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ANALYSIS OF A PRELOADED BOLTED JOINT IN A CERAMIC COMPOSITE COMBUSTOR

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ABSTRACT

This paper presents the detailed analysis of a preloaded bolted joint incorporating ceramic materials. The objective of this analysis is to determine the suitability of a joint design for a ceramic combustor. The analysis addresses critical factors in bolted joint design including preload, preload uncertainty, and load factor. The relationship between key joint variables is also investigated. The analysis is based on four key design criteria, each addressing an anticipated failure mode. The criteria are defined in terms of margin of safety, which must be greater than zero for the design criteria to be satisfied. Since the proposed joint has positive margins of safety, the design criteria are satisfied. Therefore, the joint design is acceptable.

INTRODUCTION

The Bolted Joint

The joint is a common feature in today's mechanical systems and structures. Joints allow groups of smaller parts to be brought together into a much larger assembly. In many cases, the sheer size of the item being built requires a jointed assembly. In other cases, there may be no way to manufacture the item from a single piece. Joints can also offer an economic incentive. It may be cheaper to make a large, complex part from several simpler pieces that are joined together. With ceramics, much of this cost savings comes from reduced machining. Ceramics are in general hard and consequently difficult to machine. Therefore, if the amount of machining can be minimized, via the addition of joints, the overall cost for the part can be lowered.

There are numerous methods of joining, and each has its advantages and limitations. When the joint must be repeatedly disassembled and reassembled, the bolted joint may be the only alternative. This common joint type is the subject of this research. The bolted joint does much more than hold parts together. It is a structural element that must often withstand and transmit substantial loads. It is also often required to seal against high-pressure gases and fluids.

In a bolted joint the bolts generate the clamping force that holds the joint together. To generate this clamping force, the bolts are placed in an initial tension called preload. Correct preload is critical for proper joint function. If the preload is either too high or too low, the joint is much more susceptible to failure. There are numerous methods of generating preload in the bolt. By far the most common method of creating preload is by applying a torque to the bolt. As the bolt is torqued, it is stretched and placed in tension. The clamped members, in opposition to this tensile force, are placed in compression.

Bolted Joint Analysis

There are many documented techniques for analyzing the bolted joint. Any one technique may be best depending on the situation and the criticality of the design. The various techniques vary greatly in their complexity, assumptions, and variables considered. Some of the techniques, like that detailed in the ASME Boiler and Pressure Vessel Code, are based heavily on empirical data. Others are more theoretical in nature. It is the application that dictates the appropriate approach. No one approach is necessarily the "best" for all applications.

In the aerospace industry where minimizing weight is a priority, it is necessary to thoroughly analyze each bolted joint design. Each variable that can affect the strength and reliability of the joint must be carefully considered. Since detailed analysis is conducted, relatively low factors of safety may be used (relative to what is seen in other industries). Since minimized weight is a priority, the time spent on such a detailed analysis is justifiable. In industry though, where weight is not as critical, such detailed analysis may not be warranted. It may be more economical to rely on larger factors of safety while minimizing the time spent on analysis. These larger factors of safety account for variables that may not have been explicitly considered (such as preload and preload uncertainty).

Since this research was done from an aerospace standpoint (that is, weight is considered critical), the approach has a high degree of fidelity. Therefore, most of the variables that effect joint strength and reliability are considered. The analytical approach taken here most closely resembles the guidelines defined in *Criteria for Preloaded Bolts, NSTS 08307.*[1] This guideline was modified and expanded to address the complications introduced by composite materials. The VDI approach detailed by Bickford [2] is also similar to the approach taken in this research. However, factors of safety are not addressed in the VDI approach, and the results are not stated in terms of margin of safety.

Any analysis will require a certain number of assumptions; assumptions that often require sound engineering judgment. The analysis of the bolted joint is no different. Throughout this paper, whenever assumptions are made, they are carefully noted. One important assumption about joint behavior, adopted here, should be highlighted. This assumption, as stated by Bickford [2], is "that joint behavior is fully elastic and linear. In fact, it is often neither of these things and, as a result, will not behave as we have predicted. We'll often have to assume linear behavior to estimate bolt loads or the like because the true behavior is so complex as to defy current theories, except in a few special cases." Several factors contribute to this non-linearity. First, joints inherently involve several members. Where clamped members come together, discontinuities exist. Second, joints typically involve abrupt changes in geometry. These variations in geometry generally exist in order to accommodate the jointed connection itself. A flanged pipe is a common example. In addition, phenomena like prying and embedment can lead to joint non-linearity.

The Bolted Joint, Combustion Devices, and Ceramic Composites

In order to increase the performance of today's liquid-propellant rocket engines, the engineering community is increasingly turning to advanced materials like ceramic matrix composites (CMC's). These materials, with properties and characteristics not found in traditional superalloys, can provide greater engine performance and reliability. Because of their high temperature strength, corrosion resistance, and low specific density, CMC's are well suited for the severe conditions seen in the rocket engine. But, along with the benefits, come significant design and analytical challenges. Before the full benefit of these materials can be realized, these challenges must be effectively overcome.

