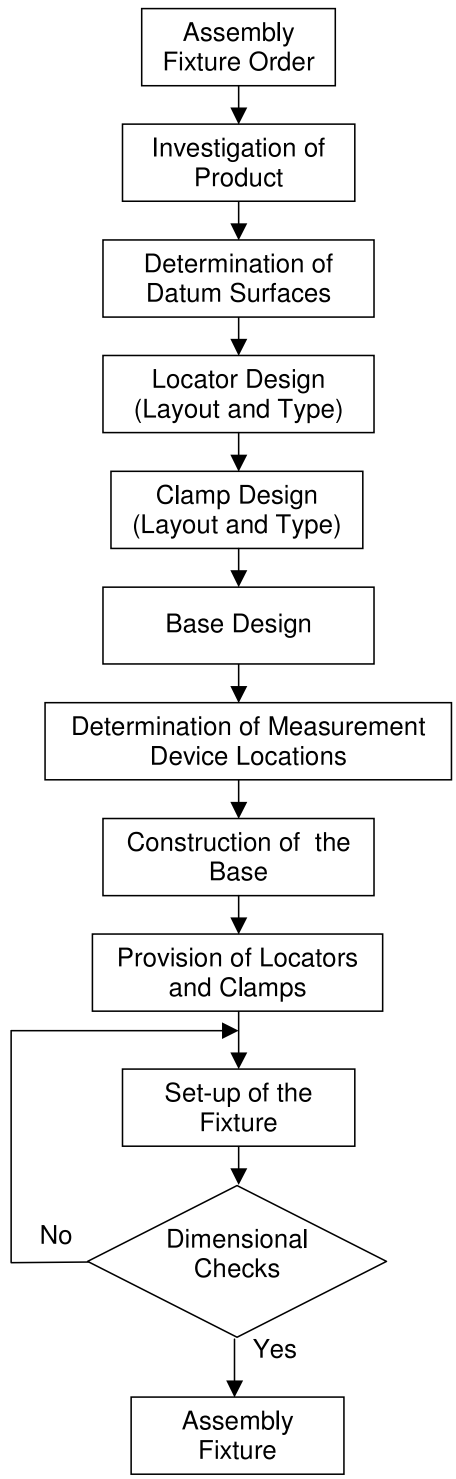


Figure 1.7 General Flow Diagram for Manufacture and Set-up of an Assembly Fixture

Design process of an assembly tool needs inputs from several disciplines. In planning of the production line, the decision about the number of the fixtures and the operations, determines the function of each assembly tool to be designed and manufactured.

Once the task of an assembly tool is assigned, the next thing to do is to determine the mating surfaces between the product and the fixture. These surfaces are generally taken as datum surfaces. The selection of datum surfaces is negotiated among the product engineer, the assembly engineer and the tool engineer, as it is a critical decision for the fit and functionality of the assembly as well as for the successful loading and joining of parts without interference. Selection of datum surfaces should also satisfy the required kinematic constraints for deterministic loading and total restraint.

After determining the datum surfaces, the locators on the mating surfaces are designed. Locator design requires two basic decisions, locator layout and property. For layout design, preventing the excessive deformation is the primary concern. The types of locators are selected due to the workpiece geometry and properties like surface roughness, contoured surfaces, workpiece stiffness, existing geometric features on the workpiece, etc.

Locators and their layout should be designed according to pre-determined mating surfaces of parts and kinematic constraints. The locating principle in fixturing is described in section 2.1. If required, locators must be removable due to part loading and unloading procedure. Locator type and layout should also meet the requirements to resist the weight of the workpiece.

Design of clamps is similar to the locator design. The layout and type of clamps should be determined according to the clamping sequence, workpiece geometry and the clamping force required to resist gravitational and operational forces.

The base should be designed to provide enough space for fastening the fixture elements (locators and clamps) onto it. The main theme is to design a supporting base platform rigid enough to resist process loads and gravitational loads where it also provides the operator an easy-to-work and secure media. Actually, this step does not make much sense from sheet metal assembly fixturing point of view.

Finally, measurement reference points and their layout should be designed according to the available dimensional measurement techniques. The layout needs to be designed in such a way that it should be possible to take measurements for each individual point required. In addition, measurement accessories should be selected convenient to the available measurement technique.

The considerations above are the basic considerations for an assembly tool. Of course, except the ones stated above, one may claim several design considerations according to the specific requirements. In this study, the “design of an assembly fixture” refers to the determination of the locations and tolerances of fixture elements.

Fixture elements and their complementary accessories (pins, bushings, fasteners, blocks, etc.) used in a sheet metal assembly fixture, are generally selected from standard design manuals. These standard parts may be purchased directly or may be manufactured according to specifications.

The base platform of the assembly fixture is constructed by welding of standard stock materials of steel alloys. Suitably selected beams are welded to achieve the aimed geometry. Then the base is subjected to a stress-relief heat treatment to reduce the residual stress from welding. Nowadays, different techniques, like resonance stress-relief, are also applicable and being popular [8]. Whatever the stress-relief technique, the main point in establishing an assembly fixture is not to apply welding or any other operation that may cause residual stresses after stress-relief treatment. In other words, all the joints and additive segments should be built-up by means of screws, pins, etc.

Once the base, fixture elements and their accessories are provided, the next step is the set-up of assembly tool. When the location of the tool is decided, a proper coordinate system is assigned for the measurements to be taken.

In aviation industry, a specific coordinate system is used for manufacturing purposes (with small changes from facility to facility), which is based on the Cartesian Coordinate System [9]. This coordinate system consists of three orthogonal planes and infinite number of imaginary parallel planes. Any critical point on the air vehicle is defined by the intersection of the three orthogonal planes passing from that point. The coordinate system of the measuring device is generally set to suit the coordinate system of the product.

The location of the measurement system tools and devices should be selected properly in order to be able to get measurements for all the necessary points without any visual access problems. Depending on the measurement system available, the orientation and adjustment of measurement devices is a time consuming and operator dependent process.

For the first set-up of an assembly tool, there are two popular methods used in industry. First method is using a “master gage”, which is properly manufactured according to the product design. Once the gage is set according to the coordinate system, this method is quite straightforward. However, it depends on an accurate and expensive gage. When all the necessary fixture elements are tied-up, an operator should perform the required dimensional checks by means of the measurement system.

Second method is directly measuring, locating and setting the fixture elements in all details. Although this method is quite time consuming and expensive as all the features of the fixture elements should be arranged one by one, in the absence of a proper gage, it is the only way for fixture set-up.

An important point for the inspection procedure of an assembly fixture is the consideration of thermal elongations. Generally the measurements are taken at a specified temperature, since thermal elongations might be quite significant when compared to the fixture tolerances allowed.

1.3.4 Problems In Sheet Metal Assembly Tooling

In aviation industry, assembling the formed sheet metal parts, from dimensional accuracy point, is generally a greater challenge than manufacturing these parts. There are several reasons for this challenge. Firstly, the parts and the final assembled body are too large when compared to most of the assembling operations performed in other industries. Although the dimensions are relatively large, tolerances specified for these parts are close tolerances. This is especially valid for the holes and supporting surfaces of structural sheet metal components, on which dynamic components lay or attach. Most of the time, the challenge in satisfying quality requirements makes the use of assembly fixtures inevitable, regardless of production quantity. Generally, production quantity and rate have a much more important effect on determining the number of stations, i.e., number of assembly fixtures.

Next problem is the compliant nature of sheet metal parts. Being compliant, sheet metal parts have the ability of compensating some dimensional variation. However, they can show some significant deformation due to operation loads if not properly hold. Supporting sheet metal parts during machining or assembling processes, requires more attention compared to dealing rigid or semi-rigid parts. The compliant nature also creates problems in deterministic loading of the sheet metal workpieces.

Another problem of sheet metal assembling arises when the assembled body is released. This is the spring-back effect and it is a serious challenge in satisfying the dimensional quality requirements of the final product. This phenomenon will be discussed in detail in the following chapters.

1.4 MOTIVATION AND PROBLEM STATEMENT

The motivation of this study depends on the problems met in industrial applications. Sheet metal assembly tools are frequently needed to control the dimensional and geometric variation of the assembly in several situations, for example; during some local or general repairs of helicopter body or for a small batch-sized manufacturing, etc. In practice, the methodology of design is almost always based on "trial and error". The results of the initial design are obtained after the first product is completed. Even after several products are manufactured there may be some negative feed-backs about the products and corrective action may be required. Considering the limited number of products, the corrective rework cost per product for any particular fixture is generally very high.

Therefore, to eliminate the design problems of assembly tooling which may cause serious corrective actions, some systematic approaches are needed. In aviation industry, especially for the overhaul and maintenance facilities or small sized companies, the approaches that will decrease the assembly tooling rework cost would mean a lot.

In previous sections of this chapter, it is explained that, although assembly fixtures have vital importance in aviation industry, most of the principles rely on practical experience. It is hard to claim general principles comprising all cases of assembly fixturing needs of the industry.

The aim of this study is to develop a systematic approach on assembly fixturing for sheet metal components of helicopters. Any knowledge on this subject will not only value from the assembly fixture design point, but also may give ideas about how to improve the product design to access full interchangeability.

The components are restricted to helicopter components. Perhaps, the fundamental approach for fixturing regarding all air vehicles might be based on similar principles. On the other hand, today's air vehicles belong to a very crowded family where products designed and manufactured are depending on very different considerations. Even rotary-wing air vehicles have several types and different applications among themselves. The term "helicopter" in this study refers to a rotary-wing air vehicle with a main rotor generating the lift and a tail rotor providing the required anti-torque for the air vehicle.

The thesis has been organized in five chapters. Chapter 1 explains the existing industrial practice and the problems of the sheet metal assembling as well as the motivation of the study. Chapter 2 reviews the fixturing concept for design. Chapter 3 discusses the methodologies for the variation analysis from sheet metal assembling point.

Chapter 4 presents design and analysis of a sheet metal assembly fixture, where part locating and clamping details are discussed as well as the effect of possible workpiece and fixture variations on the assembled product. Although there are infinite number of cases and combinations, some representative cases should be selected to get the main idea without excessive calculation effort. The analysis of some selected cases for sheet metal products and the assumptions for the analysis will be given in fourth chapter in detail.

Chapter 5 presents the conclusion, the contribution of this study and the recommendations for future work.

CHAPTER 2

FIXTURES

2.1 GENERAL FIXTURING EXPERIENCE

Fixtures are work-holding devices to locate and secure the workpiece during a process, either in manufacturing (machining and joining processes) or in inspection. Jigs differ from fixtures by guiding at least one of the process tools. Joining tools are identified by the operation they are used for: assembly fixtures, welding fixtures, soldering fixtures, riveting fixtures, etc.

Most of the knowledge about these tools comes from practical applications in industry. This experience gained has been collected in several handbooks and manuals [10-15]. Besides, newly introduced concepts like flexible and computer integrated manufacturing systems, triggered the new approaches for fixturing. Modular fixturing or automated fixtures are quite popular research topics for today [16, 17].

Although there had been some researches on fixturing before 1980 [18, 19], the basic “3-2-1” locating principle, which had been already used by the industry, was first proposed as a standard with the ANSI Y14.5M-1982 [20].

Any object in three dimensional space has six degrees of freedom (DOF); three translational and three rotational. Considering the positive and negative directions of every axis, the object needs to be restrained in total of twelve directions to be kinematically constrained.

According to “3-2-1 principle”, workpiece is located by means of three, two and one locators in three orthogonal planes, i.e., primary, secondary, and tertiary planes, respectively. The essence of the principle is to obtain deterministic loading of the workpieces, i.e., no locator interference or no redundant locators exist by means of the unique position provided by the fixture.

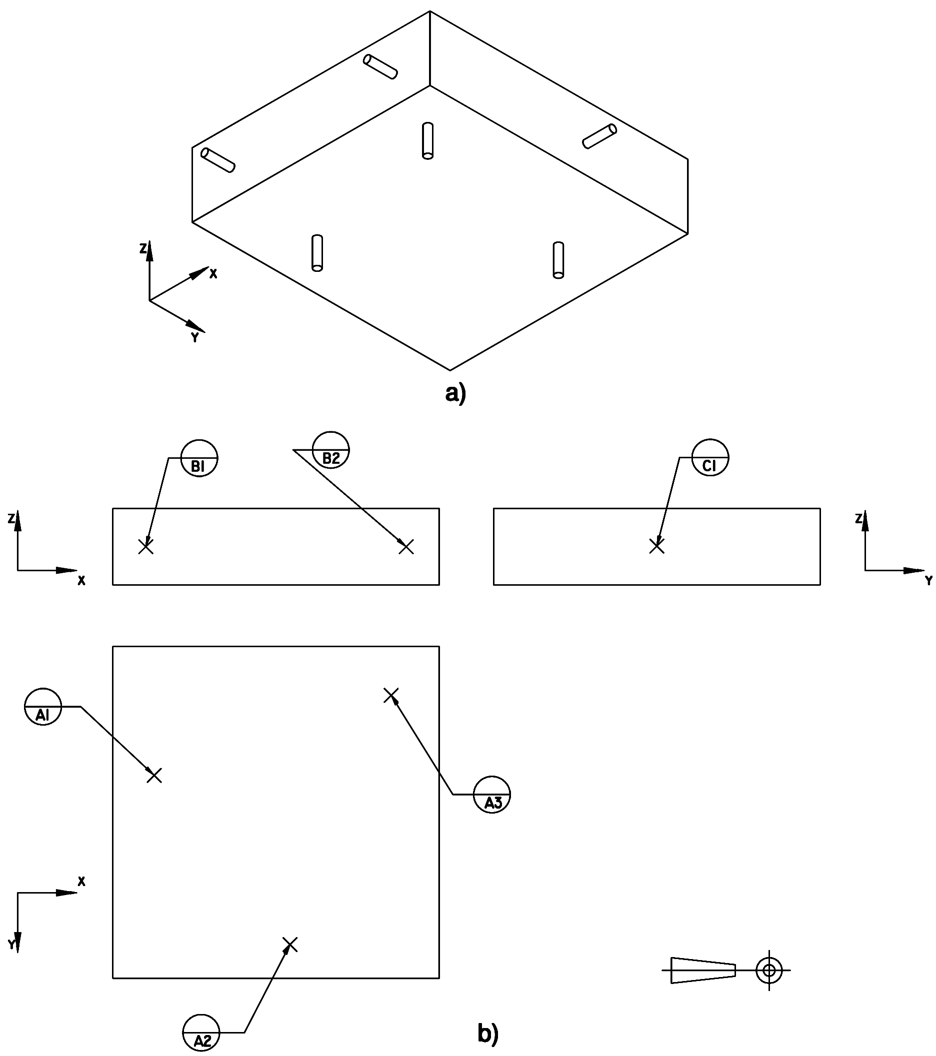


Figure 2.1 3-2-1 Locating Principle and Datum Targets

Figure 2.1 (a) demonstrates a prismatic workpiece resting on six pins according to 3-2-1 principle. The points of contact for the locators are called datum targets (given at b). As long as the workpiece is in contact with the locators, all of its freedoms for rigid body motion are restrained, which means that it is kinematically constrained.

Extra work-holding supports may be applied to prevent excessive deformation under process loads. All of the elements should be supported with a base. The schematic illustration of a machining fixture with a rectangular workpiece is given in Figure 2.2.

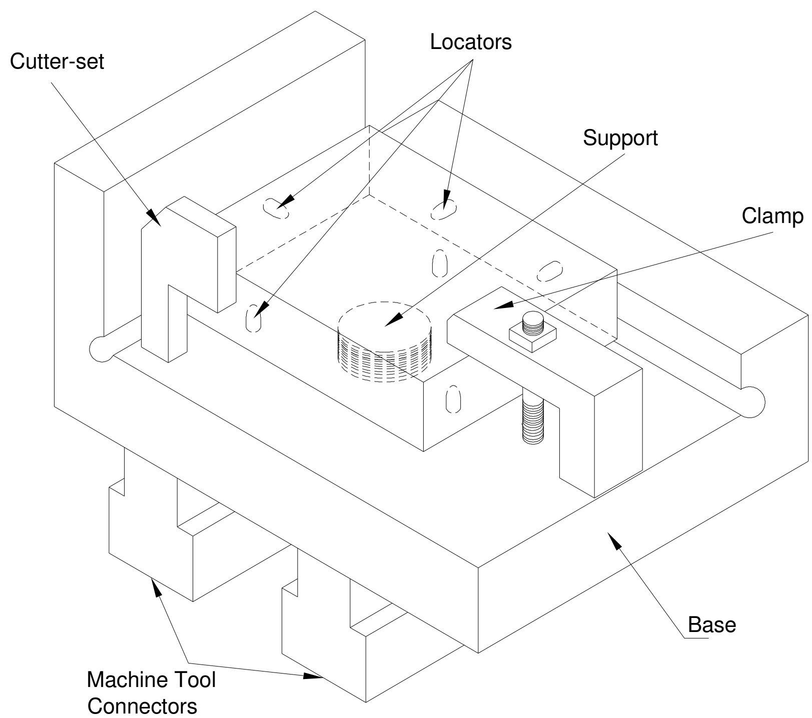


Figure 2.2 Schematic Illustration of a Machining Fixture

The workpieces which have accurate holes that are perpendicular to the primary locating surface, may be located using these holes. In such a case, the primary datum is located by means of three pins as in Figure 2.1 or by planar surfaces as in Figure 2.3. However, instead of 2-1 pins, hole features are utilized as secondary and tertiary datum features. Two pins are used, one of which is a full pin and the other is a diamond pin. This configuration is used for not violating the locating principle and preventing any pin being redundant. This method is also suitable for the locating of sheet metal workpieces.

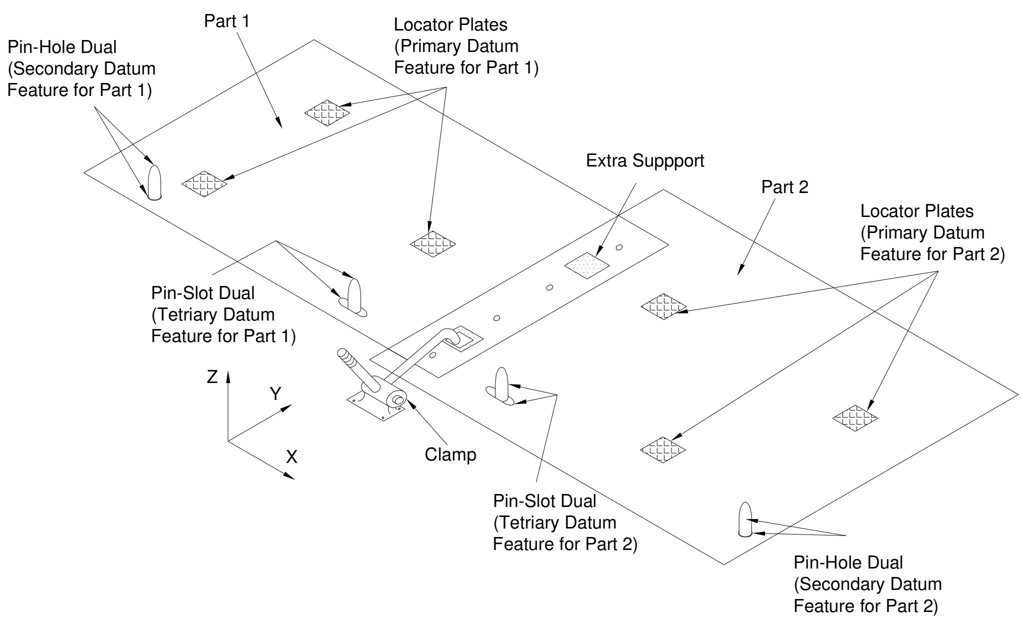


Figure 2.3 Locating and Clamping Illustration of Sheet Metal Parts on an Assembly Fixture

In sheet metal fixturing, 3-2-1 principle applies such that primary datum is the surface of the sheet metal part. Freedom of the workpiece in normal to surface direction is restricted by means of locator blocks. Pins are not very suitable for easily deformable sheet metal parts as they exert forces within a small contact area. Combinations of a full pin (four-way pin, hole-pin dual) and a diamond pin (two-way pin, slot-pin dual) are frequently used to restrain the parts in surface directions. In Figure 2.3, two planar sheet metal parts located by means of hole features, are shown. It should be noted that these holes and slots may already exist on the workpieces or may be drilled for locating purposes. In practical applications, these holes are called as “tooling holes”.

Once the work is loaded to the fixture it should be clamped by means of proper clamping devices to provide the contact to the locators during operation. Some of the popular clamping devices are screw clamps, cam clamps, toggle clamps and strap clamps [21]. Beside these clamping mechanisms, different sheet holders may be used for sheet metal joining operations. Some of these clamps are toggle mechanisms, C-clamps, or cleco fasteners. These clamps are shown in Figure 2.4. These mechanisms are applied to close the gap between two joining geometries and to secure this state during the joining process. By their nature, these mechanisms apply clamping force to both parts, equal in magnitude but opposite in direction.

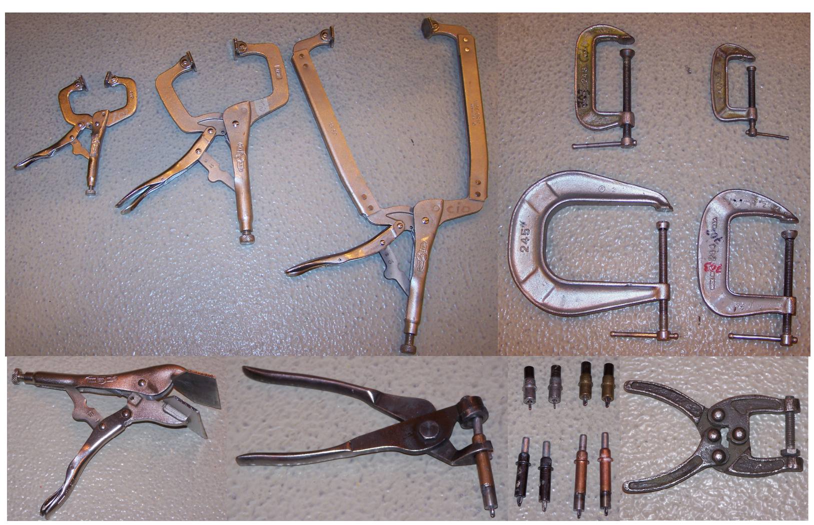


Figure 2.4 Some Clamps for Sheet Metal

2.2 LITERATURE SURVEY

As stated in the previous section, general information from practical experience about fixturing may be found in handbooks and manuals [10-15].

For today, basic problems in fixturing that studies focused on, may be summarized as: locating scheme design and optimization, clamping force optimization, contact analysis, joint characteristics, work-piece deformation analysis, finite element modeling and fixture diagnosis.

Although there is wide literature on fixtures, greater portion of the existing literature is about fixturing of bulk workpieces. Fixturing concepts on sheet metal parts are more recent but developing rapidly.

This section will be given in two subsections, one of which is devoted to literature of fixturing for 3-D bulk parts and the other is devoted to literature about sheet metal fixturing.

2.2.1 Literature for Bulk Workpiece Fixturing

Menassa and DeVries are among the firsts who treated the fixture design problem as an optimization problem. In their research [22], they proposed an optimization technique for determination of fixture support positions. The objective function was to minimize the workpiece deflection on selected nodes under process loads. Deflections were calculated by Finite Element Analysis (FEA). In their research, results of three different examples were given, reporting at least of 40 percent reduction in objective function.

Choudhuri and De Meter [23] developed a methodology based on worst-case tolerance analysis to analyze the effect of the locator error to the given datum related linear, machined surfaces. In this study, the authors also developed a simulation to indicate the relation between machined feature geometric error, locator design, and locator tolerance scheme. In this research, deformation of workpiece is not taken into consideration, which means, for the compliant workpieces, the methodology does not promise much. On the other hand, the research yielded interesting results. The authors concluded that the common rule of thumb for determining the fixture variation range (<10% of workpiece tolerance) is potentially risky and the relation between the datum variation and datum related feature variation is quite complex.

Marin and Ferreira [24] proposed a methodology to analyze the relation between the fixture locator deviation and the workpiece geometric tolerance specifications for a workpiece-fixture dual based on 3-2-1 locating principle. They extended the tolerance analysis problem by adding the curved surfaces. They proposed two problems, namely, forward problem and inverse problem. Forward problem is the determination of whether the known deviation of fixture locators violates a given tolerance of a specific feature on the workpiece or not. Contrarily, inverse problem is the determination of bounds that fixture locators should remain within in order to satisfy the given feature tolerance. The methodology assumes that the workpiece is rigid regular solid. This assumption of rigidity makes it impossible to use for compliant sheet metal parts.

Other researches on layout design or layout optimization for solid workpieces are by Lee and Haynes [25], Menessa and DeVries [26], De Meter [27], Lee and Cutkosky [28].

Satyanarayana and Melkote [29] investigated the workpiece-fixture contact modeling for FEM applications and provided an experimental verification. In their research, they modeled spherical-planar and planar-planar fixture element-workpiece contacts. They modeled the workpiece-fixture contact with different boundary conditions: nodal contact elements, surface-to-surface contact elements, nodal displacement constraints and nodal forces.

They also extended the study by modeling the compliant workpieces. The results are summarized as recommendations for such modeling studies, including recommended element sizes. The spherical-planar contact modeling results are also verified experimentally and the experimental study yielded average relative error for predicted deformation as 4.93%. According to the results, the authors also concluded that the best method for such applications is surface-to-surface contact modeling.

Fixture-workpiece interaction deformation and clamping force considerations are also studied by others. Some of the researches on the subject are by De Meter [30], Li and Melkote [31-32], Marin and Ferreira [33], Raghu and Melkote [34], Tan et al. [35].

Chen et al. [36] reported a research on optimization of measurement device placement and selection for positional certification of fixture elements to reduce the time for measuring device establishment of fixture systems.

2.2.2 Literature for Sheet Metal Fixturing

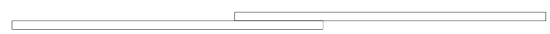
Liu and Hu [37, 38] proposed a methodology based on linear mechanics to simulate the variation of deformable sheet metal assemblies and extended their study [39] with the application of finite element Analysis (FEA). A detailed introduction of this methodology is given in Appendix A.

Chang and Gossard [40] developed a model for compliant assemblies. They identified the parts and the tooling by means of features (hole, slot, locator surface, measuring point, etc). According to feature coordinate frame, they expressed the variations, displacements, and forces as 6x1 vectors. The relationships between those during a Place, Clamp, Fasten, Release/ Place, Clamp, Measure, Release (PCFR/PCMR) cycle are represented graphically as “contact chain”. The parts and tooling are defined as object nodes where the contact between the mating features are called contact nodes. For the object nodes the constitutive relations, and for the contact nodes the geometric compatibility and force equilibrium relations are applied to satisfy. The proposed model is used to simulate the propagation of variation in the final assembly.

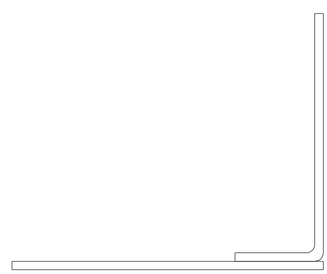
Cai, Hu and Yuan proposed a new locating principle for sheet metal fixturing [41]. They called this locating scheme “N-2-1” locating principle where N locators are applied to the primary surface of the sheet metal part. Authors also declared that two and one locators are required and sufficient to restrain in plane motion of the sheet metal parts because the major process forces are applied to a sheet metal part in out-of-plane direction. Forces in plane directions are avoided to prevent buckling. They also proposed two important conclusion: first is that the two locators in secondary datum should be as far as possible to minimize the variation due to locator error. Second is these two locators should not be applied on opposite sides, because small geometric imperfections lead large deflections in transverse direction. Authors also reported of an optimum fixture design software (OfixDesign), which uses several software for modeling, mesh generating, FEA solution and optimization. The objective function for optimization is based on minimizing the out-of-plane deformation of the sheet metal workpiece. Optimum locator layouts (minimum workpiece deformation) for an example planar sheet metal part were given for both 3-2-1 and N-2-1 principles.

Liu and Hu [42] studied the variation characteristics of three basic sheet metal joint types: lap joints, butt joints and butt-lap joints. These joint types are shown in Figure 2.5. In their study, authors used “Mechanistic Variation Simulation”. Analysis showed that the joint type has an effect on the transmission of part variation to final assembly variation. According to both analytical and experimental results, they concluded that best assembly quality is achieved by lap joints. Lap joints “absorb” the part variation while transmitting its effect, where butt joints have “magnifying” effect on part variation transmission. Butt-lap joints have a moderate effect on assembly quality between these two types.

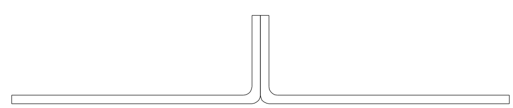
Merkley [43] developed a tolerance analysis model for flexible assemblies based on a spring model. The problem is linearized and the gaps (or interferences) between the mating parts are modeled as spring systems. Bihlmier extended that study [44] with the application of finite element analysis.



a) Lap Joint



b) Butt-Lap Joint



c) Butt Joint

Figure 2.5 Basic Joint Types

Hu et al. [45] proposed a new method for robustness evaluation of compliant assembly systems. Camelio et al. [46] developed a methodology to analyze the variation of compliant assemblies using geometric covariance.

Camelio, Hu, and Ceglarek [47] extended “Mechanical Variation Simulation” methodology for multi-station assembly systems. They defined re-location matrix by using homogeneous transformations to express the effect of re-orientation of the workpiece when it is located to the new assembly station. According to the study results, authors concluded that non-additive characteristics of parallel compliant assemblies might not be valid for certain level of fixture and joining tool variation. This conclusion means that assembly variation may increase with respect to part variations even for parallel assemblies if the levels of fixture and tooling variations are considerable.

Camelio, Hu, and Ceglarek [48] studied optimal fixture design problem for especially resistance spot welding joining of sheet metal parts by minimizing the final assembly variation as objective function. They applied an optimization algorithm composed of a FEA solution and nonlinear programming methods to optimize the locator position for the minimum assembly variation and modeled the assembly process by “Mechanistic Variation Simulation” methodology. In the study, the effects of part variation, weld gun variation, and fixture variation on the final assembly variation were analyzed and summarized.

Hoffman and Santosa [49] proposed a mathematical model to predict the change in assembly variation due to the clamping and welding sequences. They performed a Monte Carlo Simulation to evaluate the behavior of the assembly due to the clamping sequence. The result showed that, to minimize the assembly variation, clamps should be sequenced from one end to the other.

For multi-station automated assembly lines, fault diagnosis is an important consideration. Researches by Ceglarek and Shi [50, 51], Ding et al. [52] are among the several studies devoted on the fixture fault diagnosis.

Walczyk and Raju [53] proposed a simplified fixture development methodology for compliant sheet metal and composite aircraft parts to eliminate the difficulties of supporting such workpieces during machining process.

