https://ntrs.nasa.gov/search.jsp?R=19930003231 2020-03-17T10:26:33+00:00Z

IN-39
P.12

NASA
Technical
Memorandum

NASA TM - 108378

SHEAR JOINT CAPABILITY VERSUS BOLT CLEARANCE

By. H.M. Lee

Structures and Dynamics Laboratory
Science and Engineering Directorate

October 1992

(NASA-TM-108378) SHEAR JOINT CAPABILITY VERSUS BOLT CLEARANCE (NASA) 22 p

N93-12419

Unclas

G3/39 0129296

National Aeronautics and
Space Administration

George C. Marshall Space Flight Center

TABLE OF CONTENTS

	Page
INTRODUCTION	1
BACKGROUND	1
ASSUMPTIONS	2
ANALYSIS OF SINGLE BOLT	4
JOINT CAPABILITY ANALYSIS	10
CONCLUSIONS	13
REFERENCES	14
APPENDIX A – JOINT CAPABILITY POLYNOMIAL	15
APPENDIX B – SHUTTLE HARDWARE TEST CASE	16

ii INTENTIONALLY BLANK

PRECEDING PAGE BLANK NOT FILMED

LIST OF ILLUSTRATIONS

Figure	Title	Page
1.	Distribution of bolt position	3
2.	Initial bolt-to-hole configuration	3
3.	Bolt and bolt-hole geometry	4
4.	Analysis flow chart	6
5.	Calculation of V_{cl}	7
6.	Shear load for bolt diameter at 0.196 in	7
7.	Shear load for various bolt diameters	9
8.	Shear load versus bolt-hole clearance	9
9.	Joint loading history	10
10.	Joint capability (equation (13))	13

TECHNICAL MEMORANDUM

SHEAR JOINT CAPABILITY VERSUS BOLT CLEARANCE

INTRODUCTION

Almost every component designed for space flight utilizes bolted joints to transfer shear loads. Such joints are either analyzed as slip resistant ones in which shear is carried through friction imparted by the preloaded bolts, or they are analyzed as shear/bearing ones. The latter group assumes no friction and is generally limited by the bolt shear strength or the joint bearing capability.

A lot of friction testing has been done by Marshall Space Flight Center (MSFC)[1] and other reputable aerospace organizations.[2] Results of such testing have clearly shown substantial slip resistance for most aluminum joints even when the surfaces were hardened (anodized, etc.). Although friction is a fact of life for actual hardware, many are reluctant to use it because of its unpredictability. Good design practice[3] calls for tight fits around each bolt in order to transfer high shear loads. In some cases, these joints are supposed to be removable and thus must be drilled on assembly. The cost of such procedures can become prohibitive when hardware is manufactured in different locations and is assembled and reassembled numerous times. Such is the case of many Spacelab payloads. The standard analytical approach for hardware which does not meet the tight fit requirements is to analytically allow only one fastener in the joint grouping to transfer all the shear loading. Obviously this is a conservative yet simplified approach to the problem. For that reason, this report is written in an attempt to establish a practical (but still somewhat conservative) analytical method by which the capability of shear joints can be ascertained as a function of the bolt-hole clearance for those cases where high-strength steel fasteners are utilized in aluminum components. Though the results presented in the report probably should not be used for an initial design, hardware that already exists (such as short-life secondary Spacelab structure) could be shown to have greater capability.

BACKGROUND

The majority of space-flight hardware is assembled using high-strength steel fasteners with tensile ultimate strengths between 140 and 180 ksi. These bolts are not only strong but are ductile, have high fatigue life, and exhibit good fracture toughness properties. The threads are all class 3 with a pedigree that guarantees an "A-basis" yield and ultimate strength. This fact alone allows for very large preloads and associated clamping forces on the joint. If slippage does occur, the clamping force is still formidable.

Since every pound of structure affects the payload that launch vehicles can place in orbit, most flight components are constructed of aluminum. Even though the aerospace aluminum hardware is light and reasonably strong, the modulus of elasticity (E) and the bearing yield (F_{bry}) are much less in magnitude than they are for the steel bolts. Thus, the differences in material properties assure the potential for the bearing joint design to more evenly distribute shear loads to each fastener, should slippage result.

When and if slippage of a shear joint does occur, one thing that is critical is the bolt-hole clearance. As clearance around each bolt increases, the probability of a single bolt picking up a greater portion of the load also increases. This will no doubt continue until that particular bolt fails in shear and the whole joint unzips. The positive thing about such joints, however, is that if high strength fasteners are utilized with aluminum components, the bolts will begin to elongate the holes through plastic bearing deformation. When this happens, some of the hole clearance can be absorbed and the potential for other fasteners in the joint to carry load is enhanced. The purpose of this report is to analytically show how bolt-hole clearance affects shear joint capability on typical aerospace aluminum components assembled with high strength steel fasteners.

ASSUMPTIONS

The contact stresses caused by the loading of elastic bodies such as ball bearings, trunnions, rail tracks, etc., were investigated originally by H. Hertz.[4] He developed the mathematical theory for the surface stresses and deformations produced by such loading between curved elastic members. The results of his analytical work are now supported by testing. When the circular shank of a high-strength steel bolt contacts an oversized bolt hole in an aluminum part, Hertzian stress and deformation most certainly occur. The Hertzian elastic contact theory is, therefore, the first assumption made in this analysis process. Using this concept provides a conservative, yet more simplified method of dealing with a truly nonlinear problem.

The second assumption is that, as stated previously, the joint is a typical aerospace one with steel bolts and aluminum abutments. This fact leads one to assign a modulus of elasticity (E) of 30.0×10^6 lb/in^2 and 10.0×10^6 lb/in^2 for steel and aluminum, respectively. Likewise, the Poisson's ratio for these materials was taken as 0.30 and 0.33.

Another conservative assumption was used in determining the stiffness values to be assigned for each bolt to abutment in shear. It can be shown from the Hertzian theory that the effective stiffness between these two bodies decreases as the loading and associated deformation are increased. This means, in theory, that a fastener in early contact will exhibit a greater stiffness than one which has already deformed the bolt-hole material. By using a constant stiffness at all bolt attachments, the initial contact bolt will be conservatively loaded. The analysis technique will predict an earlier-than-actual bolt failure.

The next boundary condition is the assumption that the material stiffness between each bolt hole is infinite. This, in effect, again forces more load onto the bolt which was first in contact, and an earlier-than-actual failure would be expected.