Combustion devices, like the gas generator (GG), are one area of rocket engine technology that could dramatically benefit from the use of CMC's. Since CMC's maintain their strengths at much higher temperatures, the combustion device can be operated at higher temperatures without active cooling. This results in simplified designs and decreased part counts – a major benefit. But, joints will still be necessary, and joint design poses many challenges. In the case of the ceramic bolted joint, four major issues arise:

1) joint configuration and geometry
2) sealing
3) joint preload
4) interfacing with metallic materials; dissimilar materials

Each will be addressed.

Limits in Scope

Finally, it is important to realize that several factors in joint analysis are not explicitly addressed in this research. This is not to say that these factors are unimportant; in fact, many may be critical to the joint performance. But, they are excluded in order to limit the scope of the research and to concentrate in one specific area of bolted joint analysis. In addition, these factors were also negligible in the final analysis of the Light Weight Gas Generator (LWGG), the combustion device analyzed in this paper. The subject of joint analysis is vast, so it is important that the reader is aware that these factors, and others, exist. The following list provides a brief summary of often important factors in joint analysis. However, these factors are not included in this research:

1) *Various modes of joint failure*
   Two obvious examples are fatigue failure and stress corrosion cracking. If the joint sees cyclic loads or is susceptible to stress corrosion cracking, then these failure modes must obviously be addressed. In the case of the LWGG, these factors were not an issue.
2) *Shear and bending loads*
   Only tensile loads were considered in this analysis. Shear and bending loads, often a major factor in joints, are negligible in the LWGG. They were therefore excluded from this analysis. Although they would add a complicating factor, they could be incorporated into the type of analysis presented here.
3) *Behavior/response of ceramic*
   A stress analysis of the ceramic material was not conducted, per se. Although the crush strength of the ceramic and its compressive stress do play a major role in this analysis, the

full stress state of the ceramic material is not developed. Also, it is assumed that the material properties of the ceramic are fully characterized and available. For the LWGG, the CMC architecture was defined and the mechanical properties were determined through micro-mechanics models and testing.[3]

FACTORS IN JOINT ANALYSIS

The following section reviews several factors common in bolted joint analysis. It also defines four essential design criteria in bolted joint analysis -- the criteria used in the analysis of the LWGG.

Joint Preload and Preload Uncertainty

The bolted joint is distinguished from other joining methods by the presence of preload. Preload is the initial tension placed on the bolts in order to clamp the joint members together. Preload is an essential element in the structural bolted joint and is a major contributor to the joint behavior. The preload is generated at assembly as torque is applied to the nut and bolt. This torque causes the bolt to stretch, placing it in tension. Simultaneously, the joint members see an equal and opposite compression. This compression is the clamping force that holds the joint together.

Although there may be slight plastic deformation in some of the threads when the bolt is tightened, the joint members, in general, respond elastically as the bolts are tightened.[2] Therefore, the bolts behave like stiff springs that stretch as the bolts are tightened. At the same time, the joint members compress, behaving like a much stiffer spring. The bolted joint can be viewed as a group of springs-in-parallel as shown in Figure 1.[2]

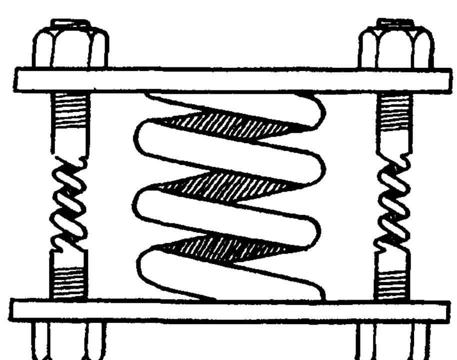

Figure 1.[2] Representation of the Bolted Joint as Springs-in-Parallel.

Preload does much more than hold parts together. It also adds significant rigidity to the joint that, in turn, adds rigidity to the structure. In addition, preload limits the amount of external load experienced by the bolts. Without preload, 100% of the external load goes into the bolts. With preload, only a small percentage of the external load goes into the bolts. This is especially important when external loads are cyclic, since large cyclic loads can dramatically limit bolt fatigue life.

Demonstrating the benefits of preload is a relatively easy task. Applying an accurate preload is another matter. The methods for establishing bolt preload include [4]

1) specifying an installation torque,
2) specifying a turn angle,
3) measuring the stretch in the bolt,
4) applying preload indicating devices, such as preload indicating washers or instrumented bolts, and
5) stretching the bolt with a loading device, such as a hydraulic ram, and subsequently setting the prescribed nut position.

Each of these methods has its advantages and drawbacks. Each has a degree of accuracy and associated cost. In general, there is a direct relationship between accuracy and the associated cost. Simply stated: the greater the accuracy, the greater the cost. Of the methods listed above, excluding the turn angle method, which is used extensively in the structural steel industry, the torque method is by far the most common. Unfortunately, it also has the lowest accuracy. Nevertheless, in order to simplify assembly and constrain costs, this method is used for preloading the LWGG joints.