CHAPTER 3

DIMENSIONAL VARIATION IN ASSEMBLING AND ANALYSIS METHODS

3.1 VARIATION PROBLEM IN ASSEMBLING

Dimensional variation of the assembled product basically comes from three sources: component variation, fixture variation and joining process variation [47]. These variation sources are given in Figure 3.1. Component (or part) variation is something related to manufacturing process. Every manufacturing method introduces its own variation. Last two sources of variations, fixture variation and joining process variation, appear during assembling process.

In literature, for a systematic approach to the variation problem of sheet metal, the assembling process is considered in four steps. These steps are given by Figure 3.2 as:

a) Placing (loading) the sheet metal parts to the work-holding fixtures,
b) Clamping the sheet metal so that the parts are deformed to nominal positions,
c) Fastening the sheet metal parts,
d) Releasing the tooling (totally or partly, depending on the next process) and the assembly springs back.

This cycle is known as PCFR cycle. If the process is an inspection process instead of an assembly process, the third step is the measurement step and the cycle is called as PCMR cycle where “M” stands for measurement [40].

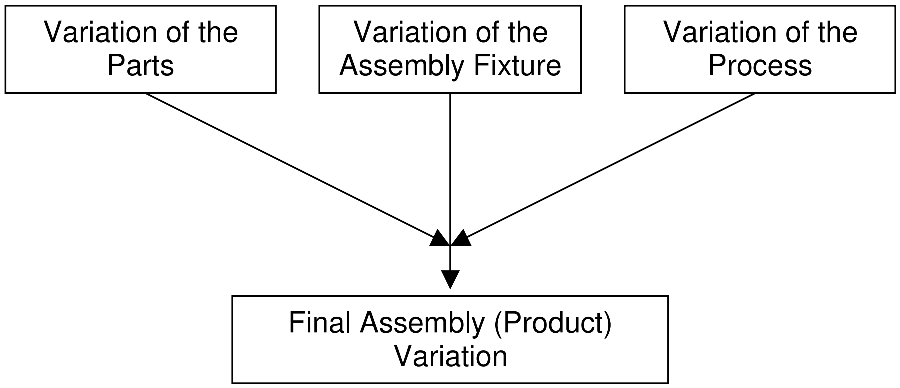


Figure 3.1 Variation Sources

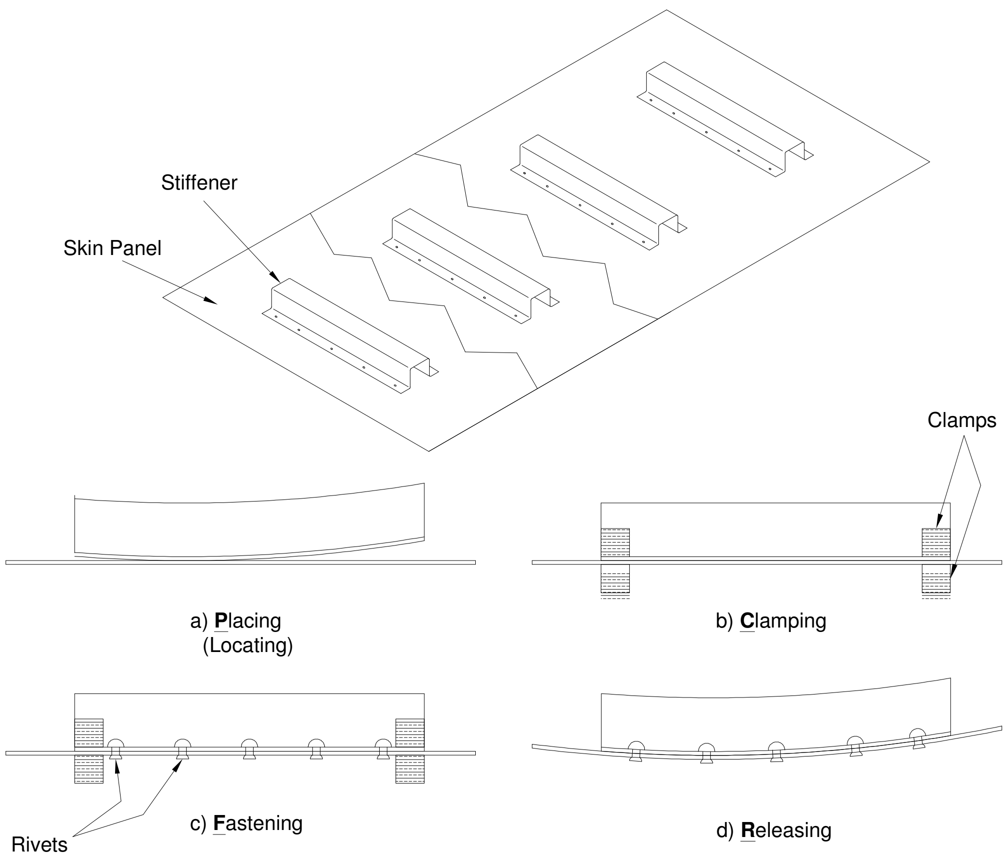


Figure 3.2 PCFR Cycle

3.1.1 Variation of the Components

The components of an assembly (parts or subassemblies) have their own variations sourced from fabrication. These variations may cause gaps or interferences between the mating features of the assembly. However, deformable sheet metal parts have the ability of compensating such cases.

The stiffener shown in Figure 3.2 (a) has a significant variation in normal-to-plane direction with respect to the skin surface. When the clamps are applied, the part is brought to its nominal state as shown in Figure 3.2 (b) assuming that fixture dimensions are perfect. Then, the parts are joined (c). However, after releasing the assembly (d), it springs back to minimize the strain energy stored during the process. This spring-back effect is one of the main problems for flexible sheet metal assemblies.

3.1.2 Variation of the Assembly Fixture

Fixture variation also contributes to the final product variation. A rule of thumb in design of such tools like fixtures, jigs, gages, etc, is to appoint the tolerances of a feature of the tool as 5-29% of the tolerances of the corresponding workpiece feature [21]. For tool engineers, taking that proportion as 10% is a very common application. However, taking a certain percentage of the workpiece feature tolerance may ignore the relation between the tool feature variation and final assembly variation. Ignoring the sensitivity of product variation to fixture variation may result in mainly two undesired cases: unnecessary tight fixture tolerances (means excessive cost), or failure in satisfying product tolerances (due to propagated effect of fixture variation).

3.1.3 Variation of the Joining Process

Joining process variation characteristics vary according to the type joining process. For resistance spot welding, the variation of weld gun directly means forcing the workpieces to a non-nominal position and joining the parts at that state. In such a case, the final assembly may spring back as in the case of part variation.

For riveting process, the variation of the rivet holes and rivets may have important effect on assembly variation. If there exists a misalignment between the corresponding holes on the parts, the parts will be forced to deform and compensate the misalignment. This causes stretching (or compression) of the sheet metal, introducing residual stresses in in-plane direction and may even result in waviness on skin.

Generally, in aviation industry, the riveting holes on corresponding parts are drilled and reamed during the assembly process, and then the riveting is done. This method allows the operator to drill the holes aligned. However, the cutting force applied during drilling may cause a similar effect as in the resistance spot welding process.

In this study, the joining process is taken as riveting. However, effects of joining process on assembly variation will be ignored due to the lack of data about this operator-dependent process.

3.2 VARIATION ANALYSIS METHODS

As tolerance is defined as the allowable variation range, to analyze the variation of an assembly, available techniques are not different from tolerance analysis. Main techniques are worst-case analysis, root sum square (RSS) analysis, Monte Carlo simulation and mechanical variation simulation [54].

In general, the relation between the variation of the assembly and the variations of each single part of an assembly having n parts, may be expressed as [38];

$$v = \sum_{i=1}^{n} s_i v_i \tag{3.1}$$

where v and v_i are deviations of the assembly and the i-th part from their nominal values respectively; s_i is the coefficient (weight) of the i-th part in the assembly.

The methodology of determining the coefficients in equation (3.1) is peculiar to the analysis method. Traditional analysis methods for rigid parts (worst case, statistical analysis, Monte Carlo Simulation) consider the geometrical relations and the kinematic constraints of the assembly system.

Considering an assembly consists of rigid parts, the variation of the assembly may be achieved by adding the variations of parts according to the kinematic and geometric relations of the parts constituting the assembly. On the other hand, when these parts are not rigid parts, as in the case of deformable sheet metal parts used in aerospace or automotive industries, calculating the assembly variation by adding up the part variations may give results different than the real production data. The reason is that; beside the rigid body motion, the parts also perform significant deformations during assembly process. Obviously, for the deformable sheet metal parts, these techniques are insufficient since they ignore the deformation of the assembly component.

A previous work [55] predicted “the conventional addition theorem of variance is no longer valid for deformable sheet metal assemblies”. For deformable assembly, part variations are compensated by the deformation of the part, and the assembly variation is dominated by the variations of relatively stiffer parts in the assembly.

The main difference of mechanical variation simulation from those methods is that mechanical variation approach considers the part deformations and mechanics of interaction among the parts when determining the coefficients of deviation of each part.

3.3 FINITE ELEMENT ANALYSIS

One of the possible methods to analyze the variation of compliant assemblies is Finite Element Analysis (FEA). There are several commercial software available, most of which have similar analysis capabilities. One of the possible methodologies for applying FEA in variation analysis is to simulate the assembly steps given in Section 3.1. In this study, such an approach is developed. According to the final distribution of interested nodes, deviation of the assembly from a given nominal may be achieved.

Main disadvantage of using FEA for variation analysis is the long run time required to perform enough number of analyses to obtain a variation distribution for the assembly according to the variation distribution from sources. The essence of the method developed by Liu and Hu [37, 38] for compliant assembly variation analysis is based on this problem. However, for a limited production case where the batch size and production rate are relatively small (for instance 2 air vehicles/month), either the distribution from source or the variation distribution obtained would not mean a lot at the beginning of a project where assembly fixtures are designed and manufactured.

During the design of assembly fixtures for a production explained above, quite valuable data may be achieved about the effect of the variation sources on final assembly variation by direct application of the FEA. For such an application, the problem is to categorize the possible variation sources and to model the complex assembly without excessive calculation effort. In chapter 4, such an approach will be used to model and analyze the design of an example assembly fixture used in aviation industry.

The finite element analysis software used in this study is ABAQUS® 6.6.1. [56]. The software has the capability of modeling the process in several steps where each step corresponds to a step of the real process. The software enables the user to define the contacts occurring between either the workpieces or the workpiece and the fixture during the assembly process. It also allows activating or deactivating these contact definitions during each analysis step, which provides an important advantage. The choices available in contact property definition, allow simulating the joints applied during the process without defining any other Multi-Point Constraints (MPC). Detailed information about the FEA model will be given in Chapter 4.

3.4 ASSEMBLY VARIATION ANALYSIS FOR 1-D BEAM CASE

In this section, an example assembly problem is examined on a 1-D beam case for simplicity. The aim is to validate the modeling strategy for the fastening and clamping methods used in sheet metal assembly fixturing for helicopter components.

Two beams having the same modulus of elasticity (E) and the same moment of inertia (I) are to be assembled as shown in Figure 3.3. EI is constant along the beams. Deviation of parts (in mm) is given by Equations (3.2) and (3.3).

The interested nodes of the beams for which the deviation values will be calculated after assembling, are also shown in Figure 3.4.

The problem is solved by two different methods. First method is analytical calculation of the deflections by classical beam deflection formulation. Second method is the FEA modeling of the case by the similar modeling strategy used in this study in Chapter 4. The solutions are given in Appendix B.

The results obtained from the solutions are given in Figure 3.5. Results predicted that the FEA approach gives quite accurate results and may be used to simulate the variation simulation for the assembly of deformable parts. In addition, the modeling strategy used in this study successfully simulates the assembly steps of such a process.

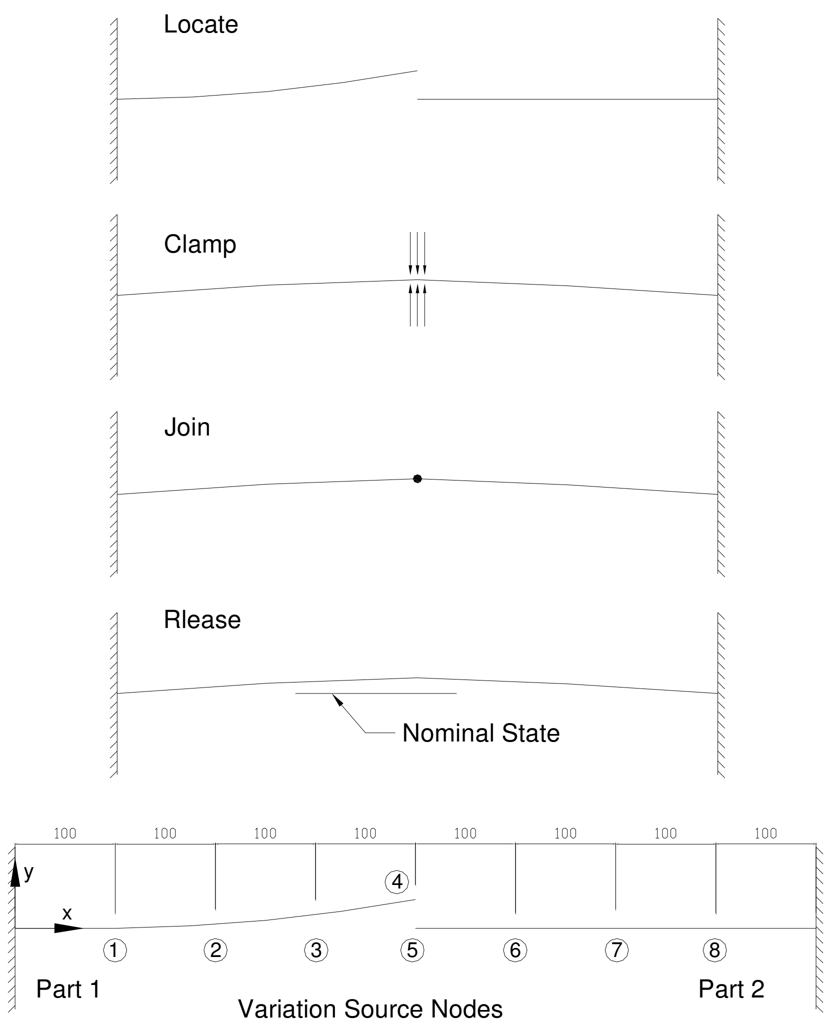


Figure 3.3 Example Problem for Two-Beam Assembly

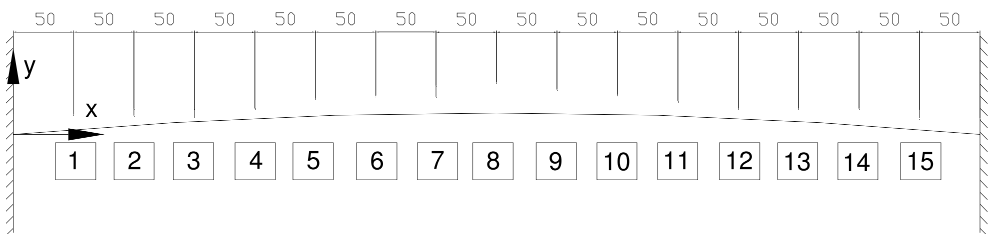


Figure 3.4 Interested Nodes of the Assembly

$$\left.\begin{array}{ll} V_1(x) = \dfrac{(x-100)^2}{90000}, & 400 \geq x \geq 100 \\ V_1(x) = 0 \quad , & 100 > x \end{array}\right\} \tag{3.2}$$

$$V_2(x) = 0 \tag{3.3}$$

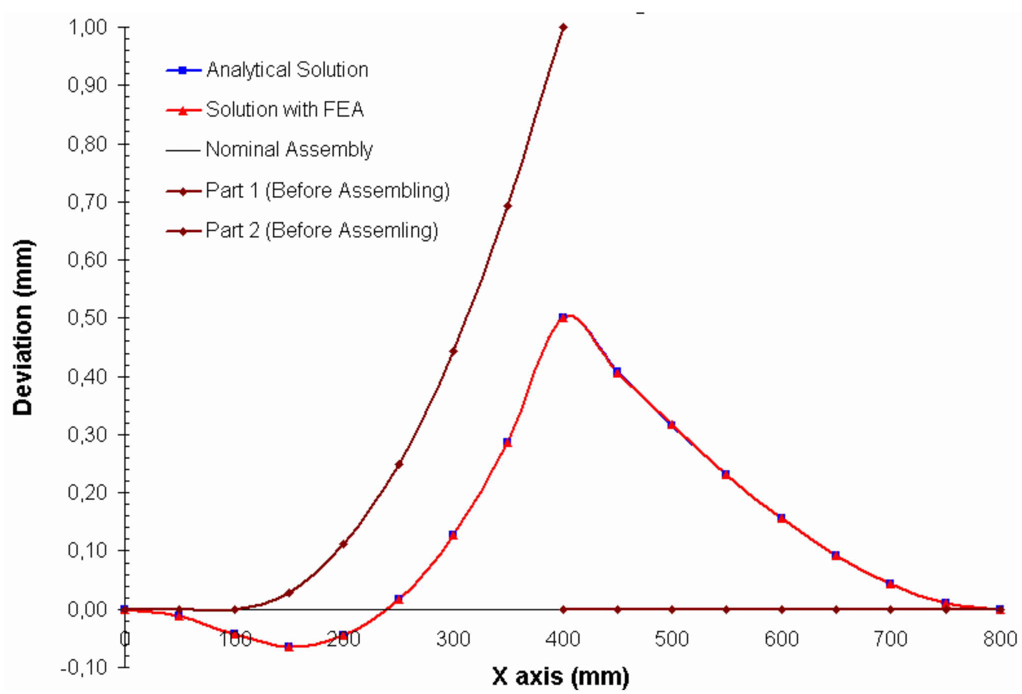


Figure 3.5 Results of 1-D Beam Assembly Problem

CHAPTER 4

DESIGN AND ANALYSIS IN ASSEMBLY FIXTURING OF HELICOPTER COMPONENTS

4.1 GENERAL

In this study, a typical helicopter tail cone structure is considered to form a basis for the design and analysis studies in fixturing of helicopter components. The particular tail cone has similar characteristics and geometry compared to the tail cone of medium weight utility helicopters currently on service for civilian or military use. However, the geometry is simplified for clarity. The geometrical features of the tail cone are given with the drawings in Appendix C.

The tail cone structure is one of the best examples of airframe structures. The fuselage, tail cone or the stabilizers all have similar structural elements as explained in Chapter 1. Beside this, for helicopters with a tail rotor, tail cone structures accommodate a long portion of the power transmission system. For the proper functioning of this power transmission system, generally close tolerances in structural assembling are required. Therefore, considering assembly fixture for a tail cone would be beneficial.

In the following section, a possible fixture design for the tail cone given is explained. Then, the finite element approach to simulate the effect of fixturing and part variations on a complex sheet metal assembly and finite element modeling strategy are introduced. Finally, the results of the analyses are given.

4.2 ASSEMBLY FIXTURE DESIGN FOR THE TAIL CONE

The design of the tail cone assembly fixture is done in accordance with the flow diagram given in Figure 1.7. When the drawings of the tail cone are examined, it may be seen that the datum features are given as forward fitting plane and the fitting holes on this plane. Datum features are shown in Figure 4.1. These features are used to install the tail cone to the fuselage and have vital importance in proper functioning of the power transmission system components are driving the tail rotor. The remaining components of

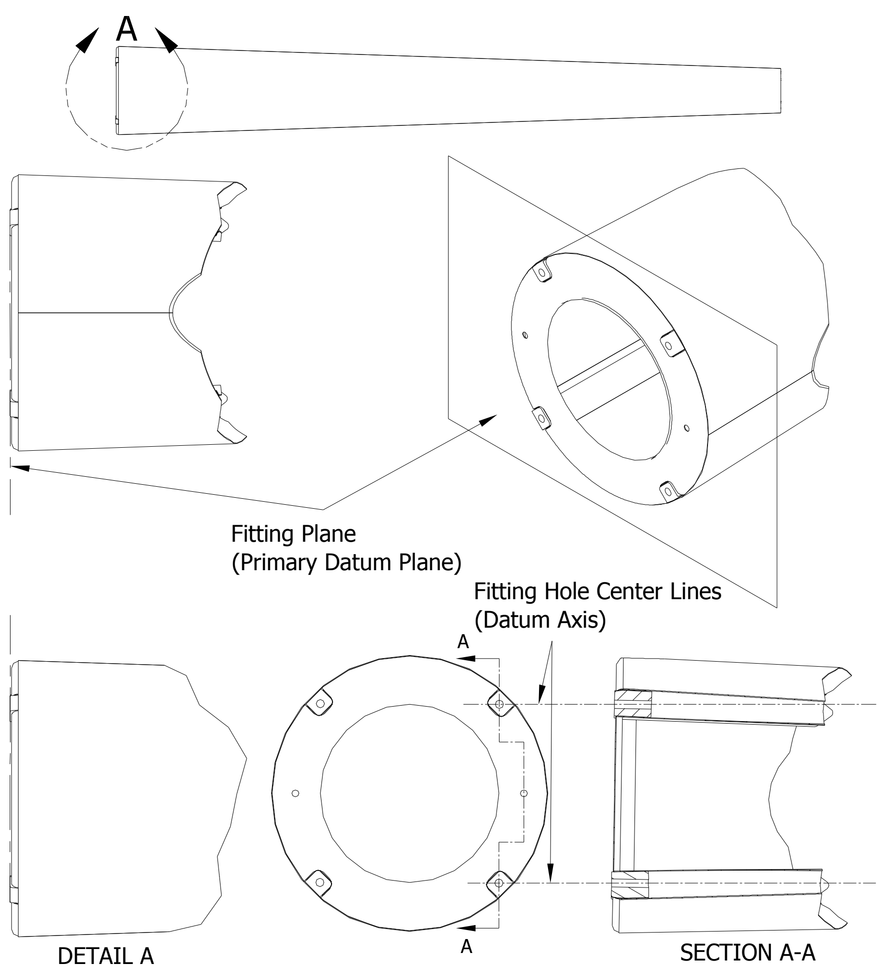


Figure 4.1 Datum Features of the Tail Cone

the tail cone should be located with respect to these features to ensure the success of the assembly. The order of assembling for the assembly components is bulkheads, longerons and surface skin. According to this order, the bulkhead locators of the fixture should be designed first.

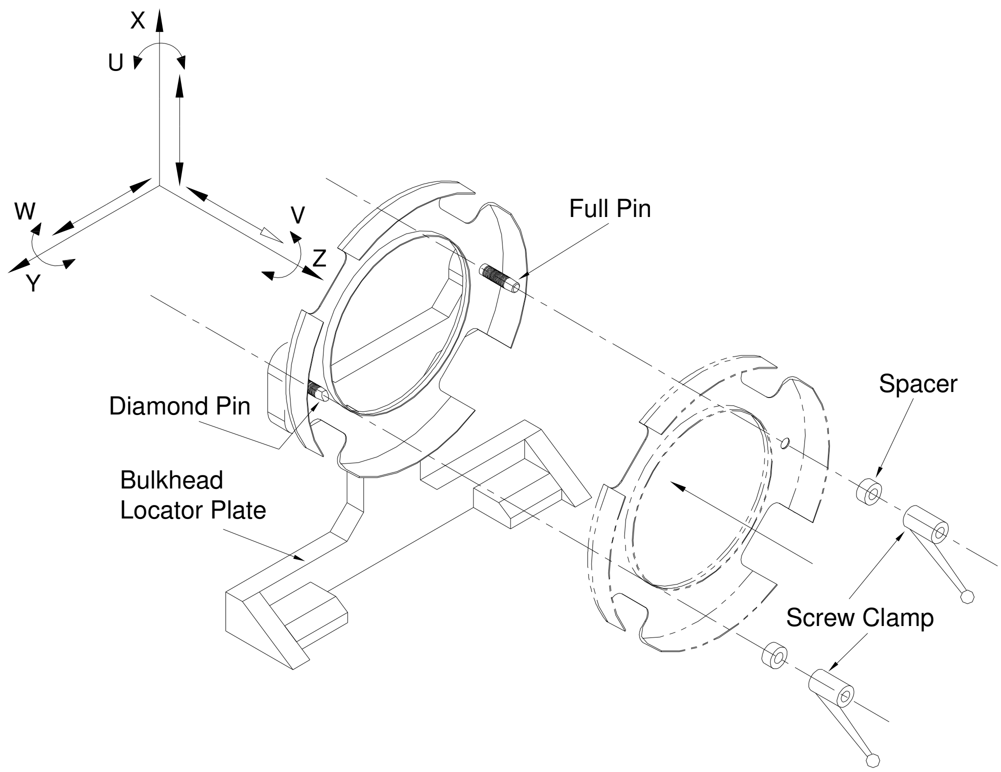


Figure 4.2 Locating and Clamping of Bulkheads

A rectangular coordinate system has been considered during the design of the fixture. The reference frame is shown in Figure 4.2. The longitudinal axis of the tail cone coincides with the Z-axis, where vertical and lateral axes of the tail cone coincide with X and Y-axes, respectively.

As shown in Figure C.1, tail cone assembly has four bulkheads to be located during assembly process. Four bulkhead locator plates are required to place these four bulkheads in the position of Z 2, Z 1000, Z 2000 and Z 3000, respectively, according to the coordinate frame given in Figure 4.1. Among these locator plates, the forward plate is designed in such a way that it has the required features to locate the four fittings.

Primary datum surface for each bulkhead locating is the surface of the corresponding locator plate, which restrains the bulkhead in five directions out of twelve directions (i.e, -Z, +U, -U, +W, -W) according to the locating principle described in Section 2.1. Two pins perpendicular to the primary surface of the locator plate are applied through the “tooling holes” of the bulkhead as seen in Figure 4.2. One of these pins is a full pin and the other is a diamond pin. The full pin restrains the bulkhead in four additional directions (i.e., +X, -X, +Y, -Y) and the diamond pin restrains another two directions (i.e., +V, -V). The remaining free direction is required to locate the part. Once the bulkhead is placed by means of the locator plate assembly, it is secured by a spacer and a screw clamp installed to each locator pins. By the way, the remaining free direction is restrained by clamping. The “full” arrowheads show the constrained directions before clamping and the “blank” arrowhead shows the direction restrained by clamping in Figure 4.2.

This procedure is applied for all of the four bulkheads by means of the designed bulkhead locator plates. The drawing of the designed locator plates for this case is given in Appendix D.

Once the fittings and the bulkheads are located to the assembly fixture, the longerons are placed by means of the flanges of the bulkheads, the fittings and the forward plate. The flanges of the bulkhead restrain the longerons in five directions. The fittings restrain another three directions and forward plate restrains one direction. By means of the remaining three directions,

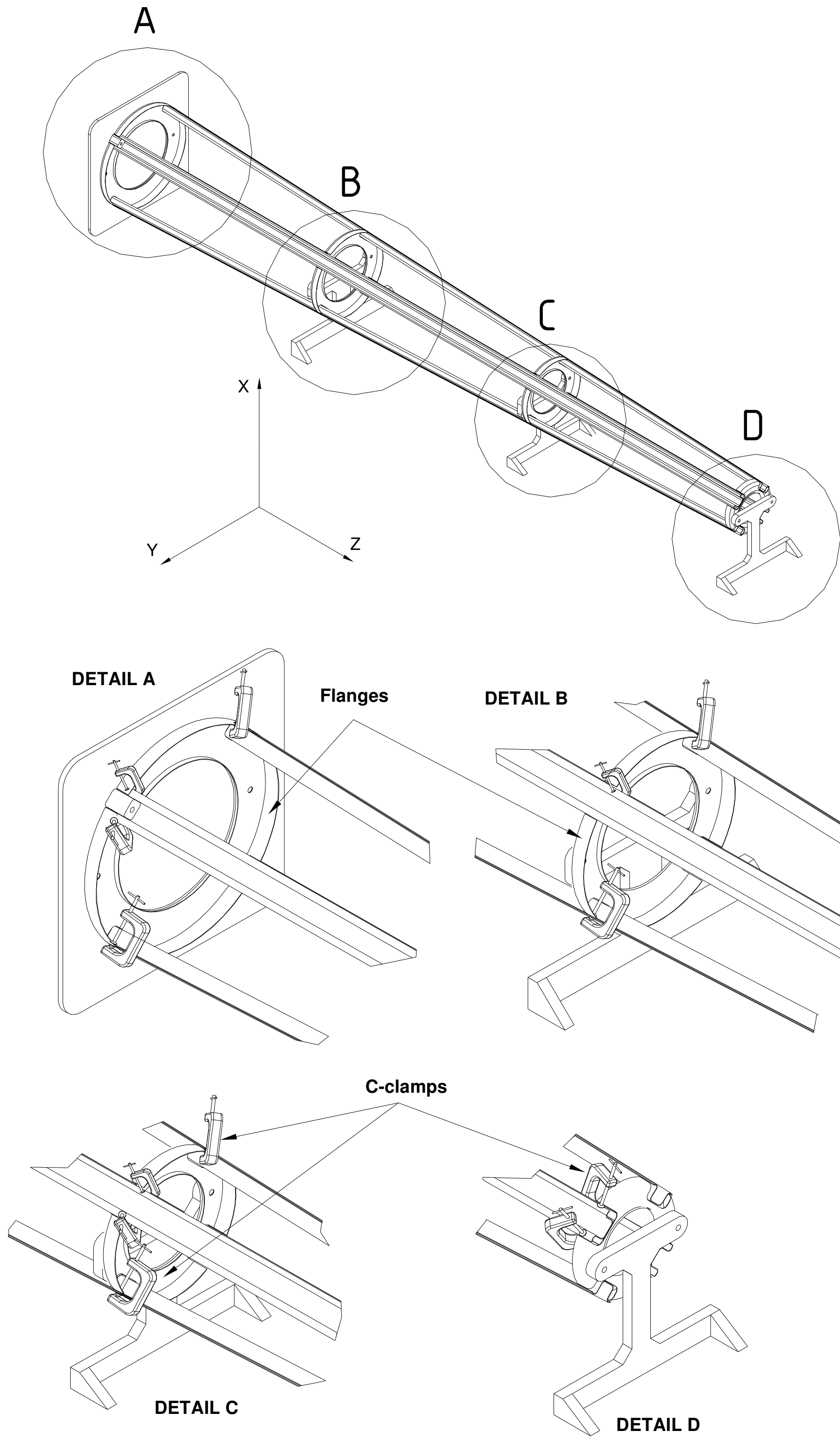


Figure 4.3 Illustration of Locating and Clamping of Longerons

longerons are located. They are secured by means of sheet holder clamps applied to bulkhead flanges and longerons, joining them temporarily. Figure 4.3 gives the placement and clamping of a longeron. Obviously, the locating procedure of the longerons utilizes the features provided by other assembling components. This means that the success of placement of longerons depends on both the success of previously done bulkhead location and the dimensional and geometric variations of bulkheads. As a result, the tolerance accumulation (build-up) is inevitable.