The final assumption deals with the initial position of the joint. Analytically, it was assumed that the joint would have two or more fasteners in the interface. If that is the case, then one fastener was stated to be initially in contact with the bolt-hole material (-3σ occurrence for a normal distribution). That fastener is referred to as the "key bolt" in later portions of this paper and requires a joint displacement of zero before picking up load. Since both abutment plates can have the same hole clearance, the "worst case" situation exists when a fastener is positioned such that a displacement of 2 CL ($+3\sigma$) is required prior to picking up shear load. Figure 1 shows graphically the normally distributed bolt positions with the "key bolt" as a -3σ assumption, and the "worst case" position as a $+3\sigma$. With this established as an estimate of bolt positioning, all fasteners except the "key bolt" were analytically located at a distance equal to the mean bolt-hole position. In other words, contact of all remaining bolts would take

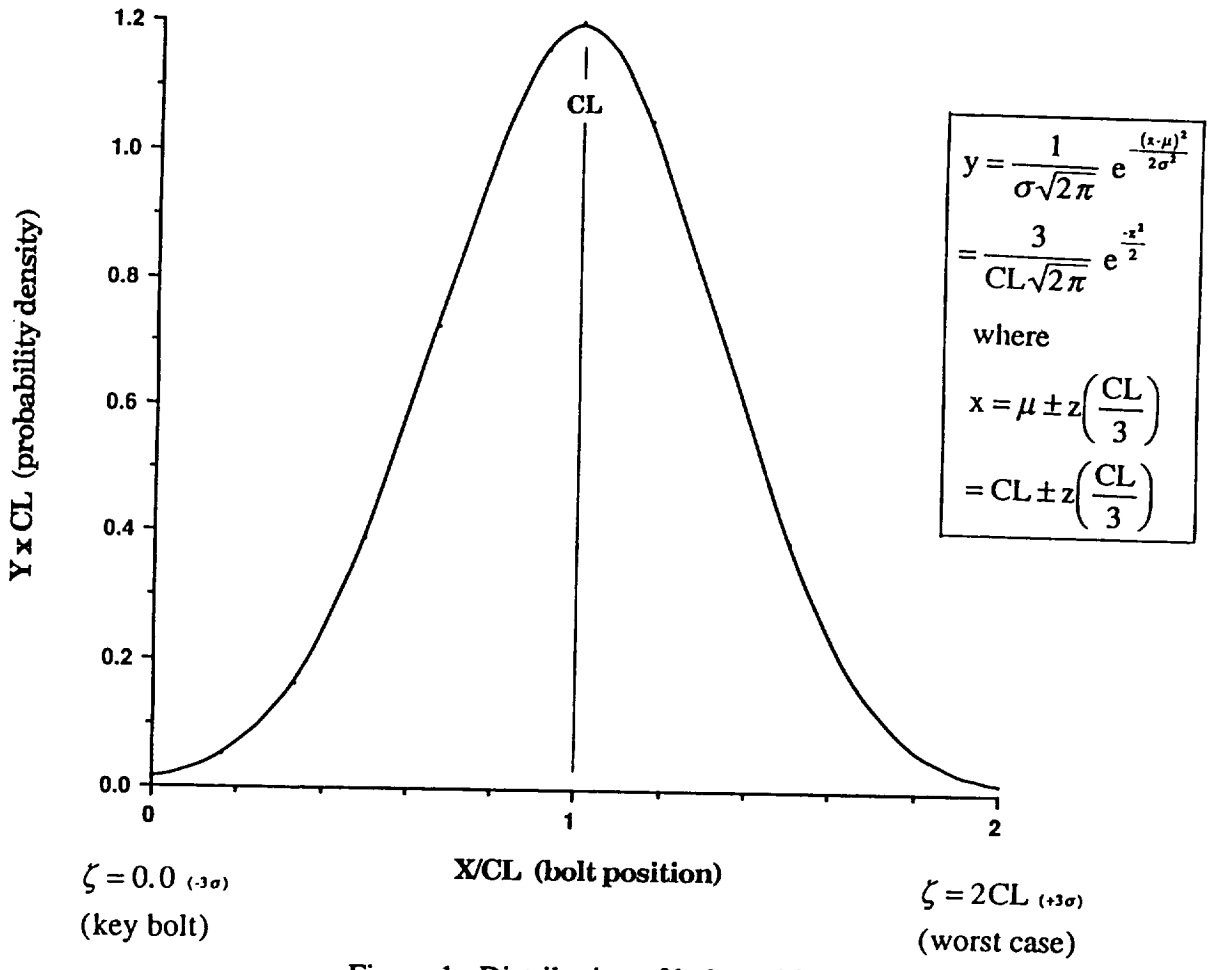


Figure 1. Distribution of bolt position.

place only after a joint slippage equal to the design clearance (*CL*) had occurred. This is certainly a reasonable approach since actual hardware does have some distribution on the position of each bolt in each hole. Figure 2 relates this pictorially.

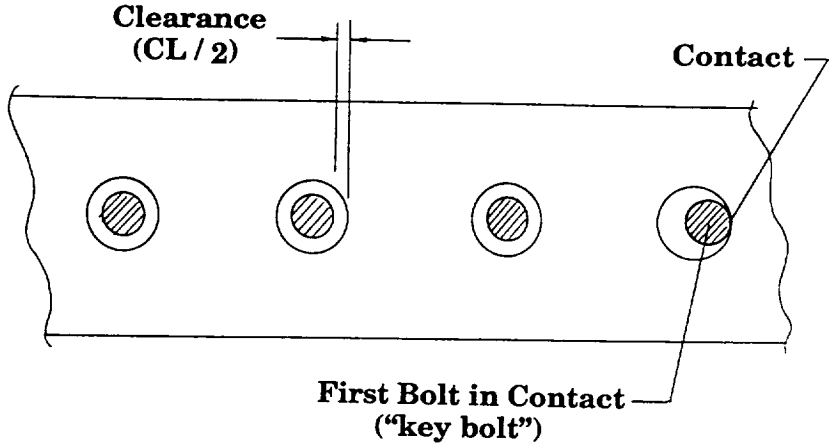


Figure 2. Initial bolt-to-hole configuration.