In the torque method, applied torque is related to bolt preload by the following equation:

$$T = KF_iD\,, \tag{1}$$

where T is the applied torque (in-lb), K is the nut factor (dimensionless), F_i is the bolt preload (lb), and D is the nominal bolt diameter (in).
Thus, for a given torque, bolt diameter, and nut factor, the preload is easily calculated. The difficulty arises in determining the nut factor (K). The nut factor is a general-purpose constant determined through experimentation. The nut factor summarizes all of the factors that affect the relationship between torque and preload in the given experiment including friction, torsion, variation in thread geometry, and plastic deformation of threads.[2] This value, however, is only a mean value based on a series of tests. There may be significant scatter in the preload generated for a given design. This scatter is discussed either tangentially or loosely in several references, e.g., Shigley, NSTS 08307, Bickford, and Sarafin.[5,1,2,4] Shigley presents an example for unlubricated and lubricated bolts. In this presentation, Shigley gives the mean

and standard deviations for the sample distributions. Sarafin subsequently reports the results from Shigley as ±43% and ±25% "scatter" for the unlubricated and lubricated bolt preloads, respectively. In a similar manner NSTS 08307 reports the "scatter" for the unlubricated and lubricated bolt preloads as ±35% and ±25%, respectively. Although it is not stated explicitly in the literature, it appears that the scatter percentage is calculated from plus or minus three times the standard deviation ($\pm 3\sigma$) and the mean value of the bolt preload. This is a reasonable approach since $\pm 3\sigma$ is used commonly in design practice to account for the random distribution of various quantities. At $\pm 3\sigma$ the range includes 99.7% of the population. In this application, since this scatter is so large (±25%), it will have a significant impact on the joint analysis.

In the real world, it is not always practical, feasible, or economical to test each new joint design. Therefore, nut factors often come from tabulated data. However, tabulated data may not always match our design and operating conditions exactly. Juvinall observed "Empirical relationships tell more about what was true, than what will be true".[6] Therefore, good engineering judgment becomes necessary.

Load Factor

Load factor plays a key role in the detailed analysis of the bolted joint. The load factor (also called joint constant and joint stiffness ratio) is a dimensionless constant that indicates the relative stiffness of the bolt and joint. Load factor is given by

$$C = \frac{K_b}{K_b + K_m}, \tag{2}$$

where C is the load factor (dimensionless), K_b is the stiffness of bolt (lb/in), and K_m is the stiffness of clamped members (lb/in).

Calculating the bolt stiffness is relatively straightforward. Bolt stiffness is given by

$$K_b = \frac{A_b E_b}{L_b}, \tag{3}$$

where A_b is the nominal cross-sectional area of the bolt (in^2), E_b is the modulus of elasticity of the bolt (psi), and L_b is the grip length (in).

Calculating member stiffness is not as straight forward or clear-cut, and is an approximation at best. The members of a joint act like springs-in-series. Therefore, the applied load is felt by each joint member. The effective stiffness (K_m) of all joint members is given by

$$\frac{1}{K_m} = \sum_{i=1}^{n} \frac{1}{K_i}, \tag{4}$$

where K_i is the stiffness of individual joint member (lb/in).

Member stiffness for a bolted joint is based on the material compressed around a single bolt, and not the joint as a whole. Several approaches have been developed to estimate member stiffness and are detailed by Bickford and Shigley.[2,5] This research takes the equivalent cylinders approach. In this approach, the member stiffness is approximated by considering a hollow cylinder of the joint material (the inner diameter of the hollow cylinder corresponds to the bolt hole, the outer diameter corresponds to the washer diameter). This equivalent cylinder represents the volume of material around the bolt compressed by the applied load. Using this approach, the stiffness of the individual members is given by

$$K_i = \frac{A_{eq} E_m}{L_m}, \tag{5}$$

where A_{eq} is the cross-sectional area of the equivalent cylinder (in^2), E_m is the modulus of elasticity of individual joint member (psi), and L_m is the thickness of individual joint member.

The load factor represents the fraction of external load carried by the bolt. Since the bolted joint acts like a group of springs-in-parallel, any external load applied to the joint will be divided between the bolts and the clamped members. The portion of external load that goes into the bolts causes an increase in bolt tension. The remaining portion of the external load relieves the clamping force in the members. Since the joint members are typically much stiffer than the bolts, a majority of the external load will go into the members. The stiffness of a typical non-gasketed joint is about five times the stiffness of the bolt.[2] This yields a load factor of approximately 0.17. This value means that for a 1000 lb external load, the bolt carries only 170 lb of the external load. The remaining 830 lb go into relaxing the clamping force in the members.

Another factor often used in joint analysis is loading-plane factor, not to be mistaken with load factor. The loading-plane factor accounts for uncertainty in the location of the applied load. A loading-plane factor of one assumes that loads are applied at the ends of the bolt. A loading-plane factor of zero assumes that loads are applied at the joint interface. For most joints, the loading plane factor falls between these extremes. Unfortunately, loading planes can only be determined experimentally, or approximated with finite element analysis.[2] For this analysis, the loading-plane factor is assumed at one.

Design Criteria

Design criteria provide a means of evaluating and assessing a design. They ensure that design requirements are met. In most cases, there are multiple criteria that must be met simultaneously. Safety criteria are a subset of the design criteria for a particular design. The safety related design criteria generally address the anticipated failure modes and service conditions seen by the design. In the following discussion, the design criteria addressed are safety related criteria.

