



**Particle Physics Division
Mechanical Department Engineering Note**

Number: MD-ENG-423

Date: 22 June 2012

Update: 9 July 2012

Project Internal References: PPD docdb 1551, NOvA docdb 7609

Project: NOVA Far Detector Installation

Title: NOvA Block Pivoter Pusher Weldment FEA

Author(s): Dave Pushka

Reviewer(s):

Key Words: Weldment, FEA, NOvA Pivoter

Abstract Summary: Design and analysis of a weldment and hydraulic cylinder assembly used to push the NOvA Far Detector Pivoter the final one inch of travel.

Applicable Codes: AISC ASD

Discussion:

Tests of the pivoter drive system with the mock block in June 2012 showed that some operators were un-able to ease the pivoter up against the booked. Other operators were able to move the pivoter with sufficient precision. Because there remains a possibility that when the pivoter is loaded with the first real detector block, it will need more control for the last inch of travel than is presently available. Therefore, this mechanism was developed to allow a hydraulic cylinder to move the pivoter.

A similar application of hydraulic cylinders was used for the FHEP block and shown to work very well.

Fermilab Engineering Manual Requirements:

Note that no written specification for this work was provided by the project physicists as described in Chapter 1 of the Fermilab Engineering Manual. Since no specifications were written, no department approval of the specifications exists. The discussion above shall serve as the 'requirements document'. Chapter 2 risk analysis has not been formally prepared. Weldments such as this have been designed and built numerous times and are considered by inspection to be a low risk. Chapter 3 requirements and specification reviews by the project have not been formally prepared. The physicist assigned as the installation manager will be provided this document and any comments, suggestions or concerns of his will be addressed, perhaps with a discussion, perhaps with a design change. This engineering note has been specifically prepared to address chapter 4 of the engineering manual. The 'reviewed by' signature on the cover page of this document addresses Chapter 5 of engineering manual. Components will be purchased using a purchase requisition, with physicist installation manager approval as required by Chapter 6. Chapter 7 requirements will be performed by the design engineer (author of this document) who will test the completed device in place. Once tested, it will be released for use by the NOvA far detector installation crew (see Chapter 8) and instructions on its use will be given verbally to the staff. Unless there are lessons learned from this effort, this engineering note and the fabrication drawings will be considered the final written Project Report as described in Chapter 9. This note will be placed in the PPD and NOvA document data bases as a means of Archiving and Control.

Pusher Capacity Determination:

Pressure gauging was installed on the drive system during testing of the pivoter with the mock block in June 2012. This gauging indicated a brief peak pressure in the hydraulic supply of 2500 to 3000 psi just as the pivoter overcame static friction and began to move. On almost all starts from a dead stop, the peak system pressure was around 2500 psi, only once did it peak at 3000 psi.

The hydraulic drive system motor and transmission data sheet indicate a nominal system pressure of 5800 psi and a nominal drive motor / planetary gear output torque of 29,661 foot-pounds.

The directly driven wheel is 30 inches (2.5 feet) in diameter.

Calculating the drive wheel force, $F_{\text{wheel}} = T/r$. $F_{\text{wheel}} = 29,661 \text{ ft-pounds} / 1.25 \text{ feet} = 23,728 \text{ pounds}$.

Reducing this force for the reduced system supply pressure yields a drive wheel force of $23,728 * (3000/5800) = 12,273 \text{ pounds}$.

There are four hydraulic motor/planetary gear/wheel drives on the pivoter, all operating in parallel. Therefore, the force required to overcome the static friction is: $F_{\text{total}} = 4 * 12,273 \text{ pounds} = 49,094 \text{ pounds}$.

Since the detector block was not on the pivoter during this testing, assume the force required to overcome the static friction is proportional to the weight. Unloaded pivoter weight as tested during the mock block exercise was approximately 750,000 pounds. Block weight is estimated at 414,000 pounds. So the ratio is $= (750+414)/750 = 1.55$

So, design the pusher capacity for $49,094 * 1.55 = 76,193 \text{ pounds}$. Assume two pushers are used; therefore the required load on each would be 38,096 pounds.