ANALYSIS OF SINGLE BOLT

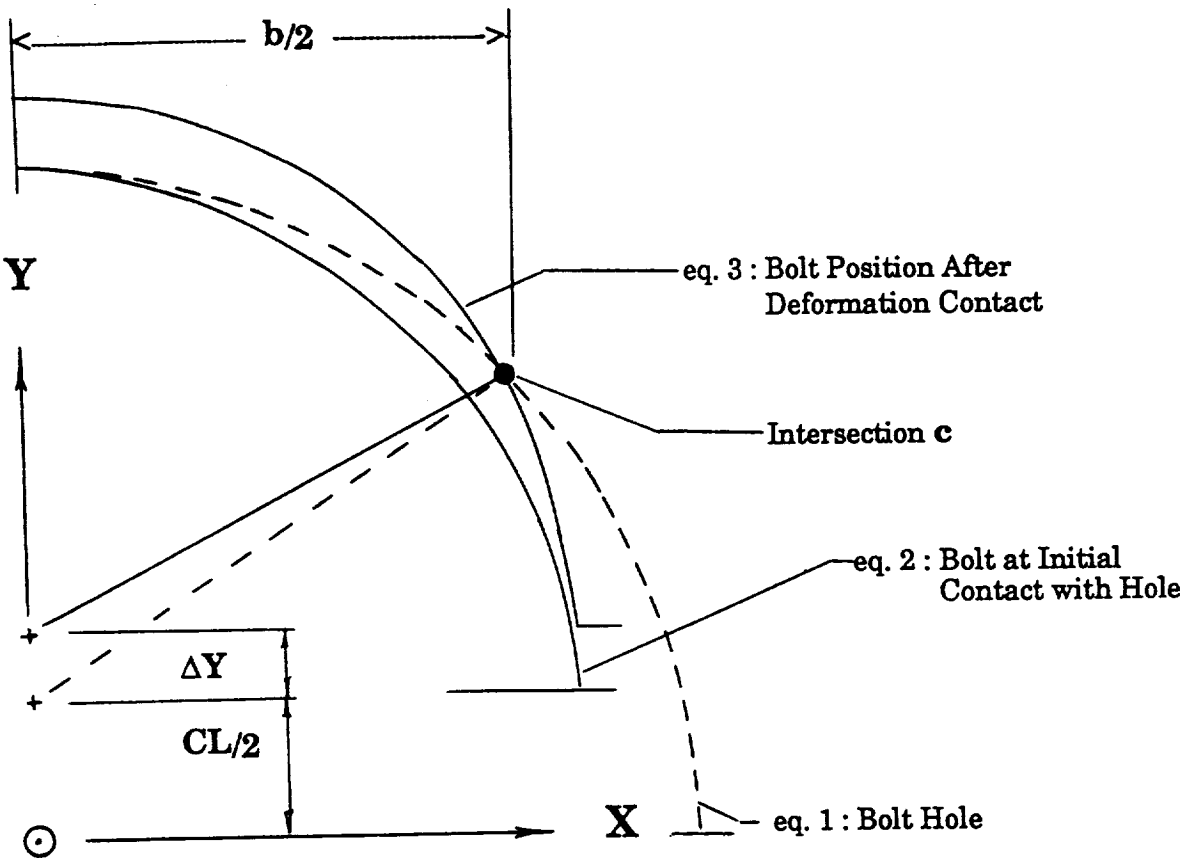
The first step in the analysis process is to develop the equations for the relative deformation and associate stiffness for the bolt-to-bolt hole material interface. Equations representing the geometric position of both items are required and are shown in figure 3.

The specific equations are shown below:

$$X^2+Y^2=D_b^2/4 \tag{1}$$

$$X^2+(Y-CL/2)^2=D_b^2/4 \tag{2}$$

$$X^2+(Y-CL/2-\Delta Y)^2=D_b^2/4 \tag{3}$$



where:
- D_b= bolt shank diameter
- D_h= bolt hole diameter
- CL= bolt hole clearance (D_h-D_b)
- ΔY= relative deformation of bolt & bolt hole
- b= width of Hertzian contact

Figure 3. Bolt and bolt-hole geometry.

Define the coordinates of the intersection point (c) in figure 3:

$$X = b/2 ;$$

this leads to

$$Y = \frac{1}{2}\sqrt{D_h^2 - b^2} .$$

Equation (3) then becomes:

$$\left(\frac{b}{2}\right)^2 + \left\{\frac{1}{2}\sqrt{D_h^2 - b^2} - \frac{CL}{2} - \Delta Y\right\}^2 = \frac{D_b^2}{4} ,$$

which can be solved for the relative deformation between the bolt and the bolt hole (ΔY):

$$\Delta Y = \frac{1}{2}\left\{\sqrt{D_h^2 - b^2} - \sqrt{D_b^2 - b^2}\right\} - \frac{CL}{2} , \tag{4}$$

from equation (4) the effective Hertzian stiffness can be simply calculated as:

$$K_{hz} = \frac{V}{\Delta Y} , \tag{5}$$

the quantity b is the width of contact and is defined by the Hertzian equation:[4]

$$b = 1.6\sqrt{\frac{VD_hD_b}{TCL}\left[\frac{(1-\nu_b^2)}{E_b} + \frac{(1-\nu_h^2)}{E_h}\right]} , \tag{6}$$

choosing the aforementioned values for modulus of elasticity (E) and Poisson's ratio (V), this quantity becomes

$$b = 0.00055297\sqrt{\frac{VD_hD_b}{TCL}} , \tag{7}$$

where T = thickness of aluminum abutment plate.

At this point, input variables such as bolt-hole diameter (D_h), bolt shank diameter (D_b), bolt-hole clearance (CL), aluminum joint thickness (T), and an initial arbitrary shear load (V_i = 100 lb.) can be placed into equation (7). This estimates the width of the initial rectangular contact area (b_i). Utilizing equation (4), the initial relative deformation (ΔY_i) can then be computed. Then through equation (8), the shear force (V_e) required to embed the bolt shank diameter (D_b) completely into the aluminum abutment plate can be determined.

$$V_e = \left(\frac{D_b}{b_i}\right)^2 \cdot V_i . \tag{8}$$

Equation (4) can again be used to determine the deformation (ΔY_e) associated with the shear force V_e, by setting $b_e = D_b$. The final step is then to calculate the shear force(V_{cl}) required to move the bolt shank a distance equal to the bolt hole clearance. This effectively equated ΔY to CL. The shear force V_{cl} now reveals the magnitude of the force it will take to move the joint to a position in which all other fasteners are in contact with each respective bolt-hole surface. Figure 4 defines in flow chart form the process of defining V_{cl} as described above.