The joint analysis in this research is built on four design criteria. Each criterion must be satisfied in order for the joint to be acceptable. The four criteria are defined in terms of the margin of safety. Each criterion is satisfied when the margin of safety is greater than or equal to zero.

The "bolt load criterion" is defined as

$$M_{BL} = \frac{S_B}{F_B P} - 1 \geq 0, \tag{6}$$

where M_{BL} is the bolt margin of safety based on the externally applied load, S_B is the bolt yield or ultimate strength (lb), F_B is the design factor of safety on bolt yield or ultimate strengths, and P is the external axial load on the bolt (lb).

The bolt load criterion ensures that bolt strength is adequate. To satisfy this criterion, the bolt strength must be greater than the external load multiplied by the design factor of safety. The bolt load criterion applies to both ultimate and yield strengths; however, only one of them will control. The one that controls depends on the relative values of strength (yield and ultimate) and relative values of factor of safety (yield and ultimate). In the case of the LWGG, yield strength controls. The bolt load criterion guarantees that, even if preload is lost, the bolts will not break (generally a catastrophic failure). In most cases, when this criterion is violated, another criterion has also been violated. But, it adds an additional layer of safety to guard against a catastrophic joint failure.

The "preload criterion" is defined as

$$M_{BP} = \frac{S_B}{P_{P\,max} + CF_B P} - 1 \geq 0, \tag{7}$$

where M_{BP} is the margin for the bolt based on preload and applied load, and $P_{P\,max}$ is the maximum preload considering preload uncertainty (lb).

The preload criterion also ensures that bolt strength is adequate. To satisfy this criterion, the bolt strength must be greater than the maximum axial bolt load multiplied by a design factor of safety. The denominator in Equation (7) represents the maximum axial load experienced by the bolt during service. It is important to note that a design factor of safety is only applied to the external load term and not to the maximum preload term. Since preload uncertainty is so great, as high as ±35% for unlubricated bolts, it is handled separately from the other uncertainties associated with the joint analysis (e.g., the uncertainty in material strength, the uncertainty in loads, etc.). As a result, the design factor of safety is only applied to the external load term. A factor of safety has, in essence, already been applied to the maximum preload term.

Interestingly, the maximum axial bolt load is higher in joints with preload than in joints without preload. This is true all the way to joint separation, at which point, the loads become the same. Although a higher axial bolt load results in a lower margin of safety, the benefits of preload more than offset this decrease in margin.

The "joint separation criterion" is defined as

$$M_{JS} = \frac{P_{P\,min}}{(1-C)F_S P} - 1 \geq 0, \tag{8}$$

where M_{JS} is the margin of safety for joint separation, $P_{P\,min}$ is the minimum preload considering preload uncertainty (lb), and F_S is the design factor of safety on joint separation.

The joint separation criterion ensures that the clamping force is sufficient to prevent joint separation. To satisfy this criterion, the minimum preload must be greater than maximum load seen by the members multiplied by the design factor of safety. The denominator in this equation represents maximum force trying to relax the clamping force in the members. Preventing joint separation in the LWGG is critical because it guarantees a proper seating load on the seals.

The "crush criterion" is defined as

$$M_C = \frac{s_C}{F_C s} - 1 \geq 0, \tag{9}$$

where M_C is the margin of safety on the compressive strength of the member, s_c is the compressive strength of weakest member (psi), F_C is the design factor of safety on ultimate compressive strength, and s is maximum member compressive stress based on bearing area (psi).

The crush criterion ensures that the compressive strength of the ceramic is not exceeded. To satisfy this criterion, the compressive strength of the members must be greater than the compressive stress in the members multiplied by the design factor of safety. For the LWGG, the maximum compressive stress occurs at assembly. This maximum stress occurs at the contact between the outer shell and metallic interface (the contact surface with the Grafoil is 67% greater). Therefore, the margin on crush is calculated at this worst-case location and condition. During operation, pressure loads actually relieve the compressive loads on the C/SiC in this area and result in a higher margin of safety.

This failure mode is generally not an issue in steel joints. Since the area in compression is much greater than the area of the bolts, the compressed steel will have more than enough compressive strength to easily handle the bolt preload. It is possible to produce failure in other materials, such as aluminum, when joining them with steel bolts. Similarly, ceramics, with potentially low compressive strengths, can be crushed by the preload generated in the bolts. The comparatively low compressive strength of the C/SiC was a major issue in the design and analysis of the LWGG.

LWGG OVERVIEW

LWGG Project

The LWGG is part of NASA's overall effort to develop new technologies for the next generation of Reusable Launch Vehicles (RLV's). These RLV's will ultimately replace NASA's aging fleet of Space Shuttles and help to significantly lower the cost of access to space. To meet the needs of the RLV program, these newly developed technologies must result in propulsion elements that are simple, reliable, and robust. The LWGG project, led by Glenn Research Center and teamed with Marshall Space Flight Center and several support contractors, addresses this need by developing and demonstrating an uncooled, lightweight, ceramic combustor.