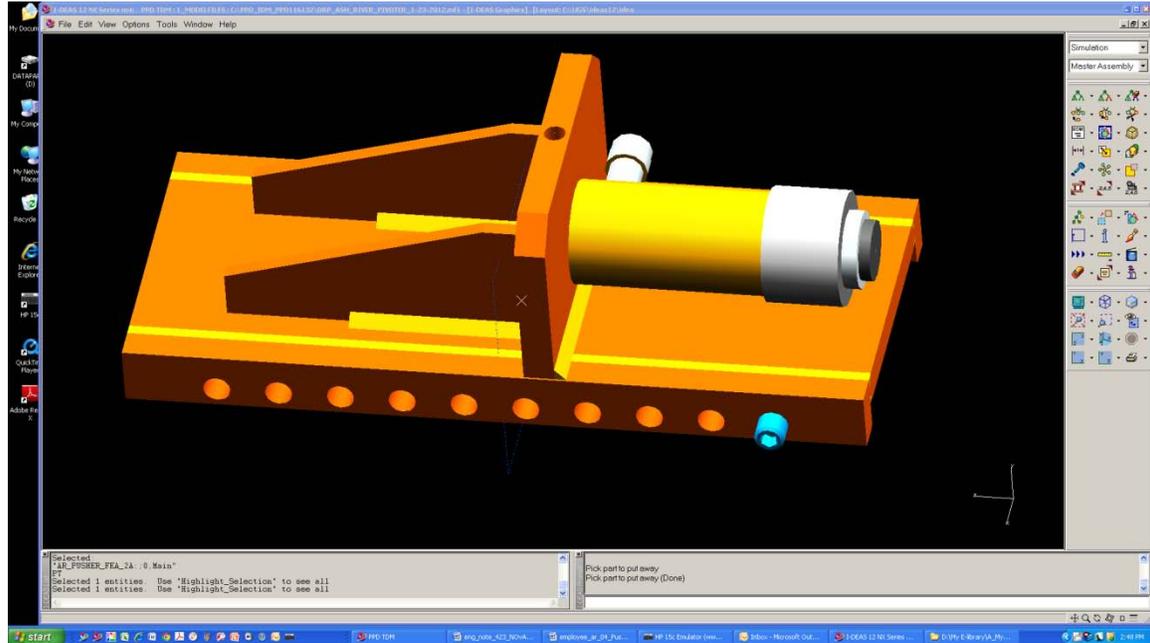
Design pusher weldment for use with a 25 ton (50,000 pound) hydraulic jack. Assume jack travel of 4 inches.

Use set screws to clamp the pusher weldment to the steel rail. The Machinery Handbook (see page 1543 of the 25th edition) lists the allowable working capacity for ¾" cup point set screws at 1300 pounds per screw. 1" set screws have an allowable working capacity of 2500 pounds. Therefore, a 50,000 pound load requires twenty (20) 1" set screws or thirty-eight (38) ¾" set screws. Reducing the capacity to 38,000 pounds reduces the number of required set screws to sixteen (16) for 1" size and twenty-nine (29) for ¾" set screws.

Unbrako (manufacture of socket head set screws) in their engineering catalog (see <http://www.unbrako.com/docs/engguide.pdf>) lists the axial holding power of a nominal one inch size set screw at 6500 pounds when torques to 5000 in-pounds (416 ft-pounds). This is a very large torque to achieve, unlikely to be achieved in practice at Ash River. 200 ft-pounds is a more reasonable torque value and this results in a holding power of about 3100 pounds per screw.

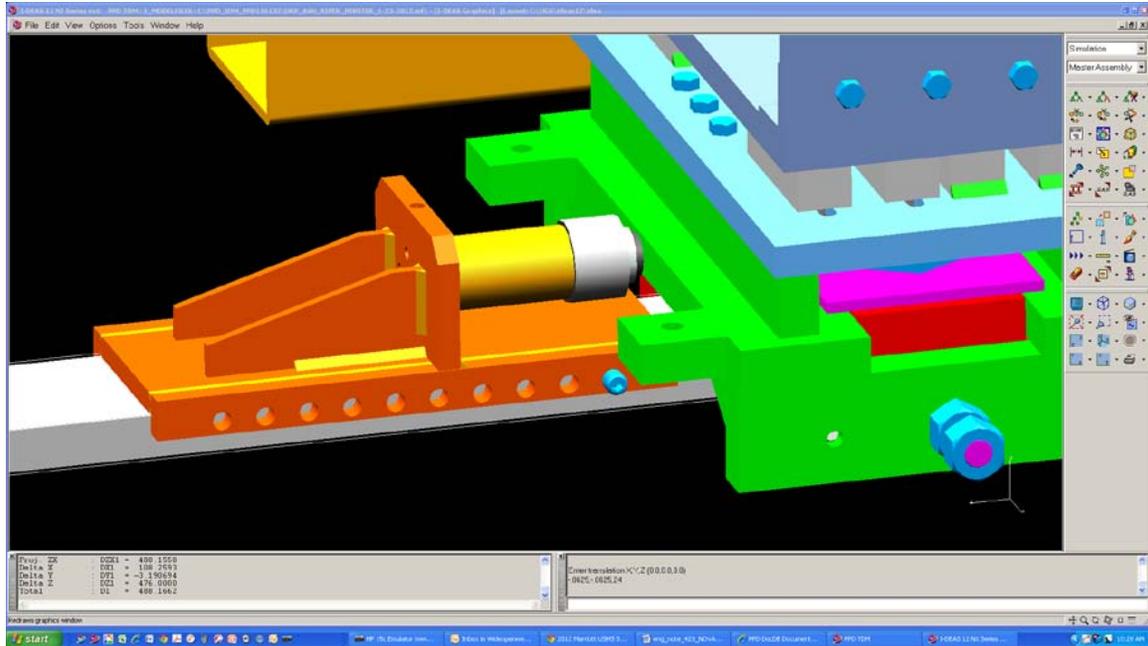
Use twenty 1” set screws, with ten on each side. Use the lower 2500 pound capacity in the design of the weldment.

