DRAFT NEXT-100 Pressure Vessel

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1 Abstract

The target of this document is fabricate NEXT100 where the designed produced must be feasible, functional and realiable.
For a given geometry, materials (titanium and Inconel) and gasket (Hellicoflex from Garlock) this paper applies the rules of design specified by ASME VIII, Division 1, Appendix 2, Appendix Y (nonmandatory) and Division 2, Part 4.16 for fixing thickness.

Contents

2 Data Design

2.1 Pressure

Table 1: Pressure

Internal Pressure,Mpa	Ext.Pressure,Mpa
1,52	0,1

2.2 Materials

Material:

- Titanium Grade 2,R50400(shell and flange)
- Inconel 718,N07718(bolts)

From ASME II :

Table 2: Table Y1

Material	Specification	Denomination	YS(MPa)	TS(Mpa)
Ti,Grade2	SB 265	R50400	275,6	344,5

Table 3: Titanium

Material	Table	Section	Division	Maximum allowable stress value,Mpa
Ti,Grade2,plate	1B	ASMEVIII	1	S=98,5
Ti,Grade2,plate	5B	ASMEVIII	1	Sm=143,3

Table 4: Inconel 750(N07750):70Ni-16Cr-7Fe-Ti-Al

Material	Table	Section	Division	Min Y.S,Mpa	Min TS, MPa	Max all.Stress,Mpa
N07750	3	ASMEVIII	1and 2	1033,5	1274,6	255

Table 5: Module of elasticity E,MPa

Material	Table	E,MPa
Ti,R50400	TM-4	106795

Table 6: Gasket,Helicoflex

Al jacket,CS	5mm
Yi=Ym1	30N/mm
Y2	220N/mm
e_2	0,9mm
P_u	63
Y_m	30
$G_g$groove depth	6,78mm
$F_g$groove width	4,1mm

2.3 Dimension Vessel

Table 7: Geometric Values

Inside diameter(mandatory)	B=1140mm
Diameter at location of gasket load reaction0	G= 1147,5mm
Bolt-circle diameter	C=1235mm
Outside diameter of flange	A=1370mm

3 Calculation

3.1 Calculation of the thickness

3.1.1 Cylindrical Shell

2007 Section VIII, Division 2 4.3.3 **Cylindrical Shell**

$$t = \frac{D}{2}\left[\exp\left(\frac{P}{SE}\right) - 1\right] = \frac{D}{2}\left[\exp\left(\frac{P}{SE}\right) - 1\right] = 6,15mm \tag{1}$$

The thickness of the shell = 10mm

Where :

- D=outside diameter of a shell or head.
- P=internal design pressure.
- S=allowable stress value from Annex 3.A evaluated at the design temperature.
- E=weld joint factor.

3.1.2 Torishperical Head

2007 Section VIII, Division 2 4.3.6 **Torispherical Head**

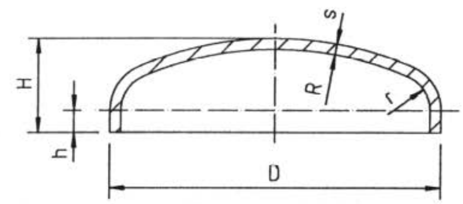
We choose Torisphericla Head DIN 28011:

Figure 1: DIN 28011

2. Decimal (torispherical type) heads acc. DIN 28011

2.1 Profile

D	=	outside dimension	
s	=	wall thickness before forming	
r	=	inside knuckle radius	(0,1D)
R	=	inside radius	(R=D)
h	=	Straight flange height	(∃3,5 x s)
H	=	total height	(–0,192D + s + h)



- **STEP 1:**Determine the inside diameter, D, and assume values for the crown radius, L , the knuckle radius, r , and the wall thickness t

  D= 1140mm

  L=D +2t=1140 +16=1156mm

  r=01D=114mm

  t=8mm

- **STEP 2:**Compute the head L/D, r/D, and L/t ratios and determine if the following equations are satisfied. If the equations are satisfied, then proceed to Step 3.
  The following eqautins are satified:

  $0,7 \leq \frac{L}{D} \leq$ 1,0

  $\frac{r}{D} \geq \ 0,006$

  $20 \leq \frac{L}{t} \leq 2000$

STEP 3:$Calculate the following geometric constants:$

$\beta th = 58,25 radians$

$\phi th = 58,25 radians$

$Rth = 913,83 radians for\ \phi th \leq \beta th$

- **STEP 4:**Compute the coefficients C1 and C2
- **STEP 5:**Calculate the value of internal pressure expected to produce elastic buckling of the knuckle
  P eth=15,18 bars
- **STEP 6:**Calculate the value of internal pressure that will result in a maximum stress in the knuckle equal to the material yield strength

  $Py = 1,752 bars$

  If the allowable stress at the design temperature is governed by time-independent properties, then C3 is the material yield strength at the design temperature, or C3= Sy = 379 Mpa
- **STEP 7:**Calculate the value internal pressure expected to result in a buckling failure of the knuckle
  $G = 8,66$

  $Pck = 13,41 bars$
- **STEP 8:**Calculate the allowable pressure based on a buckling failure of the knuckle
  $Pak = 2,27 bars$
- **STEP 9:**Calculate the allowable pressure based on rupture of the crown.
  $Pac = 2,809 bars$
- **STEP 10:**Calculate the maximum allowable internal pressure

  $Pa = mim\,(Pak, Pac) = 2,27 bars$
- **STEP 11:**If the allowable internal pressure compute from STEP 10 is greater than or equal to the design pressure, then the design is complete.

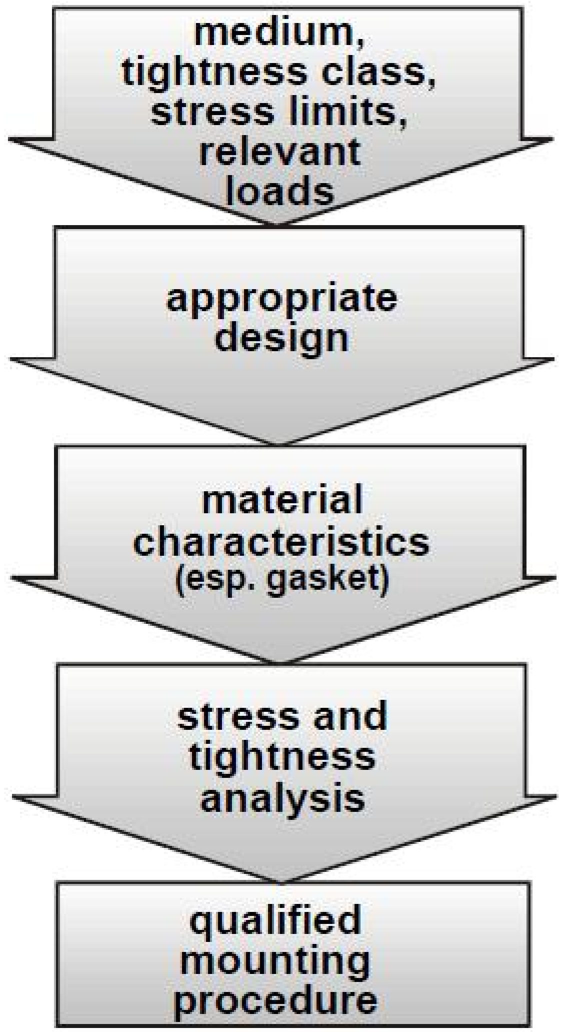
OK t= 7mm but We take t= 8mm

3.2 Calculation of the flanges

3.2.1 Background

On the whole, the steps for design a flange are:
With the vessel for the NEXT 100s project there is another stage, a previous phase: **feasibility**.

Figure 2: Step Flange



We can follow the five steps and to find a design no feasible, or feasible with a huge cost.

The pressure vessel flanges had been constructed with steel, cooper and aluminium, but not with titanium, the titanium has been used in pressure vessels as cladding material. This fact calls for great concentration and to keep a wary eye on it

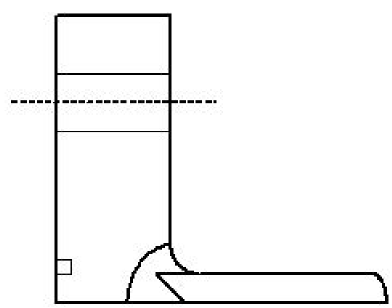
The more common flanges are hubbed flanges made by forging, steel, aluminium and cooper flanges are standardised. It is possible **to find a titanium flange with hub with a reasonable cost?** From ASM Handbook, Volume 14, Forming and Forging:

Figure 3: ASM Handbook

TITANIUM ALLOYS are forged into a variety of shapes and types of forgings, with a broad range of final part forging design criteria based on the intended application. As a class of materials, titanium alloys are among the most difficult metal alloys to forge, ranking behind only refractory metals and nickel/cobalt-base superalloys. Therefore, titanium alloy forgings, particularly closed-die forgings, are typically produced to less highly refined final forging configurations than are typical of aluminum alloys (although precision forgings in titanium alloys are produced to the same design and tolerance criteria as aluminum alloys; see the section "Titanium Alloy Precision Forgings" in this article) and to equivalent or more refined forging design sophistication than carbon or low-alloy steel forgings, because of reduced oxidation or scaling tendencies in heating. Because of the high cost of titanium alloys in comparison to other commonly forged materials, such as aluminum and alloy steels, final forging design criteria in titanium closed-die forgings are typically balanced between producibility demands and cost considerations (particularly machining costs and overall metal recovery).

We propose, bearing in mind the feasibility, an integral narrow flange without hub jointed with the cylindrical shell with a welded joint (full penetration).

Figure 4: Integral Narrow



This flange can be fabricated with plate (cutting, welding and machining). Another solution is constructing the flange with a hot rolled billet bended and welded. The first process is easier, although produces more scrap.

