



**Particle Physics Division
Mechanical Department Engineering Note**

Number: MD-ENG-461 Date: 26 February 2013
updated 21 March 2013

Project Internal Reference: PPD doc db entry 1594

Project: NOvA NDUG Vacuum lifting fixture ID 187

Title: Lifting fixture

Author(s): Dave Pushka

Reviewer(s):

Key Words: Lifting fixture

Applicable Codes: FESHM 5022, ASME B30.20, ASME BTH

Abstract Summary:

FESHM 5022 requires lifting fixture calculations to show conformance with B30.20. These are included here. Calculations to show conformance with the relatively new and far more detailed ASME BTH are also included.

Fermilab Engineering Manual Requirements:

- Chapter 1: The specification given by Dervin Allen and Ting Miao is the verbal request to prepare the FESHM 5022 required lifting fixture engineering note for this object.
- Chapter 2: Risk analysis has not been formally prepared. Lifting fixture engineering notes have been prepared previously and are deemed 'low risk'.
- Chapter 3: Written requirements and specification reviews by the project have not been formally prepared. However, a specification review meeting was held at 9 am on 26 February 2013 at the Lab F blue room at 9 am with Ting Miao, Dervin Allen, Karen Kephart, Ernie Villegas, Don Friend, and Dave Pushka. All parties agreed that the 1500 pound load specification and the geometry requirement documented herein were optimum. Meeting adjourned at 10:02 am.
- Chapter 4: This engineering note has been specifically prepared to address chapter 4 of the engineering manual.
- Chapter 5: The 'reviewed by' signature on the cover page of this document addresses Chapter 5 of engineering manual.
- Chapter 6: No Materials were purchased as part of the preparation of this document.
- Chapter 7: See the lifting fixture load test sign off.
- Chapter 8: This lifting fixture is released to operations is required after successful load test and sign off.
- Chapter 9: Unless there are lessons learned from this effort, this engineering note and the other material posted in the PPD docdb will be considered the final written Project Report as described in Chapter 9. This note will be placed in the PPD document data bases as a means of Archiving and Control.

Discussion:

This fixture was designed by staff at Argonne National Laboratory (ANL). However, the analysis prepared by ANL was for the far detector lifting fixture and did not conform to the ASME B30.20 or ASME BTH or Fermilab Environment, Health and Safety Manual (FESHM) chapter 5022. This engineering note is intended to satisfy FESHM 5022 by showing conformance with ASME BTH.



The fixture uses a commercial vacuum system shown above.

Design Capacity:

Theoretical module weight = 283.75 lbs each

UMN measures all the material (manifolds, fiber, glue, endplate, etc.) as 8577.1 g. / module (18.91 lbs)

PVC measured weight equals, on average, 514.5 lbs for 610" extrusion. Multiply by two for a module, divide by the 610", multiply by 157" (length of NDUG PVC) = 264.84 lbs

Therefore, a module will weigh 265 + 19 pounds = 284 pounds.

This fixture will also need to lift a steel compression plate. Compression plate thickness is ½ inch, width is 52 inches, and length is estimated at 160 inches. Using a steel weight of 490 pounds per cubic foot, the compression plate will weigh no more than 1179 pounds.

Theoretical compression plate weight = 1200 lbs

Therefore, the minimum fixture design capacity is 1200 pounds.

Use a fixture capacity of 1500 pounds.

Calculations for Lifters ID numbers 187

ASME BTH Lifter Design Category: Category A

ASME BTH Lifter Service Class: Class 0 (0 to 20,000 load cycles)

ASME BTH Lifter Static Design Basis, $N_d = 2.0$

Box Section Beam Calculations:

Design Inputs:

Design capacity, $P = 1500$ pounds

Span, $L = 156$ inches

Beam Material: A500 grade B carbon steel

Size: 8" by 8" by 3/16" wall Square structural steel tubing.

Qty of box sections: Qty = 1

Allowable stress per ASME B30.20: $\sigma_{\text{allowable}} = F_y / 3 = 46,000 \text{ psi} / 3$

Allowable stress from ASME BTH section 3-2.3.4 Weak Axis bending of Compact Sections, Solid Bars, and Rectangular Sections.

The second paragraph in this section indicates that the rectangular tubes, or box shaped sections with compact flanges and webs as defined by Table 3-1, with the flanges continuously connected to the webs and bent about their weak axis, the allowable bending stress is given by equation (3-6).