After the bulkheads and longerons are properly located and clamped, they need to be joined by riveting. When that is done, the frame of the structure is ready for the assembling of the pre-formed skin. For placement the sheet metal skin of the structure, bulkhead flanges, longerons and forward locating plate are utilized. Figure 4.4 illustrates the locating scheme of the skin. Previously located part features are used to locate the skin as in the case of longerons which means the tolerance stack-up propagates by the addition of the skin to the assembly.

Clamping of the skin surface is different than clamping of the other components. To eliminate the risk of undesired deformations of the skin, clamping is performed consecutively from one end to the other end of the skin. Cleco fasteners are applied one by one to the drilled pilot holes, temporarily fastening the skin to the frame. No additional clamping mechanisms are used except for the cleco fasteners.

A base structure for the fixture is needed to hold the locator and clamping devices. In design of such a base, the critical point is that it should provide the required surfaces for the fixturing elements while it is rigid enough to prevent the undesired deformations that may occur during the assembly process. The most common strategy is using the knowledge coming from the previous successful base designs, which is also the methodology used

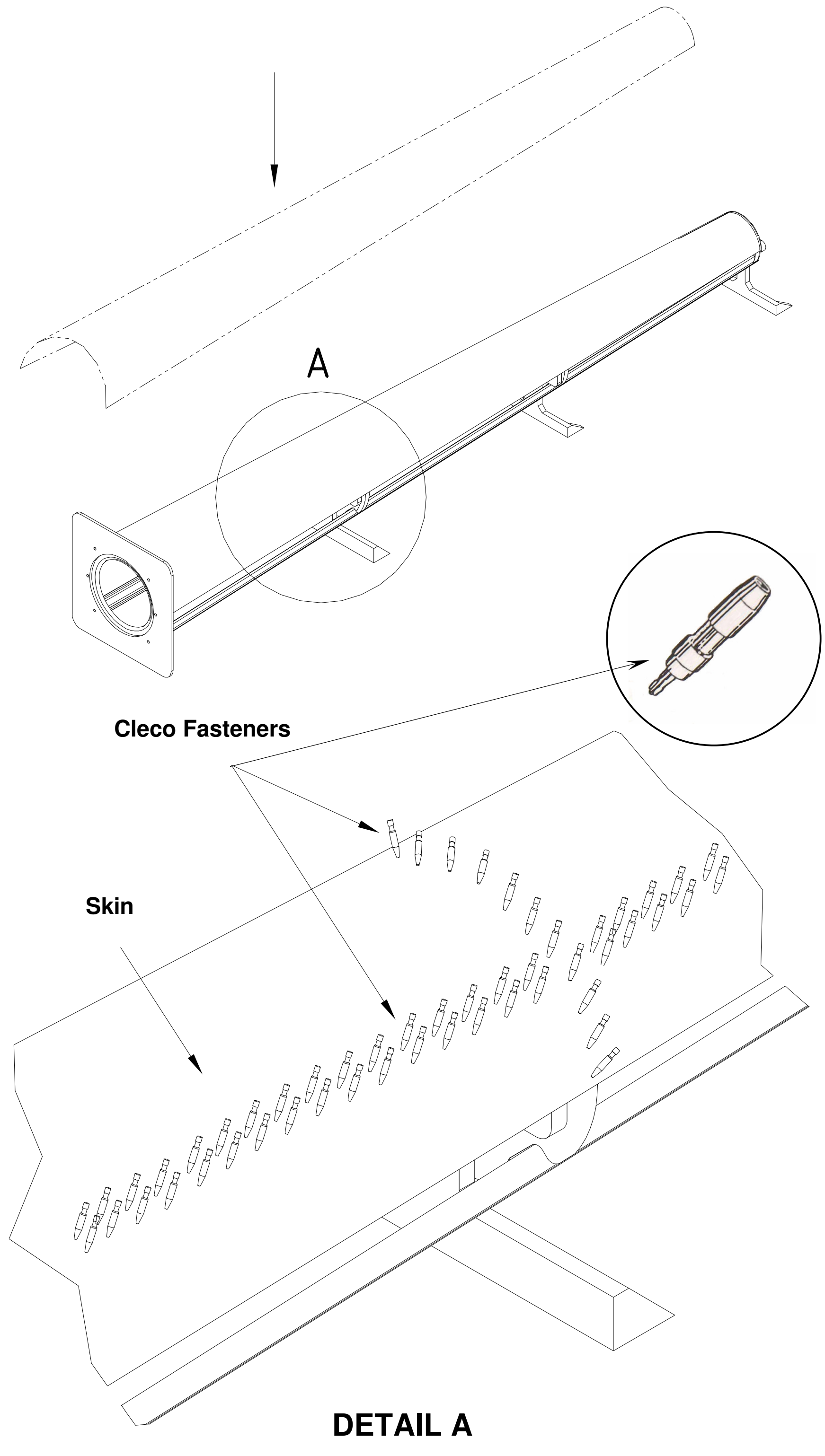


Figure 4.4 Illustration of Skin placement and temporary Fastening

in this study. Studies concerning the optimal base design problem may be done, however it is beyond the scope of this thesis.

The complete drawings of the designed fixture is given in Appendix D. In the following sections, the finite element approach to analyze the effect of possible fixture variations on product quality will be discussed.

4.3 GENERAL CONSIDERATIONS OF ANALYSIS FOR VARIATION SIMULATION

It is important to know if the designed assembly fixture will satisfy the product quality requirements or not. One of methods is waiting for the feedbacks after the completion of the first product. However, this may cause expensive rework on the fixture, delays, and costs.

Alternatively, analyses may be done to simulate the possible dimensional and geometric variation of the important features on the product after assembling. If the variations coming from fixturing and parts themselves can be properly correlated with the assembly variation by means of analyses for refining the fixture design, this may significantly lower the build-up of variations in the final product and decrease the cost.

In this section, usage of finite element analysis is introduced for prediction of assembly variation due to possible fixture and part variations.

The tail cone-fixture dual for the analyses was given in the previous section of this chapter. It is important to state the interested features of the product and the critical features of the fixture and parts clearly, in order to get the correlations between them. In this section these features are introduced.

For the product, the important feature is the projection of the tail rotor drive shaft centerline (PSCL) on the tail cone skin, which is given by Figure 4.5. This feature is controlled by the given angularity and straightness tolerances

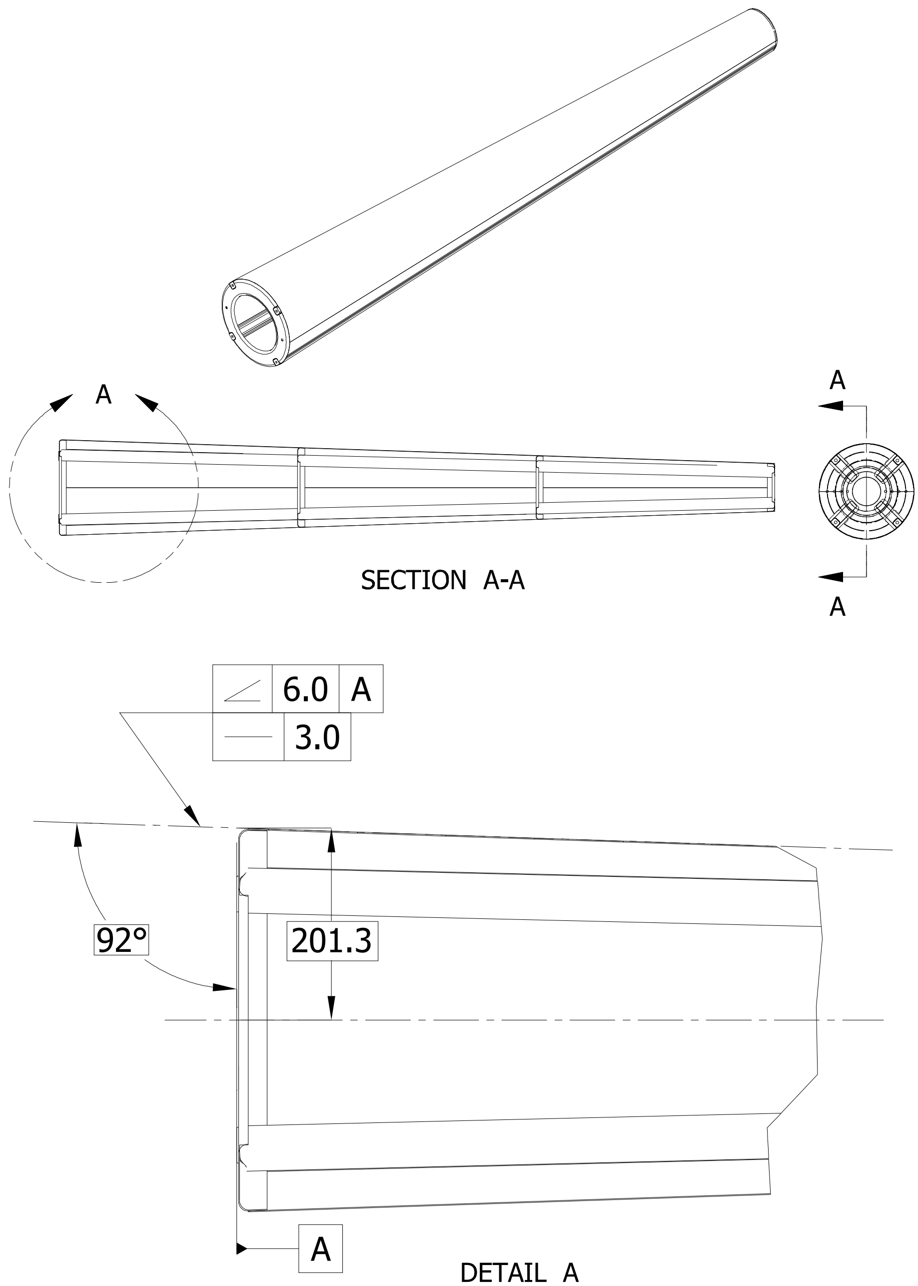


Figure 4.5 Projected Tail Rotor Shaft Centerline (PSCL)

in order to achieve a successful alignment of the tail rotor drive shaft after completion of the tail cone assembly. This feature will be referred as PSCL for rest of the thesis.

The possible sources of variations that may affect on PSCL deviation have been decided as;

a) the translational variations of the bulkhead locator plates with respect to the forward plate, given in Figure 4.6 (a),
b) the rotational variations of the bulkhead locator plates with respect to the forward plate, given in Figure 4.6 (b),
c) the translational variations in position of the locator pins with respect to the holes of fittings, given in Figure 4.6 (c),
d) possible geometric variations in the bulkheads features, given in Figure 4.7. (a),
e) possible geometric variations in the longerons features, given in Figure 4.7 (b).

It should be noted that among the variation sources given above in items a,b and c are fixture features, where d and e are part features.

When all of the possible feature variations stated above are considered, it is obvious that there are numerous cases. In a real assembly process, several of these variations may occur at the same time, which means the combinations of variations also need to be considered. Of course, it is not possible to predict all of the possible cases and their combinations within this study. Instead, typical feature variations are selected to give an idea about the effect of possible variations.

The possible feature variations stated above are applied to the analysis model for the third bulkhead of the assembly. Table 4.1 gives the deviated features and the deviation values simulated in analyses. The results of the

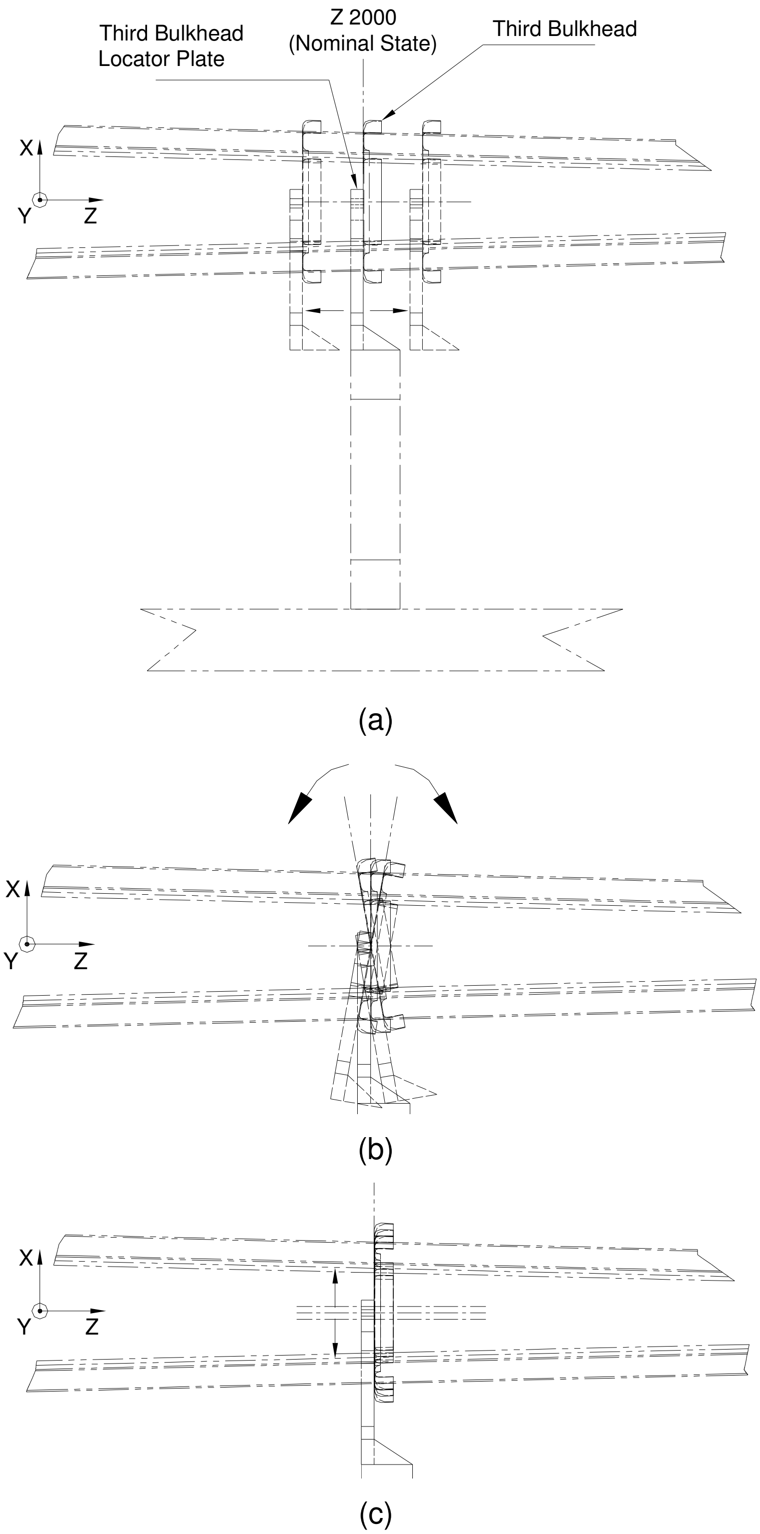
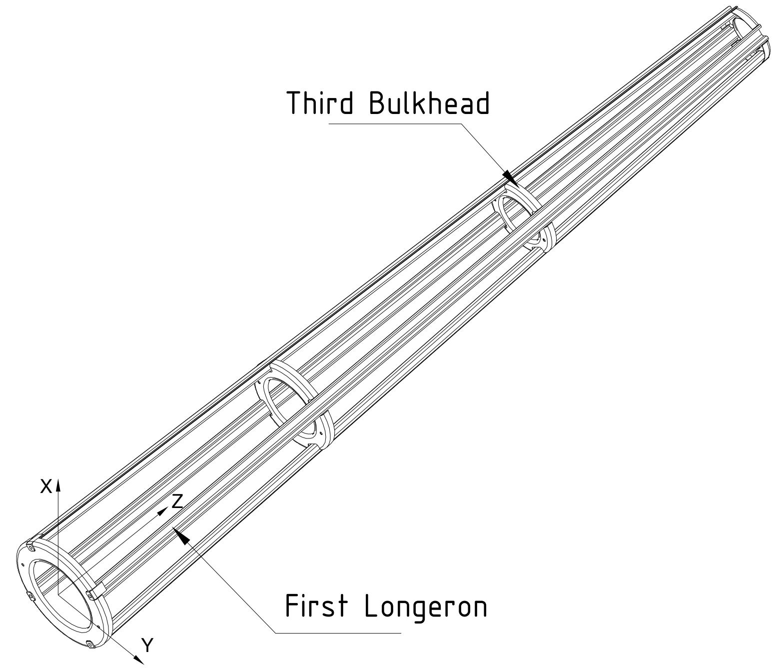
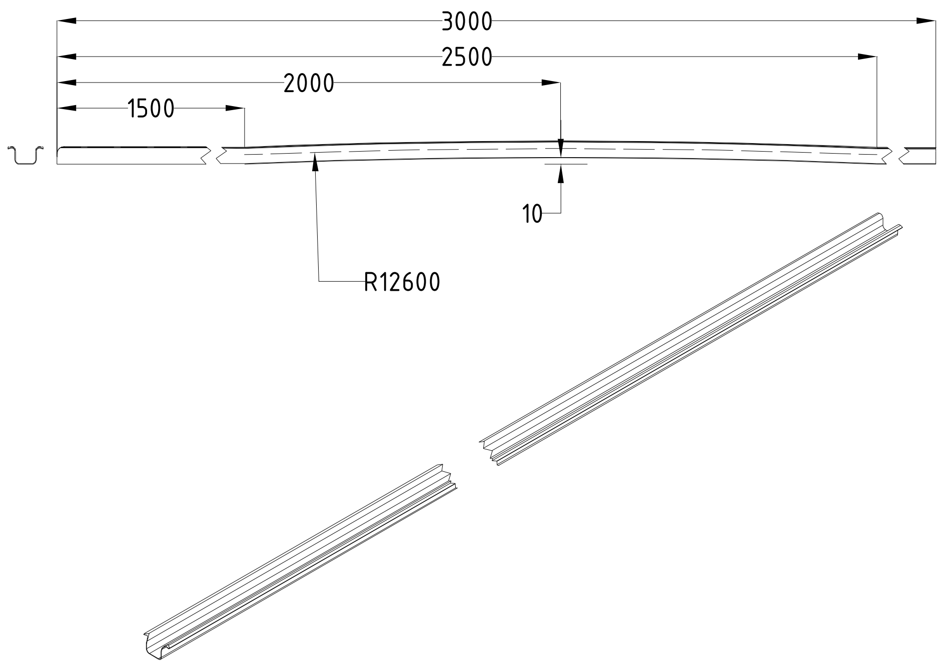


Figure 4.6 Possible Deviations on the Third Bulkhead

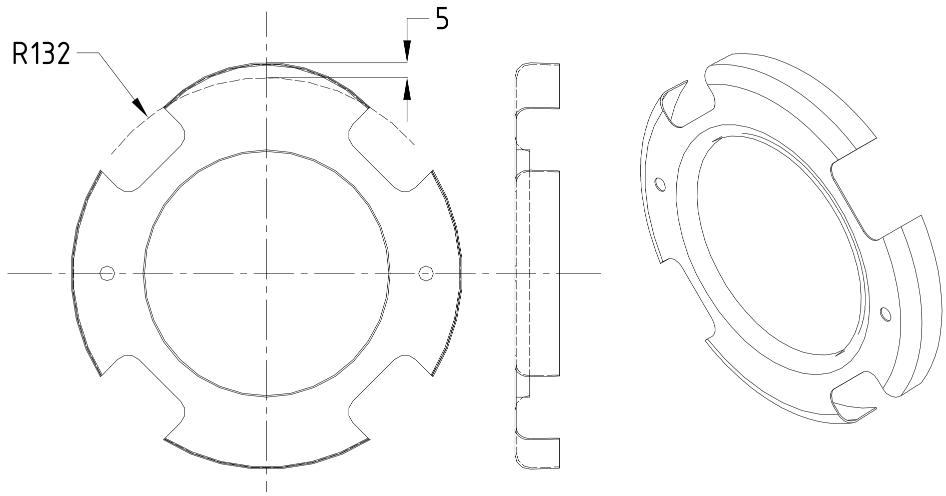


(a)



(b)

Figure 4.7 Possible Deviations on the Longeron and Bulkhead



(c)

Figure 4.7 Possible Deviations on the Longeron and Bulkhead (continued)

analyses are given in the following section to predict the effect of the variations of the selected features on PSCL. Before continuing to the analysis results, the modelling strategy for FEA and the assumptions for analyses should be discussed.

4.4 FINITE ELEMENT MODELING STRATEGY FOR ASSEMBLING PROCESS OF THE TAIL CONE

Usage of FEA for the variation simulation of an assembling process may be possible with the current commercial software. However, several problems exist. These problems may be summarized as;

a) Difficulty in simulating the steps of assembly process by means of a commercial software (for example; addition of any assembling member during the analysis),
b) Difficulty in analyzing the global model consisting of large parts as too much calculation effort and time may be required,

Table 4.1 Content of Analyses

Feature Variation	Deviation Values used in Analyses for Related Features of the Tail Cone Structure
Translation of the third bulkhead locator plate in Z-axis wrt to the forward locator plate (given in Figure 4.6)	-10, -5, 0, +5, +10 mm deviated from nominal state
Rotation of the third bulkhead in Y-axis wrt the forward locator plate (given in Figure 4.6)	-10, -5, 0, +5, +10 degrees rotated from nominal state
Translation of the locator pins on the third bulkhead locator plate in X-axis wrt the forward locator plate (given in Figure 4.6)	-5, -2, 0, +2, +5 mm deviated from nominal state
Variation of the first longeron (given in Figure 4.7)	Maximum of +5, +10 mm deviated from nominal dimensions (Maximum occurs at Z2000)
Variation of the third bulkhead (given in Figure 4.7)	Maximum of +2, +5 mm deviated from nominal bulkhead dimensions (Maximum occurs at point where projection of X-axis intersects the bulkhead flange)
Gravitational Effect	No deviation case +10 mm translation of the third bulkhead locator plate in Z +2 mm translation of the third bulkhead locator plate in X

c) Existence of high number of contact surfaces which are sources of non-linearity and convergence problems.

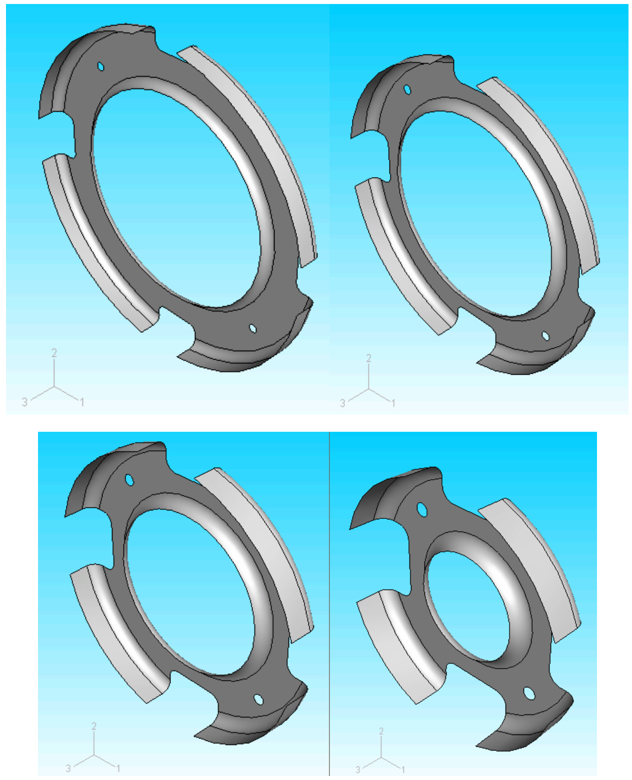
The CAD model of the assembly is created by means of AUTODESK INVENTOR® 9 [57] and imported to the FEA software using the import tools within the ABAQUS® 6.6.1. Figure 4.8 demonstrates these models.

The parts are transferred as shell parts and mid-plane approach is used, which means the representative surface is passing from neutral axis of the parts. The thicknesses of the parts are taken into account during the creation of the assembly model and defining the part interactions.

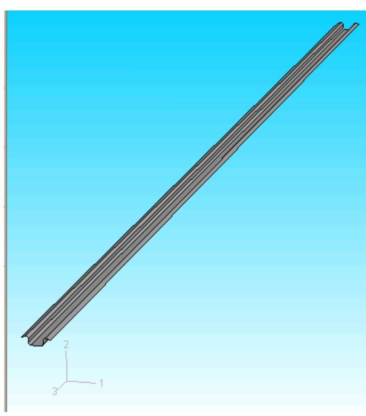
All the parts within the model have a thickness value of 1.3 mm and their material properties are defined for the aluminum alloy 2024-T3. The corresponding values of modulus of elasticity, poisson ratio and density for Al-2024-T3 are used as 72.39 GPa, 0.33, 2768 kg/m^3 respectively [5]. The assignment of material properties is illustrated in Figure 4.9.

Once the parts are modeled and material properties are defined, parts need to be "assembled" in the assembly module of ABAQUS® 6.6.1. At this stage, part thicknesses and, if exist, the deviations should taken into consideration. The "assembled" model is shown in Figure 4.10.

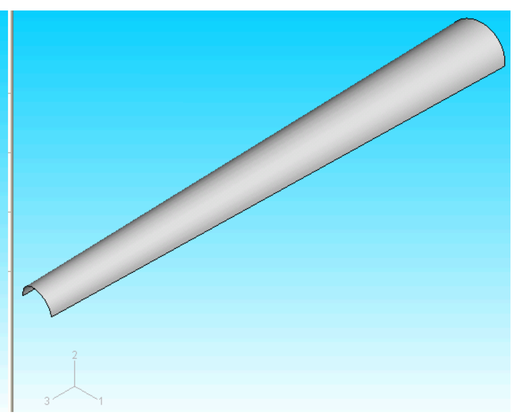
In order to simulate the assembly process, several steps are created which are corresponding to the steps in an assembly operation. For the case, total of eight steps are required to model the complete process of assembling the bulkheads, longerons, and upper skin. These step definitions are illustrated in Figure 4.11. Step 1 simulates the locating and clamping of longerons. In Step 2, longerons are fastened. In Step 3, the skin surface is located on the frame. In steps 4 and 5, it is clamped and fastened, respectively. In the remaining steps, 6-8, the assembly is released by deactivating constraints.



(a) Bulkheads



(b) Longeron



(c) Upper Skin

Figure 4.8 CAD Models for Parts

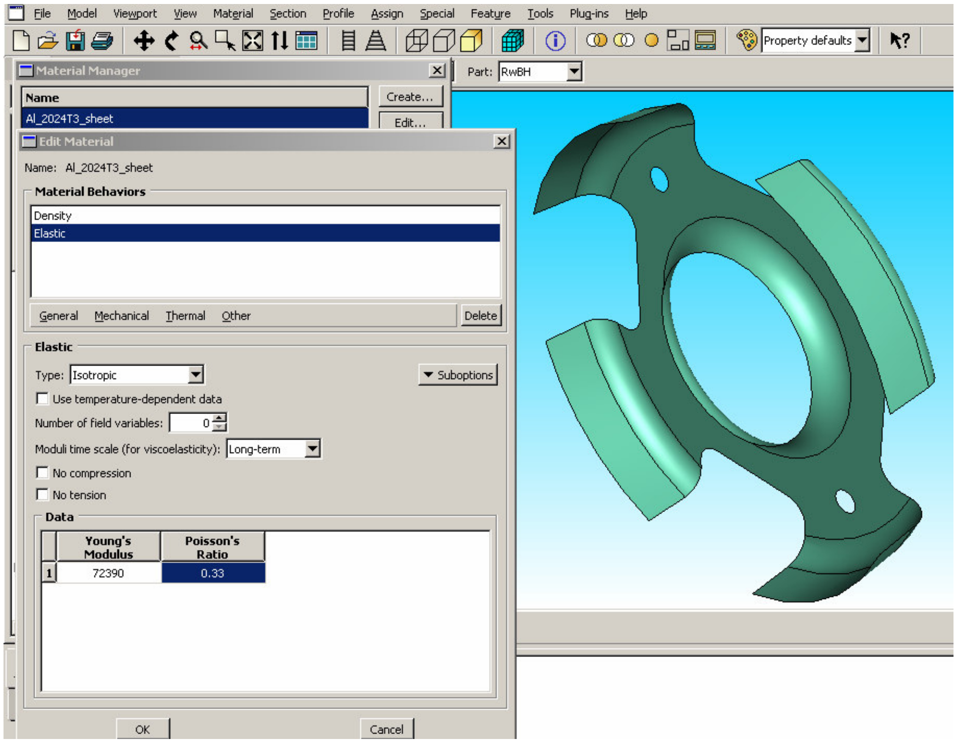


Figure 4.9 Assigning the Material Properties

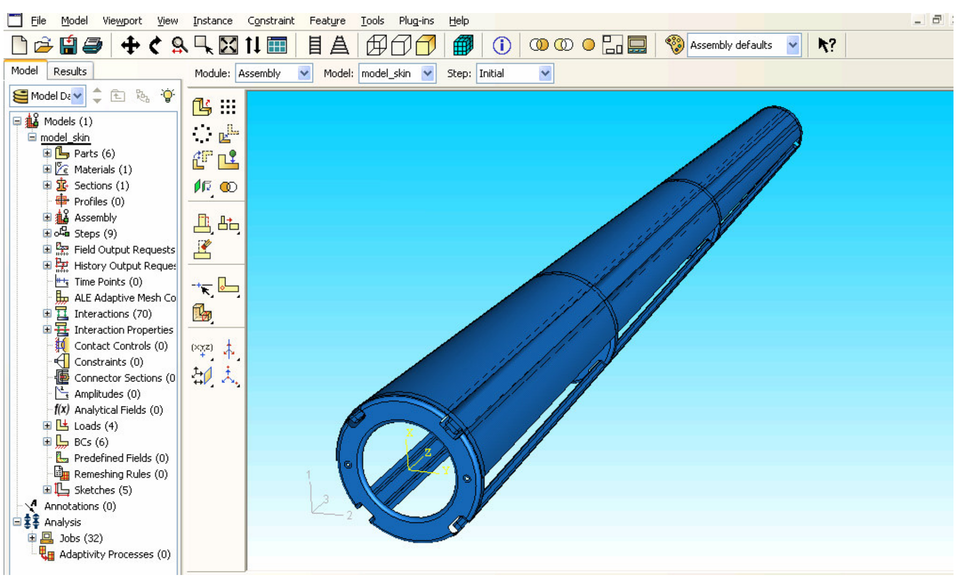


Figure 4.10 “Assembled” Model for Analysis

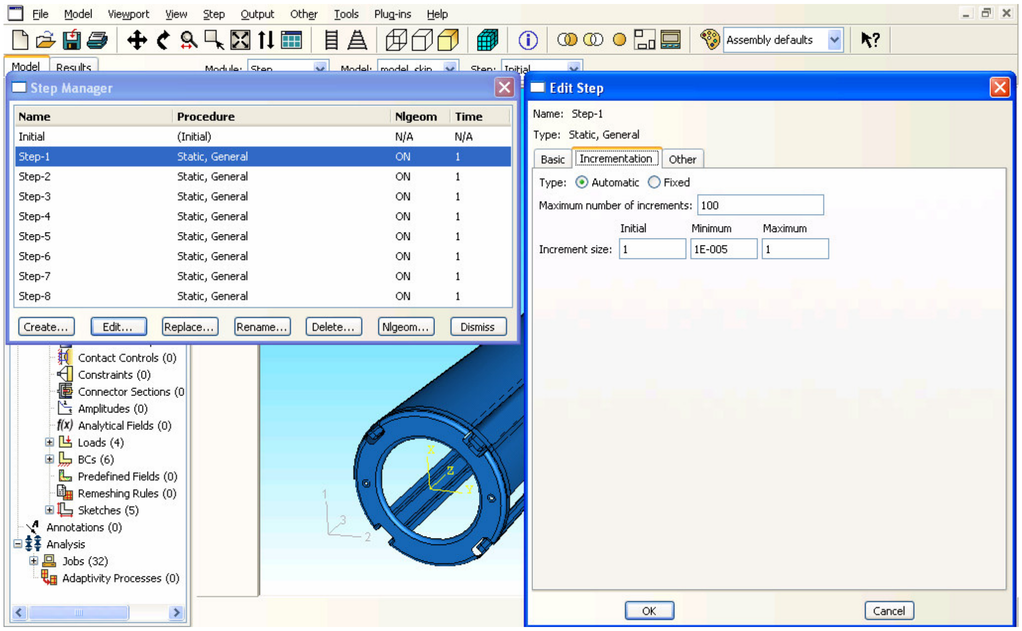


Figure 4.11 Defining the Steps of the Analysis

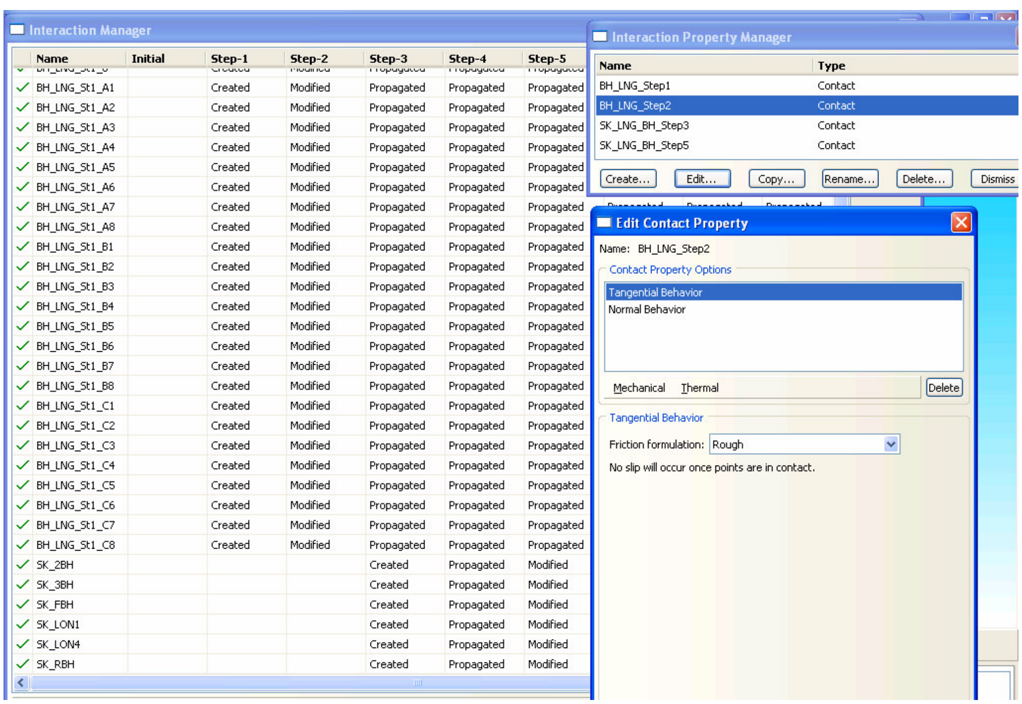


Figure 4.12 Defining the Interactions

Some interaction property definitions are needed to govern interactions of mating surfaces. Throughout the model two type of interaction properties are defined. First interaction property is defined as frictionless hard contact and this property is used until the fastening step of the assembly. For the fastening step and following steps, a second interaction property type is defined as no-slip (rough), no-separation hard contact. This property simulates the fastened joints. Alternatively, the commercial FEA software has “fastener” definitions based on Multi-Point Constraints (MPC’s) to simulate the spot welds or rivets which already exist on a structure. In ABAQUS® 6.6.1, this tool is also available, however it is not controllable through the steps of the analysis. For this reason, they are not suitable for assembly process and the aim is achieved by modificating the contact properties of interactions through the analysis steps.

