

Optimal Fastener and Preload Selection

Jim Olmstead
MDA

Paul Barker
Flexion Engineering Inc.

Jonathan Vandersluis
MDA Space Missions

ABSTRACT

Selecting fastener pre-load and strength for tension fastened joints has traditionally been an iterative tradeoff between separation, strength and slip requirements. In this paper, equations are presented for the maximum external load that a fastened joint can support and the optimal pre-load to achieve this load while simultaneously satisfying safety factors for separation, strength and slip. The equations are based on linear joint theory and joint stiffness. The optimal equations account for variations in joint stiffness due to variations in joint geometry and materials. Furthermore the equations account for variations in joint preload, load-plane factor, friction coefficients and thermal loading. The external load may be axial, shear or a combination of both. The equations can be plotted to create a 3D "Fastener Design Space" that represents the maximum acceptable joint external tension and shear loads as a function of the joint preload. The design space is normalized by fastener strength to make it applicable to any fastener. Any combination of preload, tension and shear that falls within the fastener design space represents a safe joint design. Plotting multiple load combinations provides a quick method for evaluating the safety of multiple joints and load cases. Alternatively the Fastener Design Space can be used to determine the maximum load capacity and optimal preload for a set of joints with varying properties.

CITATION: Olmstead, J., Barker, P. and Vandersluis, J., "Optimal Fastener and Preload Selection," *SAE Int. J. Mater. Manf.* 5(1):2012, doi:10.4271/2012-01-0478.

INTRODUCTION

The design and analysis of preloaded joints involves many variables. The range of these variables depends on applicable codes, internal design guidelines, past experience, joint geometry and joint materials. Typically preload selection is an iterative process because joint (1) separation, (2) strength, and (3) slip performance are independently checked with their respective safety factors.

The goal of this paper is to derive simple equations that simultaneously satisfy joint separation, strength and slip performance requirements (safety factors) while determining the optimal preload that maximizes joint load capacity. The magnitude of the maximum joint load capacity is also found. The equations will account for ranges in the following parameters:

- Joint member stiffness and associated load plane factor due to joint geometry and materials
- Safety factors

- Preload range due to preloading method, method implementation, elastic interaction and embedding
- Differential thermal expansion
- Slip coefficient of friction

REVIEW OF THREADED FASTENER LOADS

A tension fastened joint is a clamped assembly of components where the clamping force is provided by a preloaded fastener. A typical concentrically loaded axial joint is illustrated in [Figure 1](#). When a fastener is preloaded, the fastener stretches and the clamped parts compress, such that the tension in the fastener and the compression in the clamped parts are equal and opposite. The force-deflection relationship is shown in [Equation \(1\)](#).

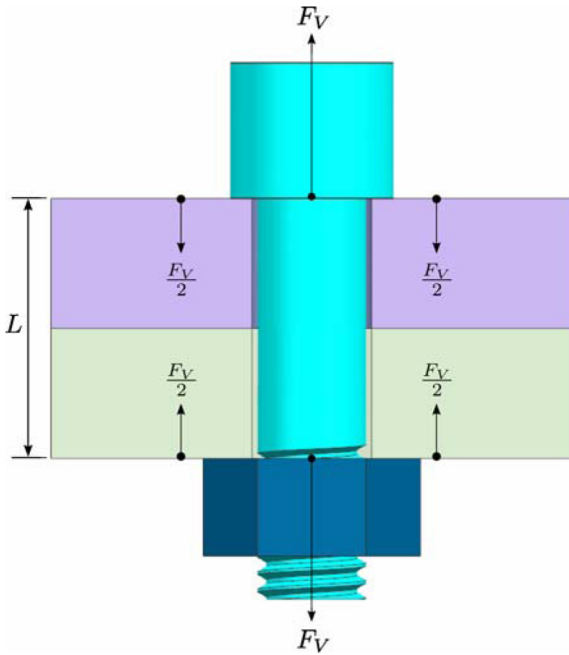


Figure 1. Fastener Geometry

$$F_V = K_b \Delta L_{b,V} = -K_c \Delta L_{c,V} \quad (1)$$

where:

F_V is the joint preload

K represents the stiffness of a component

ΔL represents the change in length of a component due to a load

b and c subscripts refer to the fastener and clamped parts respectively such that:

K_b is the stiffness of the fastener

K_c is the stiffness of the clamped parts

$\Delta L_{b,V}$ is the elongation of the fastener due to the applied preload F_V

$\Delta L_{c,V}$ is the change in length of the clamped parts due to the applied preload F_V

The relationship shown in Equation 1 can be represented by an axially loaded joint diagram. This diagram may be found in many fastener design references [1][2].