A primary objective of the LWGG project is to design and test an advanced gas generator using CMC's. Gas generators have traditionally been made of metallic materials. By incorporating CMC's into the design, it is theoretically possible to build a gas generator that is relatively simple, uncooled, and lightweight. Ceramics have frequently served as thermal liners in combustion devices. In these roles the ceramics were not integral structural members and did not carry any structural loads. But, in the case of the LWGG, ceramics are also acting as the primary container of hot gases. Therefore, in addition to carrying much of the thermal load, the outer shell of the LWGG carries the structural, or pressure, load.

CMC's are well suited for the severe conditions seen by a gas generator because of their high temperature strength and corrosion resistance. These materials, with properties and characteristics not found in traditional metals or even superalloys, can provide greater engine performance and reliability. However, along with the benefits, come significant design challenges. In the case of the LWGG, joint design provides multiple challenges including 1) sealing against high-pressure gas, 2) withstanding extremely high temperatures, and 3) incorporating multiple materials, both metallic and ceramic. Since the LWGG is a ceramic combustor with two preloaded joints, it is an ideal candidate for the joint analysis in this research.

The Gas Generator

The gas generator, as the name implies, is a device for creating hot gases. In the world of liquid propellant rocket engines, gas generators create the hot gases that drive the engine's turbopump. The turbopump delivers propellant, both fuel and oxidizer, to the engine's main combustion chamber.

The performance of gas generator cycle engines can be improved two ways: 1) by increasing combustion temperature, and 2) by increasing pressure. In metal structures, the temperature can only be increased within a limit. A point is reached where material strength begins to drop drastically. Most structural metals, including superalloys like A286 and Inconel 718, can be operated at temperatures up to 1,300 °F. Beyond this point, metal parts must be actively cooled, which results in performance losses. Designs requiring active cooling incorporate manifolds and coolant channels to help cool the metal. CMC's can operate at this temperature and higher without active cooling, which simplifies the design and dramatically reduces the overall weight of the design.

Performance can also be improved by increasing pressure. But increased pressure typically results in increased weight due to the heavier structures needed to contain the high-pressure gas. Consequently, the performance benefits gained by increasing pressure must then be balanced against the losses due to increased weight. In the case of CMC's, the specific strength of a CMC is much greater than metals. Specific strength is the ratio of material strength to density. Both strength and weight must be considered in order to get a true comparison of potential weight savings. The high specific strength of ceramics results in much lighter designs when compared to their metallic counterparts. This is important considering that every pound removed from a single-stage-to-orbit propulsion system equals a pound of additional payload.

The true benefit of a high-temperature, lightweight gas generator occurs when it is combined with the other components of the rocket engine assembly or powerpack.[7] Since these components are also acting at a higher temperature, the combined result is a significant increase in performance. Increased powerpack performance and reduced weight contribute to reducing overall vehicle size and cost by improving engine specific impulse, I_{sp}, and reducing overall engine weight. Further, since the ceramics components can operate at higher temperatures, active cooling is not required. This simplifies the design and decreases the number of parts in the design. Therefore, increased engine reliability is likely.

Joint Description

Two concentric shells of carbon/silicon carbide (C/SiC) form the heart of the LWGG. These shells act as the combustion chamber where the propellants are burned to generate hot gases. The inner shell has an outside diameter slightly smaller than the inner diameter of the outer shell. The inner shell can therefore slide inside the outer shell during assembly. The inner shell acts primarily as a thermal barrier for the outer shell. If the inner shell is damaged or eroded during operation, it can be easily removed and replaced. The inner shell also extends past the length of the outer shell to thermally protect the interface hardware. The inner shell is a straight cylinder of C/SiC approximately 8 inches in length, with a 2.3 inch outer diameter and a 3/16 inch wall thickness. The inner shell is sandwiched between an injector and a nozzle simulator. A ceramic braided rope fits into a chamfer on both ends of the inner shell to hold it in place during operation. An assembly view of the LWGG is shown in Figure 2. The outer shell is also a cylinder of C/SiC but with flared ends. The flared ends act as a clamping surface for bolting the shell to an injector and a nozzle simulator. The outer shell is approximately 6.25 inches in length, with a 2.75 inch outer diameter and a ¼ inch wall thickness. In a metallic combustion chamber, these flared ends would be standard 90-degree flanges. But in this application, limits in the braiding process restrict the flare angle to approximately 20 degrees. A metallic chamber would also typically have bolt holes machined in the flanges. Holes are avoided in fibrous composites because broken fibers result in losses in material strength.

The outer shell attaches to a gas/gas injector via a metallic interface. This interface has a machined surface that mates perfectly with the flared end of the outer shell. It also contains grooves for both the primary seal and a ceramic braided rope. The injector interface has a simple manifold to allow water-cooling of this metallic component. The manifold inlet is separated from the exit by 180 degrees. In a flight design, this interface would be integrated with the injector design. But for this program, due to limited budget and schedule, an existing gas/gas injector was selected for testing. Since the joint design was non-standard, an interface was required. Figure 3 shows the joint after the bolts have been tightened with the seals properly seated.