For the interactions of the mating surfaces of the assembly, slave-master duals should also be identified. In the selection of master surfaces, the hierarchical order is taken as longerons, bulkheads and skin. The criterion is that the surface which has relatively high stiffness is defined as the master surface.

The contact type is defined as “small sliding, surface-to-surface contact” and adjustment is allowed only to remove overclosures without creating residual stresses. The contact definitions are shown in Figure 4.12.

The restraint applied by the fixture to the workpieces are modelled as boundary conditions. For example; the screw clamps applied to bulkheads are modelled by restraining all the freedoms of a selected circular region around the pin holes on each bulkhead. Other boundary conditions are applied according to the assembly process requirements and sequence. Displacement boundary conditions are used to force a deviated feature to its nominal state, which is in practice done by the process operator.

The temporary fasteners; C-clamps and cleco fasteners are modelled as opposing uniform pressure distibutions over the corresponding areas of mating surfaces. These force definitions are required to ensure the contact of the components to achieve a stable fastening state. Once the parts are fastened, then the forces and related boundary conditions are deactivated by means of the controls available within the software. For the contact regions which were already at their nominal state initially, small pressure distributions are applied. For bulkhead-longeron contact regions, a pressure of 0.05 Mpa and for skin contact regions a pressure of 0.005 Mpa are adequate to ensure that contact occurs between corresponding regions. Once the parts are in contact, they are not allowed to separate during the other steps of the analysis, as they would be fastened. For that reason, no more clamping forces are required once mating surfaces are successfully in contact. The clamping force and boundary condition definitions are illustrated in Figure 4.13.

The mesh of the model is done with an average mesh seed of 20 mm for bulkheads and skin, and with a seed of 10mm for longerons. Meshing yielded 25126 elements. The number of elements are distributed as; 460 elements on forward bulkhead, 349 elements on second bulkhead, 344 elements on third bulkhead, 273 elements on rear bulkhead, 5100 elements on each longerons and 3300 elements on skin surface. The element type is shell element, named as “S4R” in ABAQUS® 6.6.1, which means shell elements with four nodes and reduced integration. Figure 4.14 and Figure 4.15 give the meshed model and the element type definitions, respectively.

It should be noted that an assembly model is a global model by its nature. In order to succeed in modeling of an assembly with an acceptable calculational load some details need to be sacrificed. If local results are important, then submodeling analyses should be applied for those local regions.

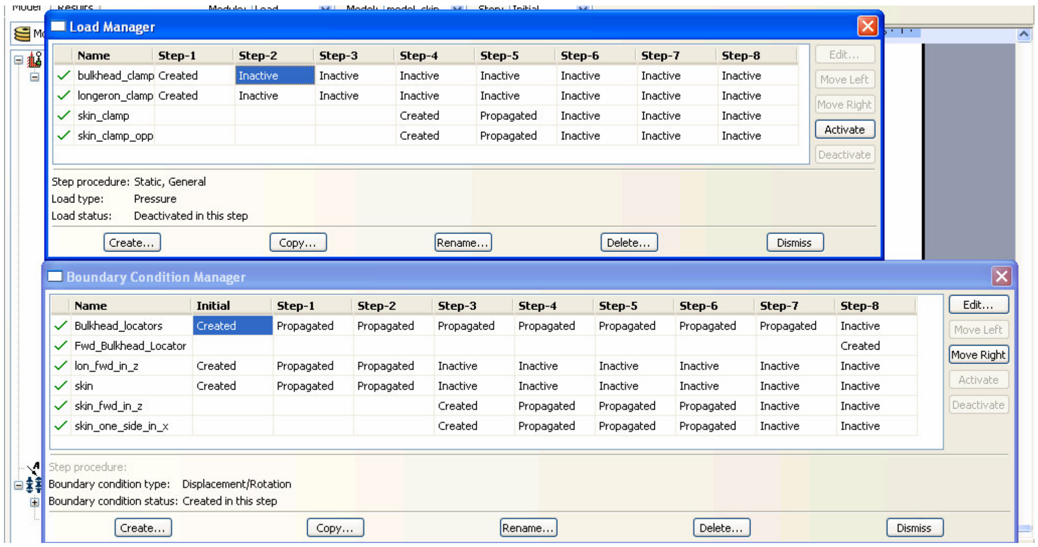


Figure 4.13 Force and Boundary Condition Definitions

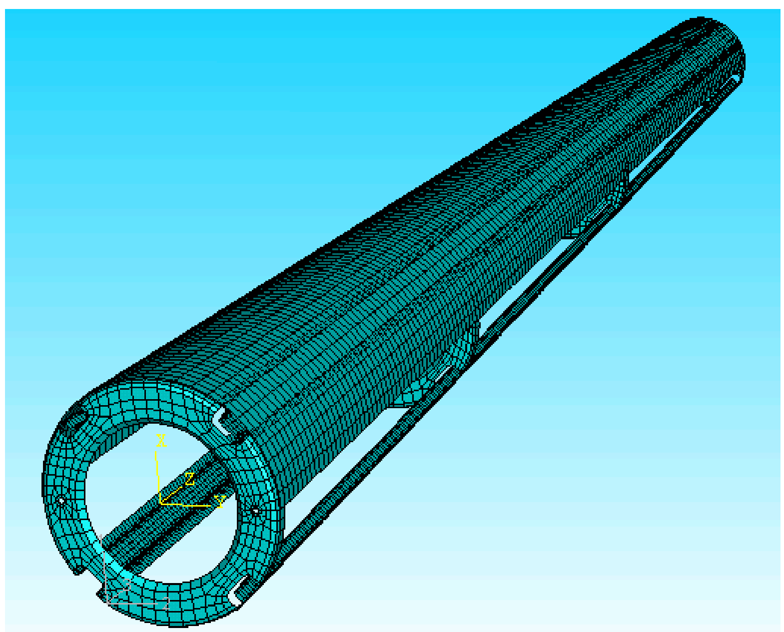


Figure 4.14 Meshed Model

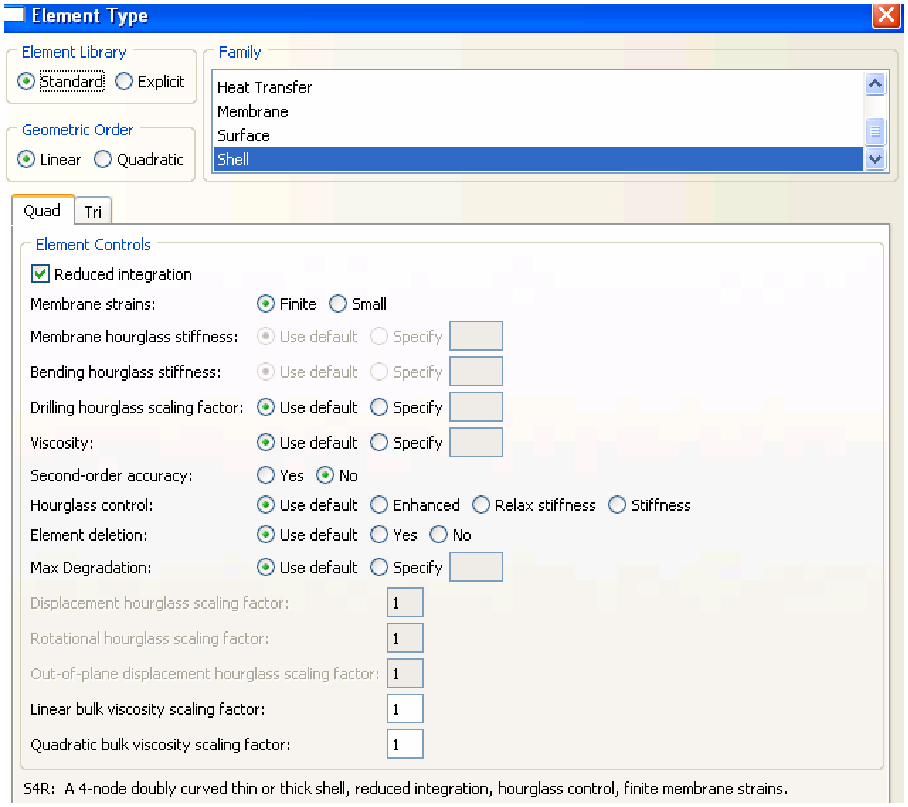


Figure 4.15 Element Type Definitions

The assumptions and restrictions for the analysis models are given below:

a) The materials are assumed to be isotropic and the deformations are in linear elastic range.
b) The dimension and geometric form of the parts are assumed to be in their hypotetically perfect state unless a deviation is specified.
c) Except for the last group of analyses where gravitational effects are investigated as given in Section 4.5.6, the effect of gravity is ignored.
d) The clamping forces and boundary displacements are assumed to be applied perfectly as ramp functions along the step in which they are initially activated. The effect of the clamping force application sequence within a single step is ignored.

e) The joining process is assumed to be performed perfectly, causing no deviation or stress concentrations during the assembling and other possible variations due to joining process are neglected.

4.5 RESULTS

4.5.1 Translation Deviation of the Third Bulkhead Locator Plate

The primary surface of a bulkhead locator plate is the geometric feature which determines the location of the bulkhead in Z direction. A deviation in the position of this plane with respect to the forward plane is directly transmitted to the related bulkhead. A series of analyses is performed in order to predict the effect of possible deviations of bulkhead locator plate position. The third bulkhead locator plate has been considered during the analyses. Five different analysis models are constructed. In these models, the third bulkhead locator plate is translated in Z direction -10, -5, 0, +5, +10 mm with respect to forward plate, respectively.

The deformations occuring during the analysis steps in X direction for +10 mm locator plate deviation in Z are given in Figure 4.16. The final position of the PSCL in X direction is achieved from the final deformed states of the nodes on the PSCL. The difference between the final position and the nominal state of nodes on PSCL gives the deviation of PSCL nodes. These deviation values are demonstrated in Figure 4.17. It should be noted that the statement “No source deviation” refers to the the analysis model where there is no deviated part, i.e., all parts are in their nominal states.

The analyses predict that resulting deviations of PSCL in X direction are much more smaller compared to the input bulkhead locator plate deviations. According to the results, it can be concluded that during the design of the fixture, the tolerances governing the bulkhead locator plate position in Z direction need not to be too close.

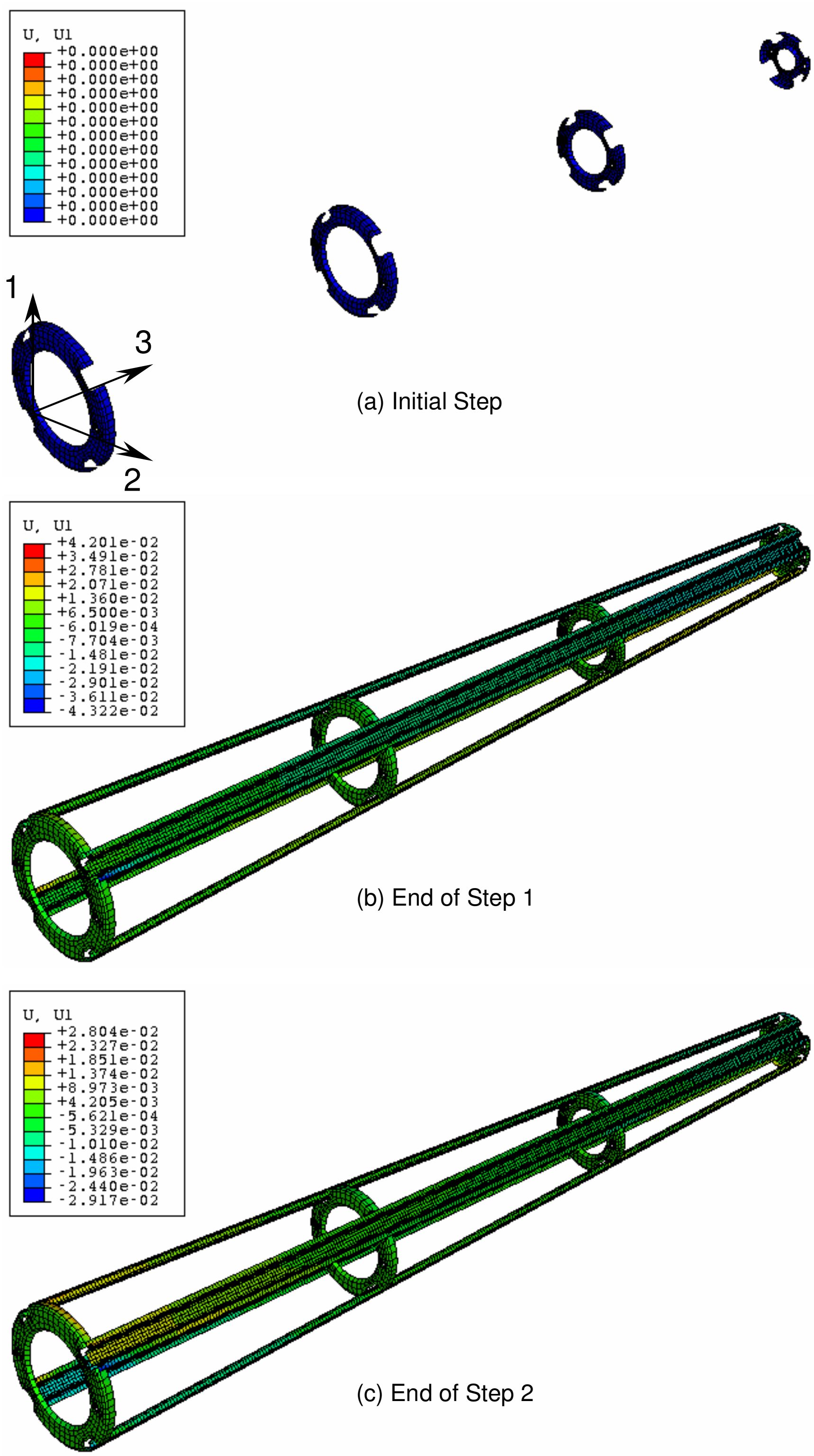


Figure 4.16 Deformations in X direction at the end of each Steps

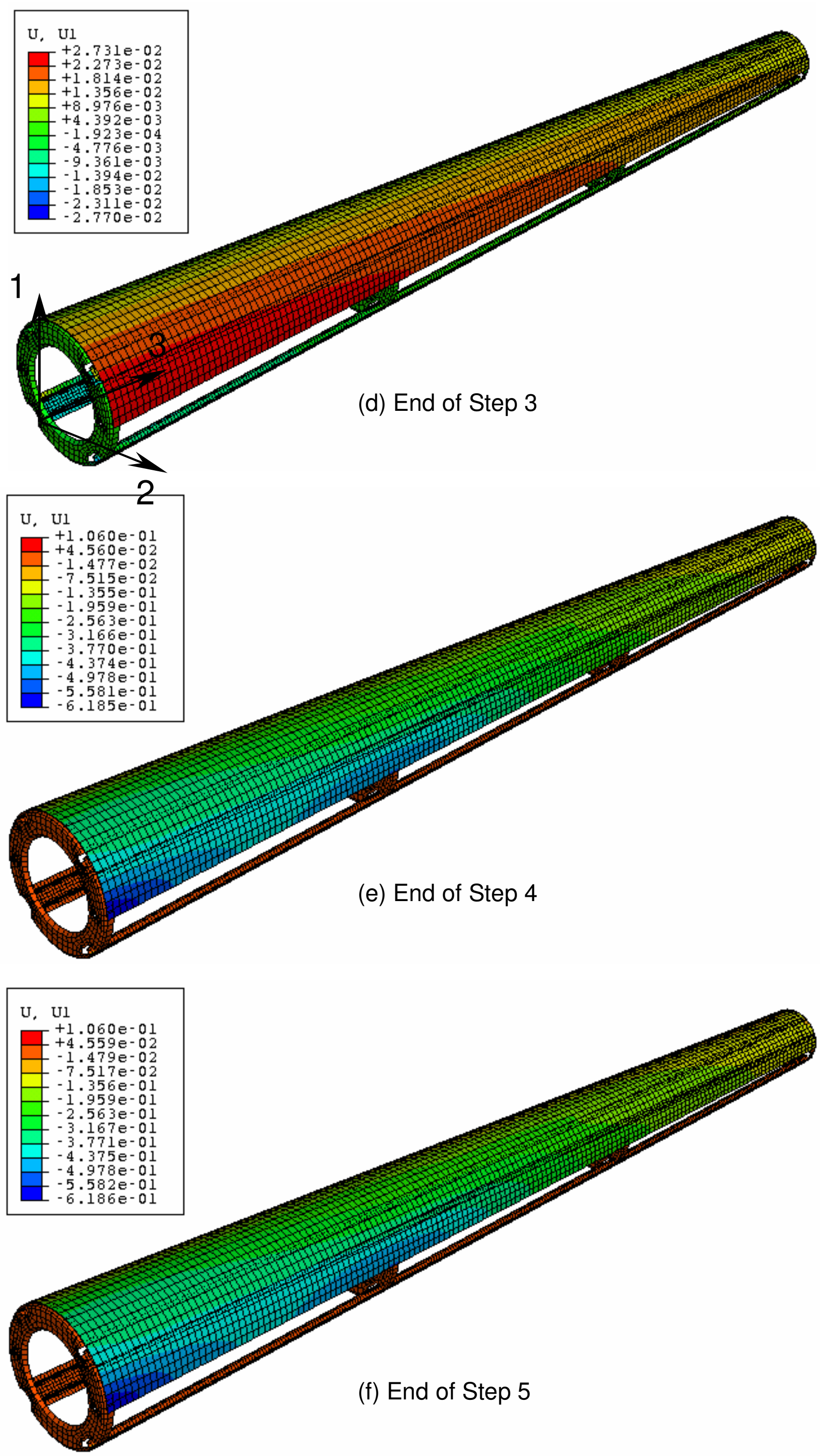
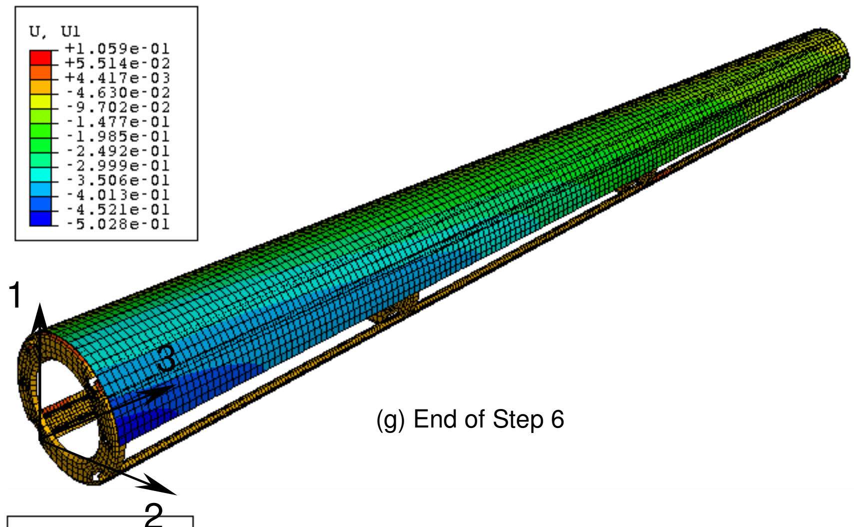
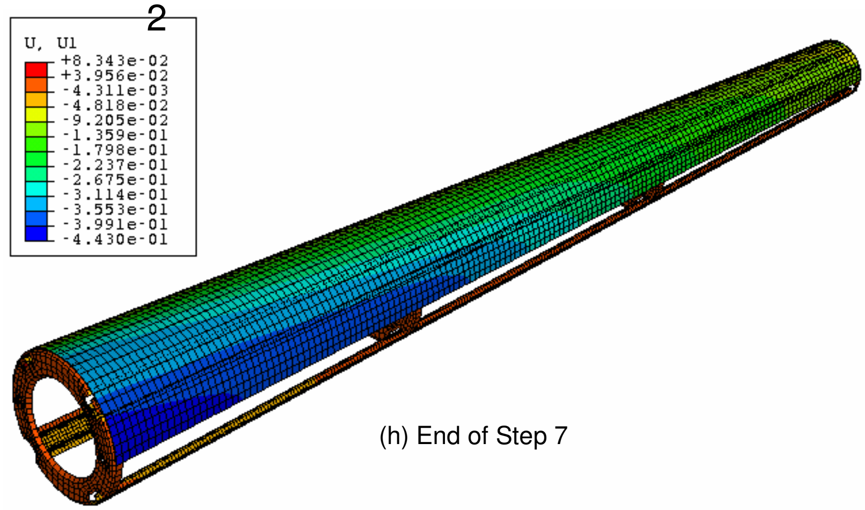


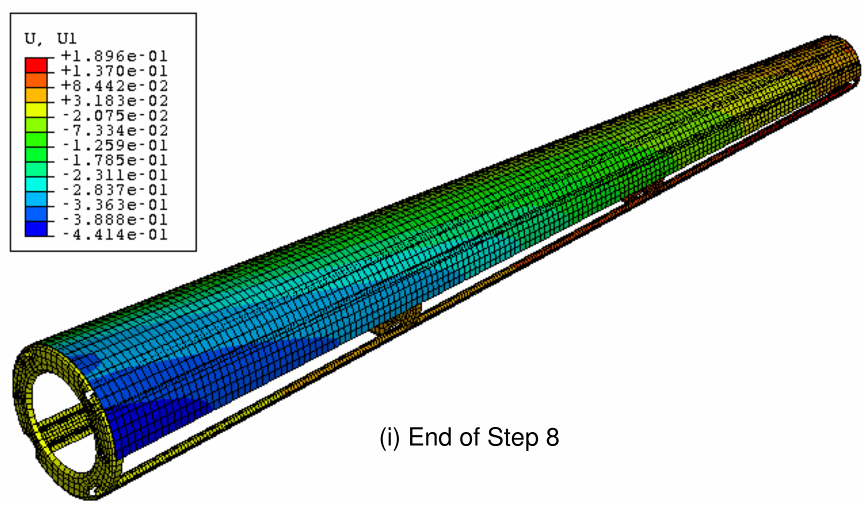
Figure 4.16 Deformations in X direction at the end of each Steps (continued)



(g) End of Step 6



(h) End of Step 7



(i) End of Step 8

Figure 4.16 Deformations in X direction at the end of each Steps (continued)

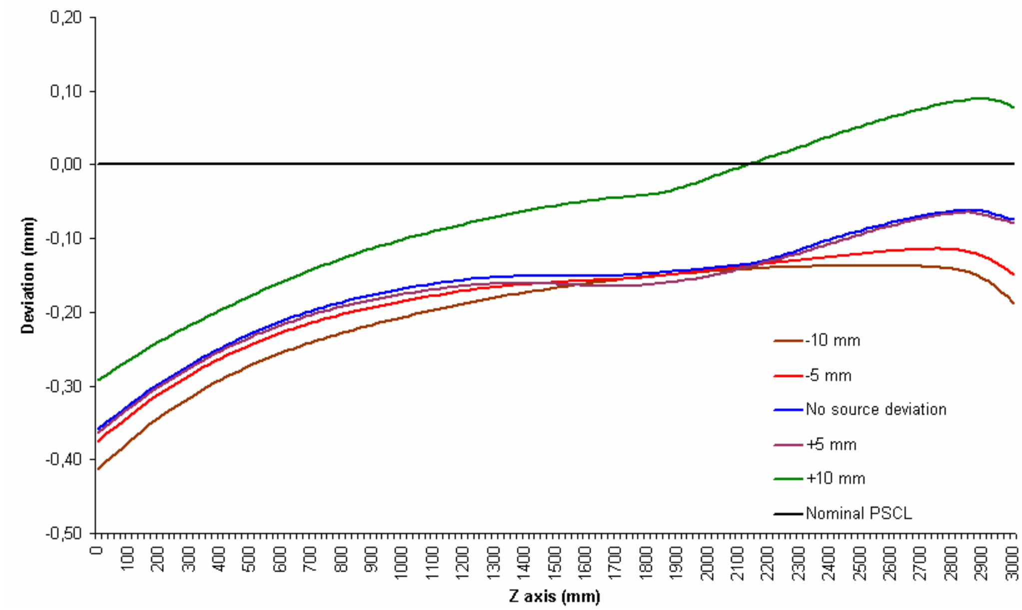


Figure 4.17 Deviation of the PSCL Due to the Translational Deviation of the Third Bulkhead Locator Plate in Z-axis

4.5.2 Rotational Deviation of the Third Bulkhead Locator Plate

Beside the position of primary surface of a bulkhead locator plate in Z direction, the parallelism of this surface may also have an effect on the final product. In order to analyze this effect, five discrete models are created. Third bulkhead locator plate is taken to apply the deviations and it is rotated with respect to the forward plate -10, -5, 0, +5, +10 degrees, respectively. The rotation axis is the line passing from the centers of the locator pins on the pirimary surface and this line is parallel to Y-axis.

The deviation of the PSCL for each case is demonstrated in Figure 4.18. The analyses results predict that rotational deviation of the primary surface of the bulkhead locator plate have relatively small effect on the PSCL node deviations. The arrangement of the parallelism requirement for a locator plate is a time consuming process. The rotational deviations in a margin of

± 10 degrees yield a maximum deviation of 0.5 mm in PSCL nodes. Actually, such rotational deviations are too high when compared with the real assembly fixturing applications in aviation industry and in general possible deviaitons are much less. According to the results, it can be concluded that parallelism tolerances do not need to be tight for bulkhead locator plates.