Compact Section Check:

$$b/t = ((8-4 \cdot .1875) / .1875) = 7.25 / .1875 = 38.66$$

$$1.12 \text{ SQRT } (E/F_y) = 1.12 * \text{SQRT } (29,000/46) = 28.12$$

$$1.40 \text{ SQRT } (E/F_y) = 1.40 * \text{SQRT } (29,000/46) = 35.15$$

$b/t >$ the compact section criteria, therefore, this is not a compact section. This is a slender section.

$$\text{Eqn (3-11): } L_r = 2 * r_y * E * \text{SQRT}(JA) / F_y * S_x$$

Where:

$$r_y = 3.18 \text{ in}$$
$$E = 29,000,000 \text{ psi}$$
$$J = 90.6 \text{ in}^4$$
$$A = 5.77 \text{ in}^2$$
$$F_y = 46,000 \text{ psi}$$
$$S_x = 14.6 \text{ in}^3$$

$$L_r = \frac{2 * 3.18 \text{ in} * 29,000,000 \text{ psi} * \text{SQRT}(90.6 \text{ in}^4 * 5.77 \text{ in}^2)}{46,000 \text{ psi} * 14.6 \text{ in}^3}$$

$$L_r = 6279 \text{ inches}$$

the maximum unbraced length criteria is much greater than the length of the fixture.

$$\text{Eqn (3-9): } F_b = F_y / N_d$$
$$F_b = 46,000 \text{ psi} / 2.0 = 23,000 \text{ psi}$$

Therefore, the allowable bending stress from BTH is 23,000 psi. B30.20 requires the bending stress remain less than 15,333 psi.

Use the lesser bending stress criteria:

$$\text{B30.20 criteria is } \sigma_{\text{allowable}} = F_y / 3 = 15,333 \text{ psi}$$

$$\text{BTH criteria is } F_b = 46,000 \text{ psi} / 2.0 = 23,000 \text{ psi}$$

Use bending stress not greater than 15,300 psi

In calculating the bending stress, assume the load is 1500 pounds at the center and the reactions occur at the ends of the beam. This is a conservative assumption and uses case 7 in AISC 9th edition.

$$M_{\text{max}} = P * L / 4 = 1500 \text{ pounds} * 156 \text{ inches} / 4 = 58,500 \text{ inch-pounds}$$

$$\sigma_{\text{bending}} = M / S_x = 58,500 \text{ inch-pounds} / 14.6 \text{ in}^3 = 4006.9 \text{ psi.}$$

$\sigma_{\text{bending}} < \sigma_{\text{allowable}}$ therefore, bending stress on the main beam is okay.

Tube Section:

Check Bending when supported by the extended 6 inch o.d. Tube:

Design capacity,	P = 1500 pounds
Span,	L = 205 inches
Beam Material:	A513 Type 5 carbon steel SAE designation 1026
Size:	6" diameter by 0.5" wall DOM round welded structural steel tubing.
Qty of sections:	Qty = 1
$S_x = 0.098175 (d_o^4 - d_i^4)/d$	10.97 in ³
Allowable stress per ASME B30.20:	$\sigma_{allowable} = F_y / 3 = 65,000 \text{ psi} / 3 = 21,660 \text{ psi}$

$$M_{max} = P * L / 4 = 1500 \text{ pounds} * 205 \text{ inches} / 4 = 76,875 \text{ inch-pounds}$$

$$\sigma_{bending} = M / S_x = 76,875 \text{ inch-pounds} / 10.97 \text{ in}^3 = 7001.9 \text{ psi.}$$

$\sigma_{bending} < \sigma_{allowable}$ therefore, bending stress on the tube is okay.

Check Cantilevered Box Section Supporting the 8 small 5.5 inch diameter Vacuum Cups:

These cups are intended to be used for lifting the extrusion. However, there is not an apparent reason why they could not be mistakenly used for lifting the heavier steel plate. So, use the weight of the steel plate in this analysis as a conservative assumption:

Consider the steel plate supported by 8 cups, each with the same load of $1500/8 = 187$ pounds.