Image of the weldment with a 25 ton jack mounted and one set screw installed:



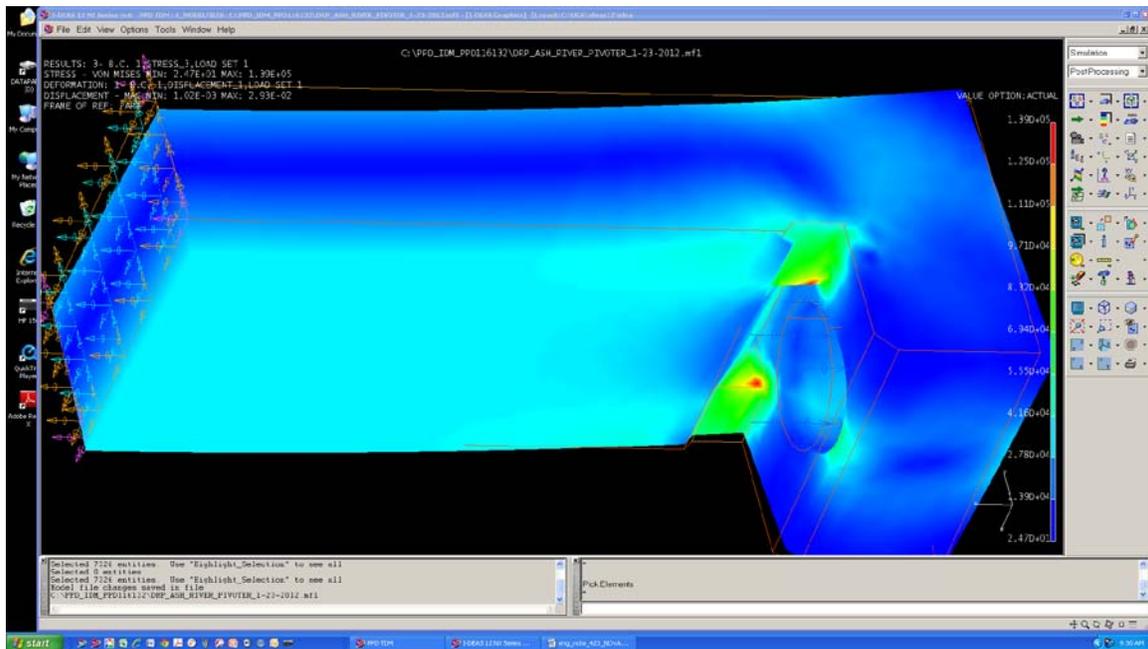
Welds are shown in yellow. The cylindrical object is a commercial hydraulic jack. The blue object is one set screw. Rail omitted for clarity. The hole in the top is for a lifting eye. Assembly weighs over 100 pounds, so manual lifting of it could lead to injury, therefore the lifting eye will facilitate use of the building crane or a strap from a portable engine lift or fork lift mounted fixture.

Image of the Pusher assembly mounted on rail and pressing against the rear cam follower assembly (note only one of the ten set screws are shown in this image):



The green object is the rear cam follower assembly and is rigidly attached to the lower weldment of the pivoter. Hydraulic pump and hose omitted for simplicity.