REVIEW OF THE TRADITIONAL JOINT DIAGRAM

The traditional joint diagram starts with individual load-deflection diagrams for the fastener and clamped parts as shown in Figure 2.

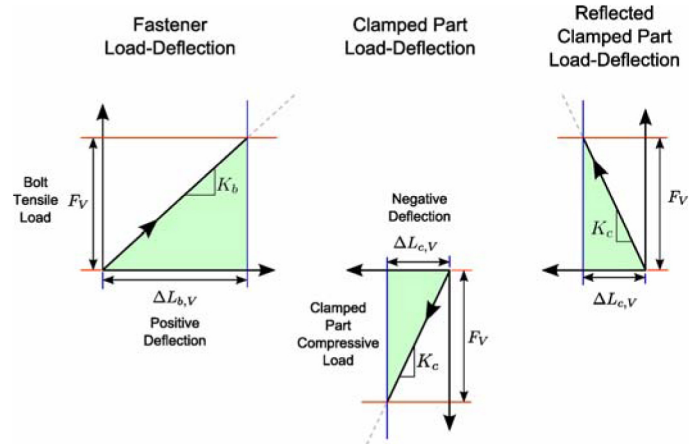


Figure 2. Fastener and Clamped Part Stiffness

In the traditional joint diagram the stiffness curves for the fastener and clamped parts are combined on the same diagram. The clamped part load-deflection diagram is first reflected as shown in Figure 2, and then the diagram's origin is moved such that the preload point coincides with the fastener diagram's preload point, F_V , as shown in Figure 3.

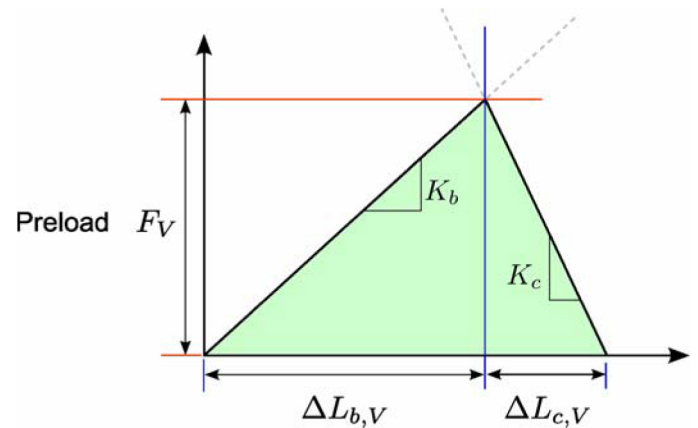


Figure 3. Traditional Preloaded Joint Diagram

When a load is applied to the joint, the load is distributed between the fastener and the clamped parts. The distribution of the load is proportional to the relative stiffnesses of the fastener and clamped parts. If the applied load is tensile, then the fastener and clamped parts are elongated. Therefore the fastener tension increases and the clamping force decreases. On the joint diagram this is shown by moving the load point along the two curves as shown in Figure 4.