Design load,	P = 187 pounds
Span,	L = 16 inches
Beam Material:	A500 grade B carbon steel
Size:	2" by 2" by 1/4" wall Square structural steel tubing.
Qty of box sections:	Qty = 1

Allowable stress per ASME B30.20: $\sigma_{\text{allowable}} = F_y / 3 = 46,000 \text{ psi} / 3 = 15,300 \text{ psi}$

$$M_{\text{max}} = P * L = 187.5 \text{ pounds} * 16 \text{ inches} = 3000 \text{ inch-pounds}$$

$$\sigma_{\text{bending}} = M / S_x = 3000 \text{ inch-pounds} / 0.766 \text{ in}^3 = 3916 \text{ psi.}$$

$\sigma_{\text{bending}} < \sigma_{\text{allowable}}$ therefore, bending stress on the cantilevered box section beam is okay.

Check when safety catches are used, the length becomes 22 inches (52 inches – 8 inches) / 2:

$$M_{\text{max}} = P * L = 187.5 \text{ pounds} * 22 \text{ inches} = 4125 \text{ inch-pounds}$$

$$\sigma_{\text{bending}} = M / S_x = 4125 \text{ inch-pounds} / 0.766 \text{ in}^3 = 5385 \text{ psi.}$$

$\sigma_{\text{bending}} < \sigma_{\text{allowable}}$ therefore, bending stress on the cantilevered box section beam is okay.

Check Torsion due to the offset of the cups from the section centerline:

Design load,	P = 187 pounds
Lever arm,	L = 3 inches
Moment, T =	T = 561 in-lbs
J (in ⁴):	1.36.
Radius, r (in)	0.694
Torsion Stress, $\tau = T*r / J$:	286 psi

$\tau_{\text{actual}} \ll \tau_{\text{allowable}}$ therefore, torsional stress on the cantilevered box section due to the offset of the cup with respect to the box section centerline is okay.

Consider combined normal and shear stresses as addressed in section 3-2.5 of ASME BTH:

$$f_{cr} = \text{SQRT}(f_x^2 - f_x * f_y + f_y^2 + 3f_v^2) < F_{cr} = F_{\text{yield}} / N_d$$

$$f_x = \sigma_{\text{bending}} = 3916 \text{ psi}$$

$$f_y = 0$$

$$f_v = \tau_{\text{actual}} = 286 \text{ psi}$$

$$f_{cr} = \text{SQRT}(3916^2 - 0 + 0 + 3 * 286^2) = 3947 \text{ psi}$$

$$3947 < 46,000 / 2 \quad \text{Therefore, this is acceptable.}$$

Note that the safety catches are in line with the 2 inch box section centerline and therefore the torsional stress does not occur when the safety catches are loaded. This is why the 3916 psi stress and not the 5385 psi stress is used in the combined stress calculation.

Consider the capacity of the small vacuum cups mistakenly used for the steel plate (the round ones):

Cup diameter:	5.5 inches
Cup Area:	23.75 square inches
Cup Load:	187 pounds
Cup load per unit area:	7.87 psi

Vacuum system is designed for 14 psi vacuum and the UPC as defined by eqn 4-18 of BTH is:

$$UPC = A * V_p = \text{Pad area} * \text{Vacuum at the pad}$$

$$V_p = 14 \text{ psi}$$

$$A = 23.75 \text{ square inches}$$

$$UPC = 332 \text{ pounds}$$

Maximum vacuum pad rating is VPR

$$VPR = UPC / N_v$$

$$N_v = 2 + 2 \sin \theta$$

$$\theta = 0$$

$$N_v = 2$$

$$VPR = UPC / N_v = 332 / 2 = 166 \text{ pounds}$$

Conclusion: Using the 8 small cups to lift a steel plate exceeds the allowable load on these cups. So if a mistake is made and these cups are used on the steel plate, the safety factor is reduced below the code specified values.

Image of the small, round vacuum cup intended to be used with the extrusions:



Check Cantilevered "I" Section Supporting the 6 large 9 inch diameter Vacuum Cups:

These cups are intended to be used for lifting the steel plates:

Consider the steel plate supported by 6 cups, each with the same load of $1500/6 = 250$ pounds.

Design load,	P = 250 pounds
Span,	L = 8 inches
Beam Material:	A36 carbon steel
Size:	S3x5.7
Qty of sections:	Qty = 1

Allowable stress per ASME B30.20: $\sigma_{\text{allowable}} = F_y / 3 = 36,000 \text{ psi} / 3 = 12,000 \text{ psi}$

$M_{\text{max}} = P * L = 250 \text{ pounds} * 8 \text{ inches} = 2000 \text{ inch-pounds}$

$\sigma_{\text{bending}} = M / S_x = 2000 \text{ inch-pounds} / 1.68 \text{ in}^3 = 1190 \text{ psi}$.