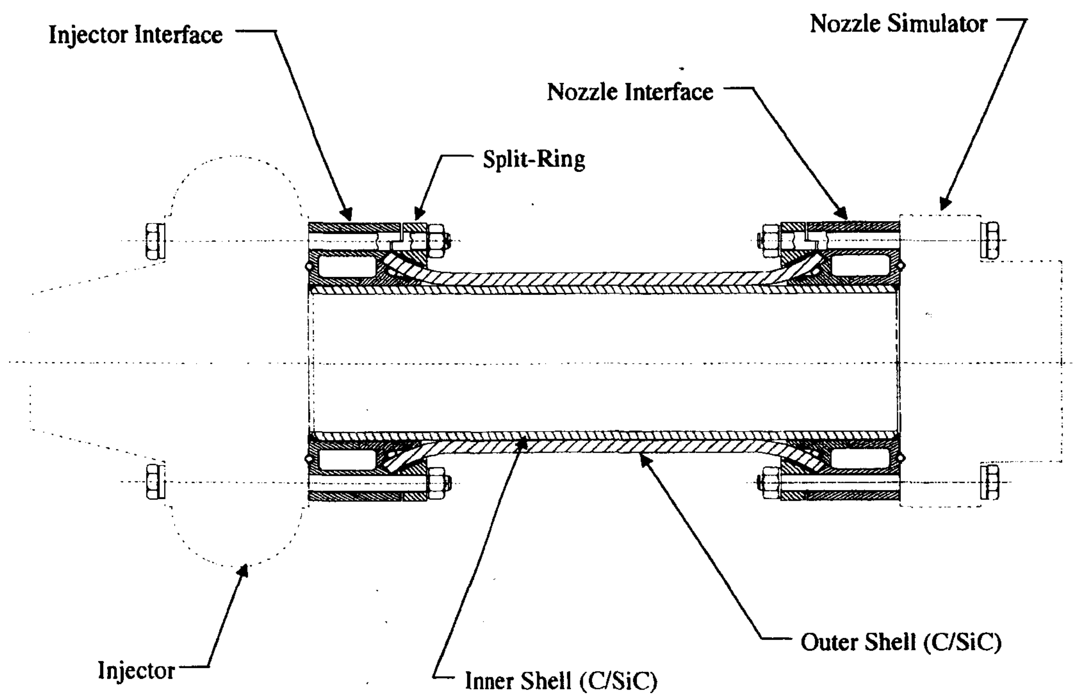

Figure 2. LWGG Assembly.

The aft end of the outer shell attaches to a nozzle simulator in a manner nearly identical to that used for the injector, i.e., via a metallic interface. The flared end of the outer shell attaches to the nozzle interface, which in turn, attaches to the nozzle simulator. The nozzle simulator represents the flow resistance produced by an engine's turbopump. Again, in the interest of budget and schedule, an existing nozzle simulator was selected for program testing. In a flight design, the nozzle interface would be integral with the turbopump attach point.

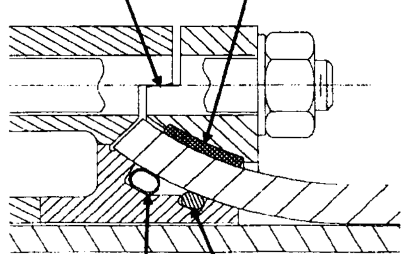

Figure 3. LWGG Joint.

A metallic split-ring clamps the flared ends of the outer shell to the two interfaces. The two-piece split-ring must be brought together around the outside of the outer shell during assembly. Otherwise, the split-ring would not pass over the flared ends of the shell. Two dowel pins help locate the two halves relative to one another. The split-ring is isolated from the outer shell's surface by a thin layer of Grafoil®. The Grafoil®, a flexible gasket of graphite, acts to evenly distribute the clamping load on the outer shell's surface. It rests in a small recess to hold it in place during assembly and operation. Although Grafoil® is often used as a high-temperature seal, in this application it only acts as a compliant member.

A thin-walled, metal o-ring acts as the primary seal for the joint. Having a thin wall minimizes the seating load for the seal. A low seating load was desired in order to minimize the line load experienced by the ceramic outer shell. The primary seal is protected from the hot gases by a braided ceramic rope. The rope, composed of Nextel fibers, acts as a first line of defense against the hot gases generated in the combustion chamber. Although the braided rope may actually provide a certain amount of sealing, it is not a design requirement for this joint.

Finally, standard metal o-rings create the seals between the injector and injector interface, and the nozzle simulator and nozzle interface. These seals are needed only for the test configuration and would not exist in a flight-type design.

Each joint is held together with eight ¼-28 NAS hex head bolts. The bolts pass through a series of through holes that are common to each piece. A high strength nut secures each bolt. Each bolt is torqued to 75-80 in-lb to generate a nominal preload of 2000 lb. A lubricant, Braycote® 602, is added to each threaded bolt to more accurately control the preload in the bolt. As the bolts are tightened, the split-ring is drawn towards the interface. A stepped surface, or pilot, helps guide the split-ring into the interface. As the joint is tightened, the o-ring compresses, and the braided rope and Grafoil® compress. Once the joint is fully preloaded, a 0.035 inch gap exists between the split-ring and the interface flange. This gap ensures that the compressive preload passes through the flared outer shell and not directly into the interface hardware. If this gap did not exist, the outer shell would not be properly clamped and the seals properly seated.