Table 8: Fabrication of flanges ring

Process	Billet	Plate	
Cutting	−	+	Abrasive water-jet cutting.Garnet and silica the most common abrasive materials used .The central circle shall be scraped.
Heating	+	−	Perhaps 815C, minimum forging temperature for C.P. titanium.
Hot forming	+		
Welding ring	+	−	End to end of billet, after forming.High risk of lack of fusion difficulties for radiographic inspection (high thickness).
Welding flange shell	+	+	
Machining	+++	+	

Note: +++ more difficult than +.
The use of plate for a flange without hub must be justified:ASME Code treats the use of plate for flanges

Figure 5: 2010 ASME VIII,Div1,Mandatory Appendix2

(d) Fabricated hubbed flanges shall be in accordance with the following:

(1) Hubbed flanges may be machined from a hot rolled or forged billet or forged bar. The axis of the finished flange shall be parallel to the long axis of the original billet or bar. (This is not intended to imply that the axis of the finished flange and the original billet must be concentric.)

(2) Hubbed flanges [except as permitted in (1) above] shall not be machined from plate or bar stock material unless the material has been formed into a ring, and further provided that:

(a) in a ring formed from plate, the original plate surfaces are parallel to the axis of the finished flange. (This is not intended to imply that the original plate surface be present in the finished flange.)

(b) the joints in the ring are welded butt joints that conform to the requirements of this Division. Thickness to be used to determine postweld heat treatment and radiography requirements shall be the lesser of

$$t \text{ or } \frac{(A - B)}{2}$$

Figure 6: 2010 ASME VIII,Div2,Part4.16

4.16.4 Flange Materials

4.16.4.1 Materials used in the construction of bolted flange connections, excluding gasket materials, shall comply with the requirements given in Part 3.

4.16.4.2 Flanges made from ferritic steel shall be given a normalizing or full-annealing heat treatment when the thickness of the flange section exceeds 75 mm (3 in.).

4.16.4.3 Fabricated flanges with hub shall be in accordance with the following:

a) Flanges with hubs may be machined from a hot rolled or forged billet or forged bar. The axis of the finished flange shall be parallel to the long axis of the original billet or bar, but these axes need not be concentric.

b) Flanges with hubs, except as permitted in paragraph 4.16.4.3.a, shall not be machined from plate or bar stock material unless the material has been formed into a ring, and further provided that:

   1) In a ring formed from plate, the original plate surfaces are parallel to the axis of the finished flange;

   2) The joints in the ring are welded butt joints that conform to the requirements of Part 6. The thickness to be used to determine postweld heat treatment and radiographic requirements shall be $\min\left[t, (A-B)/2\right]$.

The use of plate for flanges with hubs is forbidden by ASME, but noting about flanges without hubs.

The European Standard EN 13445-3: 2010, Unfired pressure vessels. Part 3, Design, as ASME, forbids (but not in the strict sense of the word) the use of plate for flanges with hubs (see 11.4.4).

On the other hand, ASME allows the use of plate in the fabrication of hubs for flat heads (not for flanges) from plate.

Figure 7: 2010 ASME VIII,Div2,Part3

3.9 Supplemental Requirements for Hubs Machined From Plate

3.9.1 General

The supplemental requirements of paragraph 3.9 are required for plate materials that are used in the fabrication of hubs for tubesheets, lap joint stub ends, and flat heads machined from plate when the hub length is in the through thickness direction of the plate.

3.9.2 Material Requirements

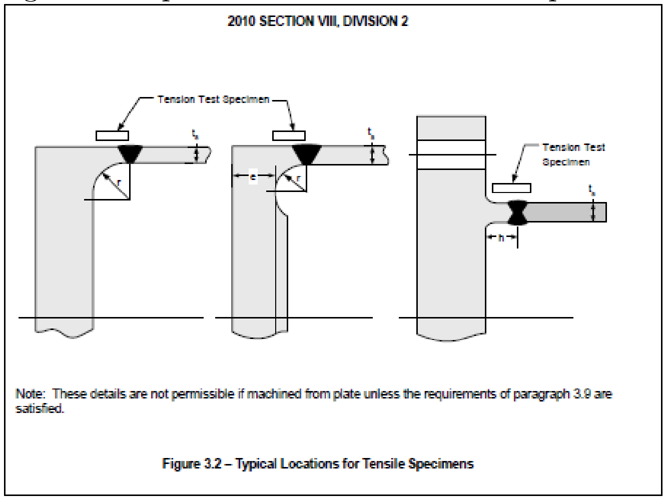
3.9.2.1 Plate shall be manufactured by a process that produces material having through thickness properties which are at least equal to those specified in the material specification. Such plate can be but is not limited to that produced by methods such as electroslag (ESR) and vacuum arc re-melt (VAR). The plate must be tested and examined in accordance with the requirements of the material specification and the additional requirements specified in the following paragraphs.

3.9.2.2 Test specimens, in addition to those required by the material specifications, shall be taken in a direction parallel to the axis of the hub and as close to the hub as practical, as shown in Figure 3.2. At least two tensile test specimens shall be taken from the plate in the proximity of the hub, with one specimen taken from the center third of the plate width as rolled, and the second specimen taken at 90° around the circumference from the other specimen. Both specimens shall meet the mechanical property requirements of the material specification. For carbon and low alloy steels, the reduction of area shall not be less than 30%; for those materials for which the material specification requires a reduction of area value greater than 30%, the higher value shall be met.

3.9.2.3 Subsize test specimens conforming to the requirements of SA-370, Figure 5 may be used if necessary, in which case the value for percent elongation in 50 mm (2 in.), required by the material specification, shall apply to the gage length specified in SA-370, Figure 5.

3.9.2.4 Tension test specimen locations are shown in Figure 3.2.

Figure 8: Tipical Locations for Tensile Specimens



The use of plate in the fabrication of the flange come up a problem: due to the fibrous structure of wrought products, the mechanical properties may be different for different orientations of the test specimen with respect to the fiber (working) direction. These phenomena (planar anisotropy) can be controlled breaking three specimens of the plate in the tension test, with 0 respect to rolling direction, 45 and 90 respectively.
Another question is to choose the appropriate design. We have three alternatives:

- 1 ASME VIII, Division 1: Mandatory Appendix 2: Rules for bolted flange connections with ring type gaskets;
- 2 ASME VIII, Division 1: Nonmandatory Appendix Y: Flat flanges with metal-to-metal contact outside the bolt circle; and
- 3 ASME VIII, Division 2: Part 4.16: Design Rules for Flanged Joints.

The rules of Appendix 2 and Part 4.16 apply specifically to the design of bolted flange connections with gaskets that are entirely within the circle enclosed by the bolt holes and whit no contact outside this circle (floating gasket)

The Nonmandatory Appendix Y applies to bolted flanged connections where the flanges are flat faced and are in uniform metal-to-metal contact across their entire face during assembly before the bolts are tightened or after a small amount of preload is applied to compress the gasket. This Appendix assume that the gasket seating loads are small and may be in most cases be neglected.

An additional advantage of flanges with metal-to metal contact is than these flanges need less thickness and less outside diameter, that is, more radiopurity, and less cost.
We have choose Appendix Y to calculate teh thickness of the flange:

Figure 9: Flange Types

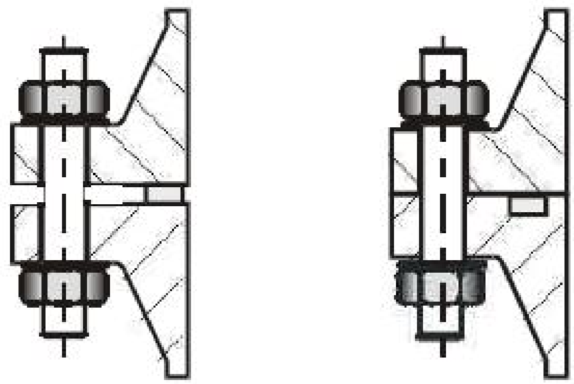


Fig. 3 : flange type with floating gasket (left) and flange type with metal-to-metal contact (right)

3.2.2 Face-to face flange,division 1,nonmandatory appendix Y, class 1,category 1

Figure 10: Face-to face flange

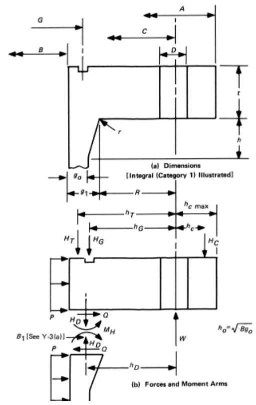
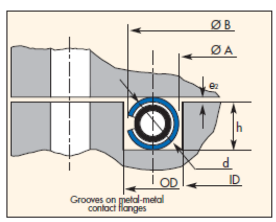


Figure 11: Gasket Helicoflex

AXIAL COMPRESSION



Internal pressure P ≥ 20 bars

• Groove assembly : at the end of compression the seal should be in contact with and supported by the groove wall opposed to the pressure side.