$\sigma_{\text{bending}} < \sigma_{\text{allowable}}$ therefore, bending stress on the cantilevered box section beam is okay.

Safety catches are not used on these members.

End view of the I Section:



Check Torsion due to the offset of the cups from the section centerline:

Note, ASME BTH does not address this loading condition, because it is generally considered bad design to use an open section (angle, channel or I beam) in torsion; therefore the formula for this come from Roark and Young – see section 10.7 in the 7th edition which applies to wide flange sections. See table 10.2, case 6. For the boundary conditions, see table 10.3, case 1a. The section used is a Standard section (S), but will approximate using the Roark and Young formula):

Design load, P =	250 pounds
Lever arm, L =	12 inches
Applied torque, T ₀ =	3000 in-lbs
Length of S section, l =	7 inches
Shear Modulus, G =	11,500,000 psi
J (in ⁴):	0.04
Radius, r (in)	1.23
b =	2.33
h =	3.0
E =	29,000,000
t =	0.26
t _w =	0.17
K =	$\frac{1}{3} * (2 * t^3 * b + t_w^3 * h)$
K =	$\frac{1}{3} * (2 * .26^3 * 2.33 + .17^3 * 3.0)$ = 0.0322
C _w =	$\frac{h^2 * t * b^3}{24}$ = $\frac{3.0^2 * .26 * 2.33^3}{24}$ = 1.23
β =	$(K * G / C_w * E)^{1/2}$ = $(0.0322 * 11,500,00 / 1.23 * 29,000,000)^{1/2}$ 0.1019
β ² =	0.0104

$$\Theta_{\max} = \frac{T_0 * I}{C_w * E * \beta^2}$$

$$= \frac{3000 * 7}{1.23 * 29,000,000 * 0.0104}$$

$$= 0.0567 \text{ radians}$$

$$= 3.24 \text{ degrees}$$

$$\Theta'_{\max} = \frac{-T_0}{C_w * E * \beta^2}$$

$$= \frac{3000}{1.23 * 29,000,000 * 0.0104}$$

$$= -0.0081$$

$$\text{Shear Stress, } \tau_{\max} = t * G * \Theta'_{\max}$$

$$= 0.26 * 11,500,000 * 0.0081$$

$$= 24,219 \text{ psi}$$

$\tau_{\max} > \tau_{\text{allowable}}$ therefore, torsional stress on the cantilevered "I" section due to the offset of the cup with respect to the box section centerline is too high and represents a design error.

Consider combined normal and shear stresses as addressed in section 3-2.5 of ASME BTH:

$$f_{cr} = \text{SQRT}(f_x^2 - f_x * f_y + f_y^2 + 3f_v^2) < F_{cr} = F_{\text{yield}} / N_d$$

$$f_x = \sigma_{\text{bending}} = 1190 \text{ psi}$$

$$f_y = 0$$

$$f_v = \tau_{\text{actual}} = 24,219 \text{ psi}$$

$$f_{cr} = \text{SQRT}(1190^2 - 0 + 0 + 3 * 24,219^2) = \text{psi}$$

41,965 >> 12,000 Therefore, this is not acceptable.

Because the original design by Argonne does not meet ASME BTH, the following few sections are added to address a modification to replace the S sections with box sections and meet the code.

Check Cantilevered Box Section to Support the 6 large 9 inch diameter Vacuum Cups:

These cups are intended to be used for lifting the steel plates:

Consider the steel plate supported by 6 cups, each with the same load of $1500/6 = 250$ pounds.

Design load,	P = 250 pounds
Span,	L = 8 inches
Beam Material:	A500 Box section carbon steel
Size:	2 x 2 x 3/16 wall A500 Box section
Qty of sections:	Qty = 1
S_x	0.668 in ³
Allowable stress per ASME B30.20:	$\sigma_{\text{allowable}} = F_y / 3 = 46,000 \text{ psi} / 3 = 15,300 \text{ psi}$

$$M_{\text{max}} = P * L = 250 \text{ pounds} * 8 \text{ inches} = 2000 \text{ inch-pounds}$$

$$\sigma_{\text{bending}} = M / S_x = 2000 \text{ inch-pounds} / 0.668 \text{ in}^3 = 2994 \text{ psi.}$$

$\sigma_{\text{bending}} < \sigma_{\text{allowable}}$ therefore, bending stress on the cantilevered box section beam is okay.