An example of how this approach can be accomplished under specific input parameters is tabulated in figure 5. Analytical results show it would take a shear force(V_{cl}*) of around 18 lb to overcome a 0.001-in bolt-hole clearance and about 1,678 lb to overcome a 0.020-in clearance for a 0.196-in steel bolt in a single 0.100-in aluminum abutment plate. Recognizing that a true shear joint will normally have two abutment plates, the shear force necessary would be reduced to one half if both plates are 0.100-in thick. To counteract this, however, is the fact that the deformation necessary to overcome the clearance in two abutment plates would be $2 \times CL/2 = CL$. The next to last column(V_{cl}*) in figure 5 can now be adjusted analytically to take into account any thickness change of the abutments. The actual value of V_{cl} (last column) for any abutment thickness will be as follows (fig. 6):

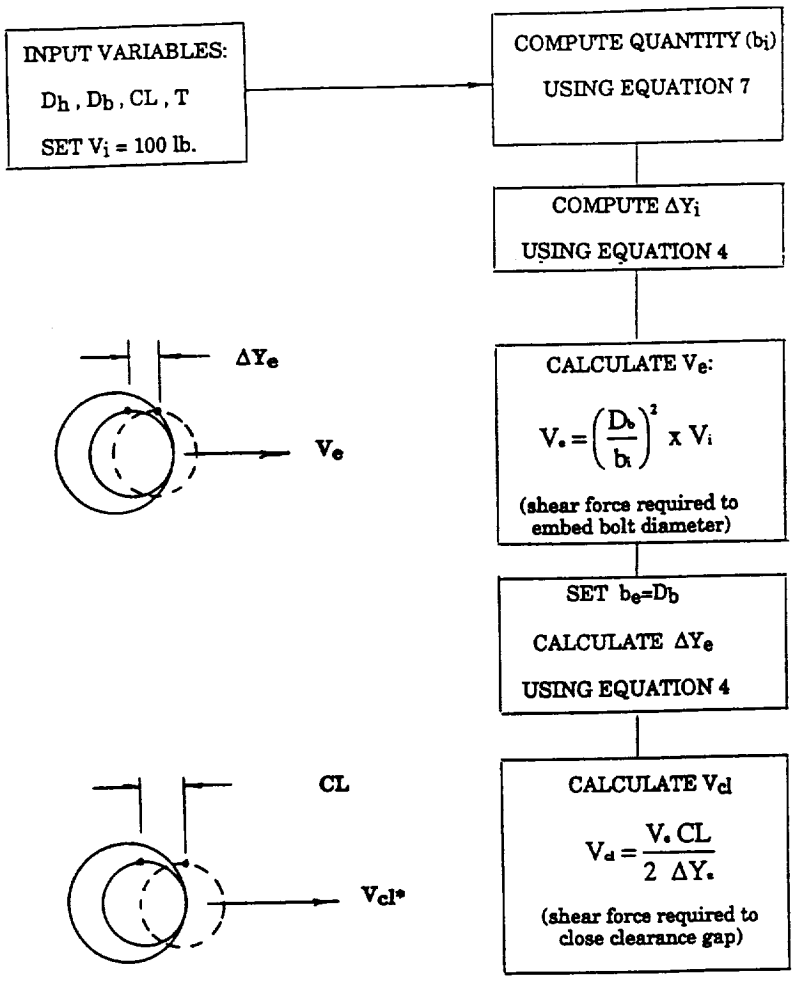


Figure 4. Analysis flow chart.

CL	D_h	V_i	b_i	ΔY_i	V_e	b_e	ΔY_e	V_{cl}*	$\frac{V_{cl}^* (T1+T2)}{(T1)(T2)}$
.001	.197	100	.1086	1.000E-04	325	.196	.009412	18	180
.002	.198	100	.0770	8.650E-05	648	.196	.013035	50	500
.003	.199	100	.0630	8.280E-05	966	.196	.015712	92	920
.004	.200	100	.0547	8.112E-05	1282	.196	.017899	144	1440
.005	.201	100	.0490	8.012E-05	1594	.196	.019776	202	2020
.006	.202	100	.0449	7.900E-05	1904	.196	.021433	266	2660
.007	.203	100	.0417	7.904E-05	2210	.196	.022924	338	3380
.008	.204	100	.0391	7.860E-05	2514	.196	.024784	414	4140
.009	.205	100	.0369	7.840E-05	2815	.196	.025537	496	4960
.010	.206	100	.0351	7.820E-05	3113	.196	.026701	583	5830
.011	.207	100	.0335	7.806E-05	3406	.196	.027790	674	6740
.012	.208	100	.0322	7.793E-05	3698	.196	.028813	770	7700
.013	.209	100	.0310	7.7781E-05	3987	.196	.029800	870	8700
.014	.210	100	.0299	7.770E-05	4274	.196	.030696	975	9750
.015	.211	100	.0290	7.760E-05	4558	.196	.031567	1083	10830
.016	.212	100	.0281	7.756E-05	4837	.196	.032398	1198	11980
.017	.213	100	.0274	7.747E-05	5117	.196	.033192	1310	13100
.018	.214	100	.0267	7.741E-05	5393	.196	.033953	1430	14300
.019	.215	100	.0260	7.737E-05	5665	.196	.034684	1552	15520
.020	.216	100	.0254	7.732E-05	5936	.196	.035387	1678	16780

V_{cl}* is calculated with one abutment plate (T=.100"), and bolt shank diameter (D_b) at .196"

Figure 5. Calculation of V_{cl}.