Internal pressure
h = d - e2
Ø B = OD - J
Ø A ≥ ID + 2e2 + 0,5

J = clearance between seal OD and Groove OD
J = f (d) (see table)

J	Ø d (C.S.)
0,3	1,5 à 3,4
0,5	3,5 à 6,9
0,7	7 à 9,9
0,9	> 10

External pressure P ≥ 20 bars

It is the groove ID which has to contact & support the seal

Pressure P < 20 bars

The seal can be centered in the groove without radial contact with a minimum groove width
w ≥ Seal cross section + 2e2 + 0,5 mm

Figure 12: Gasket Helicoflex Aluminium

	HELIUM SEALING							BUBBLE SEALING				
Jacket Material	Cross Section	e_2	e_c	Y_2 N/mm	Y_1 N/mm	Pu20°C MPa	PuΘ200°C MPa	Y_2 N/mm	Y_1 N/mm	Pu20°C MPa	PuΘ200°C MPa	Max Temp °C
Aluminum	1.60	0.60	0.70	150	20	50	n/a	90	20	35	n/a	150
	1.90	0.70	0.85	160	20	52	n/a	100	20	40	n/a	150
	2.20	0.70	0.90	165	20	53	n/a	105	20	40	n/a	180
	2.50	0.70	0.90	175	20	55	5	115	20	42	5	220
	3.00	0.80	1.00	185	25	55	10	130	20	45	10	250
	3.50	0.80	1.00	190	25	55	14	140	20	47	14	250
	4.00	0.90	1.10	200	25	60	17	150	20	50	17	280
	4.50	0.90	1.20	210	25	60	20	160	20	52	20	280
	5.00	0.90	1.40	220	30	63	22	170	25	55	22	300
	5.50	0.90	1.60	230	30	65	24	180	25	57	24	320
	6.00	1.00	1.80	245	35	67	25	195	30	60	25	340
	7.00	1.00	2.20	270	40	70	28	205	35	65	28	340
	8.00	1.00	2.60	290	50	72	32	225	40	68	31	360

The design procedure is based on the assumption that the flanges are in tangential contact at their outside diameter.
For calculation the gasket is defined by m (or Ym) and y (or Y2); when we use a Hellicoflex joint in a face-to face flange only Y2 can acts

h_c= radial distance from bolt circle to flange-flange bearing circle where tangential contact occurs.

$$h_c = (A - C) = (1370 - 1235) = 67,5mm \quad (2)$$

$$H_D = 0,785B^2P = 1571086N \quad (3)$$

$$H = 0,785G^2P = 1591827N \quad (4)$$

$$H_T = H - H_D = 20741N \quad (5)$$

Total joint contact surface compression load, Hp, in face-to-face flanges depends of Y_2 and not of Y_2

$$H_p = 3,14GY_2 = 792693N \quad (6)$$

$$W_{m1} = H - H_p = 1591827 + 792693 = 2384520N \quad (7)$$

$$H_G = W - H = 2384520 + 1591827 = 792693N \quad (8)$$

$$M_p = H_D.h_D + H_T.h_T + H_G.H_G = 102397764 \quad (9)$$

h_D	42,5	H_D	1571086
h_T	45,625	H_T	20740
h_G	43,75	H_G	792693

- **Trial values of t(flange of te thickness)and A_b**

$These values do not assure that the first trial design will meet the requeriments$

$We take$ **n=108** $bolts with a nominal diameter$ **D= 12**

$t_a = 2,45\sqrt{M_p(\pi C - nD).S_f} = 49mm(10)$

$$t_b = 0,56B_1\sqrt{PS_f} = 80mm \quad (11)$$

Trial value of t, greater os $t_a or t_b, t = 80mm$

Flange moment due to flange-hub interaction

$$M_s = -J_p F M_p t^3 + J_s F = -704870(12)$$

Where:
$F = 32633$
F from Fig.2-7.3,$F = 0,90892$
V from Fig.2-7.3,V= 0,550103
t, we check 80 mm
$AR = 0,32$
r$_B = 0,007$

Slope of flange at inside diameter times E

$$E\theta_B = 5,46\pi.t^3\left(J_s M_s + J_p M_p\right) = 37,56 \quad (13)$$

Contact force betwwen flanges at h$_c$

$$H_c = 1506561 \quad (14)$$

Bolt load at the operation conditions

$$W_{m1} = H + H_G + H_c = 3891081N \quad (15)$$

Operating bolt stress
$A_b = 31339mm^2$

$\sigma_b = \frac{W_{m1}}{A_b} = 124Nmm^2$ (16)

Design prestess in bolts

l=calculated strain length of bolt = 2t +0,5 db = 181 mm
$S_i = 121,5$

Radial flange srtess at bolt circle

$$S_R = 36,42\frac{N}{mm^2}$$

Radial Flange stress at inside diameter

$$S_R = 0,0026$$

Tangencial flange stress at insede diameter

$$S_T = 2,69$$
$$Z = 5,55$$

Longitud hub stress

$$S_H = 6,97$$

Allowable flange design stress

	t=49mm $A_b = 16321$	t=80mm $A_b = 31339$	Allowable stress,Mpa
Operating bolt stress,$\sigma_b, g1 = 10$	236,94	201	$S_b = 255$
Longitudinal hub stress,S_H	96,10	6,97	147,5
Radial stress bolt circle,S_R	1,77	36,42	98,5
Radial stress inside,S_R	5,88	0,0026	98,5
Tangencial stress,S_T	27,31	2,69	98,5
$(S_H + S_R)/2$	61,71	21,69	98,5
$(S_H + S_T)/2$	16,59	4,83	98,5

With bolted joints a good practice is to get σ_b near of allowable stress $(0,75\sigma_b)$.

For this reason we change the area of bolts, taking 72 bolts $M22x1,5$, $Ab = 18504mm^2$, and the minimum thickness is 49mm

The welding between the plate of the flange and the shell produces a small hub, we suppose h=10mm that produce a change of f, F and V. From tables:

g1/g0=2, and h/h0=0,09
f=3,28
V=0,44
F=0,9

With the new values the minimum thickness is 47mm.
We take to **the thickness of the flange = 50 mm**

3.2.3 This gasket load may produce a plastic strain in the flange?

Assuming a cylinder of aluminium, the semi-width of contact between the gasket cylinder and the flange is (Shigley, 2-21):

$$b = \sqrt{\frac{2F}{\pi l} \frac{\left(1 - v1^2\right)/E_1 + \left(1 - v2^2\right)/E_2}{1/d1}} \tag{17}$$

F/L=220 N/mm^2

And the maximum pressure

$$P_{max} = \frac{2F}{\pi bl} = 1158Mpa \tag{18}$$

,than excess the yield strength of the flange

$E_1, aluminium$	70967
$E_2, titanium, g2$	106795
v_1	0,33
v_1	0,32

We will find the same situation with the others design rules; an advantage with face-to face design is than the value Y2 (220N/mm) is greater than Ym (30N/mm) and therefore greater tightening after the local strain in the flange

With $F/l = 30N/mmm$

,

newline $b = 0,045$ and $Pmax = 429MPa$

Assuming a modulus of elasticity Y2/e2 for the gasket, E1= Y2/e2 =220/0,9=244 MPa

b=1,6mm

$Pmax = 87,5MPa, withF/l = 220N/mm$ **and therefore not local strain.**

4 Summary

- Flange metal to metal contact(Appendix Y):With these flanges need less thickness and less outside diameter, that is, more radiopurity, and less cost
- Gasket :Garlok type HNDE , but We have to ask Garlok Company because maybe the best would be these gaskets when NEXT100 would be operate for long time(At first there is a possibility that the chamber is opened and closed frequently) , but are the best to control of the leaks
- We need to have radiopure measurements of : Titanium Grade 2, Inconel 718 and Titanium Grand 2 with TIG Welding but with these materials and this welding, we get :

	Thickness mm
Cylindrical Shell	10
Torispherical Cover	8
Flanges	50

5 Specifications

6 Drawings

Specification Vessel

1 Specification Vessel ASME VIIIDivision 2

1.1 User´s Design Specification

a) Installation Site

1. **Localation**
   Spain , Hueca ,Canfranc: Laboratorio Subterráneo de Canfranc

b) Vessel Identification

1. **Vessel Number o indentification**
   NEXT 100
2. **Service fluid**
   Xenon gas 136 : flow rate of 200 slmp at 15 bar.

c) Vessel Congifuration and Controlling Dimensions

1. **Outline drawings**
   See Document *NEXT 100 Drawings*
2. **Position**
   Horizontal
3. **Openings and size**
   See Document *Next100 Drawings*
4. **Support method**
   See Document *Shielding Design* from Lluís Ripoll, Jordi Torrent y Miquel Batallé

d) Design Conditions

1. **Specified design Pressure**
   - Internal Pressure : Maintain Xenon gas equivalent to 100 Kg at a internal Pressure of 1,54 MPA(MAWP : Maximum Allowable Working Pressure)
   - External Pressure : 0,1 Mpa
2. **Design Temperature**
   - Minimun Temperature : 20-25 C
   - Maximun Temperature :

e) Operating Conditions

1. **Operating pressure**
   1,50 bars

2. **Temperature** 20-100C

f) Design Fatigue Life

g) Materials

1. **Material Specification** All the components shall be according with the rules for construction of pressure vessels ASME, 2010.

   - **Cylindrical vessel:** Titanium grade 2,Inside diameter 1140mm. The thickness shall be conforming to ASME, Section VIII, Division 1.
   - **Flanges (integral type):** Titanium grade 2, The calculations shall be conform with ASME Nonmandatory Appendix Y: Flat face flanges with metal-to-metal contact outside the bolt circle.
   - **Joints:** Garlok type HNDE, aluminium jacket, CS 5mm.
   - **Bolts:** Inconel 718 (radiopurity test)
   - **Heads:** Torispherical heads, DIN 28011 type, titanium, grade 2. Calculation according ASME VIII, Division 2.
   - **Nozzles:** Titanium, grade 2.
   - **Pressure relief valve**
   - **Supports:** Titanium grade 2.