Check Torsion due to the offset of the cups from the section centerline:

Design load,	P = 250 pounds
Lever arm,	L = 3 inches
Moment, T =	T = 750 in-lbs
J (in ⁴):	1.15.
Radius, r (in)	0.726
Torsion Stress, $\tau = T*r / J$:	473 psi

$\tau_{\text{actual}} \ll \tau_{\text{allowable}}$ therefore, torsional stress on the cantilevered box section due to the offset of the cup with respect to the box section centerline is okay.

Consider combined normal and shear stresses as addressed in section 3-2.5 of ASME BTH:

$$f_{cr} = \text{SQRT}(f_x^2 - f_x f_y + f_y^2 + 3f_v^2) < F_{cr} = F_{\text{yield}} / N_d$$

$$f_x = \sigma_{\text{bending}} = 2994 \text{ psi}$$

$$f_y = 0$$

$$f_v = \tau_{\text{actual}} = 473 \text{ psi}$$

$$f_{cr} = \text{SQRT}(2994^2 - 0 + 0 + 3 \cdot 473^2) = 3104 \text{ psi}$$

$$3104 < 46,000 / 2 \quad \text{Therefore, this is acceptable.}$$

Note that if safety catches were used for the steel plates and they were installed at the ends of the 2 inch box sections, the resulting bending stress would be:

Design load,

$$P = 250 \text{ pounds}$$

Span,

$$L = 20 \text{ inches}$$

$$M_{\text{max}} = P * L = 250 \text{ pounds} * 20 \text{ inches} = 5000 \text{ inch-pounds}$$

$$\sigma_{\text{bending}} = M / S_x = 5000 \text{ inch-pounds} / 0.668 \text{ in}^3 = 7485 \text{ psi.}$$

$\sigma_{\text{bending}} < \sigma_{\text{allowable}}$ therefore, bending stress on the cantilevered box section beam is okay.

So, for the steel plates, safety catches could be added and safely supported by these 2 inch box sections.

New section added 3-20-2013 – Vendor failed to provide identical vacuum cups and so a new size will have to be ordered. This section addresses the capacity of the new cups.

Consider the capacity of the alternate vacuum cups used for the steel plate (the blue round ones) Anvar part number PS70S-NBR200 at \$206 each:

Cup diameter:	7.13 inches
Cup Area: inches	39.92 square
Cup rated Load:	160 pounds
Cup load per unit area:	4.0 psi
Number of cups per compression plate section:	4
Weight of a compression plate section:	326 pounds
Weight per cup:	81.67 pounds

Vacuum system is designed for 14 psi vacuum and the UPC as defined by eqn 4-18 of BTH is:

$$UPC = A * V_p = \text{Pad area} * \text{Vacuum at the pad}$$

$$V_p = 14 \text{ psi}$$

$$A = 39.92 \text{ square inches}$$

$$UPC = 558 \text{ pounds}$$

Maximum vacuum pad rating is VPR

$$VPR = UPC / N_v$$

$$N_v = 2 + 2 \sin \theta$$

$$\theta = 0$$

$$N_v = 2$$

$$VPR = UPC / N_v = 558 / 2 = 279 \text{ pounds}$$

Conclusion: Using the 12 large cups to lift a steel plate is acceptable

The steel plate is made of three pieces, each 48 inches square by ½ inch thick with 4 cups per plate piece (or section).