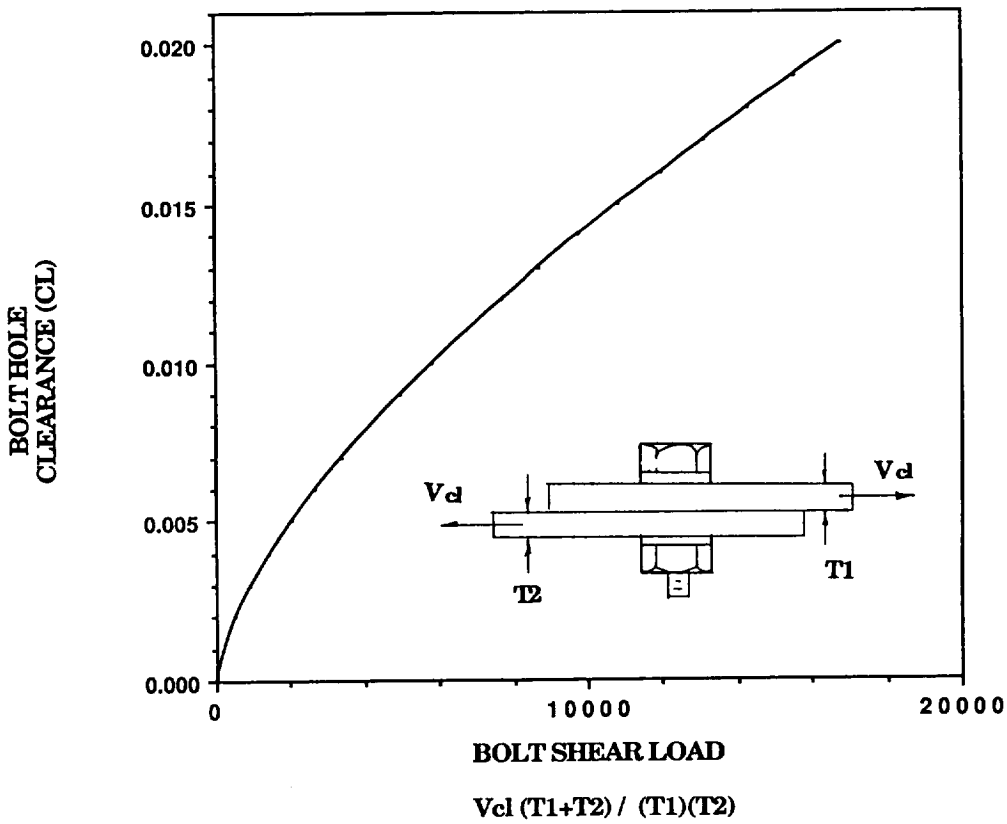


Figure 6. Shear load for bolt diameter at 0.196 in.

$$V_{cl} = \frac{(T1)(T2)V_{cl}^*}{(T1+T2)}$$

Tabulating this it becomes:

CL	$V_{cl}\frac{(T1+T2)}{(T1)(T2)}$
0.001	180
0.002	500
0.003	920
0.004	1,440
0.005	2,020
0.006	2,660
0.007	3,380
0.008	4,140
0.009	4,960
0.010	5,830
0.011	6,740
0.012	7,700
0.013	8,700
0.014	9,750
0.015	10,830
0.016	11,980
0.017	13,100
0.018	14,300
0.019	15,520
0.020	16,780

Figure 6 shows graphically the magnitude of the parameter $V_{cl}(T1+T2)/(T1)(T2)$ versus the bolt-hole clearance (CL). In an attempt to account for any bolt shank diameter, the next step is to calculate the same term for various diameters such as 0.190 in (No. 10),0.250 in (1/4), 0.3125 in (5/16), 0.375 in (3/8), 0.4375 in (7/16), and 0.5000 in (1/2). A plot of these data is shown in figure 7. It was found that the bolt shank load term $V_{cl}(T1+T2)/(T1)(T2)$ is proportional to the square root of the ratio of bolt diameters. In other words, it can be expressed in the following example equation:

$$(D_b = 0.190 \text{ in}) \qquad (D_b = 0.250 \text{ in})$$

$$\left\{\frac{V_{cl}(T1+T2)}{(T1)(T2)}\right\} = \sqrt{\frac{0.190}{0.250}}\left\{\frac{V_{cl}(T1+T2)}{(T1)(T2)}\right\}$$

Figure 8 shows the culmination of these efforts and relates the bolt shear load to the bolt-hole clearance for any aluminum abutment thicknesses and any steel bolt shank diameter. This generic graph allows one to compute the shear load on the bolt that will cause joint movement equal to the bolt-hole

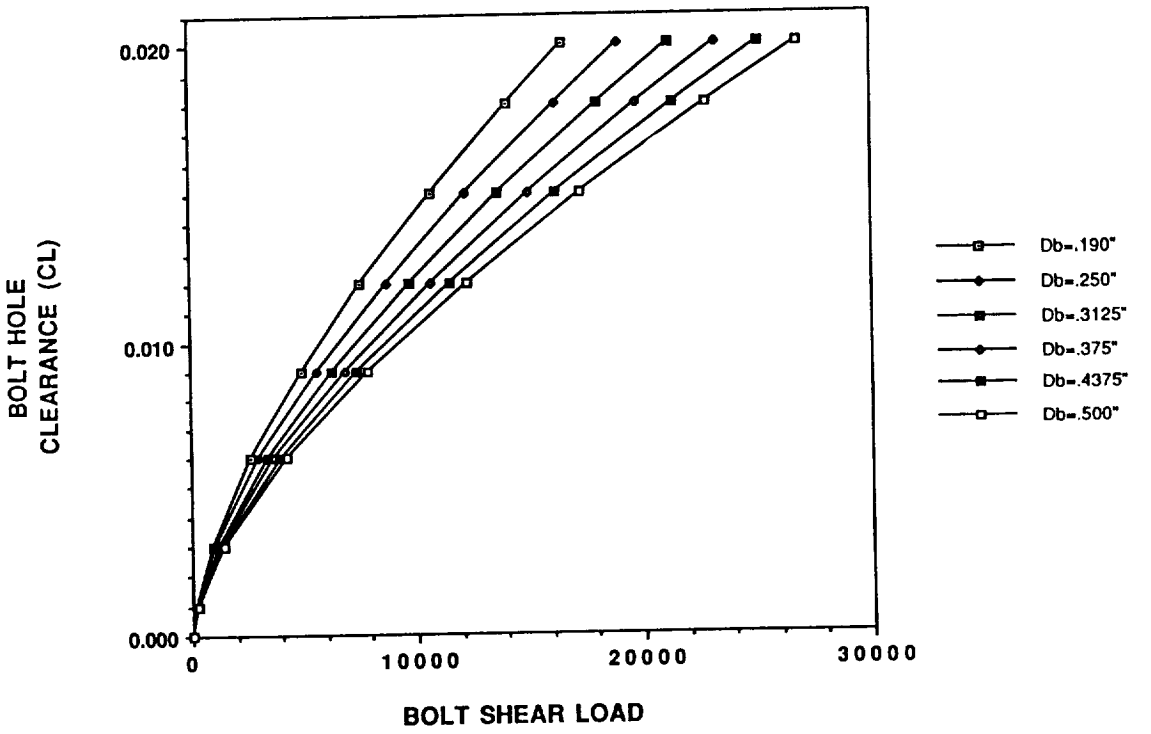
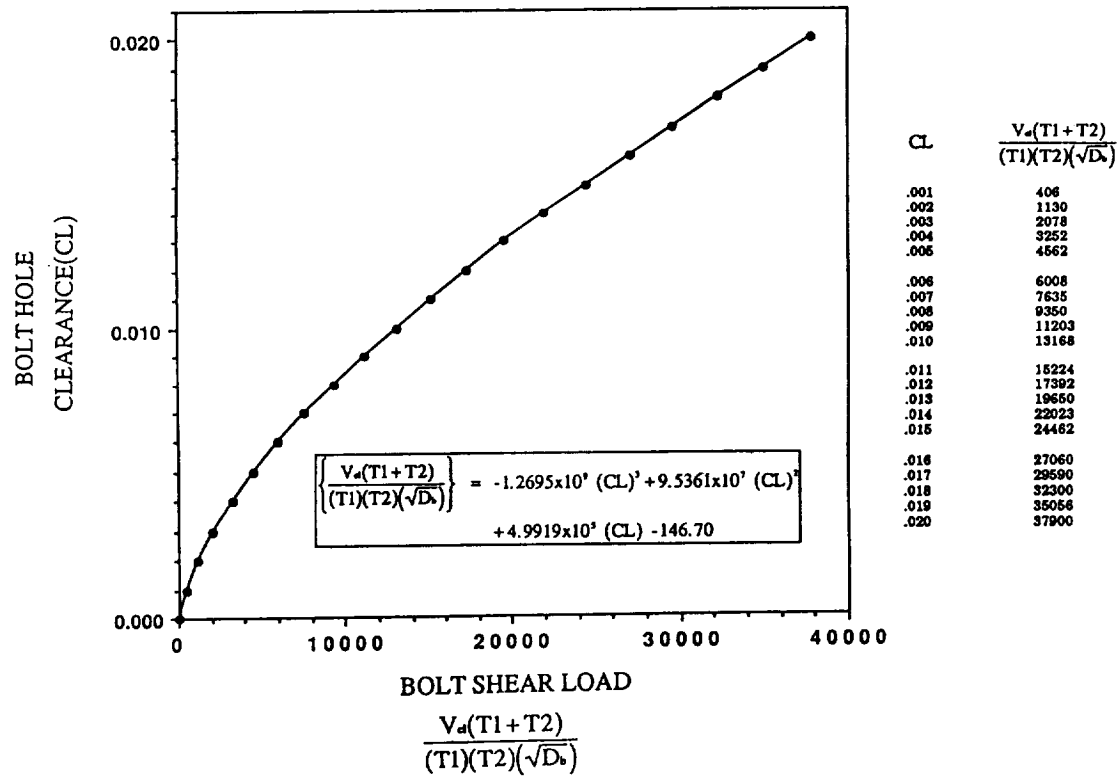


