

Body Join Drilling for One-Up-Assembly

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ABSTRACT

Over 1,200 large diameter holes must be drilled into the side-of-body join on a Boeing commercial aircraft's fuselage. The material stack-ups are multiple layers of primarily titanium and CFRP. Due to assembly constraints, the holes must be drilled for one-up-assembly (no disassembly for deburr).

In order to improve productivity, reduce manual drilling processes and improve first-time hole quality, Boeing set out to automate the drilling process in their Side-of-Body join cell. Implementing an automated solution into existing assembly lines was complicated by the location of the target area, which is over 15 feet (4 meters) above the factory floor.

The Side-of-Body Drilling machines (Figure 1) are capable of locating, drilling, measuring and fastening holes with less than 14 seconds devoted to non-drilling operations. Drilling capabilities provided for holes up to $\frac{3}{4}$ " in diameter through stacks over 4.5" thick in a titanium/CFRP environment. Using high precision servo control, each layer could be customized with specific drill parameters optimized to improve hole quality and decrease drill cycle time. Drill life was improved by tracking depth drilled for each drill bit.

The drilling process is stabilized by rigid support structure which is optimized for both stiffness and natural frequency resulting in deflections no greater than 0.020 inches. Each drilling machine is light-weight and mobile to accommodate multiple work zones in multiple assembly lines. One-up assembly was achieved by using custom doweling/clamping fasteners automatically installed by the machine in strategic locations to provide the proper part clamping.

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INTRODUCTION

Over 1,200 large diameter holes must be drilled into the side-of-body join of a Boeing commercial aircraft's fuselage. The thick material stack-ups are multiple layers of primarily titanium and CFRP. Once the fuselage sections are aligned, the holes at the Side-of-Body fitting must be drilled for one-up-assembly.

In order to improve productivity, reduce manual drilling processes and improve first-time hole quality, Boeing set out to automate the drilling process in their Side-of-Body join cell. Implementing an automated solution into existing assembly lines was complicated by the location of the target area, which is over 15 feet (4 meters) above the factory floor.



Figure 1. Side-of-Body Drilling Machine

To automate the drilling operations, the focus was first on stabilizing the drilling process by overcoming the limitations for support structure. The structure needed to be stiff but also needed to have a high a natural frequency for the best stability so there was a trade-off between these two parameters.

To improve hole quality, the drilling process itself needed to be controllable over multiple layers of stack-up to be able to vary and optimize drilling profiles through each layer.

To achieve One-Up-Assembly, a part clamping scheme was developed to provide clamp-up during and after drilling with specialized temporary fasteners developed by Centrix.

SUPPORT STRUCTURE DESIGN

Mobile Machine

The first stage of the structure design process was to design the mobile machine portion of the automated equipment. The mobile machine essentially acts as a mass (m) at the end of a cantilevered beam. This beam (the x-axis beam is supported on two columns shown in [Figure 2](#)) had some unknown stiffness (k) that was optimized once the machine was designed.

$$\text{Natural frequency: } f_n = \frac{1}{2\pi} \sqrt{\frac{k}{m}}$$



Figure 2. Mobile Machine



Figure 3. Machine Clearance

For this reason, a lower mobile machine mass is desirable in order to maximize the natural frequency but at the same time, the machine needs to have minimal deflection. The stiffest beam was not the most stable solution in this case. The mobile machine can be seen in [Figure 2](#). The main structure of the mobile machine is a single weldment that integrates all of the machine's axes and electronics and interfaces with a sled on the beam. As a starting point, the existing factory crane limited the target machine weight to less than 11,000 pounds. Also, the machine had to clear existing tooling and work platforms used for accessing the top of the barrel and had to fit within the length of the existing cell. The main machine structure was optimized through many iterations of finite element analysis, adjusting dimensions and plate thicknesses to minimize its weight, while maximizing its stiffness. With a goal of deflection less than 0.010 inches of deflection for the mobile machine portion of the assembly and the length and height constraints, the resulting structure came in at 2,000 pounds and pushed the limits of the dimensional constraints. [Figure 3](#) shows the machine's physical space constraints and how close it came to the existing bridge structure (in yellow). [Figure 4](#) shows the finite element analysis of the mobile machine deflection at the clamp pad with a 1,200 pound clamp force assuming it was attached to a rigid beam. This allows the beam to deflect some to give a total machine deflection of under 0.020 inches. The resulting total machine weight was less than 7,000 pounds which was helpful in increasing the natural frequency of the system and also facilitated shipping the machine and moving it around the factory. Another consideration was the machine tower deflecting under X acceleration moves. This drove the X length of the machine with the addition of triangular gussets that help to resolve some of the X acceleration loads. [Figure 5](#) shows the

deflection of the machine tower under an acceleration of 0.1g. This analysis was run in the worst case with the process head at the top of travel. The goal was to minimize this deflection so that the machine would not need to wait for it to settle out while doing pitch moves in X from one hole to the next.