LWGG JOINT ANALYSIS

This section summarizes the results of the LWGG joint analysis. The detailed calculations for this analysis are presented by Hissam.[8] Steps 1-5 generate and define the information needed for calculating the joint margins of safety. Step 6 calculates the four margins of safety that define the design criteria for the LWGG. These steps are

1. determine service loads on the joint and bolts,
2. calculate load factor,
3. calculate preload losses,
4. calculate maximum and minimum preloads,
5. determine strengths (allowables) for bolts and ceramic, and
6. calculate margins of safety.

Determine Service Loads on the Joint and Bolts

The joints of the LWGG see two primary loads: 1) a blow-off load and 2) a seal-seating load. Since both loads are tensile in nature, and since both act along the central axis of the joint, this step is relatively straightforward. If shear loads had been present, or if the loads had acted off-axis, then this step becomes much more complicated.

Chamber pressure, acting on the LWGG injector, creates the blow-off load. A 1000 psi chamber pressure acts on a surface defined by the injector seal diameter of 2.88 inches. The result is a blow-off load of 6514 lb.

The seal-seating load is the force required to properly compress the seals. Without this compression, the seals will not function properly. The percent compression and stiffness of the seal determine the magnitude of this force. Since each LWGG joint contains four seals, each must be considered. The seal seating load for the four seals is 2679 lb. The combined blow-off load and seal seating load for each LWGG joint is 9193 lb (or 1149 lb/bolt).

Calculate Load Factor

The load factor for a bolted joint is given by Equation (2). In order to calculate the load factor, both the bolt stiffness and the member stiffness must be determined. The stiffness of the bolt is given by Equation (3). For a NAS 6704 ¼-28 bolt, the stiffness is 4.96E5 lb/in. The effective stiffness of all joint members (K_m) is given by Equation (4). The stiffness values for the individual members (K_i) are based on the equivalent cylinder approach and are calculated with Equation (5). The Grafoil stiffness is the only exception. It is based on the total area of contact between the Grafoil® and the metallic split-ring. Since the Grafoil® has a much lower stiffness than the other members, it dominates the overall stiffness of the joint members (spring stiffnesses add by reciprocals). The result is a relatively low joint stiffness of 4.56E5 lb/in. A low joint stiffness, in turn, leads to a relatively large load factor of 0.52. As a result, when an external load is applied to the joint, a greater percentage of this load will go into the bolts instead of reducing the compressive load in the clamped members. If the Grafoil® had not been present in the joint, the load factor would have been significantly lower.

Calculate Preload Losses

A bolted joint will commonly experience losses in preload after the joint has been assembled. These losses can adversely affect joint performance and must be accounted for. Some of these losses occur immediately after assembly. Others occur over time and during operation of the joint. Three such factors that result in preload loss are embedment, creep, and elastic reactions. A full description of these types of losses is provided by Bickford.[2] For this analysis, these losses are approximated at 30%. The presence of the Grafoil®, which acts like a gasket material, accounts for much of this potential preload loss.

Another possible source of preload loss occurs when the joint experiences temperature changes. Differences in the CTE between the bolts and clamped members result in this preload loss. This can lead to either an increase or decrease in preload depending on the relative values of CTE. In the case of the LWGG, the bolts lose preload when the joint temperature increases. Since the NAS 6704 ¼-28 bolts (composed of A286 stainless steel) have a greater CTE than the C/SiC, the bolts expand more when heated. The result is a loss in preload. The LWGG joint sees a maximum temperature increase of 400 °F.[9] This temperature increase occurs only in a section of the joint. The remainder of the joint remains cool due to its proximity to the coolant manifold. Assuming a maximum temperature increase of 400 °F over a length of 0.335 inches, each bolt then experiences a 133 lb preload loss.

Calculate Maximum and Minimum Preloads

The maximum and minimum preloads are calculated from the torque equation, Equation (1). The specified installation torque for the LWGG is 75 to 80 in-lb. The high value (80 in-lb) is used to calculate the maximum preload. The low value (75 in-lb) is used to calculate the minimum preload. Since a lubricant is applied to the bolts at installation, a ±25% preload uncertainty is applied. With a nominal bolt diameter of 0.25 inches, a nut factor of 0.16, and a +25% preload uncertainty, the maximum preload is 2500 lb. With a nominal bolt diameter of 0.25 inches, a nut factor of 0.16, and a −25% preload uncertainty, the minimum preload is 1406 lb. However, when calculating minimum preload, the preload losses must also be considered. When these losses (calculated in step #3) are included, the minimum preload drops to 851 lb.

Determine Strengths (Allowables) for Bolts and Ceramic

The bolt strength is found by multiplying the bolt material yield and ultimate strengths (psi) by the tensile stress area. This provides the bolt strengths (lb) at room temperature (70 °F). Since the LWGG bolts experience a maximum temperature of 470 °F, the bolts will experience approximately a 10% loss in strength.[9] For a NAS 6704 ¼-28 bolt at this temperature, the yield strength is 3931 lb and the tensile strength is 5242 lb. Both of these strengths are bolt allowables; however, only one will control. The controlling value is determined by dividing the strengths by their respective design factors of safety. The lowest value controls. For the LWGG, this is yield strength. Therefore, the bolt allowable for this analysis is 3931 lb. The design factors of safety for the LWGG analysis, 1.25 for yield strength and 1.4 for ultimate strength, are obtained from *NASA-STD-5001, Structural Design and Test Factors of Safety for Spaceflight Hardware.*[10]

The compressive strength for the C/SiC material is 12,500 psi, a value that was determined from testing. Since C/SiC maintains its strength at very high temperatures, there is no loss in strength at the LWGG operating temperatures. The maximum compressive stress on the C/SiC is found by dividing the maximum bolt preload (of all eight bolts) by the contact area between outer shell and nozzle interface. The result is a compressive stress of 8801 psi.