2. **Corrosion and / or erosion allowance**

h) Overpressure Protection
The type over pressure protection intended for the vessel are :

1. **Buffer Gas/Vacuum (RGA)**

2. **Rupture Discs**

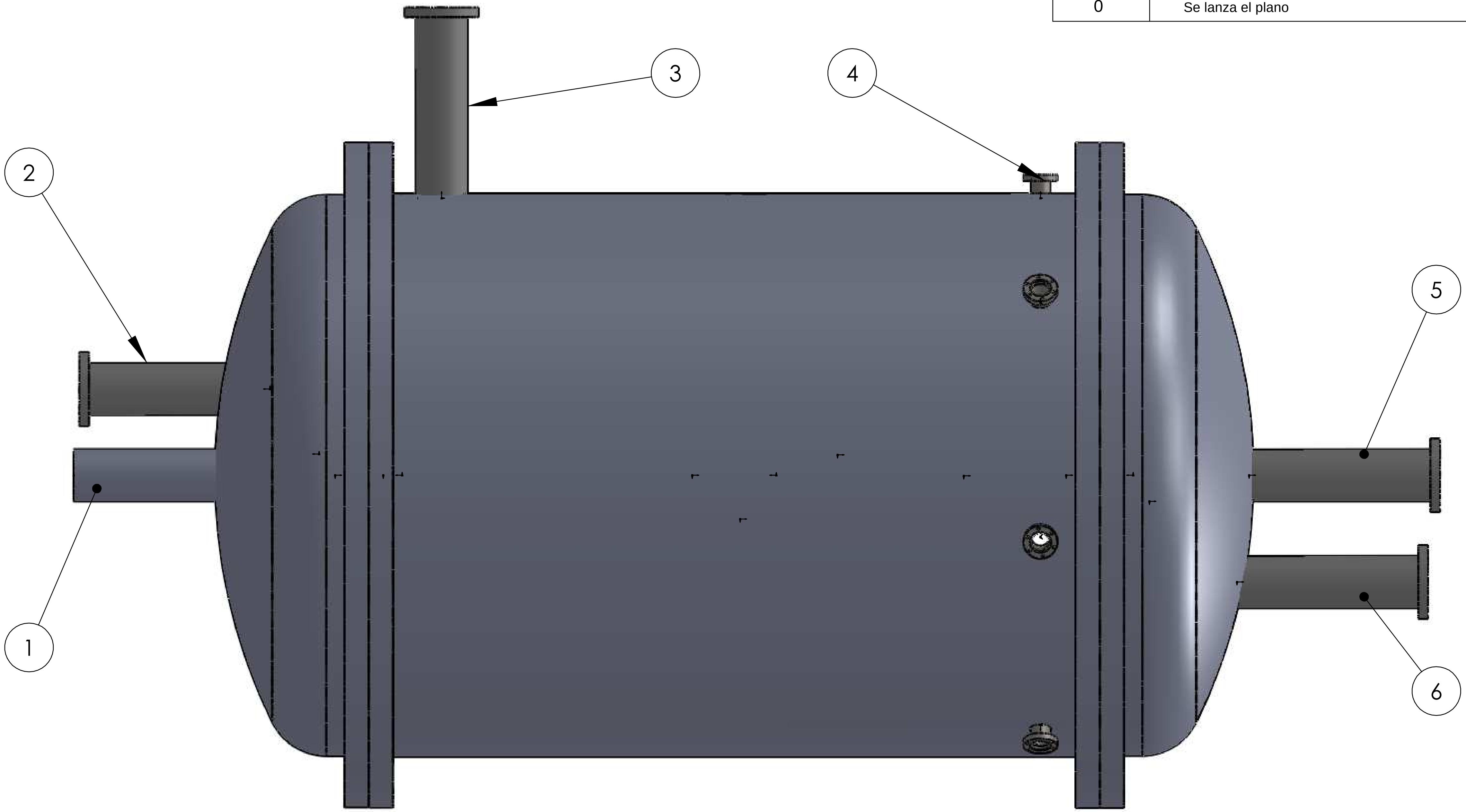
3. **etc etc**

i) Final Test Report

1. **Hydrostatic Test (OCA).**

2. **Declaration of Conformity with Pressure Directive 97/23/CE (OCA).**

Revision	Description	Date	Approved
0	Se lanza el plano		

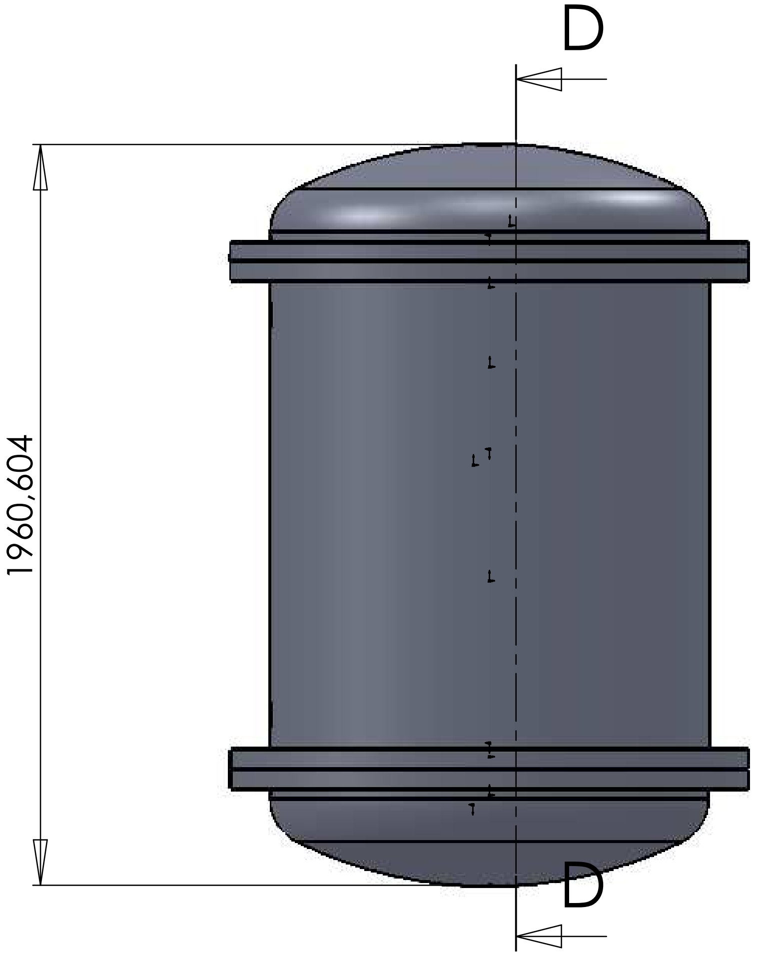


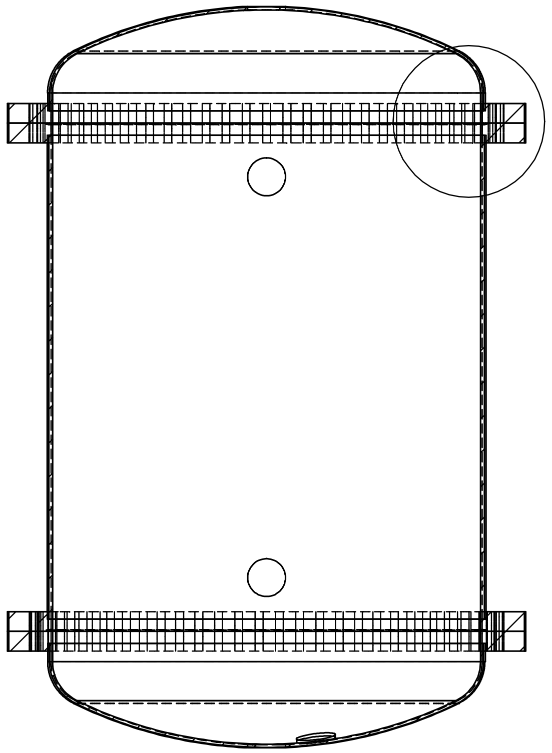
Nº	Size	Description	Quantity
1	DIN 100	PMT Connexions	1
2	DIN 100	Pumping_Gas	1
3	DIN 100	HV Cathode Mesh	1
4	DIN 40	HV EL Mesh	7
5	DIN 100	SIPMsConnexions	1
6	DIN100	Pumping_Gas	1

NEXT100	UNITS: mm			
PROYECT	NORM	MATERIAL	PROCESSING/FINISH	WEIGHT
DRAWN	CHECKED	APPROVED		SCALE 1:10
INSTITUTO DE FÍSICA CORPUSCULAR LAWRENCE BERKELEY NATIONAL LABORATORY	TITTLE Next100			DATE 01/10/2011
	DWG NO. DRWP0001	EDITION	FORMAT A3	SHEET SHEET 0 OF 7

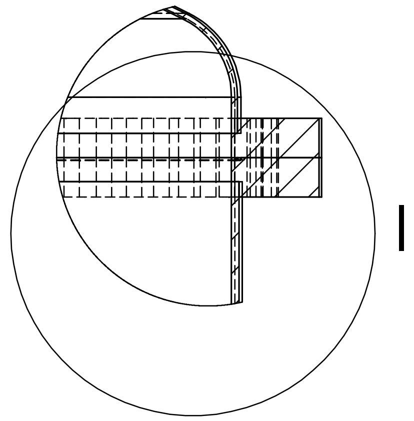
TOLERANCIAS GENERALES: +- 0.1 (NO ACUMULABLES)
NO TOMAR MEDIDAS DIRECTAMENTE DEL DIBUJO
DIMENSIONES EN MILIMETROS