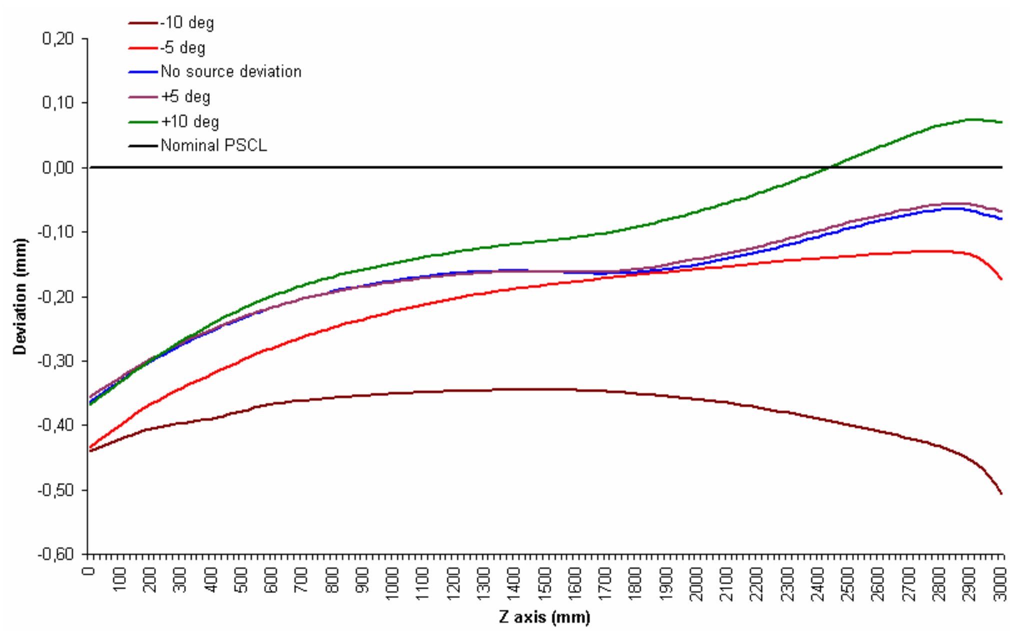


Figure 4.18 Deviation of the PSCL Due to the Rotational Deviation of to the Third Bulkhead Locator Plate

4.5.3 Deviation of the Third Bulkhead Locator Pins

The locator pins on a bulkhead locator plate are the fixture elements which determine the position of the corresponding bulkhead in X direction. Possible deviations in the position of these pins may result in a significant deviation of the skin surface. This effect is analyzed by means of five discrete models created. The pins are deviated -5, -2, 0, +2, +5 mm from their nominal state, respectively. The deviation of the PSCL in X direction is shown in Figure 4.19.

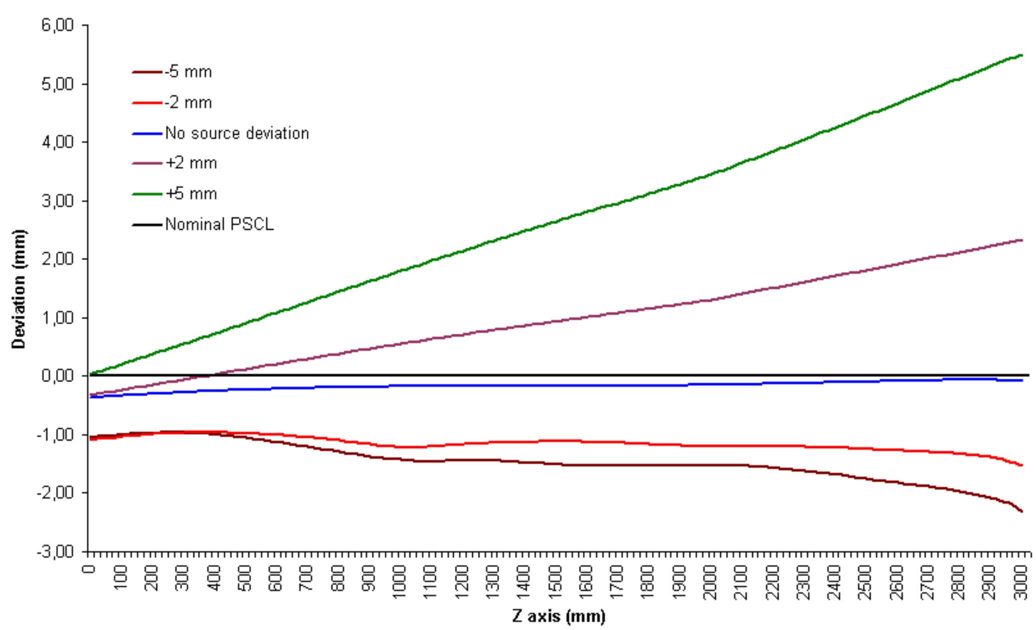


Figure 4.19 Deviation of the PSCL Due to the Translational Deviation of the Bulkhead Locator Pins in X-axis

The results predict that any deviation in position of locator pins in X direction has significant effect on the PSCL deviation. This means that the locator pin position should be controlled by means of relatively close tolerances to succeed in keeping the PSCL deviation in limits.

4.5.4 Longeron Deviation

Deviation of a longeron feature is one of the possible type of part variation for airframe structure. Especially, the deviation of the longeron in directions normal to its longitudunal axis may result in quite significant variations.

Two models are constructed to analyze the effect of a deviated longeron including three identical longerons and a fourth deviated longeron. The deviation scheme of the longeron is given by Figure 4.7. The maximum deviation occurs at Z 2000. The analyzed deviation values are 5 mm and 10 mm. The position of the PSCL is shown in Figure 4.20.

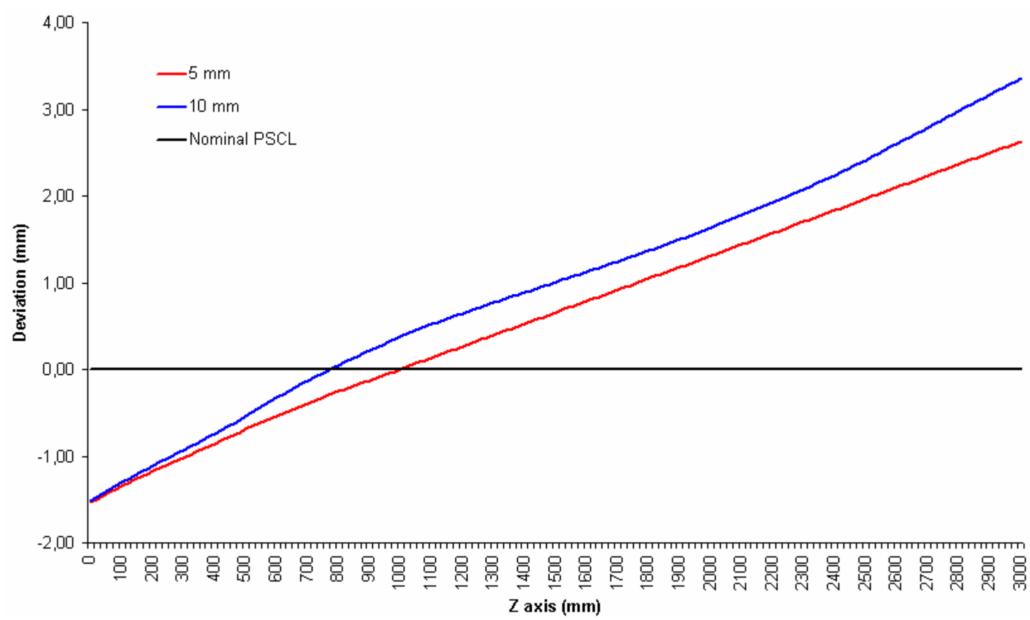


Figure 4.20 Deviation of the PSCL Due to the First Longeron Deviation

The results predict that the deviation of a single longeron may have quite significant effect on PSCL deviation. It should be noted that longeron variation comes originally from the manufacturing process of the longeron. In other words, it is an inevitable input variation from assembly fixturing point. Analyzing the effect of a part deviation does not directly give any idea about the fixture element design. However, there is still some inferences. In case of inevitable and relatively “large” part variations, the designer should avoid close tolerances as it would mean nothing but additional cost.

4.5.5 Bulkhead Flange Deviation

Beside its possible positional deviations, which have been discussed in the previous sections, a bulkhead may have an effect on the final assembly due to its own deviated features. Especially, the effect of deviation of a bulkhead flange may be significant since it provides the surfaces used to locate the longerons and the skin. Two models are created in order to analyze the effect of possible bulkhead flange deviations. The deviation scheme is shown in Figure 4.7 and the maximum deviation values for these two models are 2 mm and 5 mm for the flange of the third bulkhead. The position of PSCL is demonstrated by Figure 4.21. The results predict that flange variation has a similar effect on PSCL as the locator pin deviation.

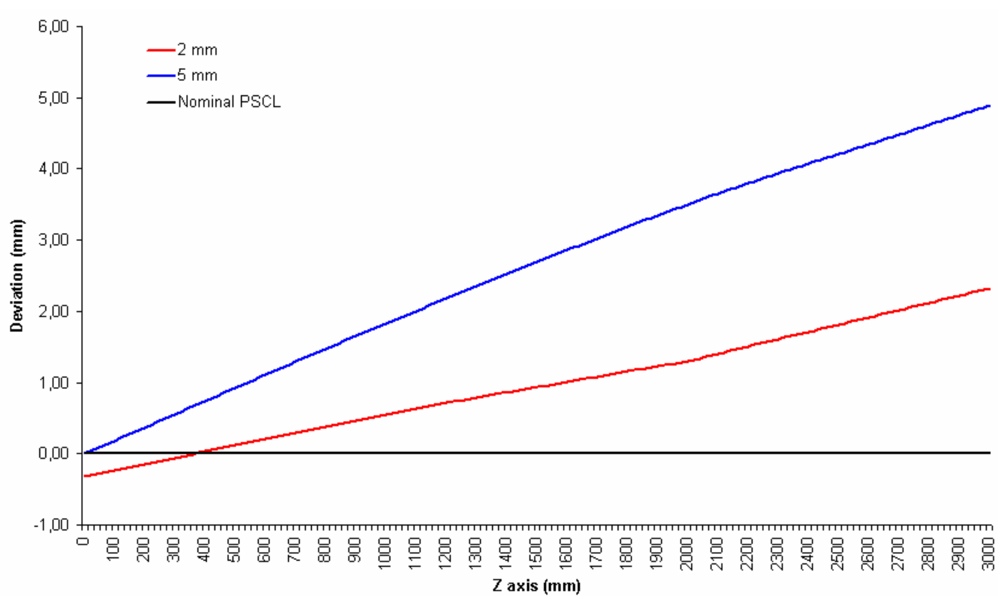
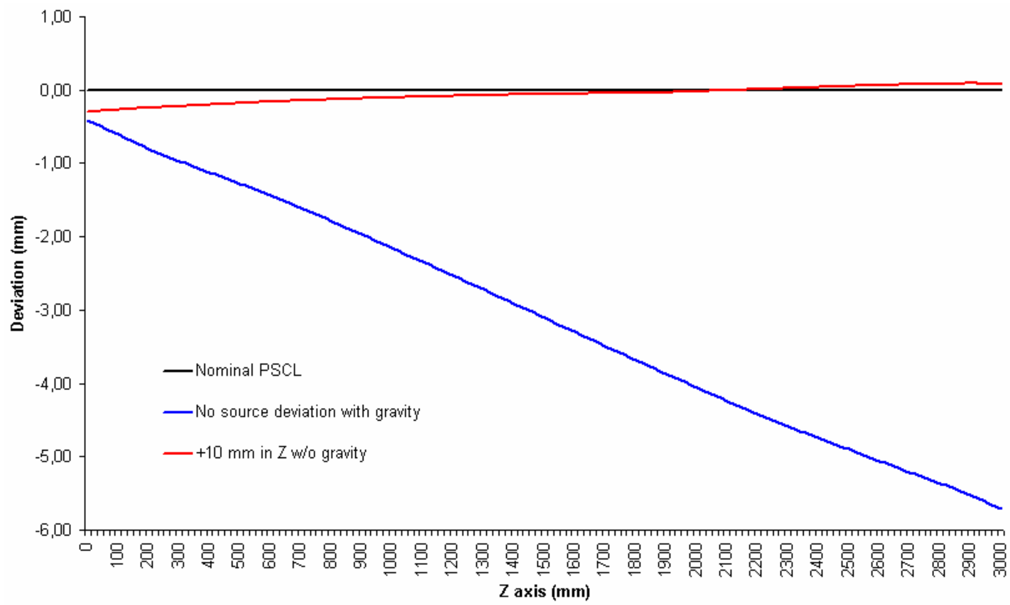


Figure 4.21 Deviation of the PSCL Due to the Flange Deviation of The Third Bulkhead

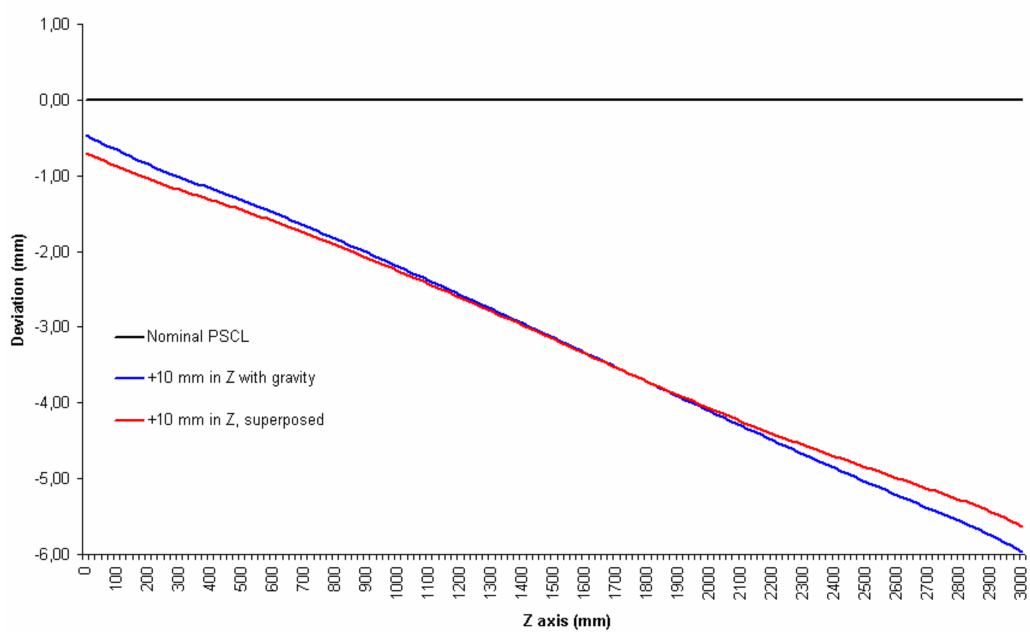
4.5.6 The Gravitational Effect

Gravitational force should also be taken into consideration as it may cause significant deformations which may result in deviation of the critical features. In order to predict this effect, gravitational force is added to three discrete models and calculations are performed. These models are "No source deviation", where all parts are in their nominal states as explained in Section 4.5.1, "+10 mm translation in Z-axis" and "+2 mm translation in X-axis". These cases have been individually studied in previous sections.

Analyses predict that gravity has some quite significant effect on the final product and should be considered during the design of assembly fixture. If not, there probably occurs a significant deviation when clamps are opened and the product is released. Analyses also predict that the effect of gravity may be obtained by superposing the related results. For instance, the gravity-included deviation of the PSCL nodes for "+10 mm translation in Z-axis" case, can be obtained by superposing the gravity-ignored result of this case and the gravity-included results of "No source deviation" case. For any case, the gravity-included results may be obtained similarly. Figure 4.22 and Figure 4.23 compare the results obtained by superposing with the results obtained by directly including the gravity into the models. In Figure 4.22 (a), resulting PSCL deviations in X-axis are given for two cases. One of the cases is +10 mm translational deviation of the third bulkhead locator plate in Z-axis and, in this case, gravity is not taken into consideration. In the other case, no deviation exists but the gravity is included. In Figure 4.22 (b), gravity-included results for +10 mm deviation in Z are given. One of the curves is obtained by adding the gravitational force to the analysis model and second curve is obtained by superposition of the two curves in Figure 4.22 (a). In Figure 4.23, a similar comparison is done for +2 mm translational deviation of the third bulkhead locator pins in X-axis. As shown in the figures, superposing gives satisfactory results with only little loss of accuracy.

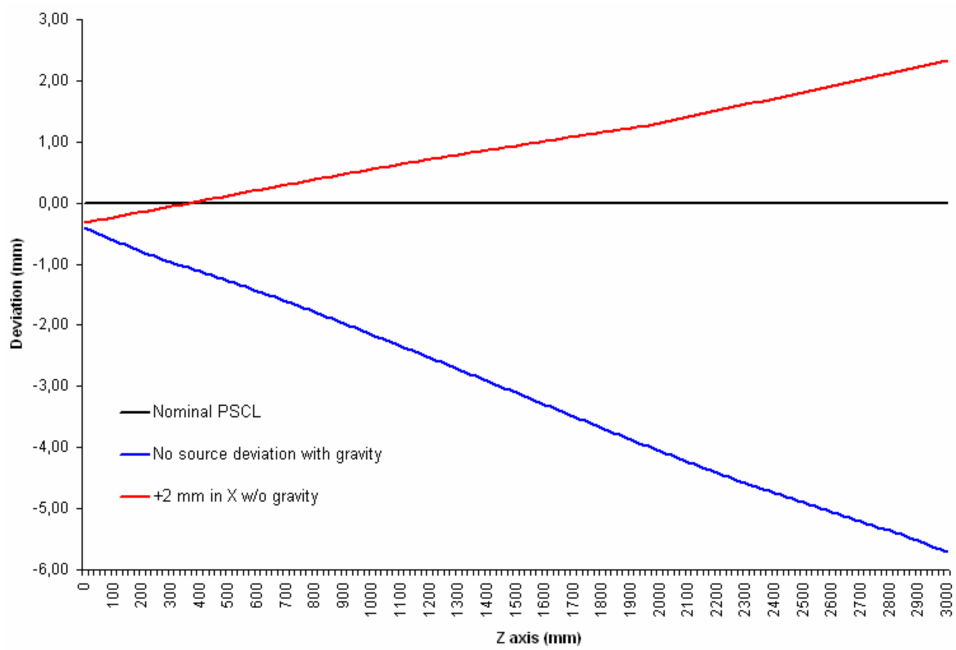


(a)

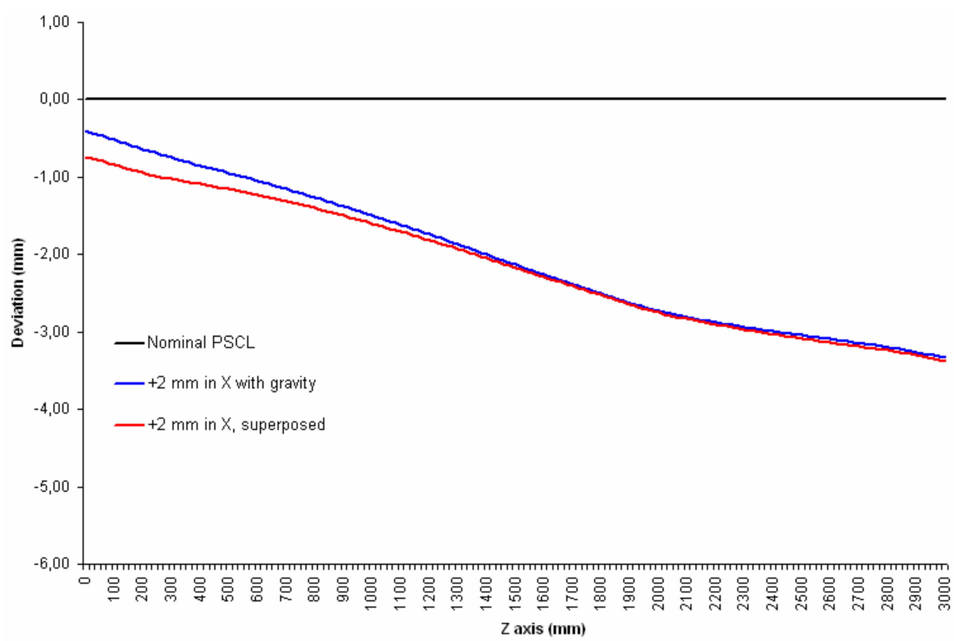


(b)

Figure 4.22 Deviation of the PSCL due to the +10 mm translation of the Third Bulkhead Locator Plate in Z-axis with the comparision of superposed and directly calculated results.



(a)



(b)

Figure 4.23 Deviation of the PSCL due to the +2 mm translation of the Third Bulkhead Locator Pins in X-axis with the comparision of superposed and directly calculated results.

This advantage is important because of two reasons. First is that, one do not need to run the models again to obtain the effect of gravity. This means a great reduction in the time required to obtain results.

Second is that the gravity-included models may require more time to converge and sometimes may have difficulty in converging, especially for the steps simulating the release of the product.

CHAPTER 5

CONCLUSION AND DISCUSSION

5.1 CONCLUSION AND DISCUSSION

Fixturing for compliant sheet metal assemblies is a challenging effort. This study has aimed to demonstrate a systematic approach for sheet metal assembly fixturing. The problem has been constrained as fixturing for assembly of sheet metal components of helicopters. An assembly fixture design study has been done for a particular tail cone assembly. Within the study;

a) typical aerostructural parts have been introduced,
b) the assembling order has been defined,
c) locating and clamping of these parts have been illustrated,
d) a possible assembly fixture design has been given for the particular tail cone.
e) the critical feature of the assembly and the features that may affect the critical feature have been stated.
f) Finite Element Analysis has been used to predict the effect of possible fixture deviations on the deviation of the critical feature of the product.

The correlation between the deviation source features and the critical product features is hard to predict for complex geometries. The variation simulation in this study depends on Finite Element Analysis (FEA). In the case of compliant assemblies, beside the relative rigid body motion of the assembly components, their deformations during the assembly process are important. This study demonstrated that FEA may be used to simulate the assembling process for complex sheet metal assemblies of helicopters. Such a simulation gives significant results which allow the tool designer to correlate the possible input and final assembly variations.

In simulating a compliant assembly process with a commercial FEA software, some of the problems that may exist have been stated and solved within the study. However, it should be noted that both the determination of the interested features and the analysis modelling approach should consider the characteristics peculiar to the product.

The analyses predict that variation of the final product depends on the input variations and the correlation between them may vary. It is not always the right choice to assign close tolerances to the fixture element features in order to guarantee a successful assembling. Instead, the determination of the critical features which have significant effect on the final assembly by analyses, may be a more practical solution. Especially, knowing that the CAD model of the product is available for most of the cases in todays manufacturing applications, the direct usage of FEA to simulate the assembly variation may be a successful alternative for tolerance analysis of compliant assemblies.

From practical applications, it is known that riveting operation causes stretching or compressing effects on the assembled sheet metal parts. Unfortunately, for most of the cases, riveting is an operator dependent process and it is difficult to predict the effect on the final assembly. For this reason, the effect of the joining process has not been included in this study.

5.2 FUTURE WORK

The design and analysis approach given in this study may be confirmed by an experimental or pilot study. Such a study may answer the following questions;

a) Is the real production data validate results from analyses?
b) For the combination of possible input variations, is it possible to superpose the results of discrete analyses?
c) Is FEA approach provide a reduction in fixturing cost? Is the total cost of analyses less than the cost of assigning close tolerances in order to guarantee a successful assembling?
d) Is it possible to develop an optimal fixture design approach using FEA for complex sheet metal assemblies, by defining an objective function to minimize the deviation(s) on the interested feature(s)?

In order to evaluate the results from analyses and to compare to the experimental results, the analyses may be done for all the possible deviations. For instance, instead of analyzing the effect of deviations of a single bulkhead, all the bulkheads may be taken into consideration. This would permit the designer to revise or refine the fixture tolerances according to the feedback from complete analyses.

This study may also find an application for assembly problems of compliant parts either in aviation industry or in other industries.

REFERENCES

[1] Norcross, C., Quinn, J.D., 1942, How to Do Aircraft Sheet Metal Work, 1942, McGraw-Hill, New York.

[2] 1998, Airframe Depot Maintenance Work Requirements, DMWR-1-1520-237-1, United Technologies Sikorsky, Connecticut.

[3] Peery, D.J., Azar, J.J., 1982, Aircraft Structures, Mc-Graw Hill, New York.

[4] Bray, J.W., 1995, Aluminum Mill And Engineered Wrought Products, A.S.M Handbook Vol. 2 Properties and Selection of Non Ferrous Alloys and Special–Purpose Materials, American Society For Metals, pp. 29-60, Ohio.

[5] 2003, Metallic Materials and Elements for Aerospace Vehicle Structures, MIL-HDBK-5J, pp. 3-1/3-325.

[6] Özbay, O., 2002, Havacılık Sanayisinde Kullanılan Yüksek Mukavemetli Alaşımların Seçimi, MS Thesis, Gazi Üniversitesi, Fen Bilimleri Enstitüsü, pp. 1-66.

[7] Private communucation with 5-th Main Maintenance Center personnel.

[8] Alpine Aerotech LTD., Light Tailboom Repair Fixture, http://www.alpineaerotech.com/Aerotech_Products_Light_Tailboom_Repair_Fixture.htm, Last Accessed Date 12.01.2007.

[9] 1985, Drafting Room Manual, Bell Helicopter Textron, pp. 3A-6.

[10] Wilson, F.W., Harvey, P.D., 1959, Tool Engineers Handbook (SME), Mc-Graw Hill, New York.

[11] Wilson, F.W., Holt J.M., 1962, Handbook of fixture design (SME), Mc-Graw Hill, New York.

[12] Jones, E.J.H., M.I.P.E., 1963, Production Engineering Jig and Tool Design, George Newnes Limited, London.

[13] Harold, S., 1970, Jigs and Fixtures for limited production, Society of Manufacturing Engineering, Michigan.

[14] John, G.N., 1998, Fundamentals of Tool Design, Society of Manufacturing Engineers, Michigan.

[15] Franklin, D.J., 1955, Jig and Fixture Design, The Industrial Press, New York.

[16] Wu, Y., Rong, Y., Ma, W., ReClair, S.R., 1998, Automated Fixture Planning: Geometric Analysis, Robotics and Computer-Integrated Manufacturing, Vol.14, pp. 1-15.

[17] Kumar, A.S., Fuh, J.Y.H., Kow, T.S., 2000, An Automated Design and Assembly of Interference-free Modular Fixture Setup Computer-Aided Design, Vol. 32, pp. 583-596.

[18] Shawki, G.S.A., Abdel-Aal, M.M., 1965, Effect of Fixture Rigidity and Wear on Dimensional Accuracy, International Journal of Machine Tool Design and Research, Vol. 5, pp. 183-202.

[19] Shawki, G.S.A., Abdel-Aal, M.M., 1967, Rigidity Considerations in Fixture Design-Contact Rigidity for Eccentric Clamping, International Journal of Machine Tool Design and Research, Vol. 7, pp. 195-209.

[20] Lowell, W.F., 1982, Geometrics II: Dimensioning and Tolerancing, ANSI/ASME Standard, Y13.5M. pp. 35-52.

[21] Karabay, M., 2004, Tool Design Lecture Notes, METU pp.136-153.

[22] Menassa, R.J., DeVries, W.R., 1991, Optimization Methods Applied to Selecting Support Position in Fixture Design, ASME Journal of Engineering for Industry, Vol. 113, pp. 412-418.

[23] Choudhuri, S.A., De Meter, E.C., 1999, Tolerance Analysis of Machining Fixture Locators, ASME Journal of Manufacturing Science and Engineering, Vol. 121, pp. 273-281.

[24] Marin, R.A., Ferreira, P.M., 2003, Analysis of the Influence of Fixture Locator Errors on the Compliance of Work Part Features to Geometric Tolerance Specifications, ASME Journal of Manufacturing Science and Engineering, Vol. 125, pp. 609-616.

[25] Lee, J.D., Haynes, L.S., 1987, Finite Element Analysis of Flexible Fixturing Systems, ASME Journal of Engineering for Industry, Vol. 109, pp. 134-139.

[26] Menassa, R.J., DeVries, W.R., 1989, Locating Point Synthesis in Fixture Design, CIRP Annuals, Vol. 38/1, pp. 165-170.

[27] De Meter, E.C., 1997, Fast Support Layout Optimization, International Journal of Machine Tools & Manufacture, Vol. 38, pp. 1221-1239.

[28] Lee, S.H, Cutkosky, M.R., 1991, Fixture Planning with Friction, ASME Journal of Engineering for Industry, Vol. 113, pp. 320-327.

[29] Satyanarayana, S., Melkote, S.N., 2004, Finite Element Modeling of Fixture-Workpiece Contacts: Single Contact Modeling and Experimental Verification, International Journal of Machine Tools and Manufacture Vol. 44, pp. 903-913.

[30] De Meter, E.C., 1994, Restraint Analysis of Fixtures Which Rely on Surface Contact, ASME Journal of Engineering for Industry, Vol. 116, pp. 207-215.

[31] Li, B., Melkote, S.N., 1999, An Elastic Contact Model for the Prediction of Workpiece-Fixture Contact Forces in Clamping, ASME Journal of Manufacturing Science and Engineering, Vol. 121, pp. 485-493.

[32] Li, B., Melkote, S.N., 2001, Fixture Clamping Force Optimization and its impact on Workpiece Location Accuracy, The International Journal of Advanced Manufacturing Technology, Vol. 17, pp. 104-113.

[33] Marin, R.A., Ferreira, P.M., 2002, Optimal Placement of Fixture Clamps: Minimizing the Maximum Clamping Forces, ASME Journal of Manufacturing Science and Engineering, Vol. 124, pp. 686-694.

[34] Raghu, A., Melkote, S.N., 2003, Analysis of the Effects of Fixture Clamping Sequences on Part Location Errors, International Journal of Machine Tools & Manufacture, Vol. 44, pp. 373-382.

[35] Tan, Y.T., Kumar, A.S., Fuh, J.Y.H., Nee, A.Y.C, 2004, Modeling, Analysis, and Verification of Optimal Fixturing Design, IEEE Transactions on Automation Science and Engineering, Vol.1 No.2, pp. 121-132

[36] Chen, H., Hu, J., Woo, T.C., 2001, Visibility Analysis and Synthesis for Assembly Fixture Certification Using Theodolite Systems, ASME Journal of Manufacturing Science and Engineering, pp. 83-89.

[37] Liu, S.C., Hu, S.J., 1995, An Offset Element and its Applications in Predicting Sheet Metal Assembly Variation, International Journal of Machine Tools & Manufacture, Vol. 35, pp. 1545-1557.

[38] Liu, S.C., Hu, S.J., 1995, Variation Simulation for Deformable Sheet Metal Assemblies Using Mechanistic Models, Transactions of NAMRI, Vol. XXII, pp. 235-240.

[39] Liu, S.C., Hu, S.J., 1997, Variation Simulation for Deformable Sheet Metal Assemblies Using Finite Element Methods, ASME Journal of Manufacturing Science and Engineering, Vol. 119, pp. 368-374.

[40] Chang, M., Grossard, D.C., 1997, Modeling the Assembly of Compliant, Non-ideal Parts, Computer-Aided Design, Vol. 29, No. 10, pp. 701-708.

[41] Cai, W., Hu, S.J., Yuan, J.X., 1996, Deformable Sheet Metal Fixturing: Principles, Algorithms, and Simulations, ASME Journal of Manufacturing Science and Engineering, Vol. 118, pp. 318-324.

[42] Liu, S.C., Hu, S.J., 1998, Sheet Metal Joint Configurations and Their Variation Characteristics, ASME Journal of Manufacturing Science and Engineering, Vol. 120, pp. 461-467.

[43] Merkley, K.G., 1998, Tolerance Analysis of Compliant Assemblies, PhD Dissertation, Brigham Young University, pp. 69-91.

[44] Bihlmaier, B.F., 1999, Tolerance Analysis of Flexible Assemblies Using Finite Element and Spectral Analysis, MS Thesis, Brigham Young University, pp. 7-21.