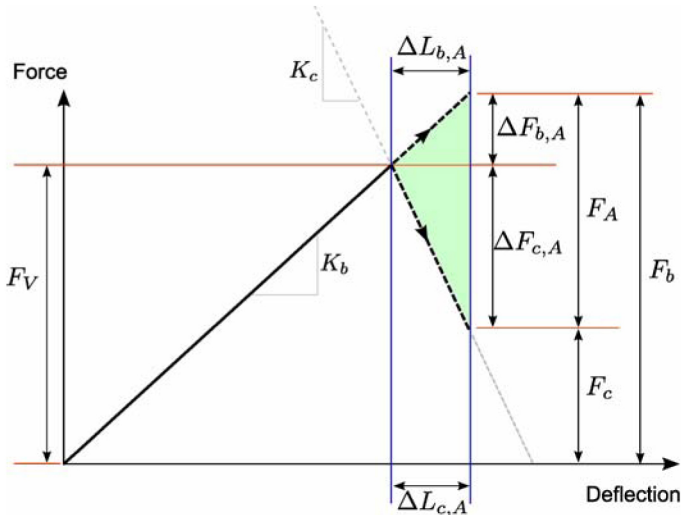


Figure 4. Traditional Joint Diagram with Applied Load

Where

F_A is the load applied to the joint

F_b is the load on the fastener

F_c is the load on the clamped parts

$\Delta F_{b,A}$ is the change in fastener load due to F_A

$\Delta F_{c,A}$ is the change in clamping load due to F_A

$\Delta L_{b,A}$ is the fastener elongation due to F_A

$\Delta L_{c,A}$ is the change in length of the clamped parts due to F_A

The total load on the fastener is:

$$F_b = F_V + \Delta F_{b,A}$$

The total load on the clamped parts is:

$$F_c = F_V - \Delta F_{c,A}$$

To calculate $\Delta F_{b,A}$ and $\Delta F_{c,A}$, the slopes of the joint diagram lines are used:

$$\Delta F_{b,A} = K_b \Delta L_{b,A}$$

$$\Delta F_{c,A} = K_c \Delta L_{c,A}$$

However:

$$\Delta L_{b,A} = \Delta L_{c,A} = \Delta L$$

Therefore:

$$F_A = \Delta F_{b,A} + \Delta F_{c,A}$$

$$F_A = (K_b + K_c) \Delta L \quad (2)$$

The joint force ratio, Φ , is defined as the ratio of the incremental load induced in the fastener due to the applied external load, divided by the applied load:

$$\Phi = \frac{\Delta F_{b,A}}{F_A}$$

This ratio can also be expressed in terms of the joint stiffnesses:

$$\Phi = \frac{K_b}{K_b + K_c}$$

Using the above definition:

$$\Delta F_{b,A} = \Phi F_A \quad (3)$$

$$\Delta F_{c,A} = (1 - \Phi) F_A \quad (4)$$

The above equations are only valid for joints where the point of application of the applied external load is the outer surface of the flanges.

TRADITIONAL JOINT DIAGRAM WITH LOADING PLANE FACTOR

The above derivation of Φ assumes that F_A acts on the outer surfaces of the flanges. In practice, the applied force acts below the surface as shown in Figure 5.

By acting below the clamped part's outer surfaces the length of the fastener load path increases and the length of the clamped part load path decreases. This decreases the effective fastener stiffness and increases the clamped part stiffness.

The result is a decrease in $\Delta F_{b,A}$ and an increase in $\Delta F_{c,A}$. To represent this, the loading plane factor is defined as n where:

$$\Delta F_{b,A} = n \Phi F_A \quad (5)$$

$$\Delta F_{c,A} = (1 - n\Phi)F_A \quad (6)$$

The loading plane factor, n , represents the percentage reduced incremental fastener force, due to the applied load acting on the load plane. The reduction in applied load taken by the fastener is accompanied by an equal increase in the load taken by the clamped parts.

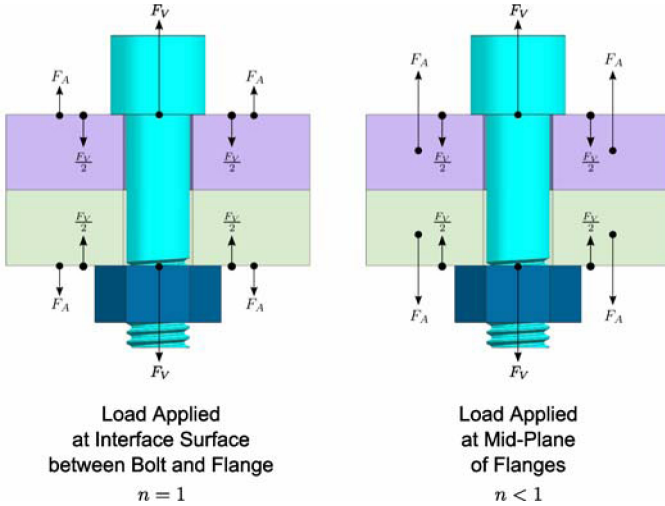


Figure 5. Joint with Load Applied Below Flange Surfaces

The loading plane factor decreases the stiffness of the bolt force curve K_b and increases the stiffness of the clamped part curve K_c on the joint diagram, as illustrated in Figure 6.