Revision	Description	Date	Approved
0	Se lanza el plano		

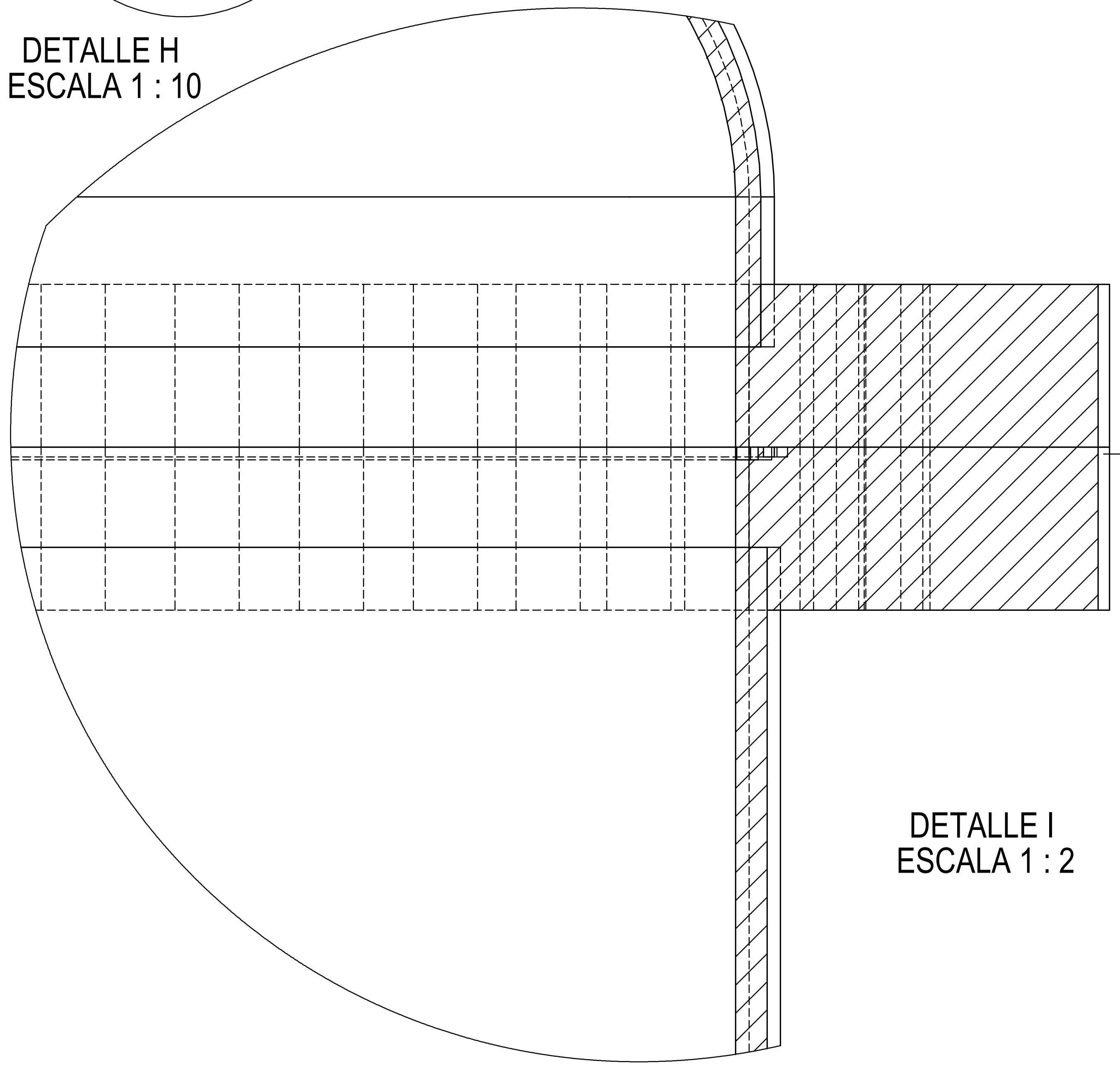




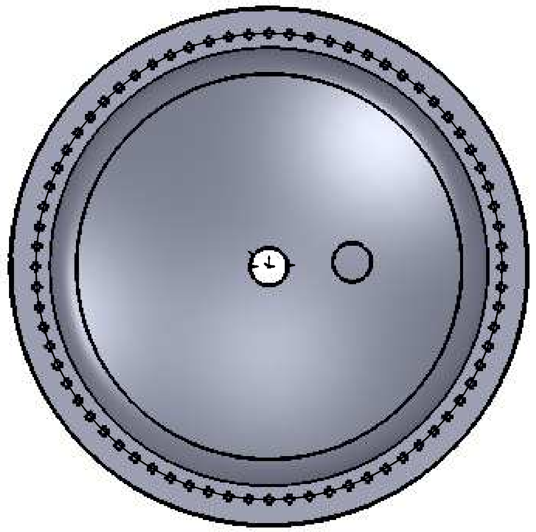
SECCIÓN D-D



DETALLE H
ESCALA 1 : 10



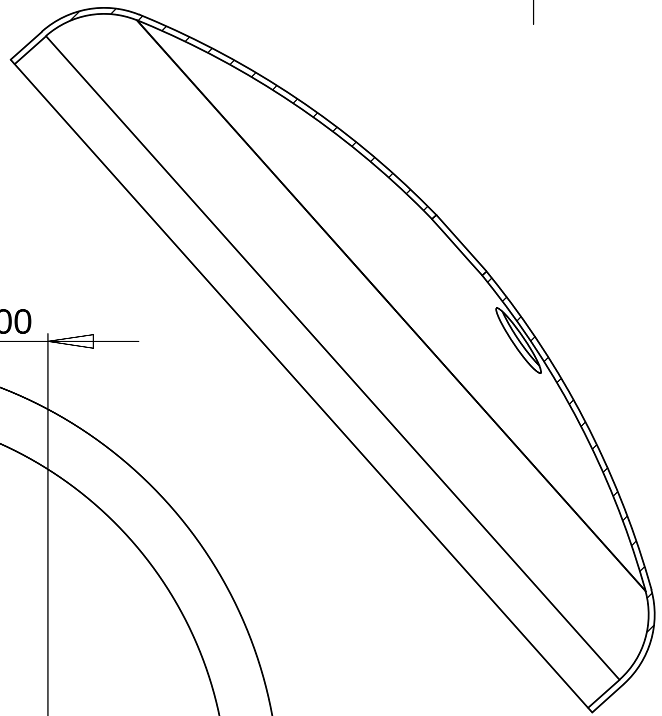
DETALLE I
ESCALA 1 : 2

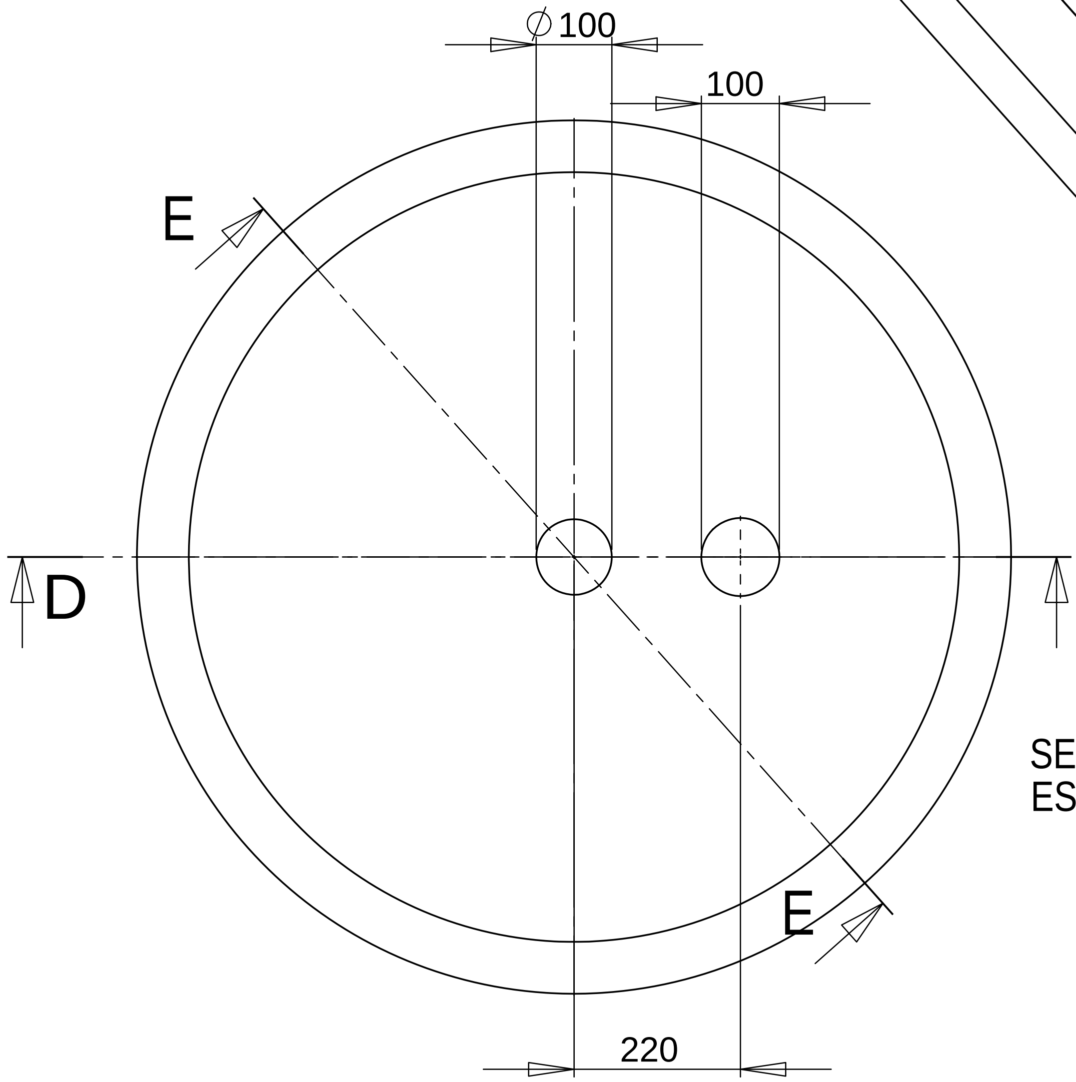


TOLERANCIAS GENERALES: +- 0.1 (NO ACUMULABLES)
NO TOMAR MEDIDAS DIRECTAMENTE DEL DIBUJO
DIMENSIONES EN MILIMETROS

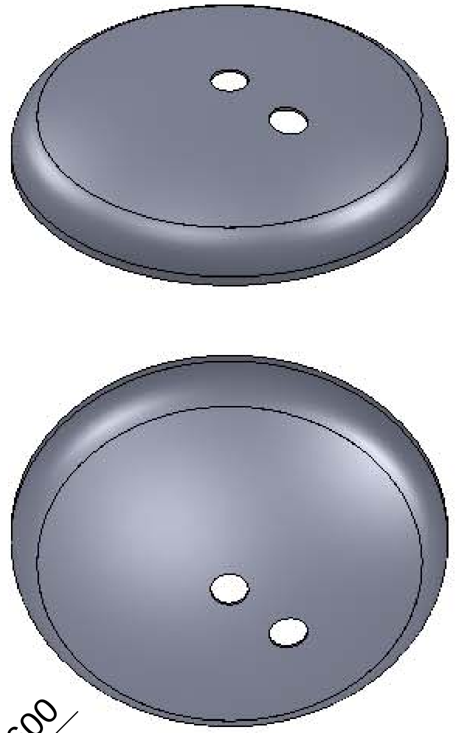
NEXT100	UNITS: mm			
PROYECT	NORM	MATERIAL	PROCESSING/FINISH	WEIGHT
DRAWN	CHECKED	APPROVED		SCALE 1:10
INSTITUTO DE FÍSICA CORPUSCULAR LAWRENCE BERKELEY NATIONAL LABORATORY	TITTLE: Next100 Without HUB			DATE 01/10/2011
	DWG NO. DRWP0001	EDITION	FORMAT A3	SHEET: SHEET 1 OF 7