Consider the capacity of the large vacuum cups used for the steel plate (the blue round ones) PS90S-NBR200 at \$195 each:

Cup diameter:	9 inches
Cup Area:	63.6 square inches
Cup Load:	250 pounds
Cup load per unit area:	3.92 psi

Vacuum system is designed for 14 psi vacuum and the UPC as defined by eqn 4-18 of BTH is:

$$UPC = A * V_p = \text{Pad area} * \text{Vacuum at the pad}$$

$$V_p = 14 \text{ psi}$$

$$A = 63.6 \text{ square inches}$$

$$UPC = 890 \text{ pounds}$$

Maximum vacuum pad rating is VPR

$$VPR = UPC / N_v$$

$$N_v = 2 + 2 \sin \theta$$

$$\theta = 0$$

$$N_v = 2$$

$$VPR = UPC / N_v = 890 / 2 = 445 \text{ pounds}$$

Conclusion: Using the 6 large cups to lift a steel plate is acceptable

However, the steel plate is made of three pieces, each 48 inches square by ½ inch thick. Original design used two cups for each compression plate section. Ideally, three or four cups should be used with each plate to reduce the rotation of the compression plate due to differences between the plate center of gravity with respect to line drawn between only two cups.

If only two cups remain used per compression plate, a means to determine the maximum rotation of each compression plate is needed so that the cup rating can be de-rated per ASME BTH section 4-10. This is not possible as one cannot predict the maximum offset that a technician may inadvertently apply. So, use four cups per plate, this puts the center of gravity of the plate between the cups.

Therefore the plate rotation is near zero and the de-rating factor for the rotation angle θ in section 4-10 is negligible. Loads calculated above are reduced by a factor of two and remain acceptable.

Image of the large, round vacuum cup intended to be used with the steel plates:



These cups have a model or part number clearly shown on them:



Comparing these to the Anver vacuum catalog (see: <http://www.anver.com/document/vacuum%20components/vacuum%20cups/suspensions.htm>) these appear to be a ball mount, swivel suspension with a part number of PS90S-NBR200 and the vacuum cup is a part number VP90-4.

Vendor rated load for this cup and swivel mount assembly is 270 pounds.



The center lug (shown above) is part of the commercial vacuum lifting device and so calculations to show conformance with the BTH code are omitted from this document.

Manufacture's published literature indicates the capacity of this Vack Pack is 10,000 pounds. See:

http://www.anver.com/document/vacuum%20components/vacuum%20generators/vpf3_vacpack.htm

Bolts connecting the commercial portion of the fixture to the box beam:



A total of eight bolts attach the commercial vacuum lifting device Anver Vack Pack to the weldment. These bolts are 3/8 inch diameter and are loaded in shear. Check the bolt shear.

Total load is conservatively the weight of the fixture (1200 pounds) plus the weight of the fixture capacity (1500 pounds) less the weight of the Vack Pack (247 pounds):

$$P = 1200 + 1500 - 247 = 2453 \text{ pounds}$$

Bolts appear to be grade 8 socket head cap screws with a minimum tensile strength of 120,000 psi.

From ASME BTH section 3-3.2, the allowable shear stress is:

$$F_v = 0.62 * F_u / 1.20 * N_d$$

$$F_{v \text{ allowable}} = 0.62 * 120,000 / 1.20 * 2.0 = 31,000 \text{ psi}$$

It is not clear if the shear plane passes thru the bolt shank or passes thru the threaded portion. Therefore, assume the shear plane goes thru the threaded portion and uses the root area.

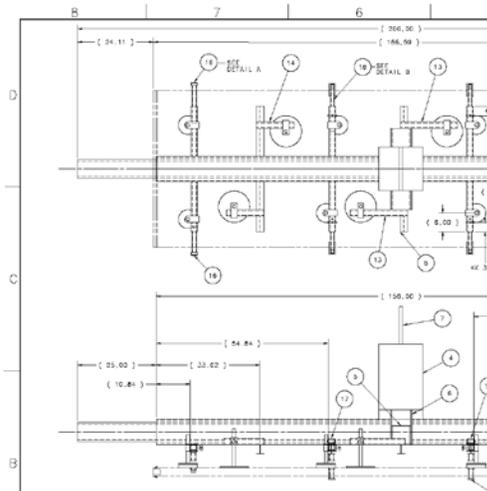
$$A = 0.0678 \text{ in}^2$$

Number of bolts = 8

$$F_{v \text{ actual}} = 2453 / 8 * 0.0678 = 4522 \text{ psi.}$$

$F_{v \text{ actual}} \ll F_{v \text{ allowable}}$ therefore, okay.

Documentation drawing for As Received condition of Lifting Fixture with ID number 187:



A PDF of this drawing is available at: http://www-admscad.fnal.gov/MSDMain/cgi-bin/TP_PPDifind-web.pl

Once this page is opened, type in the last six digits of the drawing number (493448) in the search box to access this drawing.