Figure 7. Shear load for various bolt diameters.



CL	$\frac{V_{cl}(T1+T2)}{(T1)(T2)(\sqrt{D_b})}$
.001	406
.002	1130
.003	2078
.004	3252
.005	4562
.006	6008
.007	7635
.008	9350
.009	11203
.010	13168
.011	15224
.012	17392
.013	19650
.014	22023
.015	24462
.016	27060
.017	29590
.018	32300
.019	35056
.020	37900

Figure 8. Shear load versus bolt hole clearance.

clearance simply by inputting the abutment thicknesses and bolt shank diameter. An algebraic equation to represent this graph is shown below:

$$\frac{V_{cl}(T1+T2)}{(T1)(T2)\left(\sqrt{D_b}\right)} = -1.2695\times10^{9}(CL)^{3}+9.5361\times10^{7}(CL)^{2}+4.9919\times10^{5}(CL)-146.70 \ .$$

Tabular values for this expression are also depicted in figure 8.

JOINT CAPABILITY ANALYSIS

In order to extend the analysis results obtained from a single bolt to a joint with multiple fasteners, the initial position of each fastener as shown in figure 2 must be assumed. This is a relatively conservative approach and any other boundary condition would need to be measured data. As long as there is no slip between the abutment plates, the shear joint acts as though it were solid, with a relatively smooth transfer of stress from one member to another. Once slip occurs, however, a more complex stress pattern emerges.[5] Load transfer may vary in individual bolts, especially in long joints. This can be accounted for in most cases by using a fitting factor (say 1.15). Figure 9 shows pictorially how the ever

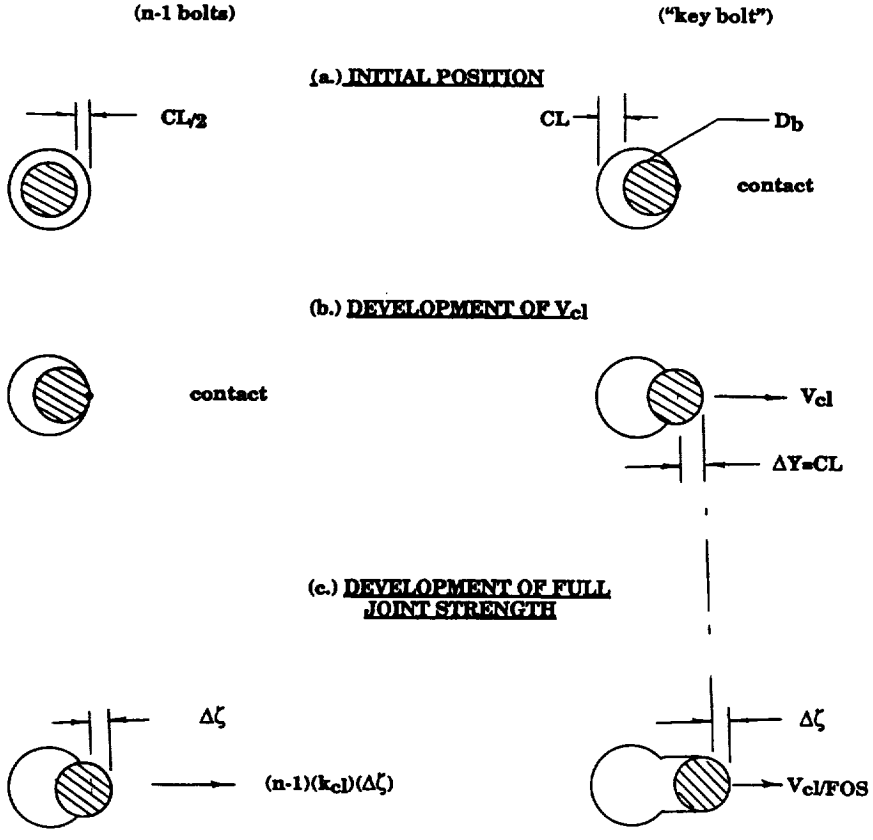


Figure 9. Joint loading history

increasing shear load (V) is transmitted through such a joint with "n" bolts. In summary, the initial position (a) places the "key bolt" in an immediate loading situation, while all other fasteners (n–1) must move a distance equal to one half the bolt-hole clearance (CL) before contact will occur. The next position (b) will exist after enough shear load (V_{cl}) has developed on the "key bolt" to deform the joint a magnitude equal to twice that distance ($2\times CL/2 = CL$). The distance is doubled because there are two abutment plates each with the identical hole clearance. At this point in the joint history, the "key bolt" will transmit the shear load V_{cl} and all other fasteners will just come into contact with the abutment plates. From this time on, any motion of the joint will result in additional shear load being carried by every fastener. The final position (c) represents development of the full strength of the joint. Making the conservative assumption that the shank-to-bolt-hole stiffness is equal at all "n" bolts, the joint carrying capability can be written as:

$$\text{Joint Carrying Capability} = \frac{V_{ult}}{FOS} + (n-1)(k_{cl})(\Delta\zeta) \ , \tag{9}$$

where:

V_{ult} = ultimate shear strength of fastener
FOS = desired factor of safety (including a fitting factor)
n = number of fasteners in joint
k_{cl} = Hertzian stiffness of shank on bolt hole (V_{cl}/CL)
$\Delta\zeta$ = deformation of bolt relative to hole (from load V_{cl} to V_{ult}/FOS).

In order to solve for the joint carrying capability desired in equation (10), the load seen by the "key bolt" must be developed. Equation (10) shows the limit of the shear load on the "key bolt."

$$\frac{V_{ult}}{FOS} \le V_{cl}+(K_{cl})(\Delta\zeta) \tag{10}$$

$$\le V_{cl}+\frac{V_{cl}}{CL}(\zeta_t-CL)$$

$$\le V_{cl}+\frac{V_{cl}\zeta_t}{CL}-V_{cl}$$

$$\le \frac{V_{cl}\zeta_t}{CL} \ ,$$

where $\Delta\zeta = \zeta_t - CL$. This leads to

$$\zeta_t \ge \frac{V_{ult}(CL)}{(FOS)(V_{cl})} \ . \tag{11}$$

Placing equation (11) into equation (9), a unique expression for the joint carrying capability can be seen.

$$\text{Joint Carrying Capability} = \frac{V_{ult}}{FOS} + (n-1)(K_{cl})(\Delta\zeta)$$

$$= \frac{V_{ult}}{FOS} + (n-1)\left(\frac{V_{cl}}{CL}\right)(\zeta_t - CL)$$

$$= \frac{V_{ult}}{FOS} + (n-1)\left(\frac{V_{cl}}{CL}\right)\left\{\frac{V_{ult}(CL)}{FOS(V_{cl})}\right\}$$

$$\boxed{\text{Joint Carrying Capability} = (n)\left(\frac{V_{ult}}{FOS}\right) - (n-1)(V_{cl}) \ .} \tag{12}$$

To determine the analytically predicted capability of the joint relative to its potential capability with a no clearance design, equation (13) was constructed as:

$$\text{Capability (\%)} = \frac{\frac{(nV_{ult})}{FOS} - (n-1)(V_{cl})}{\frac{(nV_{ult})}{FOS}}$$

$$\boxed{\text{Capability(\%)} = 1 - \frac{(n-1)(FOS)(V_{cl})}{(n)(V_{ult})} \ ,} \tag{13}$$

where:

n = number of bolts in joint load path
V_{cl} = from figure 8
V_{ult} = $(\pi D_b{}^2 F_{su})/4$
FOS = desired factor of safety (including a fitting factor).

Figure 10 depicts the plot of joint capability in percent of its potential versus the term $(V_{cl} \cdot FOS)/V_{ult}$. In addition, it shows the effect of having various numbers of fasteners in the shear joint. From this plot, it is easy to see that percent capability definitely decreases as the number of fasteners in a joint increases. It also reveals that the capability will decrease as the magnitude of V_{cl} increases. Since V_{cl} is directly proportional to the bolt hole clearance, it stands to reason that increased bolt clearance does indeed decrease the shear carrying capability of a joint. The one positive conclusion from this is that even though the bolt-hole clearance is greater than a desired tight fit, the joint is much stronger than when assuming that only one fastener will carry the entire shear load. Assuming one fastener in shear for such joints is a customary analytical approach and is conclusively overly conservative.

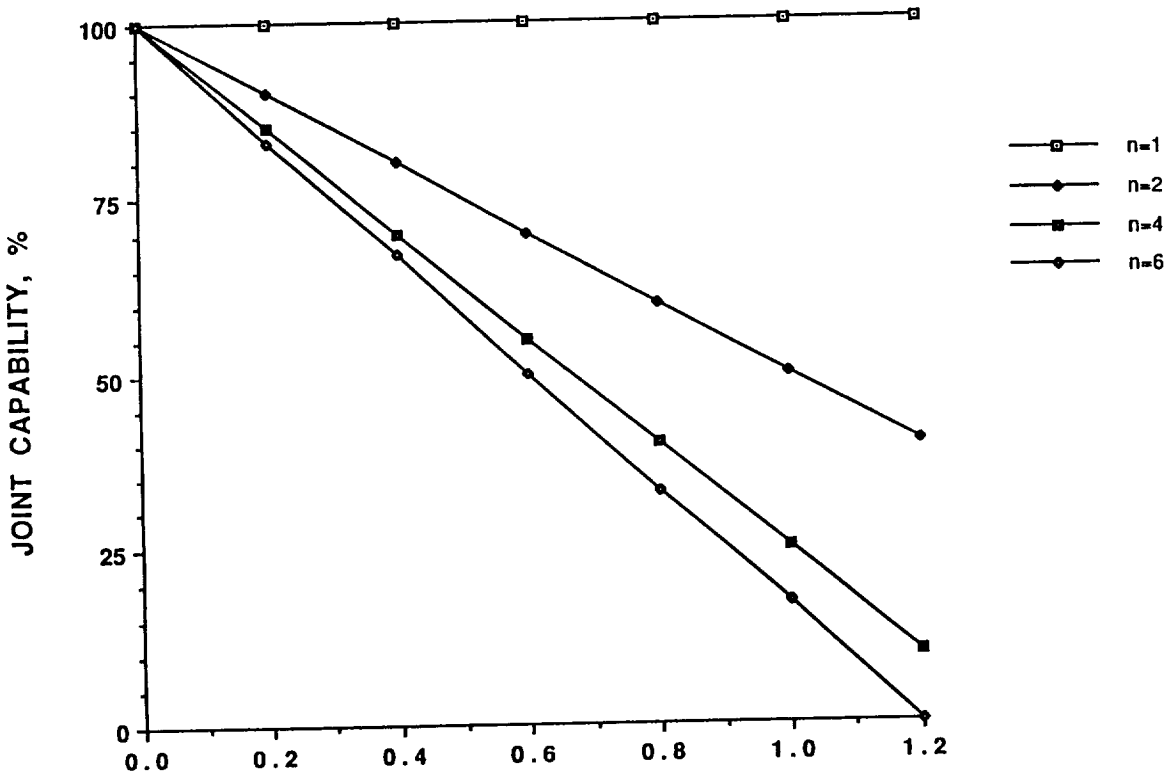


Figure 10. Joint capability (equation (13)).