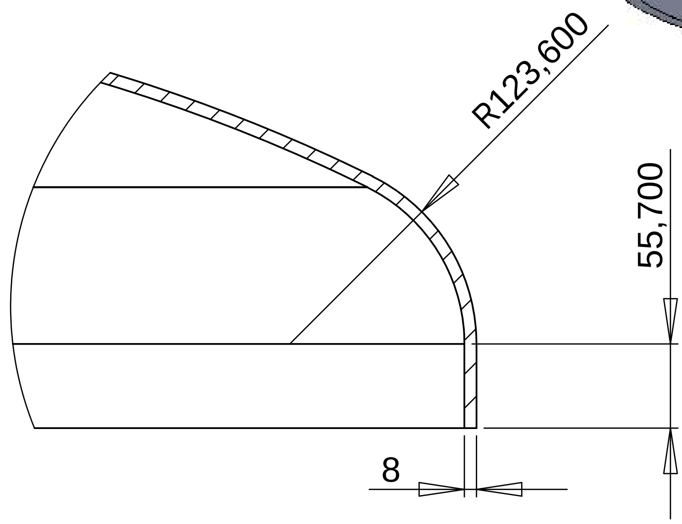
Revision	Description	Date	Approved
0	Se lanza el plano		



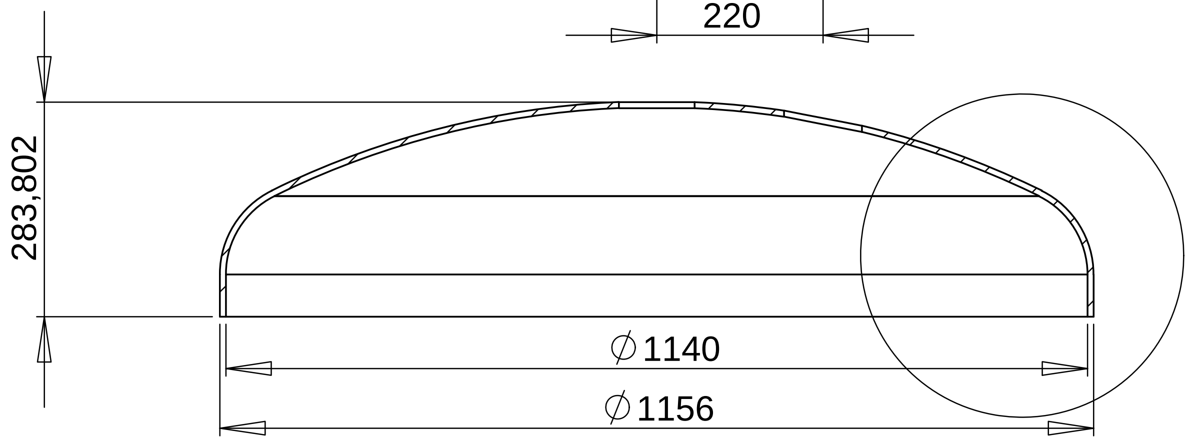


SECCIÓN E-E
ESCALA 1 : 10





DETALLE F
ESCALA 1 : 5

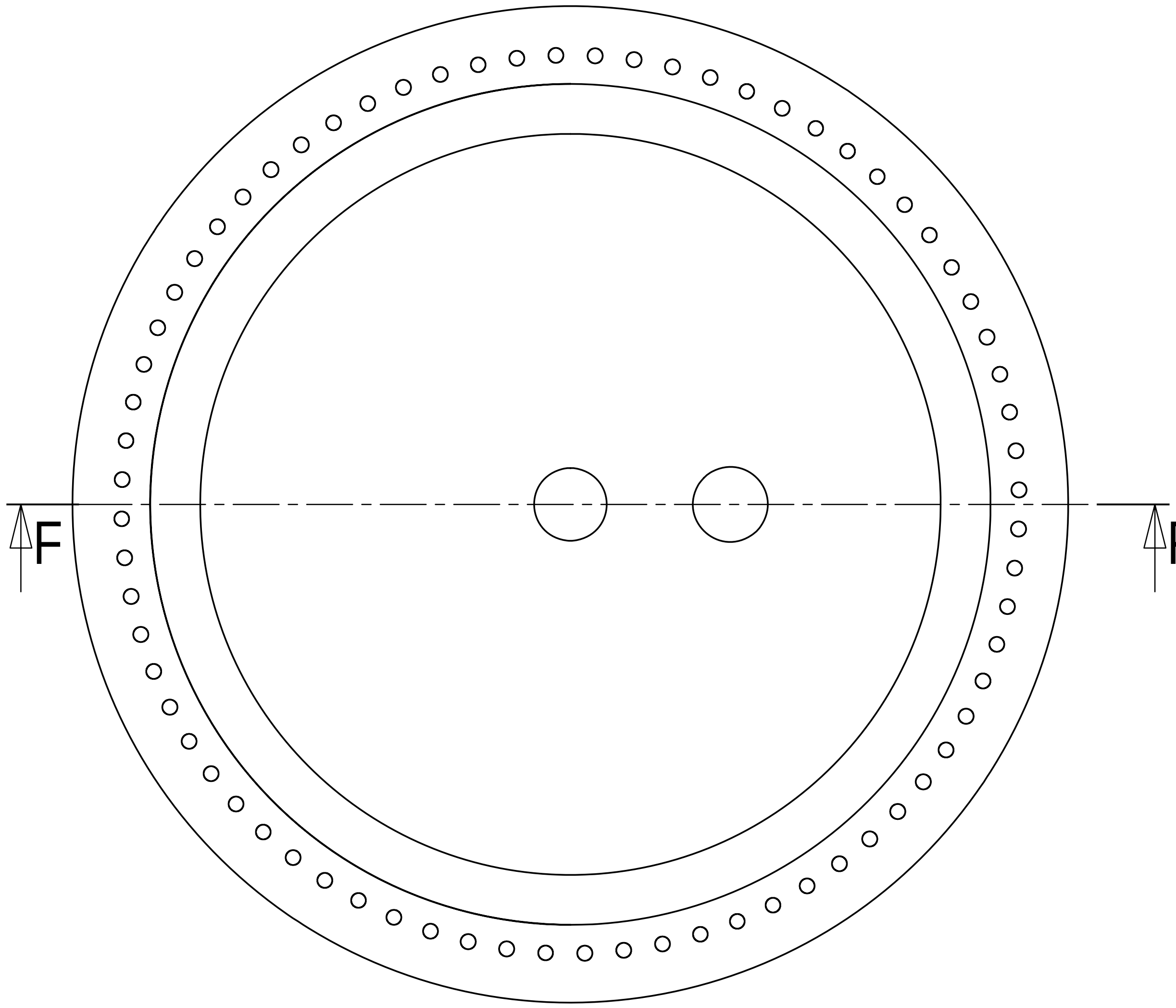


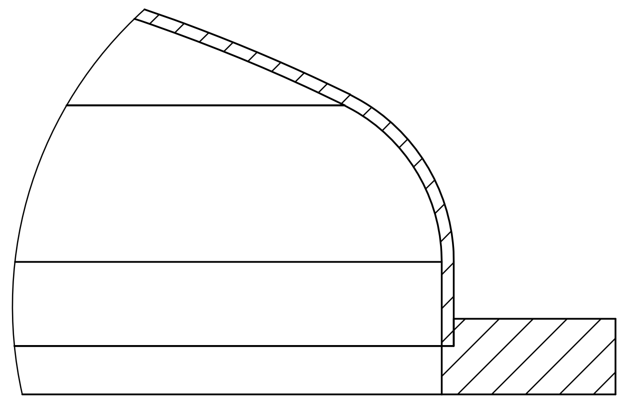
SECCIÓN D-D
ESCALA 1 : 10

TOLERANCIAS GENERALES: +- 0.1 (NO ACUMULABLES)
NO TOMAR MEDIDAS DIRECTAMENTE DEL DIBUJO
DIMENSIONES EN MILIMETROS

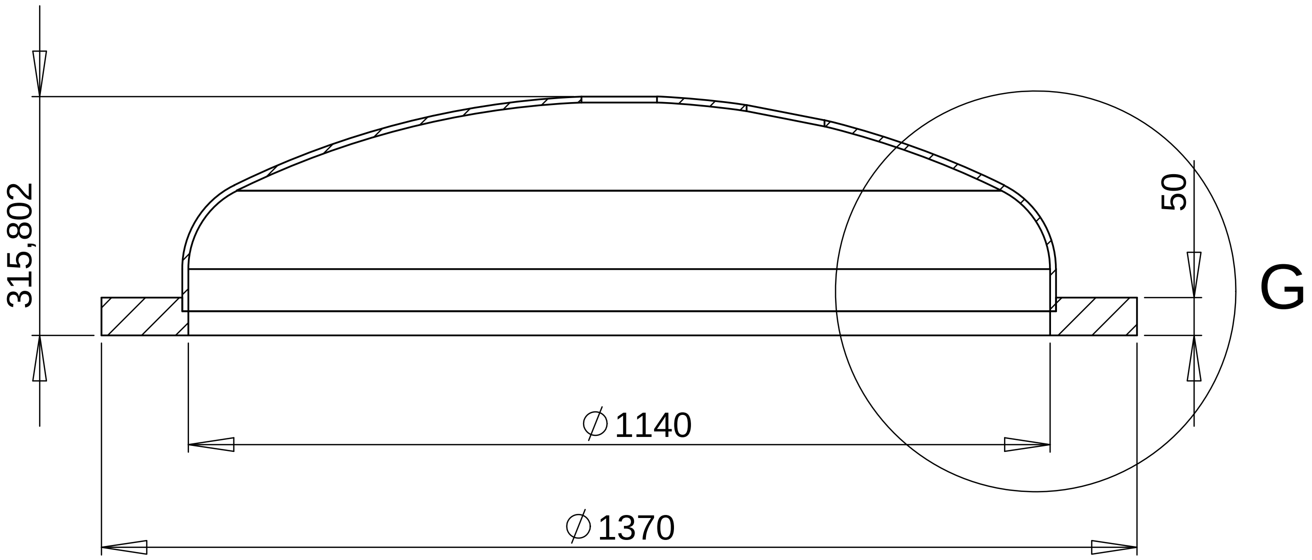
NEXT100	UNITS: mm			
PROYECT	NORM	MATERIAL	PROCESSING/FINISH	WEIGHT
DRAWN	CHECKED	APPROVED		SCALE 1:10
INSTITUTO DE FÍSICA CORPUSCULAR LAWRENCE BERKELEY NATIONAL LABORATORY	TITTLE Next100_TorisphericalHead_no_Hub			DATE 01/10/2011
	DWG NO. DRWP0002	EDITION	FORMAT A3	SHEET SHEET 2 OF 7