Calculate Margins of Safety

Applying Equations (6-9) and inserting the values found in steps 1-5, the calculated margins of safety for the LWGG joints are

$M_{BL} = 1.74$
$M_{BP} = 0.21$
$M_{JS} = 0.10$
$M_C = 0.01$

Since each margin is greater than zero, each design criterion is satisfied. The low margin of safety on crush is a result of the relatively low compressive strength of the C/SiC. This is often the case with composite materials. Therefore, it is important that the crush strength is checked when composite materials are present in the joint.

Because margin of safety values must be interpreted in relationship to the design factors of safety, the realized factors of safety were computed and are

bolt load = 3.42 $\geq$ 1.25
preload = 2.40 $\geq$ 1.25
joint separation = 1.54 $\geq$ 1.4
crush = 1.42 $\geq$ 1.4

These values verify that the design requirements for the LWGG joints are satisfied.

Although the current design satisfies all margins of safety, this was not always the case. Early in the design cycle, a negative margin on crush was continuously a problem. In order to obtain a positive margin, three design changes were required:

1) a lubricant was added to the bolts in order to decrease preload uncertainty. This decreases the value of the maximum preload.
2) the joint design was modified to increase the bearing area on the C/SiC. The loads are therefore spread over a greater area.
3) the maximum specified applied torque was lowered from 90 in-lb to 80 in-lb. This change also decreases the value of the maximum preload. The minimum specific bolt torque (75 in-lb) could not be lowered because joint separation becomes an issue.

These changes ultimately resulted in a positive margin of safety on crush.

CONCLUSIONS

The proposed joint for the LWGG has positive margins of safety. Consequently, the design criteria are satisfied, and the joint design is acceptable. The comparatively high load factor (0.52) calculated for the LWGG joint is due to the presence of the Grafoil® in the joint. Originally, Grafoil® was added to the joint to act as a compliant member. In this capacity, it accounted for tolerance mismatch in the parts and helped to evenly distribute load of the C/SiC surface. But, it also resulted in the relatively large load factor. This had the unexpected result of increasing the range of acceptable external loads for the joint. If the Grafoil® had not been present, the load factor would have been lower and the margin on joint separation likely not satisfied. In most cases, a lower load factor is desired. But, for the LWGG, since the bolts had ample margin, the larger load factor is actually a benefit. If the bolts had not had such a large margin, this may not have been the case.

Although critical to joint performance, preload adds a significant complication to joint analysis. This complication is the result of the preload uncertainty associated with the bolted joint and the potential of preload losses. Nevertheless, it is still possible to design joints that are safe and meet design requirements. For aerospace applications, where low factors of safety are generally required, a successful design solution requires a detailed and accurate analysis that considers the many variables affecting joint behavior. Such a detailed analysis minimizes the overall uncertainty associated with the joint, thereby allowing the use of relatively small factors of safety. When larger factors of safety are acceptable, a less comprehensive analysis may be appropriate. The larger factors of safety account for the greater degree of uncertainty associated with the joint.

REFERENCES

[1]NASA Johnson Space Center, *Criteria for Preloaded Bolts, NSTS 08307*, 1998.

[2]Bickford, J.H., *An Introduction to the Design and Behavior of Bolted Joints*, 3rd ed., Marcel Dekker, New York, 1995.

[3]Casey, S., and Buesking, K., *Braid Architectures for C/SiC Gas Generators*, NASA Glenn Contract NAS3-98073, 2000.

[4]Sarafin, T. P., *Design and Analysis of Fastened Joints for Aerospace Engineers–Short Course*, Rev. D, 1999.

[5]Shigley, J.E., and Mischke, C.R., *Mechanical Engineering Design*, 5th ed., McGraw-Hill, New York, 1989.

[6]Juvinall, R. C., *Engineering Considerations of Stress, Strain, and Strength*, McGraw-Hill, New York, 1967.

[7]Lewis Research Center, *Project Plan for the Light-Weight Gas-Generator Combustor Assembly for RLV Tasks in NRA 8-21*, 1998.

[8]Hissam, D.A., *Characterization and Analysis of a Preloaded Bolted Joint in a Ceramic Composite*

Combustor, Thesis, University of Alabama, Huntsville, 2001.

[9]Marshall Space Flight Center, *Light Weight Gas Generator Stress Analysis Final Report,* Structural Analysis Group Report No. ED22-00-125, 2000.

[10]National Aeronautics and Space Administration, *NASA-STD-5001, Structural Design and Test Factors of Safety for Spaceflight Hardware,* 1996.