Analysis of the set screw loading on the pusher weldment – set screw induced bending:



An axial load of 12,500 pounds was applied to the cylindrical surface of the hole where the set screw is to be installed. The magnitude is calculated as such: An assumed 0.2 coefficient of friction exists between the rail and the set screw surface. A static friction force needed to resist the 25 ton load applies along the rail long axis results in a load per set screw of:

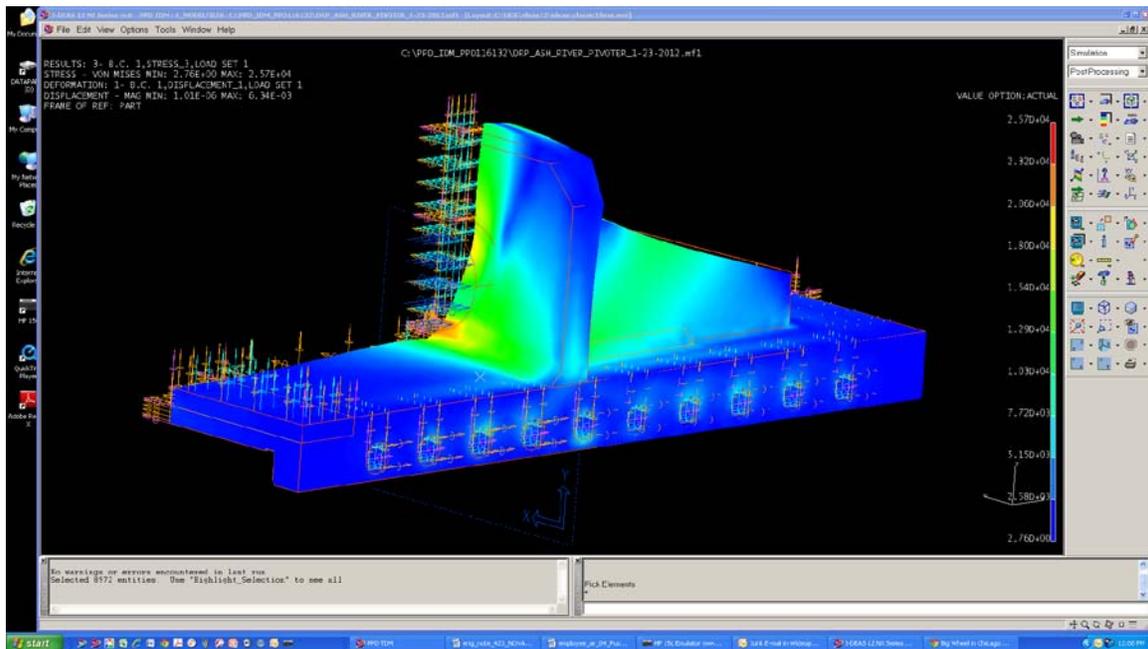
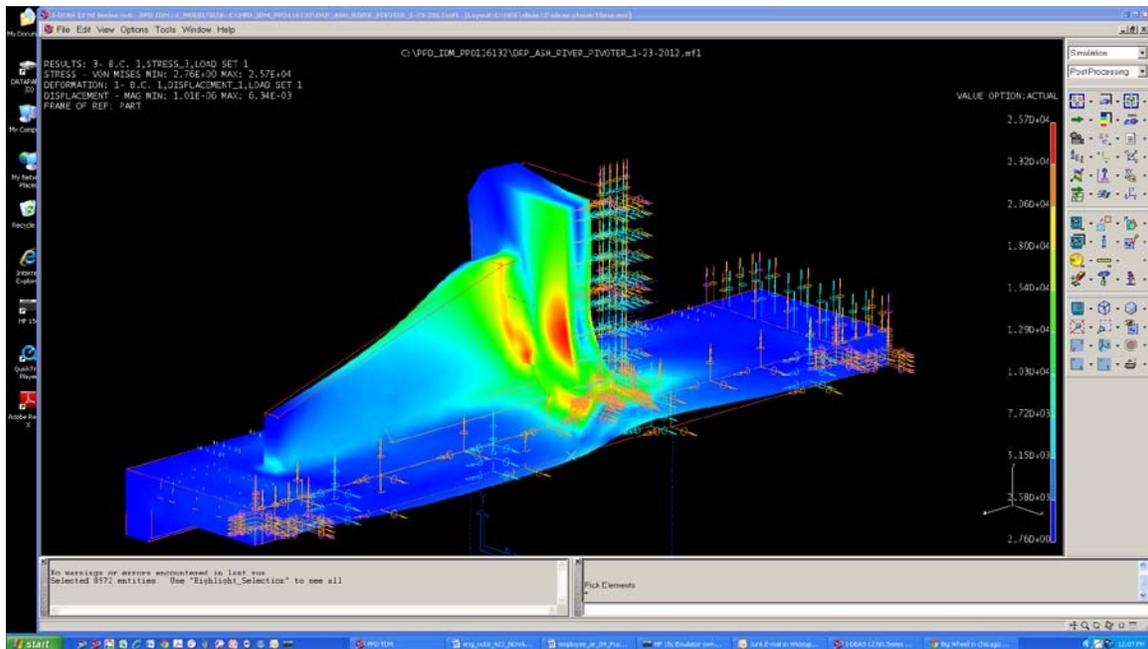
$$(50,000 \text{ pounds} / 0.2) / 20 \text{ set screws} = 12,500 \text{ pounds}$$

This is the worst case loading and does not consider the beneficial effect of the set screw cup point imbedding into the rail material.