Revision	Description	Date	Approved
0	Se lanza el plano		





DETALLE G
ESCALA 1 : 5

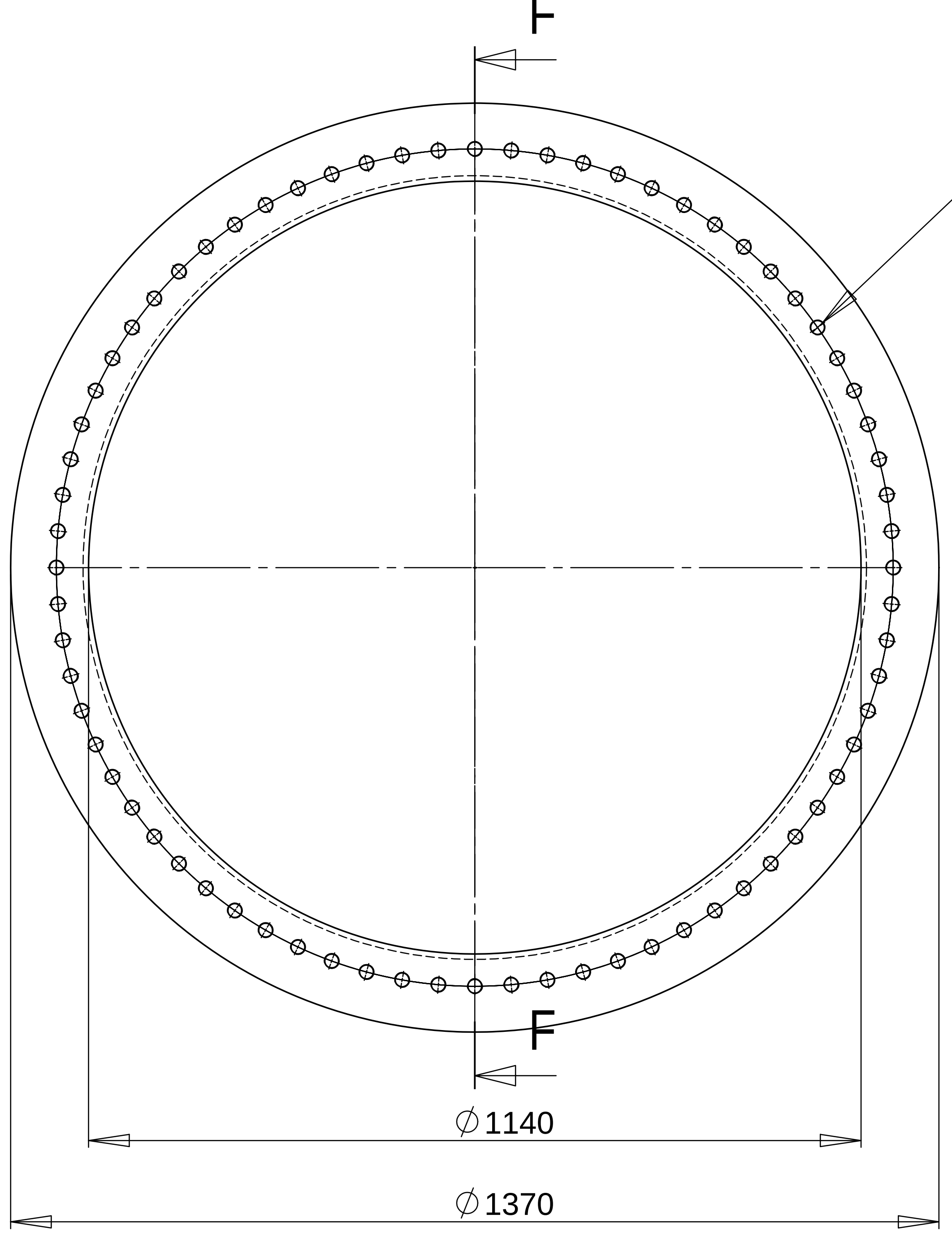


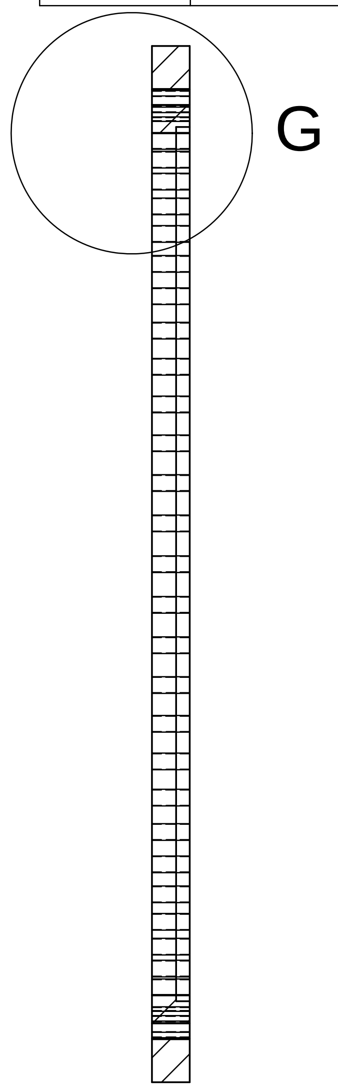
SECCIÓN F-F
ESCALA 1 : 10

TOLERANCIAS GENERALES: +- 0.1 (NO ACUMULABLES)
NO TOMAR MEDIDAS DIRECTAMENTE DEL DIBUJO
DIMENSIONES EN MILIMETROS

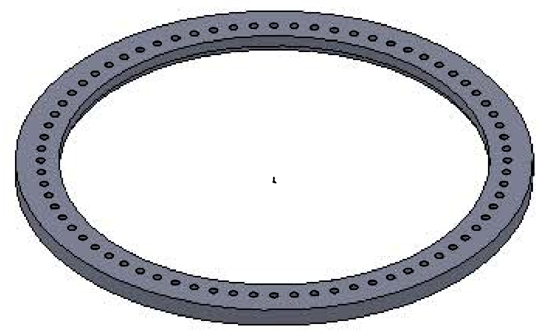
NEXT100	UNITS: mm			
PROYECT	NORM	MATERIAL	PROCESSING/FINISH	WEIGHT
DRAWN	CHECKED	APPROVED		SCALE 1:10
INSTITUTO DE FÍSICA CORPUSCULAR LAWRENCE BERKELEY NATIONAL LABORATORY	TITTLE **Next100_TorisphericalHeadCover_no_hub**			DATE 01/10/2011
	DWG NO. **DRWP0003**	EDITION	FORMAT A3	SHEET **SHEET 3 OF 7**

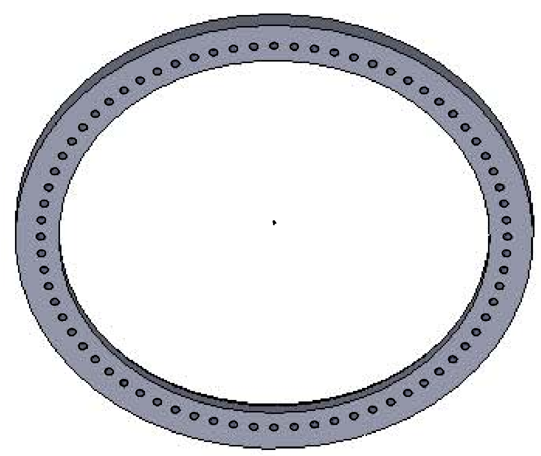
Revision	Description	Date	Approved
0	Se lanza el plano		