[45] Hu, S.J., Webbink R., Lee, J., Long, Y., Robustness Evaluation for Compliant Assembly Systems, ASME Journal of Mechanical Design, Vol.125, pp. 262-267.

[46] Camelio, J.A., Hu, S.J., Marin, S.M., Compliant Assembly Variation Analysis Using Component Geometric Covariance, ASME Journal of Manufacturing Science and Engineering, Vol. 126, pp. 355-360.

[47] Camelio, J., Hu, S.J., Ceglarek, D., 2003, Modeling Variation Propagation of Multi-Station Assembly Systems with Compliant Parts, ASME Journal of Mechanical Design, Vol.125, pp. 673-681.

[48] Camelio, J., Hu, S.J., Ceglarek, D., 2003, Impact of Fixture Design on Sheet Metal Assembly Variation, SME Journal of Manufacturing Systems, Vol. 23, No. 3, 182-193.

[49] Hoffman, K., Santosa, F., 2003, A Simple Model of Sheet Metal Assembly, Society for Industrial and Applied Mathematics Review, Vol. 45, No. 3, pp. 558-573.

[50] Ceglarek, D., Shi, J., 1996, Fixture Failure Diagnosis for the Autobody Assembly Using Pattern Recognition, ASME Journal of Engineering for Industry, Vol. 118, pp. 55-66.

[51] Ceglarek, D., Shi, J., 1999, Fixture Failure Diagnosis for Sheet Assembly with Consideration of Measurement Noise, ASME Journal of Manufacturing Science and Engineering, Vol. 121, pp. 771-777.

[52] Ding, Y., Gupta, A., Apley, D.W., 2004, Singularity Issues in Fixture Fault Diagnosis for Multi-Station Assembly Process, ASME Journal of Manufacturing Science and Engineering, Vol. 126, 200-210.

[53] Walczyk, D.F., Raju, V., 2000, Simplifying the Development and Usage of Fixtures for Sheet Metal and Composite Aircraft Parts, ASME Journal of Manufacturing Science and Engineering, Vol. 122, 370-373.

[54] Long, Y., 2000, Variation Simulation for Compliant Sheet Metal Assemblies with Applications, Ph.D. Dessertion, University of Michigan, pp. 16-58.

[55] Takezawa, N., 1980, An Improved Method for Establishing the Process Wise Quality Standard, JUSE Reports of Statistical and Applied Research, Vol. 27, No.3, pp. 63-76.

[56] ABAQUS® 6.6.1, 2006, ABAQUS/CAE User's Manual, ABAQUS, Inc.

[57] AutoDesk Inventor®, 2004, Tutorials, Autodesk, Inc.

[58] Chase, K.W., Parkinson, A.R., 1991, A Survey of Research in the Application of Tolerance Analysis to the Design of Mechanical Assemblies, Research on Engineering Design, Vol.3 No.1, pp. 63-75.

[59] Hibbeler, R.C., 1994, Mechanics of Materials, Prantice Hall, Inc., London, pp. 581-656.

APPENDIX A

MECHANICAL VARIATION SIMULATION AND METHOD OF INFLUENCE COEFFICIENTS

Mechanical variation simulation is first proposed by Liu, Lee and Hu, and combines the engineering structural models with statistical models [38]. Two of the same authors, Liu and Hu, also introduced the application of finite element analysis for “Mechanical Variation Simulation Analysis” technique to calculate the assembly stiffness [39].

Under the assumption of isotropic material, small deformations (stiffness matrix remains constant for the part) and all deformations are in linear elastic range, the force required to bring a deviated part to its nominal may be written in the form;

$$\{F\} = [K_p]\{V_p\} \tag{A.1}$$

Where $\{F\}$ is the force vector; $[K_p]$ is the stiffness matrix of the parts before joining, and $\{V_p\}$ is the deviation vector of parts.

Remembering the steps of sheet metal assembly process given in the third chapter, the formulation above stands for the second step (b) and the components of vector $\{F\}$ represent the forces provided by the fixture elements (clamps).

In step (d), the clamps are released from the joined structure (assembly). The assembly will spring back from its nominal position after the clamps are released. This spring-back issue may be expressed similarly;

$$[K_a]\{V\} = \{F\} \tag{A.2}$$

where $\{V\}$ is the spring-back of the assembly. From (A.1) and (A.2);

$$\{V\} = [K_a]^{-1}[K_p]\{V_p\} = [S]\{V_p\} \tag{A.3}$$

Matrix $[S]$ is called the sensitivity matrix (Chase and Parkinson, 1991) [58]. This linear relation obtained is called "Mechanistic Variation Model".

To obtain the sensitivity matrix, Liu and Hu (1997) developed a methodology called "Method of Influence Coefficients".

Before continuing into the detail of the methodology, some inferences from the developed relation given by equation (A.3), should be underlined. Those may be summarized as; if $[K_p] < [K_a]$, then $\{V\} < \{V_p\}$, the assembly process "absorbs" the part deviation or if $[K_p] > [K_a]$, then $\{V\} > \{V_p\}$, the assembly process "magnifies" the part variation, i.e, if parts of the assembly are compliant (relatively low stiffness) and the assembly itself is realtively rigid (higher stiffness), then the assembly variation is reduced with respect to the part variations.

Derivation of the matrices of the linear relation between component variations (variation sources) and assembly variation is given below:

A stiffened panel, as given in Figure A.1 (a), composed of several stiffeners and a skin sheet can be considered for the purpose of discussing the method. The variation sources of the panel are defined as nodes. The direction of the variation, the the variation sources and their number should be identified by the user. For the case, assuming N number of variation sources and applying unit force to these nodes (variation sources), an NxN "matrix of influence coefficients" is obtained;

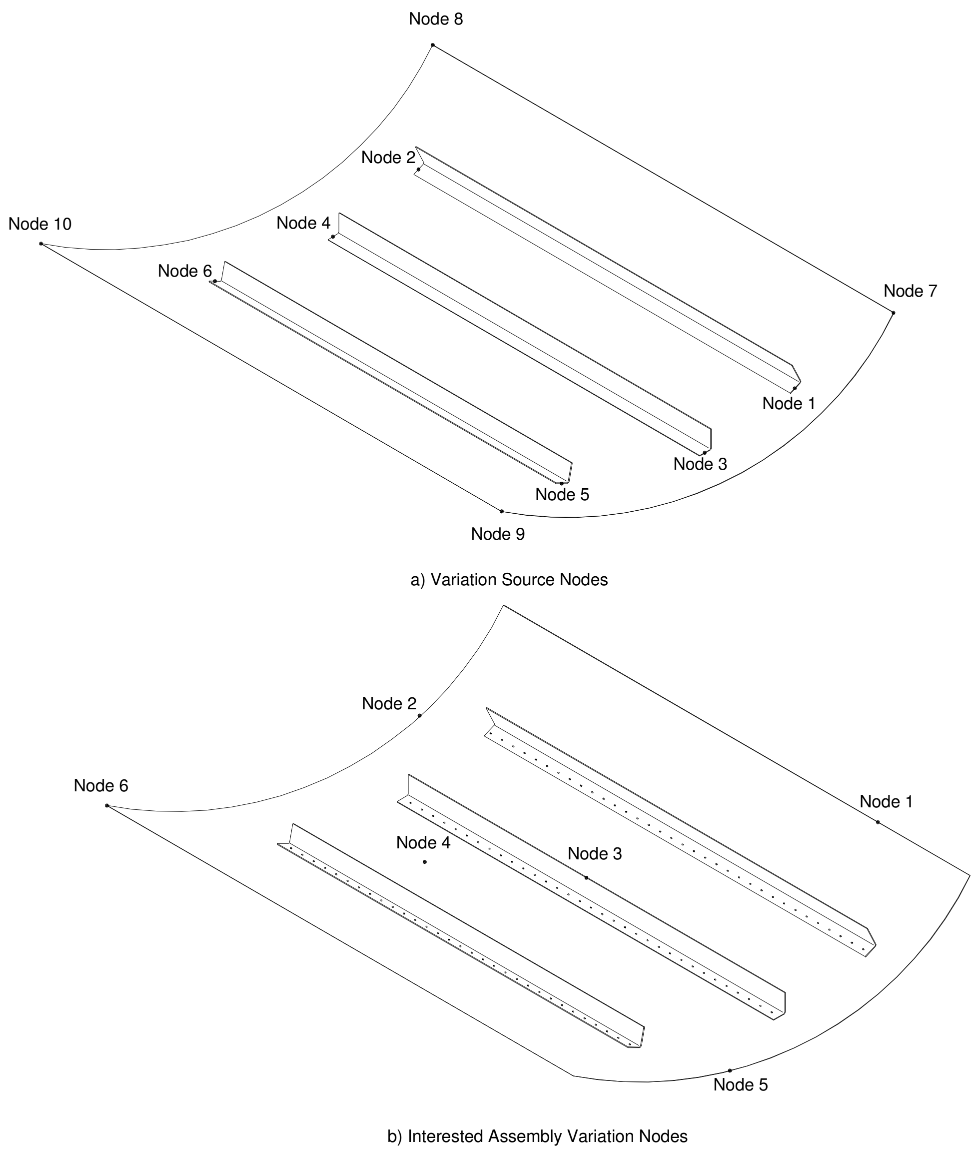


Figure A.1 Variation Nodes

$$C = \begin{bmatrix} c_{11} & c_{12} & c_{13} & \cdots & c_{1N} \\ c_{21} & \cdots & \cdots & \cdots & \cdots \\ c_{31} & \cdots & \cdots & \cdots & \cdots \\ \cdots & \cdots & \cdots & \cdots & \cdots \\ c_{N1} & \cdots & \cdots & \cdots & c_{NN} \end{bmatrix} \tag{A.4}$$

where c_{ij} is the deformation of the i-th variation source under the unit force applied to the j-th variation source. The variations of source nodes (coming from part variations) may be written in vector form as;

$$V = \begin{bmatrix} v_1 \\ v_2 \\ v_3 \\ \cdots \\ v_N \end{bmatrix} = \begin{bmatrix} c_{11} & c_{12} & c_{13} & \cdots & c_{1N} \\ c_{21} & \cdots & \cdots & \cdots & \cdots \\ c_{31} & \cdots & \cdots & \cdots & \cdots \\ \cdots & \cdots & \cdots & \cdots & \cdots \\ c_{N1} & \cdots & \cdots & \cdots & c_{NN} \end{bmatrix} . \begin{bmatrix} f_1 \\ f_2 \\ f_3 \\ \cdots \\ f_N \end{bmatrix} \tag{A.5}$$

F is the force vector and f_i is the force required to close the tooling, i.e., the clamping force required to bring the variations back to the nominal values. To obtain F;

$$[F] = [C]^{-1}.[V] \tag{A.6}$$

Once the clamps are closed and the joining process is completed, the stiffness of the assembly changes. When the clamps are released, the assembly springs back. The relation between the variations of the interested nodes on the assembly after releasing the clamps, may be written as;

$$[U] = \begin{bmatrix} u_1 \\ u_2 \\ u_3 \\ \cdots \\ u_M \end{bmatrix} = \begin{bmatrix} c_{11}^{t} & c_{12}^{t} & c_{13}^{t} & \cdots & c_{1N} \\ c_{21}^{t} & \cdots & \cdots & \cdots & \cdots \\ c_{31}^{t} & \cdots & \cdots & \cdots & \cdots \\ \cdots & \cdots & \cdots & \cdots & \cdots \\ c_{M1}^{t} & \cdots & \cdots & \cdots & c_{MN}^{l} \end{bmatrix} . \begin{bmatrix} f_1 \\ f_2 \\ f_3 \\ \cdots \\ f_N \end{bmatrix} \tag{A.7}$$

Here, c_{ij}^{t} is the deformation of the i-th interested node on the assembly when a unit force is applied on the j-th variation source.

Substituting for the force vector F but in opposite direction, one can equivalently write,

$$[U]_{Mx1} = -[C^{t}]_{MxN} . [C]_{NxN}^{-1} . [V]_{Nx1} \tag{A.8}$$

or,

$$[U] = [S].[V] \tag{A.9}$$

This expression gives the relation between the input part variation and the output assembly variation. It should be noted that $[S]$ is the sensitivity matrix and the inverses of matrices $[C]$ and $[C^{t}]$ are the stiffness matrices for components and assembly, respectively. These matrices are obtained by FEM for complex geometries. The advantage of the method is that once the coefficient matrices (or stiffness matrices) are obtained, it is possible to use the statistical data and achieve the assembly variation distribution without performing FEA for each sample data.

APPENDIX B

SOLUTION OF 1-D BEAM ASSEMBLY PROBLEM

B.1 SOLUTION BY BEAM DEFLECTION FORMULATION

For the cantilever beam shown in Figure B.1, derivation of the deflection expression is given below [59];

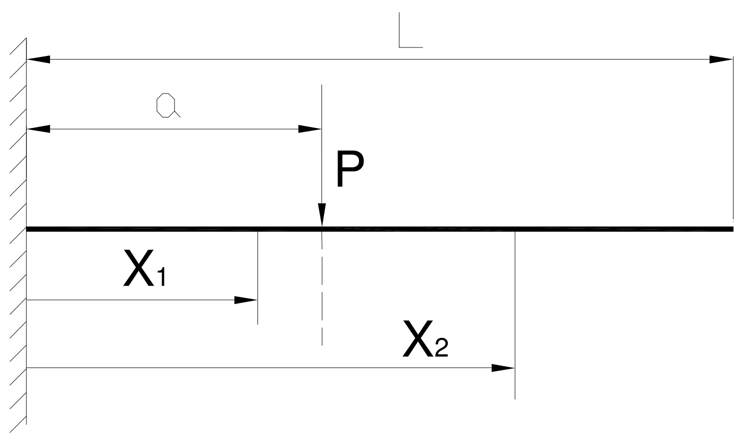


Figure B.1 Cantilever Beam before Assembling

$\underline{0 < x < a}$

$$M_1 = Px - Pa = P(x - a) \tag{B.1}$$

$$EI\frac{d^2 v_1}{dx^2} = P(x - a) \tag{B.2}$$

$$EI\frac{dv_1}{dx} = \frac{P}{2}(x - a)^2 + C_1 \tag{B.3}$$

$$EIv_1(x) = \frac{P}{6}(x-a)^3 + C_1 x + C_2 \tag{B.4}$$

<u>Applying Boundary Conditions;</u>

1- $x = 0$, $v_1 = 0$ (B.5)

2- $x = 0$, $\frac{dv_1}{dx} = 0$ (B.6)

From BC's;

$$C_1 = -\frac{Pa^2}{2} \tag{B.7}$$

$$C_2 = \frac{Pa^3}{6} \tag{B.8}$$

<u>$a < x < L$</u>

$$M_2 = Px - P(x-a) - Pa = 0 \tag{B.9}$$

$$EI\frac{d^2v_2}{dx^2} = 0 \tag{B.10}$$

$$EI\frac{dv_2}{dx} = C_3 \tag{B.11}$$

$$EIv_2(x) = C_3 x + C_4 \tag{B.12}$$

<u>Applying Continuity Conditions;</u>

1- $x = a$, $v_1 = v_2$ (B.13)

2- $x = a$, $\frac{dv_1}{dx} = \frac{dv_2}{dx}$ (B.14)

From CC's;

$$C_3 = -\frac{Pa^2}{2} \tag{B.15}$$

$$C_4 = \frac{Pa^3}{6} \tag{B.16}$$

Deflection formulation for the beam is;

$$v(x) = \frac{P}{6EI}\left[(x-a)^3 - 3a^2x + a^3\right] \quad , \qquad 0 \le x \le a \tag{B.17}$$

$$v(x) = \frac{P}{6EI}\left[-3a^2x + a^3\right] \quad , \qquad a < x \le L \tag{B.18}$$

Since the variation of the first beam is small, the stiffness values for the beams before joining can be taken as equal;

$K_1 = K_2 = K$; (B.19)

and the force to close the gap is applied to both beams equally in magnitude but in opposite direction,

$\delta_1 = \delta_2$ (B.20)

As $\delta_0 = \delta_1 + \delta_2 = 1.00$ mm; (B.21)

$\delta_1 = \delta_2 = \delta = 0.50$ mm (B.22)

Force applied by the clamps to close the gap can be achieved by the relation,

$\delta = FxC_8$ or $F = \dfrac{\delta}{C_8}$ (B.23)

$[C]_{15x1}$ is the deformation vector constituting deformations of each interested node under a unit load applied to the eighth node. It should be obtained from the beam deflection formulation given above. C_8 is the deformation of eighth node under a unit force applied to the same node.

$[U] = [V_{IN}] + [\delta]$ (B.24)

$[\delta] = FxC_8$, (B.25)

$[U] = [V_{IN}] + FxC_8$, (B.26)

or, $[U] = [V_{IN}] + \dfrac{\delta}{C_8}[C]$ (B.27)

Initial variation of the parts are defined as $[V_{IN}]$ and it can be achieved by applying the variation expressions for all interested nodes.

$[V_{IN}] = [0 \quad 0 \quad 0.028 \quad 0.111 \quad 0.250 \quad 0.444 \quad 0.694 \quad 1.000 \quad 0 \quad 0 \quad 0 \quad 0 \quad 0 \quad 0 \quad 0]^T$

(B.28)

$[C] = [2.875 \quad 11 \quad 23.625 \quad 40 \quad 59.375 \quad 81 \quad 104.125 \quad 128...$

$...104.125 \quad 81 \quad 59.375 \quad 40 \quad 23.625 \quad 11 \quad 2.875]^T x\left(\dfrac{10^6}{6EI}\right)$ (B.29)

$[U]$ is calculated as;

$$[U] = [-0.011 \quad -0.043 \quad -0.064 \quad -0.045 \quad 0.018 \quad 0.128 \quad 0.287 \quad 0.5 \ldots$$
$$\ldots 0.407 \quad 0.316 \quad 0.232 \quad 0.156 \quad 0.092 \quad 0.043 \quad 0.011]^T \qquad \text{(B.30)}$$

It should be noted that for the common node of the two beams, i.e., node 8, the variation of either part one or part two may be written in $[V_{IN}]$. Hovewer, the force F should have a minus or plus value convenient to that entry, yielding the same result for both cases.

B.2 SOLUTION BY FINITE ELEMENT ANALYSIS MODEL

For this case, the beams are modeled with 2-D wire option in ABAQUS CAE 6.6.1 and meshed as suitable to the nodes given in the problem. In other words, each node of the FEA model coincides with the corresponding node in the problem. To model the clamping of beams, displacement boundary conditions are applied –0.5 mm in y direction to the node 4 of part 1 at x=400; and, 0.5 mm in y direction to the node 5 of part 2 at x=400 in Step 1. In the next step, the contact between the beams are modified by not allowing any separation in normal direction and no slip in tangential direction. This modification enforces the contacting nodes to behave as if joined. Also displacement boundary conditions are deactivated to simulate the spring-back effect when the clamps are opened.

APPENDIX C

DRAWINGS OF THE TAIL CONE ASSEMBLY

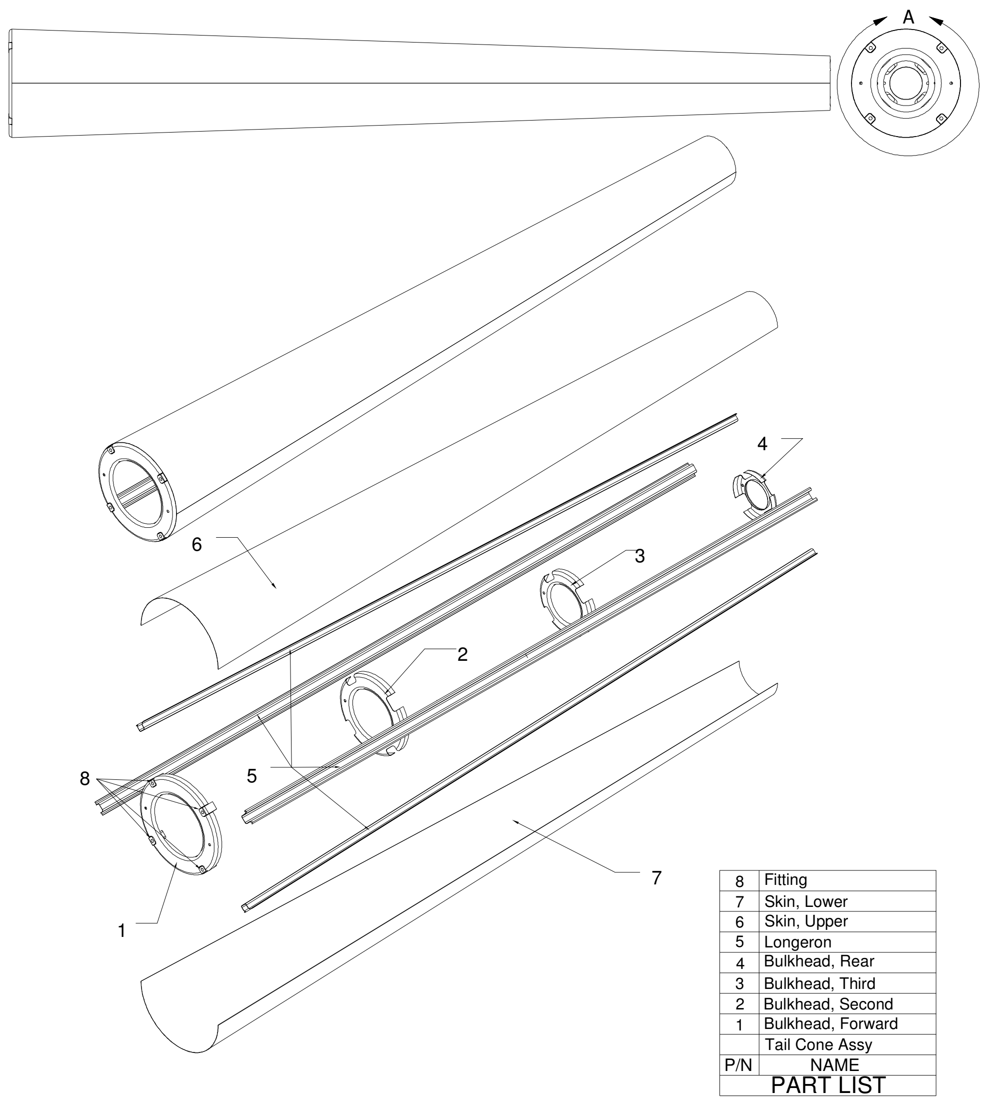


Figure C.1 Tail Cone Assembly Drawings

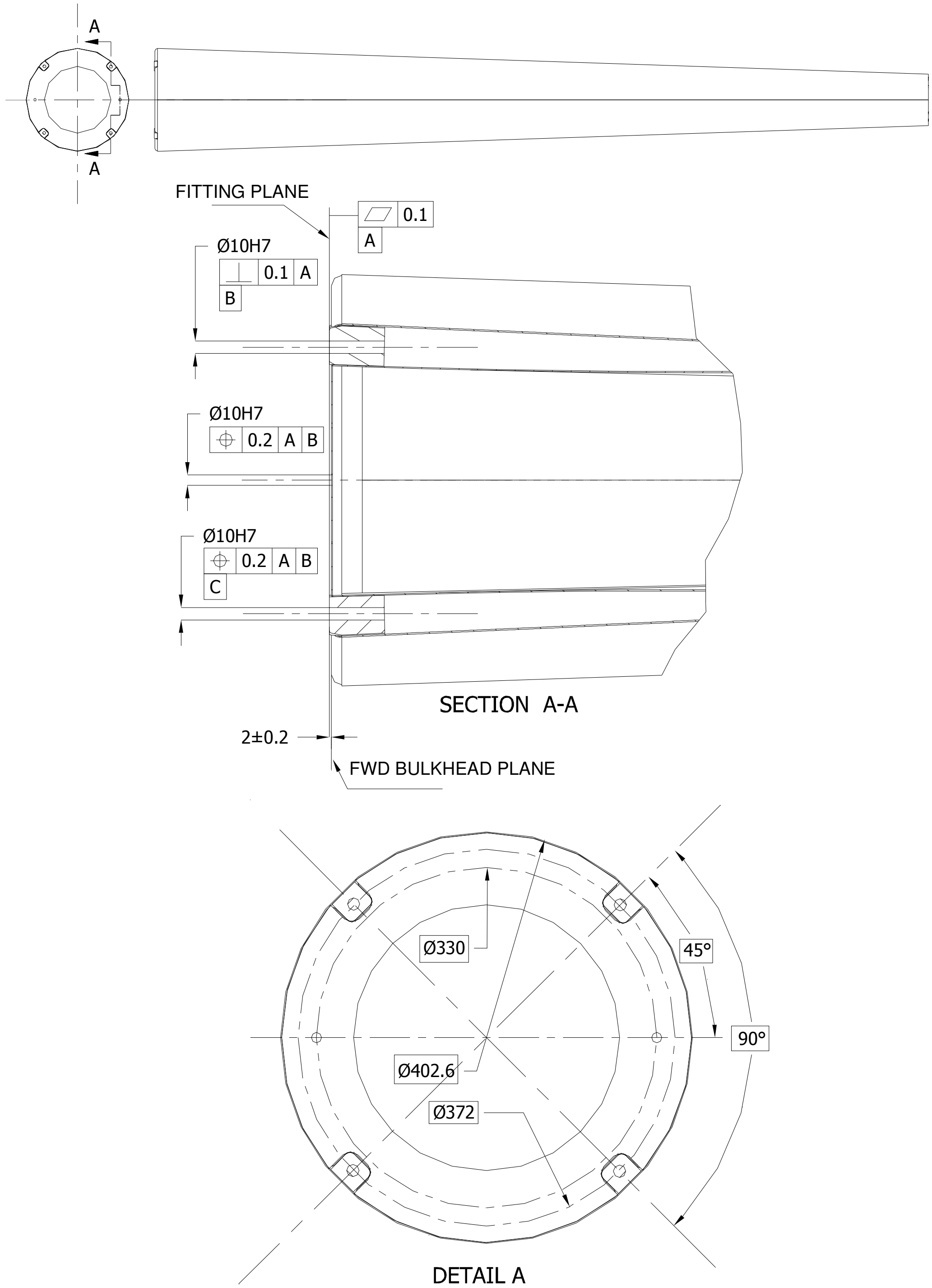
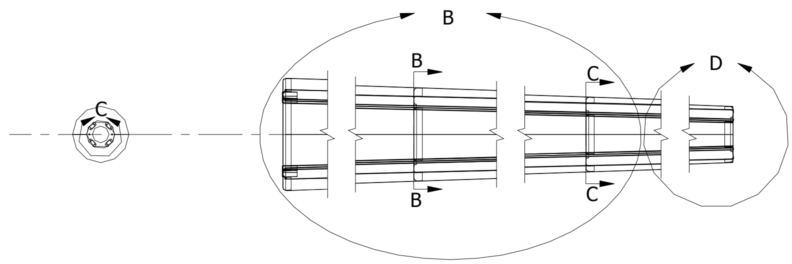
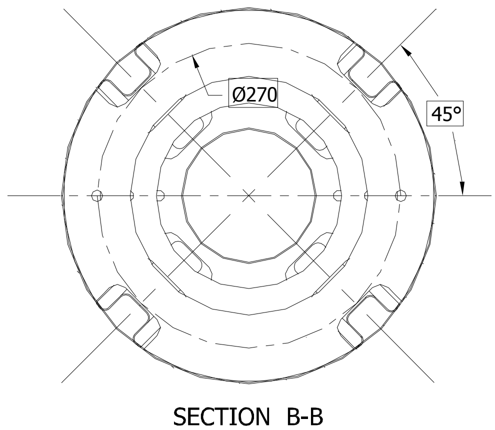


Figure C.1 Tail Cone Assembly Drawings (continued)





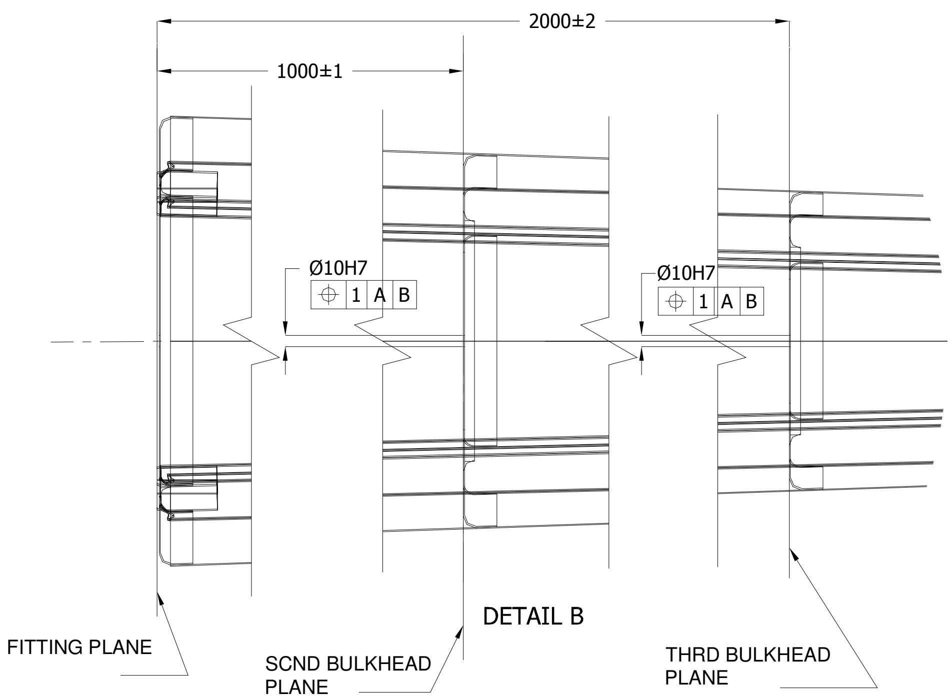
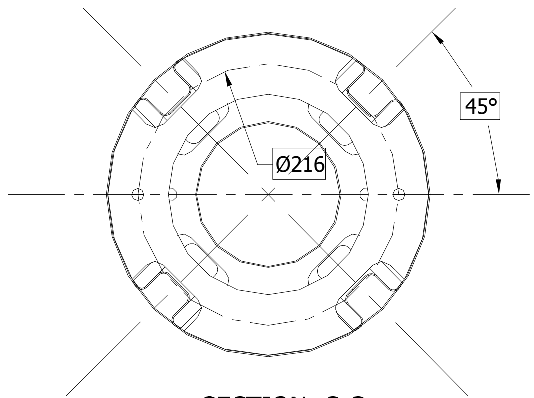
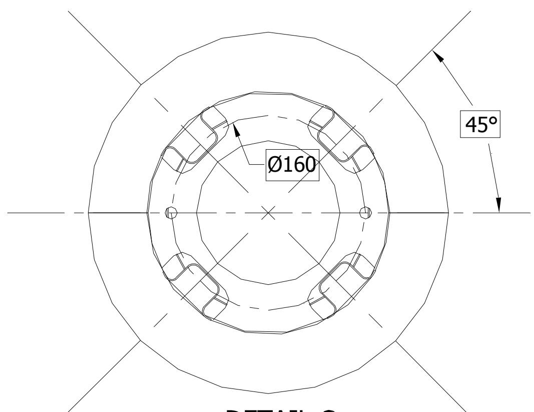