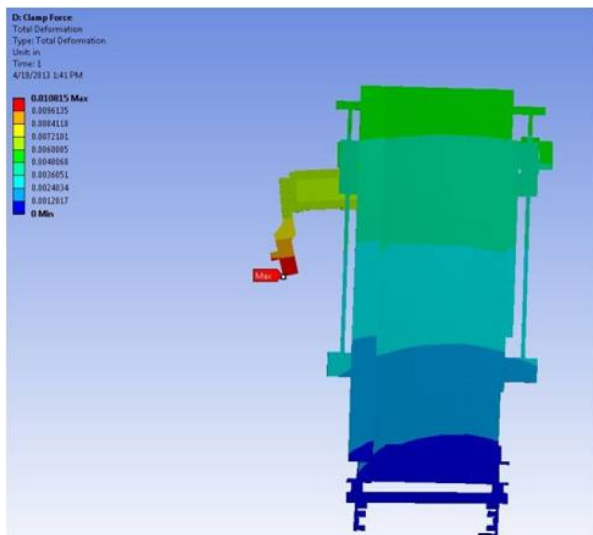


Figure 4. Machine Deflection under 1200# Clamp Force

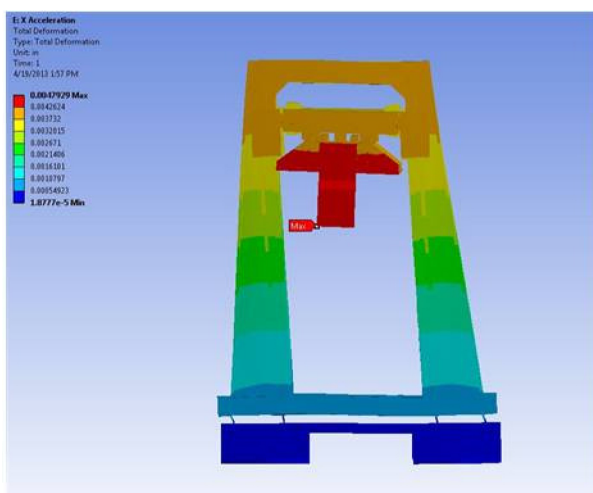


Figure 5. Machine deflection under 0.1g acceleration

Support Structure

With a total weight of less than 7,000 pounds for the machine, the overall stiffness and natural frequency of the system could be analyzed. The torsional rigidity of the x axis beam as well as the stiffness of the columns both contribute to the natural frequency of the system. Boeing allowed three options for the beams and columns: use the existing beams and columns to support the machine, use the existing columns and build new beams or build all new beams and columns. With another round of finite element analysis, it was

determined that the existing beams and columns were not stiff enough for the new machine. The existing cell posed some significant space constraints, particularly in the area directly below the machine and this created some issues with achieving the desired stiffness. Electroimpact first started by optimizing the space available for a beam with a new, stiffer beam. Figure 6 shows that the new beam alone did not achieve the desired natural frequency and came in around 8 hertz, well below the goal of reaching 10 hertz. It also shows the shape of the columns that support the beam. The thin area of the column below the beam is to allow for the existing automated guided vehicle used to transport the barrel from cell to cell. Figure 7 shows the same analysis but with new, optimized columns as well. The natural frequency was increased to over 11 hertz. The new columns were designed to use as much of the available space as possible and to be as stiff as possible. This was necessary because a direct load path was not available. The two worst case positions were when the machine was in the middle of the beam between 2 columns and also at the end of a beam. The ends of the beam, however, were less important because they are both over-travel areas. The second natural frequency was also considered. This mode is excited by the X axis acceleration and shuttle moves. The new beams and columns did not have much of an effect on the mode as it was primarily due to the tower as shown in Figure 8. The natural frequency was greatly driven by the weight of the mobile machine and the height of the process heads. All analyses were run with the head at the top of travel to find the worst cases.

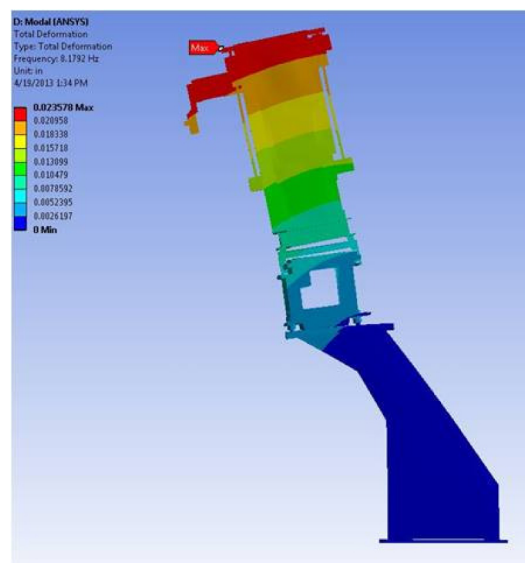


Figure 6. Modal Analysis of New Beam on Old Columns

With the individual components of the system stiff enough and the natural frequency improved, the system as a whole needed to be analyzed for deflections. Finite element analyses were run for many different scenarios. All analyses were performed at both the middle of the beam and the end of the beam and with the process head at the way up in both