Result is a nominal tensile field on the bottom side of the weldment horizontal plate on the order of 30 ksi and a compressive stress on the top of the weldment horizontal plate that is slightly less. Stress in the weld area is solidly in the 60 to 70 ksi range.

Assuming use of A36 material (36 ksi yield) and E70XX weld filler metal (70 ksi yield), this design puts the weldment stresses right at yield for the conservative loading condition with a low coefficient of friction between the set screws and rail.

Analysis of the jack loading on the pusher weldment – Hydraulic jack induced bending:



A half model was analyzed with a 25,000 pound load applied to the area under the hydraulic jack base. At the plane of symmetry, x displacements and y and z axis rotations were set to zero. At each of the set screw holes, the y and z displacements were set to zero, and on the bottom, near the back

end, the vertical (y) displacement was set to zero (implies contact with the top of the rail).

Result of the hydraulic jack loading on the weldment show bending stresses in the vertical plate to which the hydraulic jack bears at around 25 ksi. This value is below yield but above to the AISC allowable bending stress for A36 material of 60% of 36 ksi. Local peak stresses in the weld area also approach 25 ksi, well below the yield in E70XX filler metal. Other stresses are under 20 ksi.

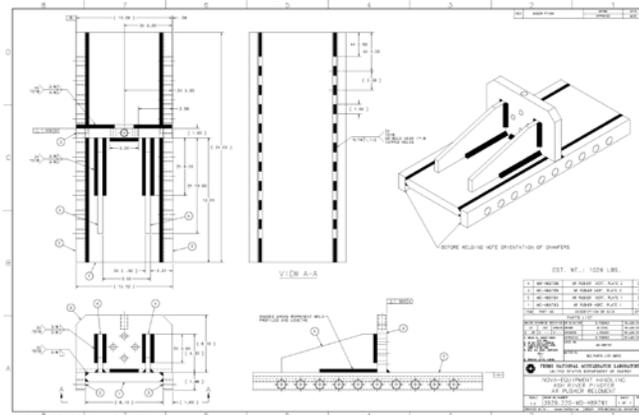
AISC ASD 9th Edition Commentary Chapter A, paragraph A5 Section 1 includes the sentence "It is not intended that highly localized peak stresses that may be determined by sophisticated computer-aided methods of analysis, and which may be blunted by confined yielding, must be less than the stipulated allowable stresses".

Conclusions:

Peak stresses in this weldment are above the AISC allowable values based on a very conservative loading condition used in this analysis. However, stresses in the bulk volume of the material are below the AISC allowable for the jack loading. Loading from the set screws results in bulk volume stress approximately equal to A36 material yield stress. Considering that this weldment is designed to be used on a floor mounted steel rail and is not important as a life safety device, local yielding is deemed acceptable.

Assembly drawing is 3929.220-MC-489767

Weldment drawing is 3929.220-MC-489761



Test Results:

Prior to shipping these fixtures to Ash River, they were tested in Lab F using the set up shown in the photograph below:



Test set up consisted of a short section of carbon steel bar (8 inches by 2 inches by 48 inches long) upon which both weldments were mounted. An existing (old) 25 ton, 8 inch stroke, Enerpac hydraulic cylinder was mounted on one weldment and pushed against the other (via the aluminum block shown in the photograph). All fixture set screws were tightened to approximately 100 foot-pounds.

Hydraulic cylinder was pressurized with an electric powered, Enerpac pumping unit.

Pressure was raised in 1000 psi increments until 9000 psi was reached (pressure gauge shown in the photograph was taken at a pressure of 9000 psi). This corresponds to a cylinder load of 22.5 tons. At 9000 psi, the leakage in the internal relief valve of the pumper unit prevented the pressure from going up further. However, the 22.5 ton load exceeds the calculated required load to move the pivoter, and the test was concluded.

Incidentally, the 2 by 8 inch bar showed noticeable deflection during the test due to the moment induced by the cylinder load being applied at an eccentric