CONCLUSIONS

An analytical study for typical space-flight hardware shear joints was accomplished in this report. The joints studied consisted of high-strength steel bolts clamping aerospace aluminum abutments together. Utilizing conservative assumptions and Hertzian contact theory, a general analytical expression was developed relating bolt-hole clearance to the bolt shear load required to overcome the clearance and place all fasteners into a shear transfer position. The equation takes into consideration the potential thicknesses of the abutment plates and the diameter of the bolt shank.

The analytical results mentioned above were then extended from the single fastener condition to a shear joint with multiple fasteners. The ensuing work developed a unique expression for the joint ultimate load carrying capability as a function of the number of bolts in the joint, the shear strength of the bolt shank, the bolt hole clearance, and the desired factor of safety. In order to more fully appreciate the effects of bolt hole clearance on the joint load-carrying capability, an equation was formulated which divides this predicted load capability by the potential capability of the joint if it had a clearance of zero.

It is quite evident from the analytical results obtained that, even when a conservative approach is taken, a shear joint can exhibit healthy loading capacities when less-than-ideal bolt-hole tolerances are utilized in the design of high strength steel bolts in aluminum joints.

REFERENCES

1. MSFC letter ED25 (92-40): "Spacelab Slip Resistance Tests for Rack Analyses," March 28, 1992.

2. British Aerospace paper: "Slip Coefficients for Shear Joints: the Effects of Dynamic Loading and Surface Treatment," September 18, 1991.

3. MSFC Engineering Drafting Manual

4. Roark, R.J.: "Formulas for Stress and Strain," 4th edition, p. 318.

5. Bickford, J.H.: "An Introduction to the Design and Behavior of Bolted Joints," 1981, pp. 322–325.

APPENDIX A

JOINT CAPABILITY POLYNOMIAL

An algebraic third-order polynomial expression was developed in figure 8 for the graphical data bolt clearance versus bolt shear load. For information only, this equation can be placed into the joint carrying capability formula of equation (12) with the following results:

$$\text{Joint Carrying Capability (lb)} = \left\{\frac{(n)(V_{ult})}{FOS}\right\} - \left\{\frac{(n-1)(T1)(T2)\left(\sqrt{D_b}\right)}{(T1+T2)}\right\} [\Omega] \ ,$$

$$\Omega = \{-1.2695\times10^{9}(CL)^{3} + 9.5361\times10^{7}(CL)^{2} + 4.9919\times10^{5}(CL) - 146.7\} \ .$$

Likewise, the same polynomial can be placed into equation (13) to give a more complete expression for joint capability in percent. Doing this results in:

$$\text{Joint Capability (\%)} = 1 - \left\{\frac{(n-1)(FOS)(T1)(T2)\left(\sqrt{D_b}\right)}{(n)(V_{ult})(T1+T2)}\right\} \{\Omega\} \ .$$

APPENDIX B

SHUTTLE HARDWARE TEST CASE

Reference is made to the space shuttle solid rocket booster (SRB) external tank (ET) ring fastener static structural test reported in MSFC document SRB-QUAL-ET87-056, dated December 16, 1987.

The SRB ET attach ring is connected to the SRB with 3/8 - 24 high-strength MP35N fasteners. The tolerance on the bolt holes of the SRB tang is large (0.016 in) in order to facilitate assembly. Under applied load, the bolted configuration experiences joint slippage resulting in load sharing among the row of bolts. The test was designed to determine how the applied load would be distributed as an increasing number of bolt holes have a maximum clearance of 0.16 in.

The test consisted of four 0.375-in fasteners of MP35N(F_{su} = 145 ksi) material with a modulus (E) of 34×10^6 lb/in^2 and a Poisson's ratio of 0.34. The abutments were constructed of 4130 steel. One representing the ET ring was 0.375-in thick, while the other simulated the SRB motor case at 0.400-in thick.

Utilizing the equations outlined in this report, the value for V_{cl} was determined to be 3,476 lb. Using equation (12) as shown below,

$$\text{Joint Carrying Capability} = (n)\left(\frac{V_{ult}}{FOS}\right) - (n-1)(V_{cl}) \ ,$$

with FOS = 1.0, and V_{ult} calculated at 16,015 lb, the following table shows a comparison of actual test failures and those analytically predicted.

TEST No.	TEST FAILURE	ANALYTICAL FAILURE	TEST CONDITIONS	
1&10	70,500 lb	64,060 lb	4 tight fit	or 4 with CL=0.016 in
2 – 5	63,825 lb	60,584 lb	3 tight fit	1 with CL=0.016 in
6&7	57,900 lb	57,108 lb	2 tight fit	2 with CL=0.016 in
8	53,000 lb	53,572 lb	1 tight fit	3 with CL=0.016 in

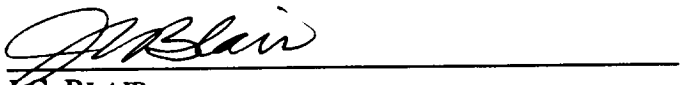
Although this tested joint is not comprised of what has been previously defined as a typical aerospace aluminum joint, the use of very high-strength MP35N fasteners with 4130 steel appears to fall into the same category as a bearing shear joint.

APPROVAL

SHEAR JOINT CAPABILITY VERSUS BOLT CLEARANCE

By H.M. Lee

The information in this report has been reviewed for technical content. Review of any information concerning Department of Defense or nuclear energy activities or programs has been made by the MSFC Security Classification Officer. This report, in its entirety, has been determined to be unclassified.



J.C. BLAIR
Director, Structures and Dynamics Laboratory

☆ U.S. GOVERNMENT PRINTING OFFICE 1992—773-050/602008