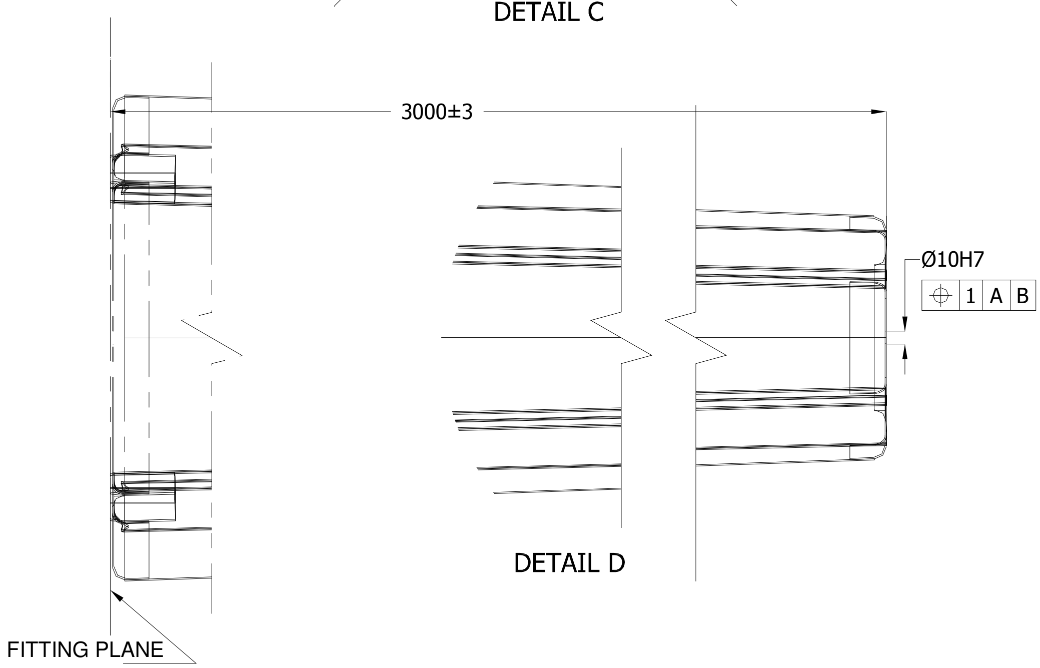
Figure C.1 Tail Cone Assembly Drawings (continued)



SECTION C-C

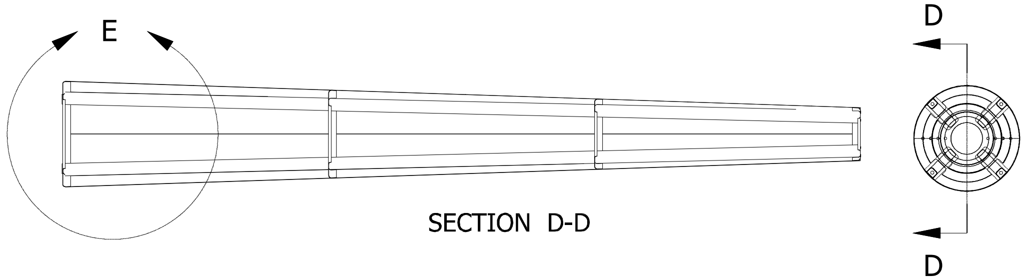


DETAIL C



DETAIL D

Figure C.1 Tail Cone Assembly Drawings (continued)



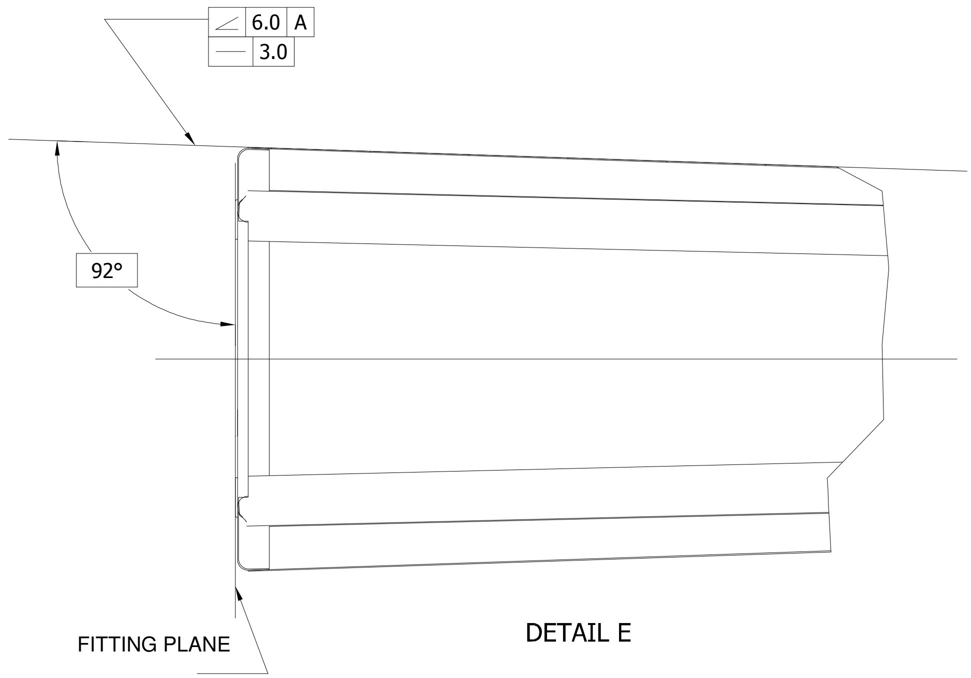
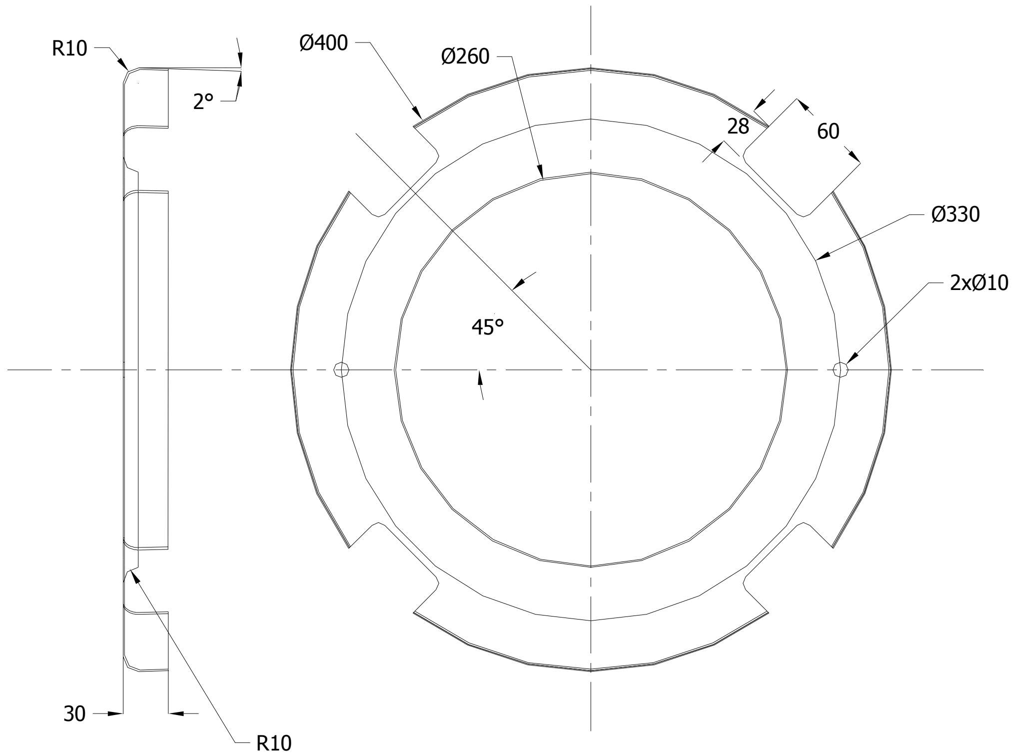
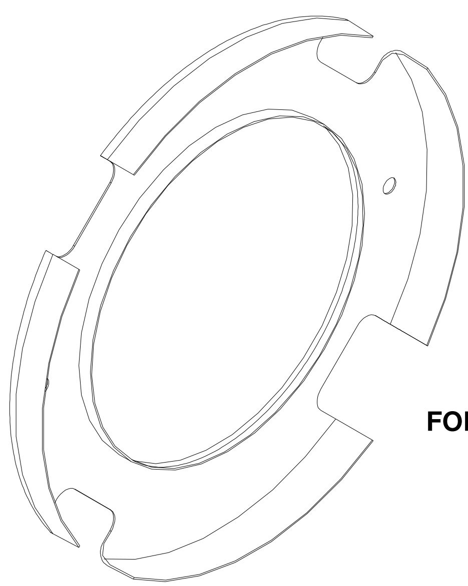


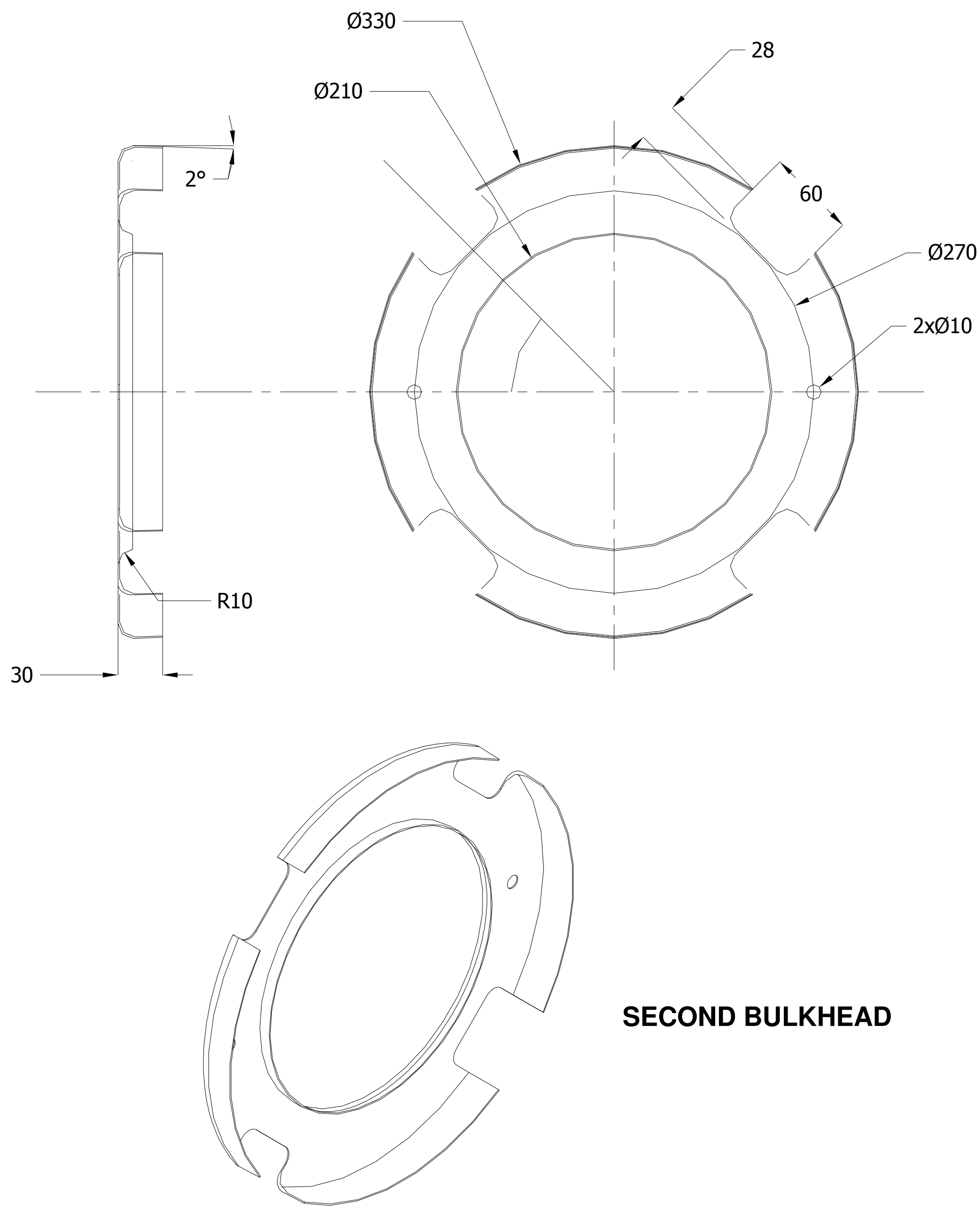
Figure C.1 Tail Cone Assembly Drawings (continued)





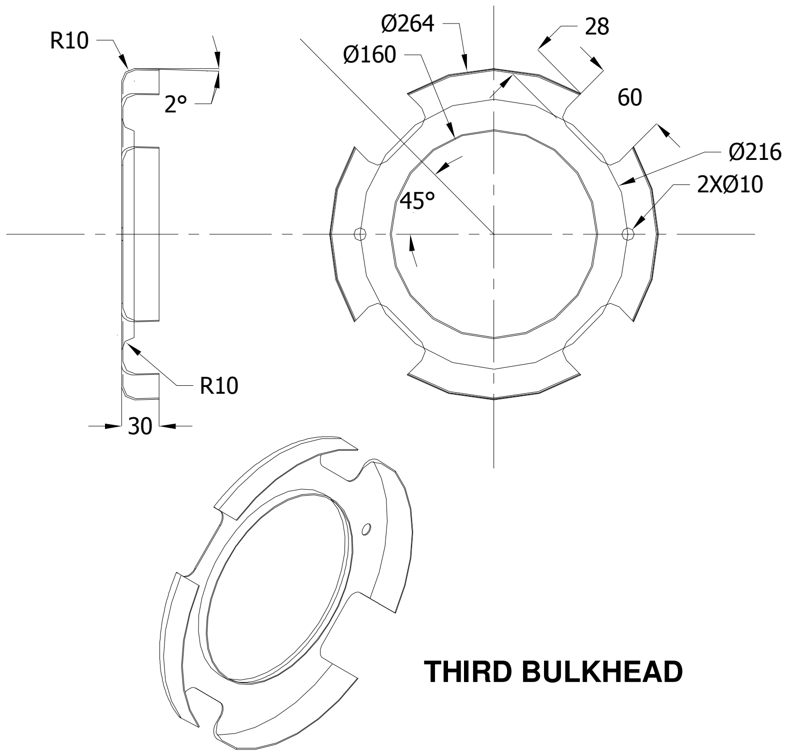
FORWARD BULKHEAD

Figure C.2 Tail Cone Component Drawings

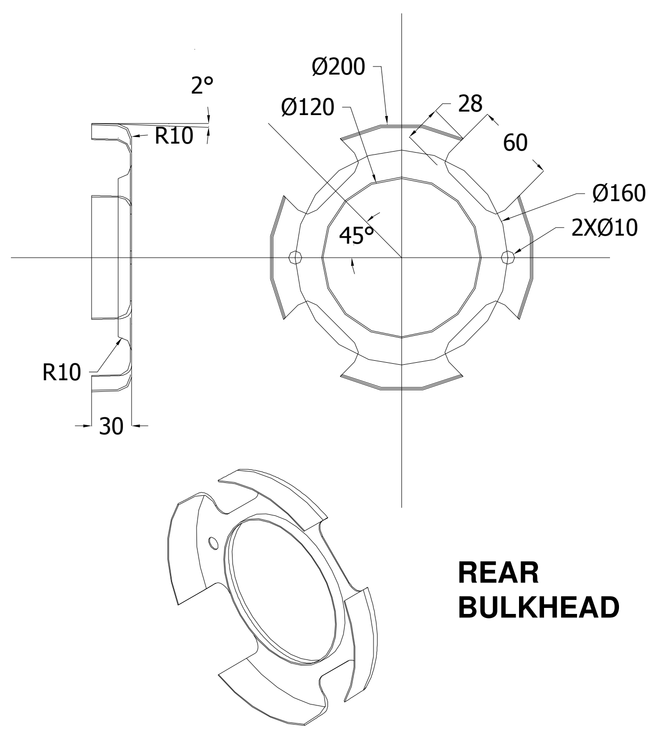


SECOND BULKHEAD

Figure C.2 Tail Cone Component Drawings (continued)

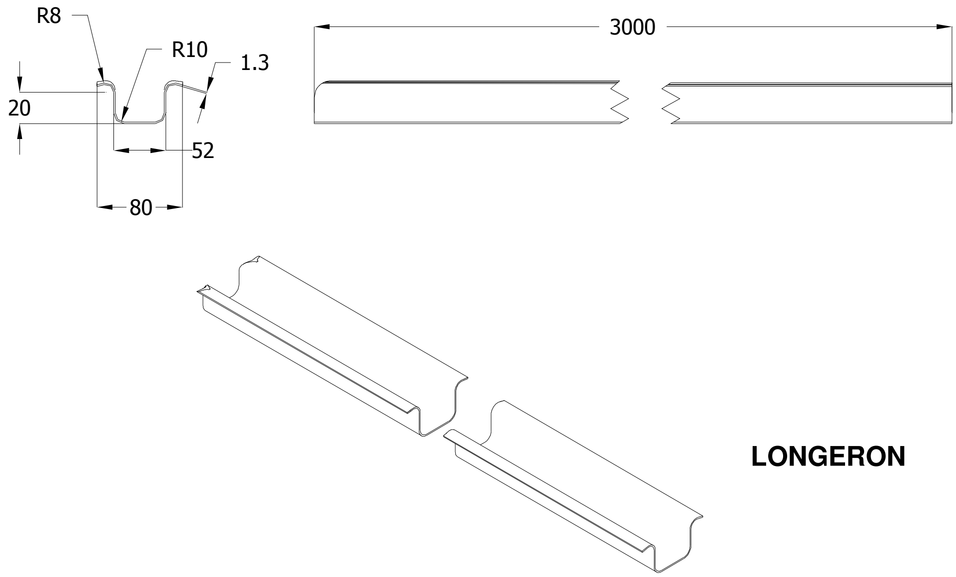


THIRD BULKHEAD

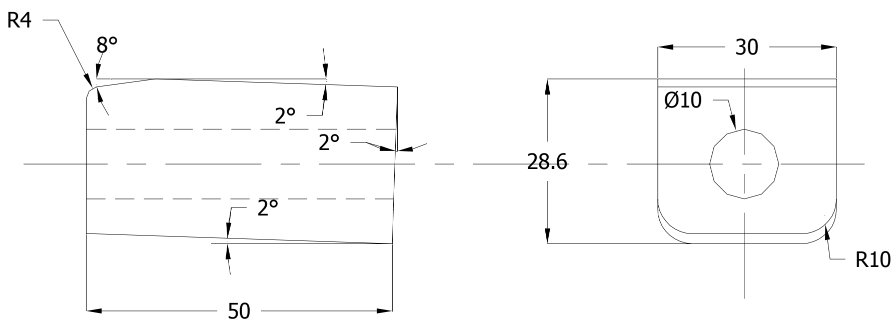


REAR BULKHEAD

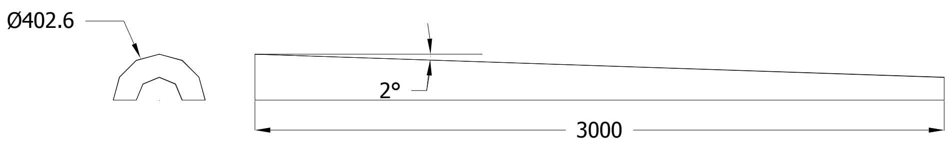
Figure C.2 Tail Cone Component Drawings (continued)



LONGERON



FITTING

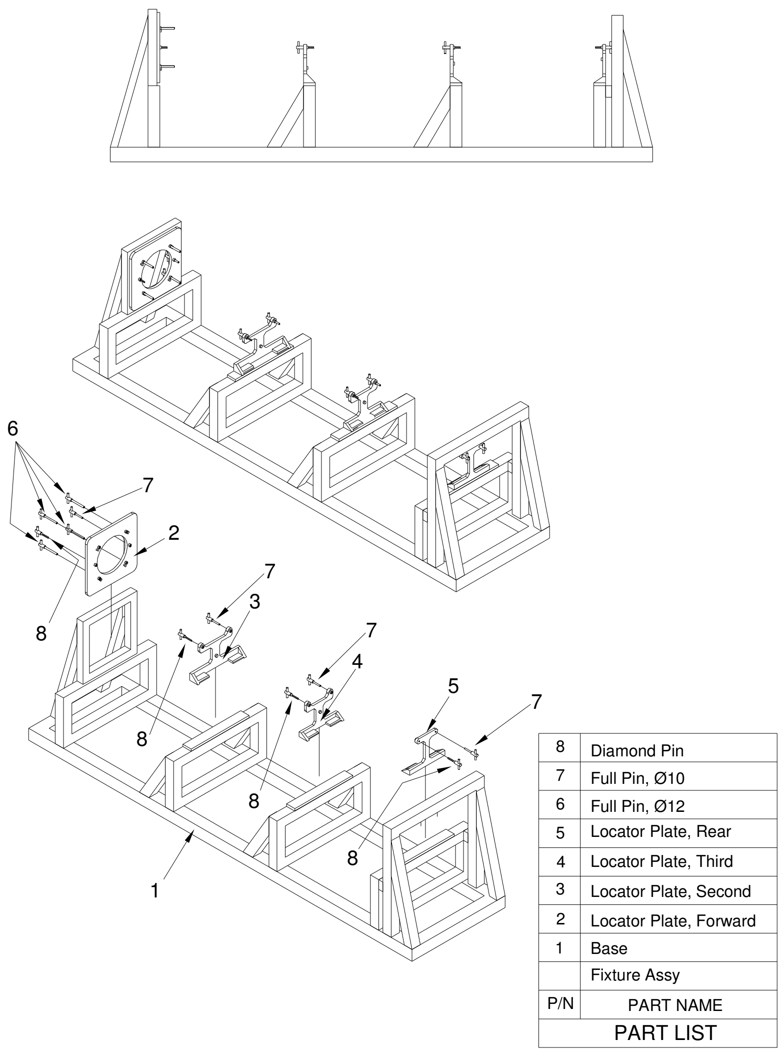


SKIN

Figure C.2 Tail Cone Component Drawings (continued)

APPENDIX D

DRAWINGS OF THE TAIL CONE ASSEMBLY FIXTURE



P/N	PART NAME
8	Diamond Pin
7	Full Pin, Ø10
6	Full Pin, Ø12
5	Locator Plate, Rear
4	Locator Plate, Third
3	Locator Plate, Second
2	Locator Plate, Forward
1	Base
	Fixture Assy

PART LIST

Figure D.1 Tail Cone Assembly Fixture, Assembly Drawings

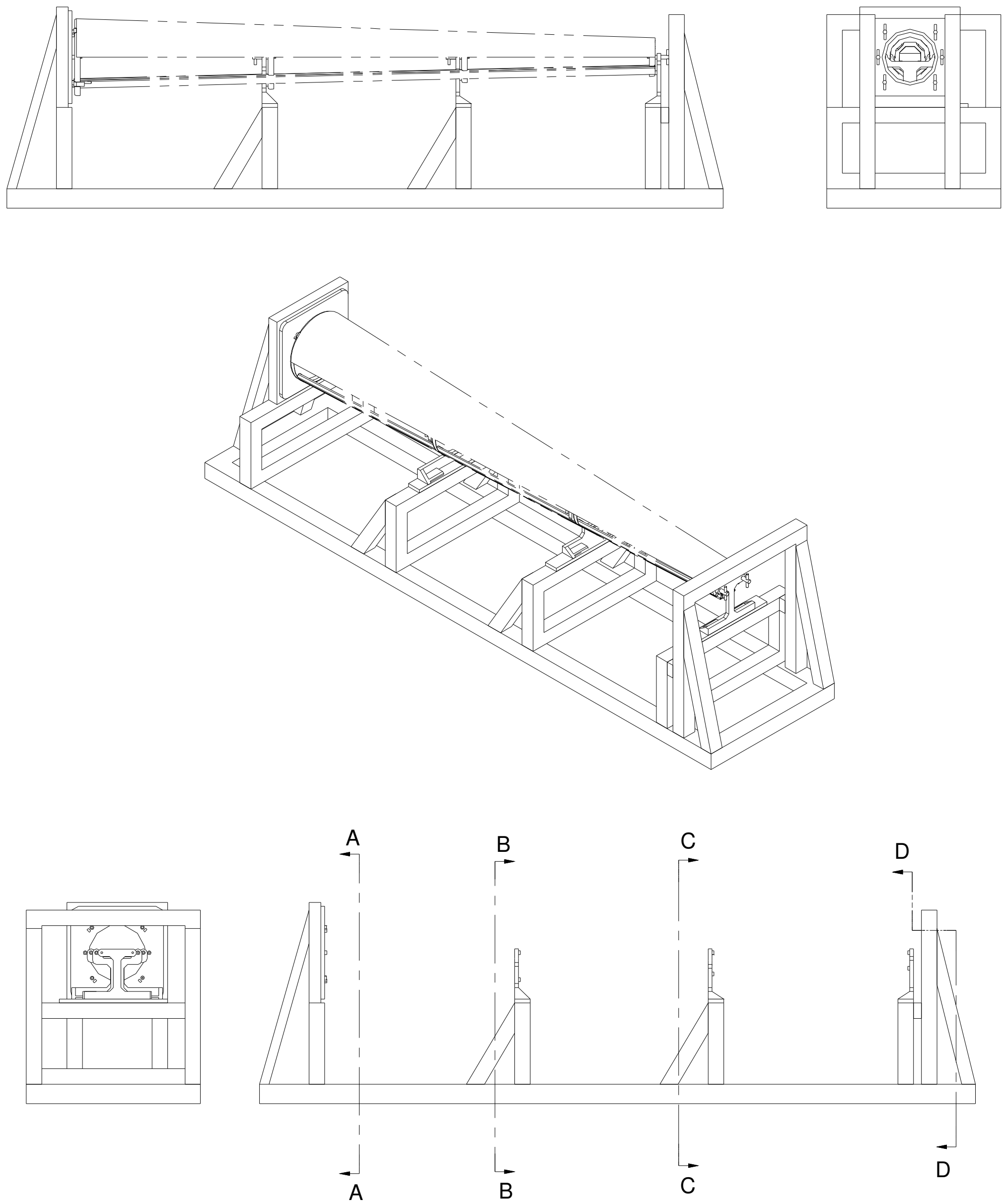
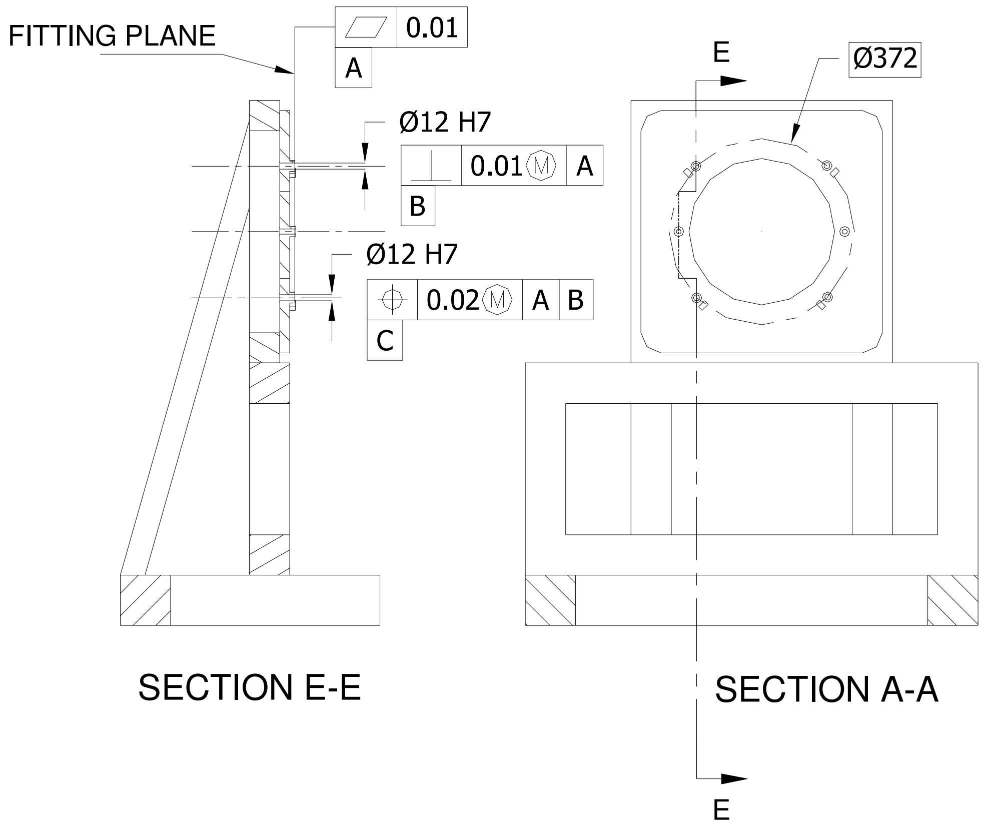
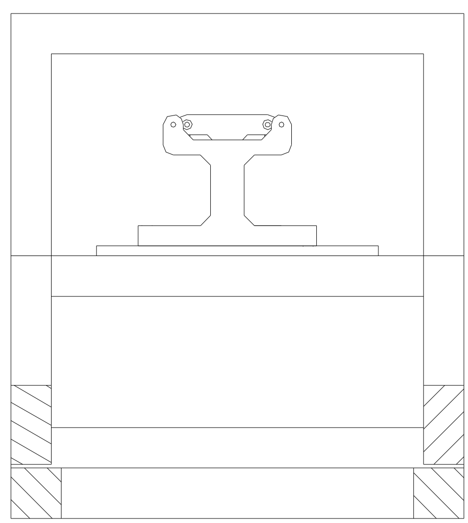


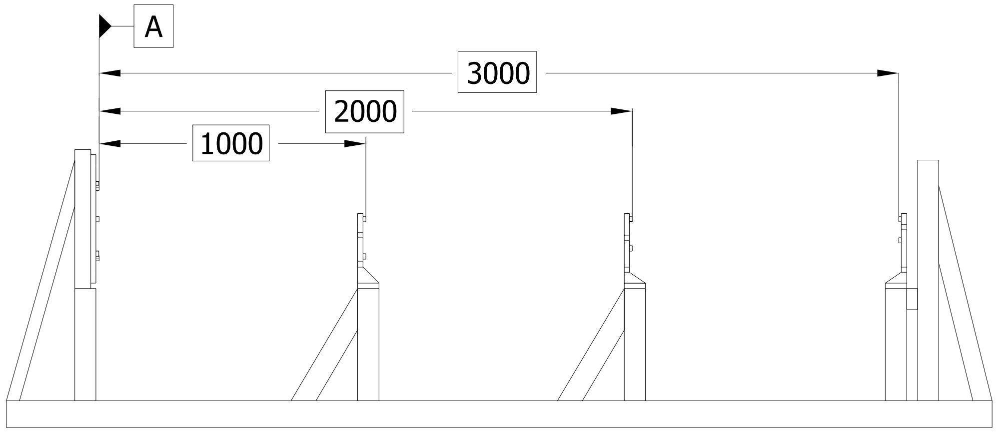
Figure D.1 Tail Cone Assembly Fixture, Assembly Drawings (continued)