Note that during fastener preloading and thermal loading, the load acts at the outer surfaces of the flanges. Accordingly, the loading plane factor is not needed for preloading and thermal loading.

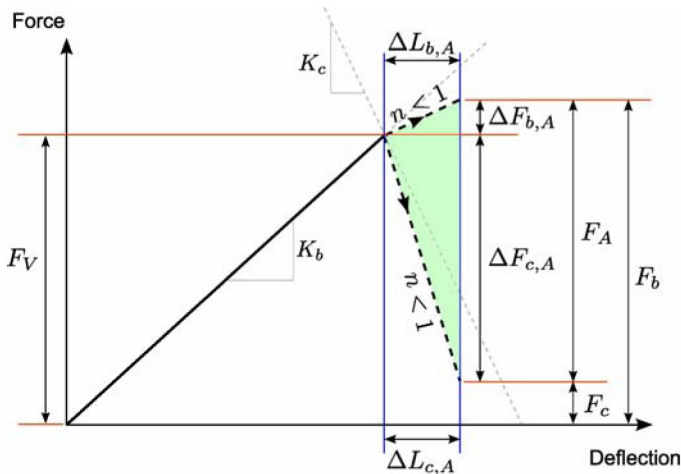


Figure 6. Joint Diagram Including Load Plane Factor n

FASTENER ANALYSIS WITH TRADITIONAL JOINT DIAGRAM

Using the traditional joint load diagram, fastener strength is checked by comparing the fastener load to the fastener strength as shown in Figure 7. Fastener failure is assumed to occur when:

$$F_V + \Delta F_{b,A} \geq F_y$$

$$F_V + n\Phi F_A \geq F_y \quad (7)$$

where:

F_y = yield load of fastener

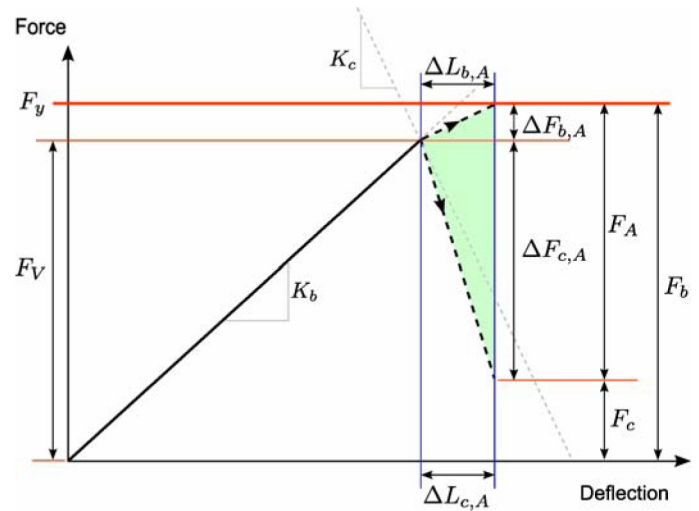


Figure 7. Joint Diagram showing onset of Fastener Tension Failure

Using the traditional joint load diagram, joint separation is checked by ensuring that the clamping force F_c remains positive as shown in Figure 8. Separation is assumed to occur when:

$$F_V - \Delta F_{c,A} \leq 0$$

$$F_V - (1 - n\Phi)F_A \leq 0$$

$$F_V \leq (1 - n\Phi)F_A$$

$$F_A \geq \frac{F_V}{(1 - n\Phi)} \quad (8)$$

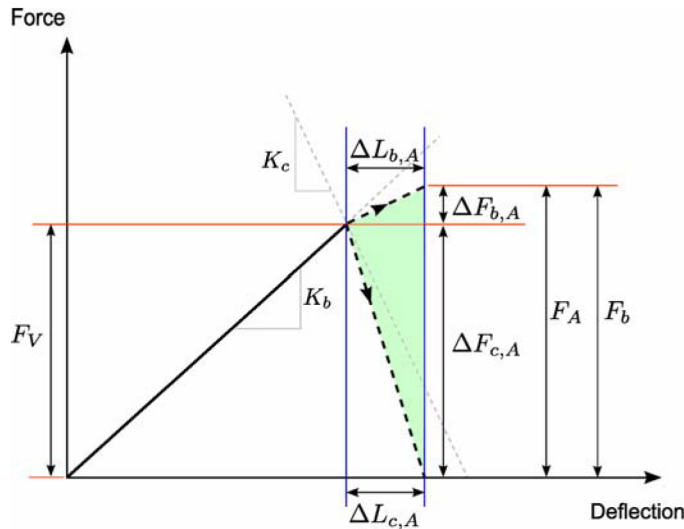


Figure 8. Joint Diagram showing onset of Separation Failure

OPTIMIZED TRADITIONAL JOINT DIAGRAM

By comparing Figure 7 and Figure 8 it can be seen that for a particular joint geometry (i.e. for specific values of K_b , K_c , and n) the determination of whether a joint fails by fastener strength or joint separation, is dependent on the joint preload. A low preload will cause joint separation to be critical. A high preload will cause fastener strength to be critical. There is a single preload where these two limits will occur simultaneously as shown in Figure 9. This “optimal” preload maximizes the allowable external load on the joint, F_A , and can be calculated using the following equations:

$$F_V = \Delta F_{c,A}$$

$$F_V = (1 - n\Phi)F_A$$

$$F_A = F_y$$

$$F_V = (1 - n\Phi)F_y$$

(9)

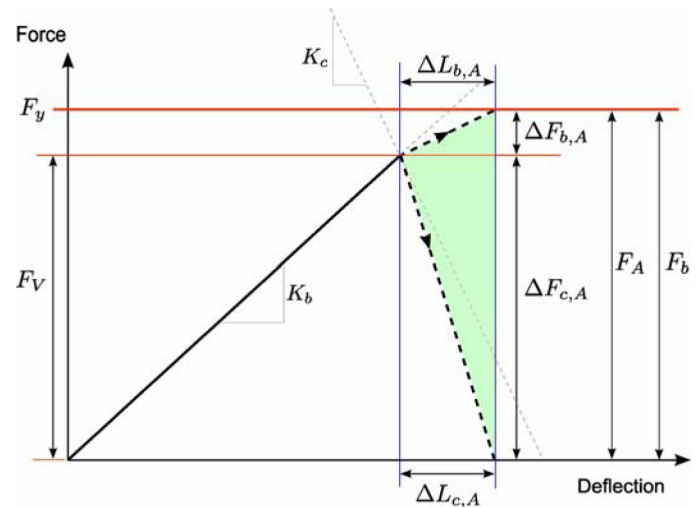


Figure 9. Optimized Joint Diagram

The above joint diagrams assume single values for the joint parameters n and Φ . Furthermore, the diagrams do not incorporate safety factors nor have axes that allow direct measurement of allowable separation or fastener strength loads. Each of these limitations will be discussed and removed in the following sections.

FASTENER DESIGN SPACE FOR TENSION JOINTS

Traditionally, the joint diagram is used for fastener analysis. A nominal preload will be determined based on past experience or design guidelines that account for variations in joint parameters such as preload, stiffness, thermal expansion and friction. The preload will then be iteratively altered in an attempt to improve, but not necessary optimize the joint. The objective of this study is to modify the traditional optimized joint diagram, transforming it such that it incorporates range variations in joint parameters, can directly measure the allowable separation or fastener strength loads and can be used to provide optimum preloads and allowable external loads.

To find the optimal preload and external load, the concept of a Fastener Design Space (FDS) is introduced. The FDS expresses the maximum acceptable joint external load as a function of the joint preload. Both the preload and the external load are expressed as percentages of fastener strength. An example fastener design space graph is shown in Figure 10. For any joint, the preload and applied external load can be plotted on the graph. If the value is contained within the safe shaded region of the FDS, then the joint and fastener will be safe for both separation and strength.

An FDS can be created quickly for any joint using a series of equations for the vertices of the FDS triangle. These equations are developed in the following sections.