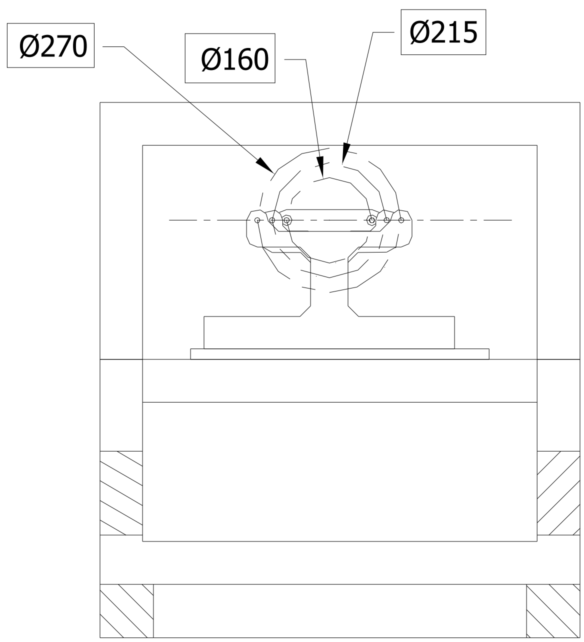




SECTION C-C

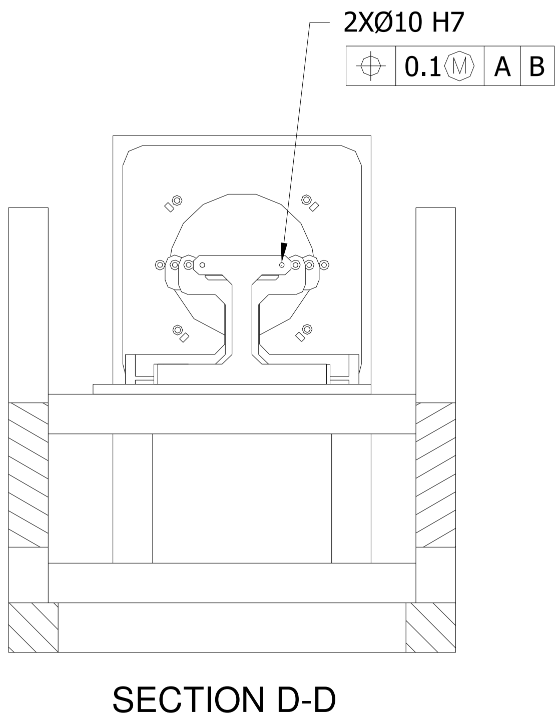
Figure D.1 Tail Cone Assembly Fixture, Assembly Drawings (continued)





SECTION B-B

Figure D.1 Tail Cone Assembly Fixture, Assembly Drawings (continued)



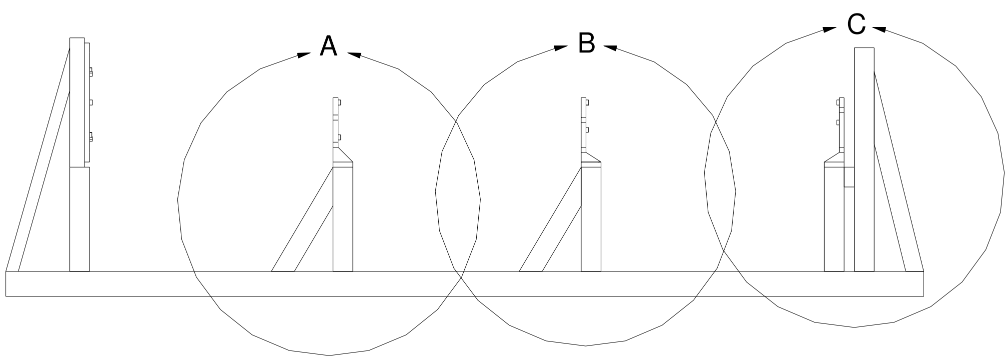
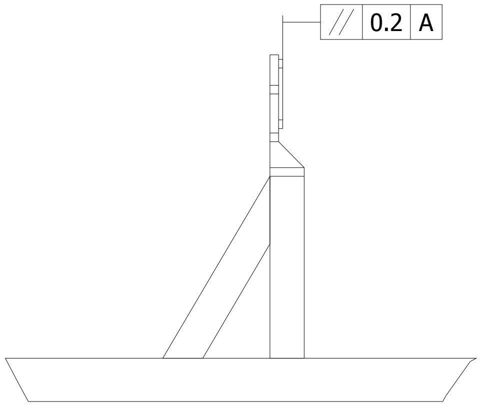
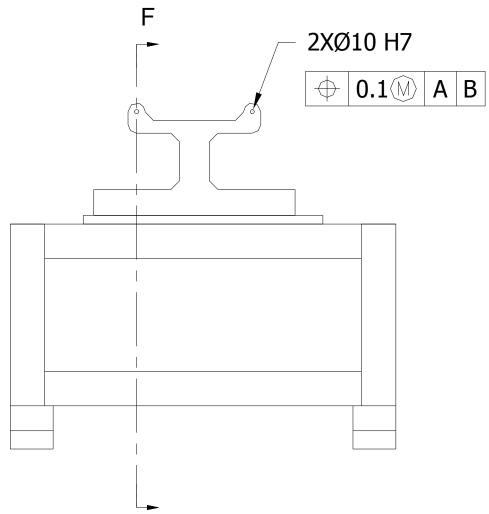
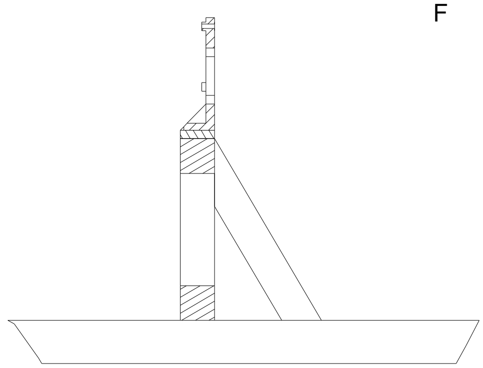


Figure D.1 Tail Cone Assembly Fixture, Assembly Drawings (continued)



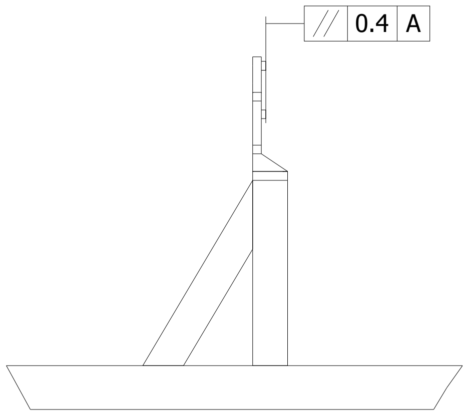
DETAIL A



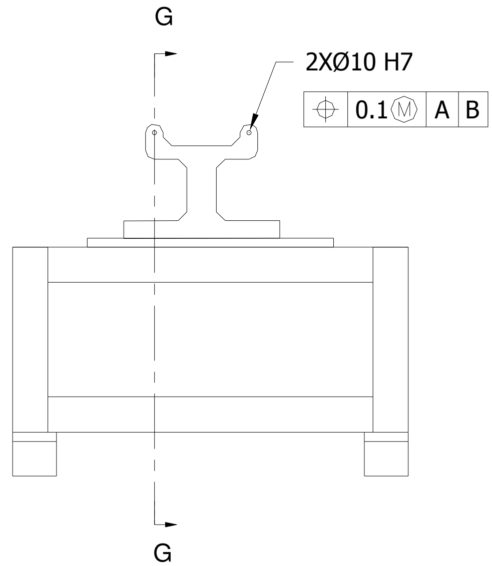


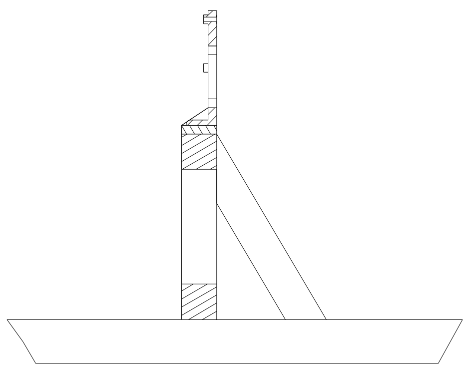
SECTION F-F

Figure D.1 Tail Cone Assembly Fixture, Assembly Drawings (continued)

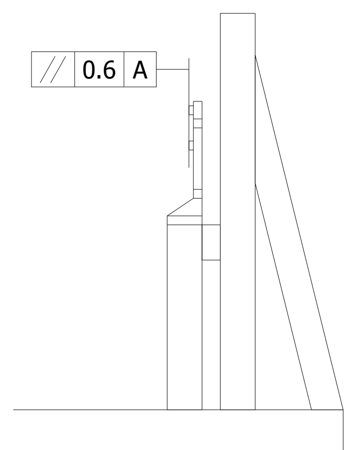


DETAIL B





SECTION G-G



DETAIL C

Figure D.1 Tail Cone Assembly Fixture, Assembly Drawings (continued)

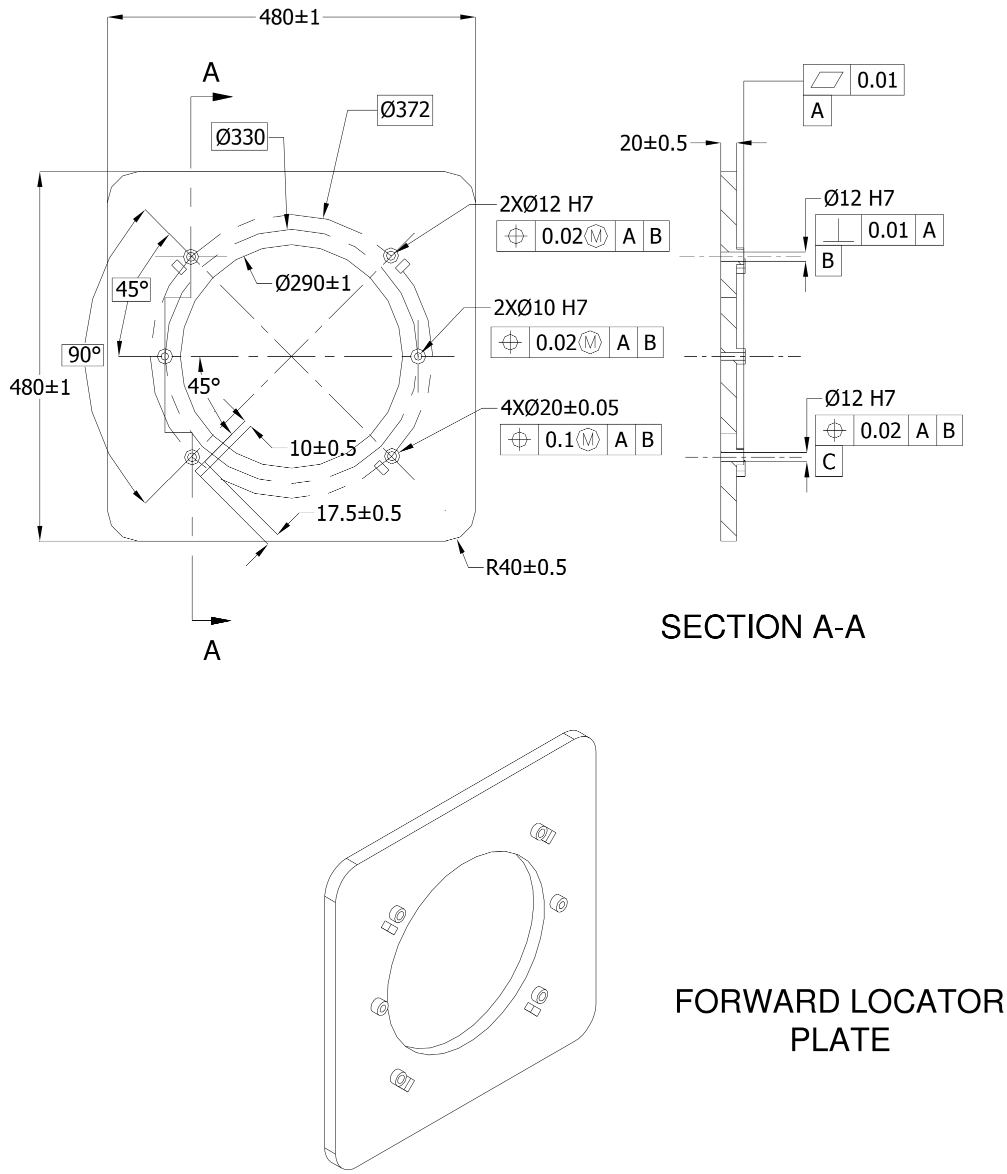


Figure D.2 Assembly Fixture Component Drawings

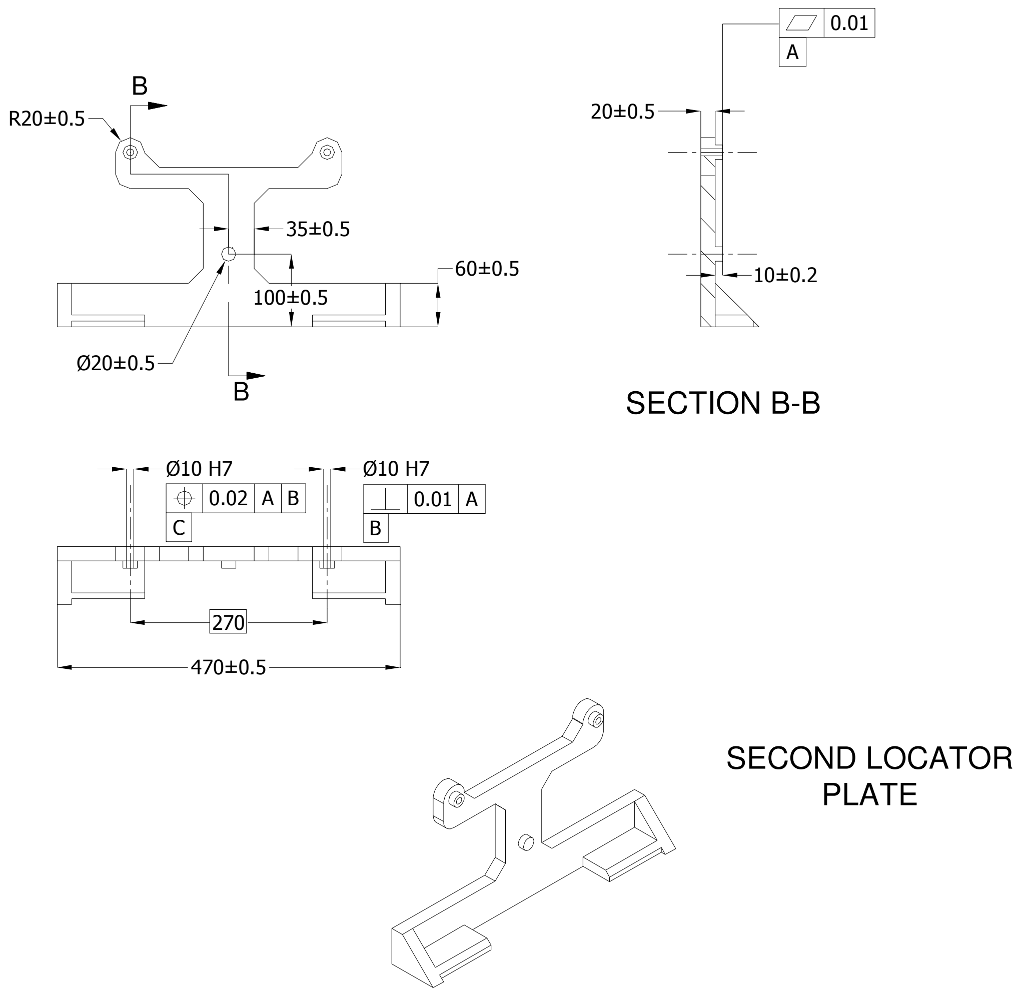
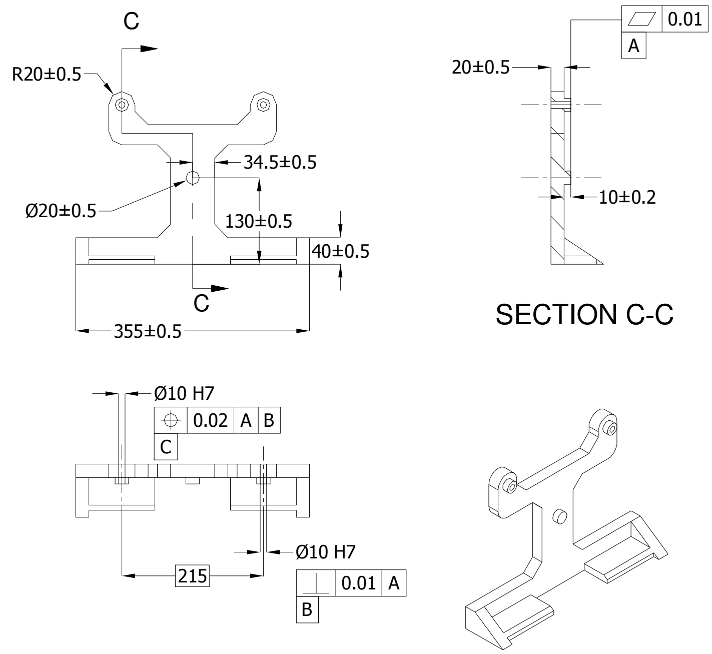
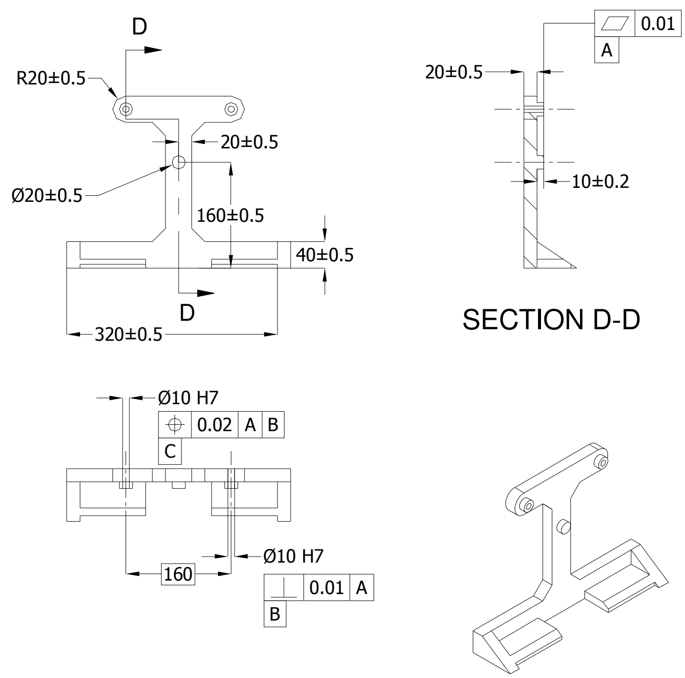


Figure D.2 Assembly Fixture Component Drawings (continued)

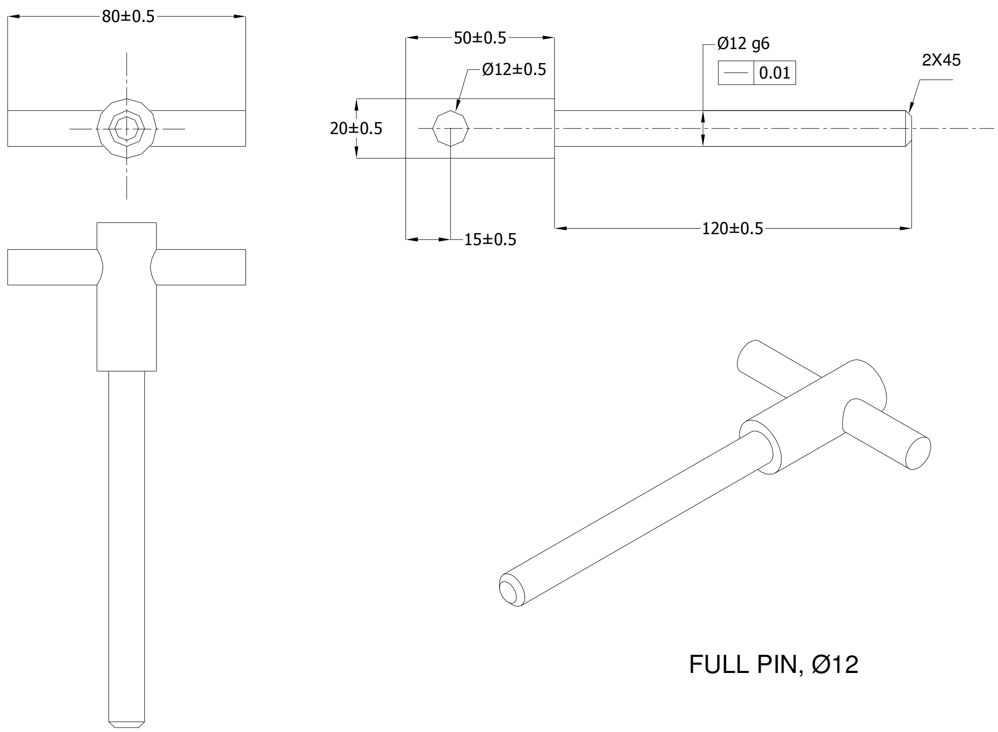


THIRD LOCATOR PLATE

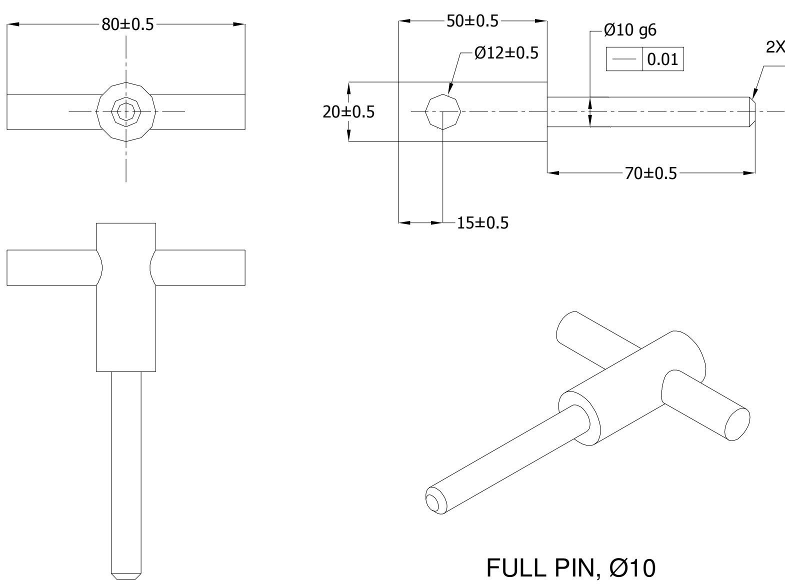


REAR LOCATOR PLATE

Figure D.2 Assembly Fixture Component Drawings (continued)

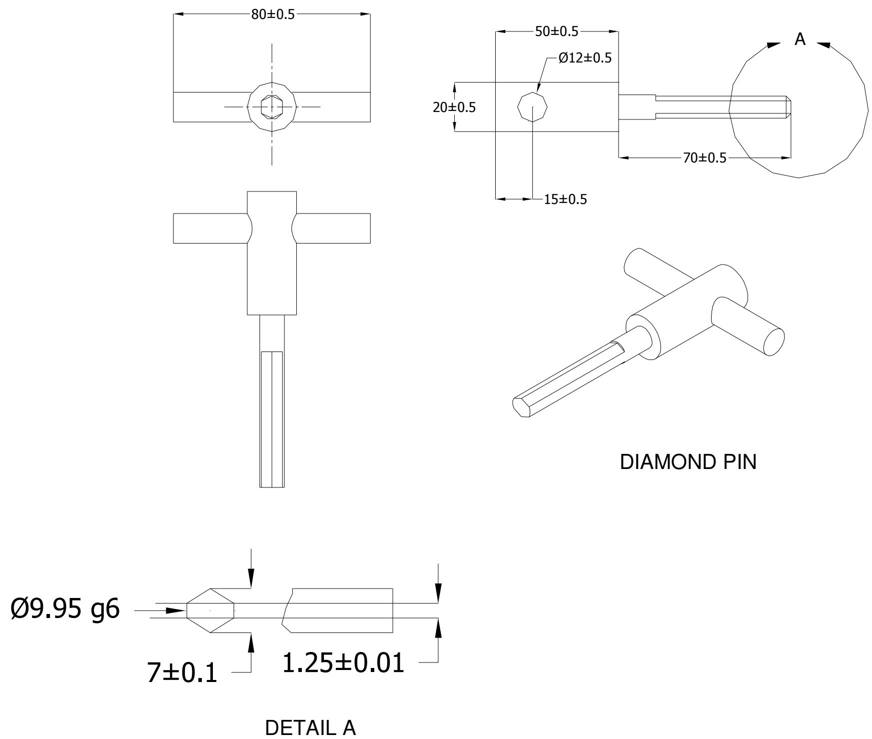


FULL PIN, Ø12



FULL PIN, Ø10

Figure D.2 Assembly Fixture Component Drawings (continued)



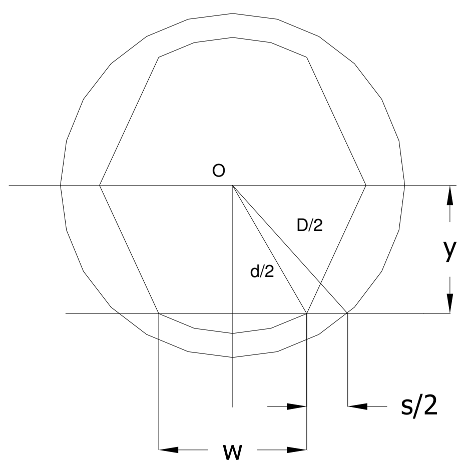


Figure D.2 Assembly Fixture Component Drawings (continued)

Design of the diamond pin is given below;

from geometric relations;

$$y^2 = \left(\frac{d}{2}\right)^2 - \left(\frac{w}{2}\right)^2 \tag{D.1}$$

$$y^2 = \left(\frac{D}{2}\right)^2 - \left(\frac{w}{2} + \frac{s}{2}\right)^2 \tag{D.2}$$

From equations (D.1) and (D.2);

$$2sw = D^2 - d^2 - s \tag{D.3}$$

$$w = \frac{(D-d)(D+d)}{2s} - \frac{s}{2} \tag{D.4}$$

$$w = \frac{[2D-(D-d)](D-d)}{2s} - \frac{s}{2} \tag{D.5}$$

$$w = \frac{2D(D-d)-(D-d)^2}{2s} - \frac{s}{2} \tag{D.6}$$

Since $(D-d)$ is small, $(D-d)^2$ is smaller and can be neglected;

$$w = \frac{D(D-d)}{s} - \frac{s}{2} \tag{D.7}$$

From practice, for w, 1/8 of the nominal D is reasonable unless w is smaller than 0.8 mm. Let s be equal to the position tolerance for hole centers than;

$w = \frac{D}{8}$, $D = 10mm$, $s = 0.02mm$,

Calculating d;

$d = 9.997mm$

Smaller or equal values for the calculated d, are sufficient to provide enough clearance. For the case, $d = 9.995mm$ is assumed.

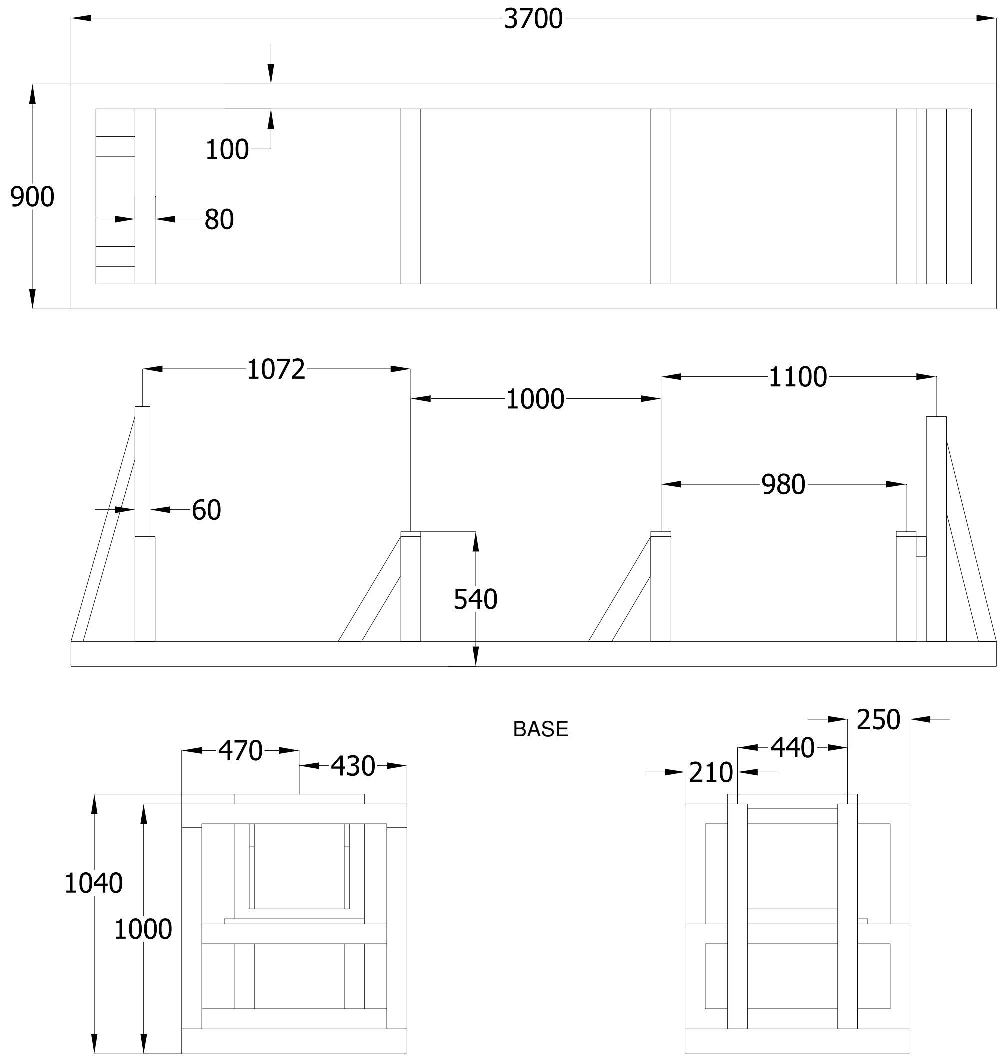


Figure D.2 Assembly Fixture Component Drawings (continued)