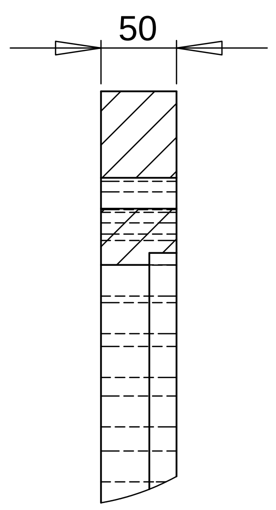




SECCIÓN F-F
ESCALA 1 : 10





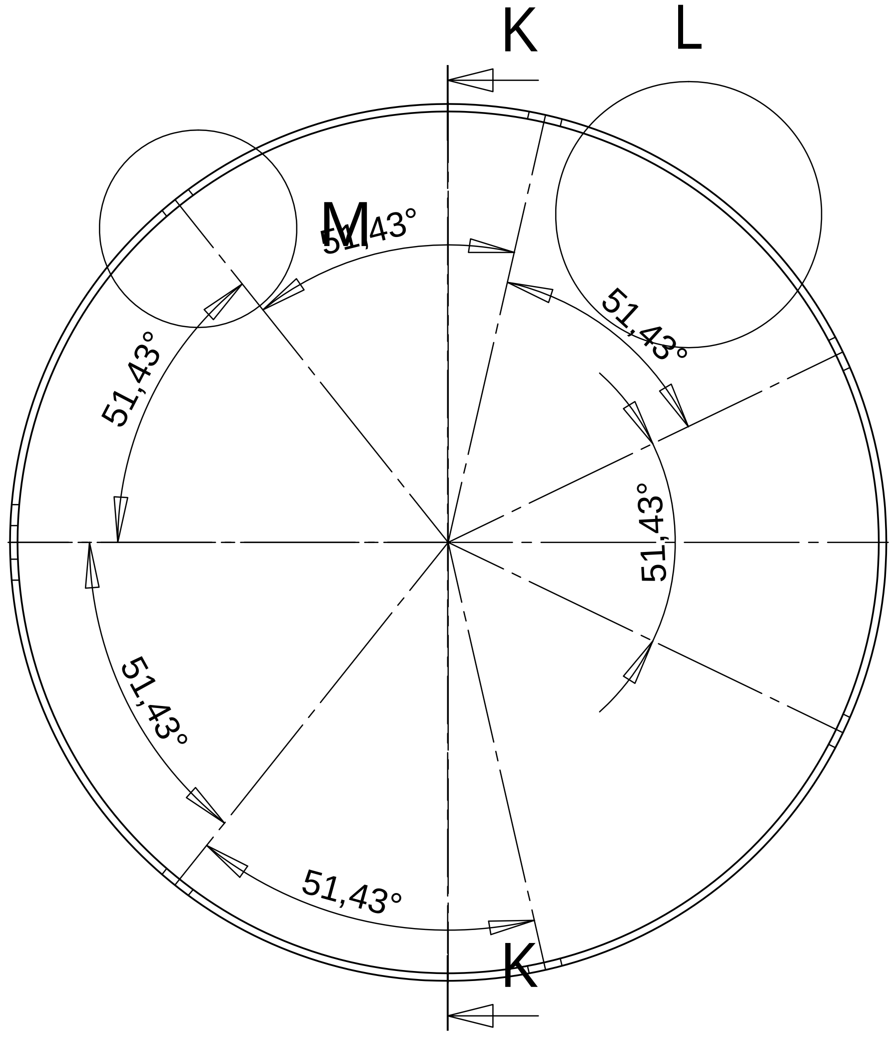


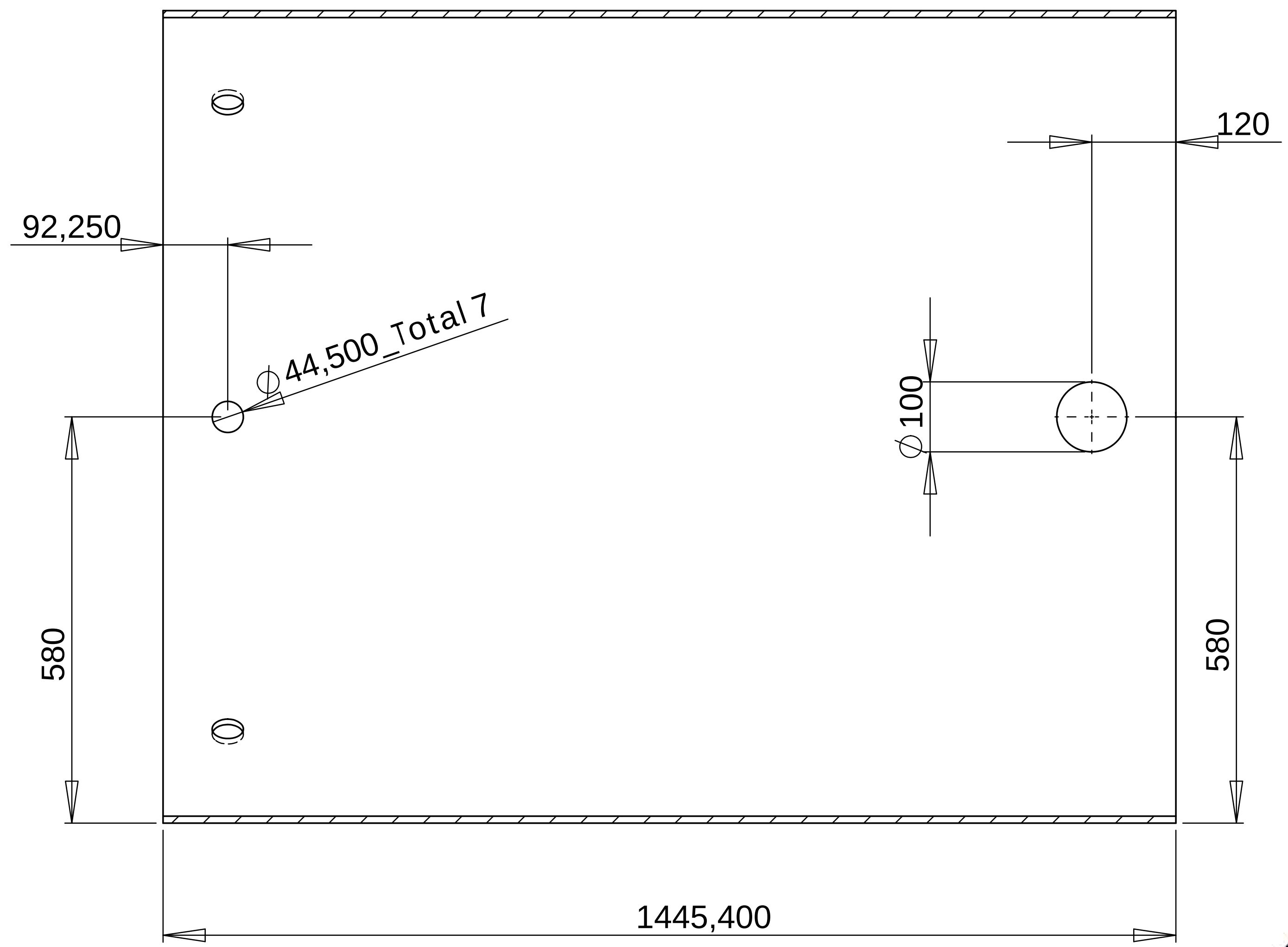
DETALLE G
ESCALA 1 : 5

TOLERANCIAS GENERALES: +- 0.1 (NO ACUMULABLES)
NO TOMAR MEDIDAS DIRECTAMENTE DEL DIBUJO
DIMENSIONES EN MILIMETROS

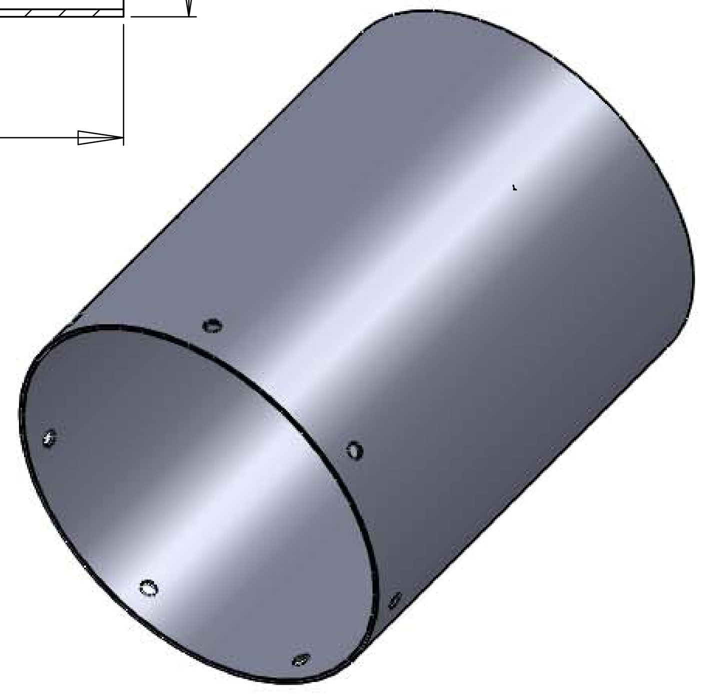
NEXT100	UNITS: mm			
PROYECT	NORM	MATERIAL	PROCESSING/FINISH	WEIGHT
DRAWN	CHECKED	APPROVED		SCALE 1:10
INSTITUTO DE FÍSICA CORPUSCULAR LAWRENCE BERKELEY NATIONAL LABORATORY	TITTLE Next100_TorisphericalFlangeHeadCover_no_Hub			DATE 01/10/2011
	DWG NO. DRWP0004	EDITION	FORMAT A3	SHEET SHEET 4 OF 7

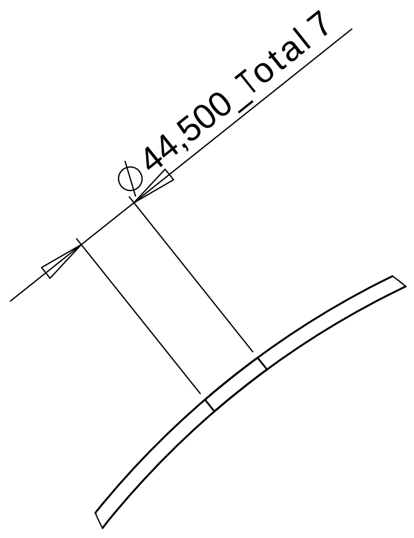
Revision	Description	Date	Approved
0	Se lanza el plano		

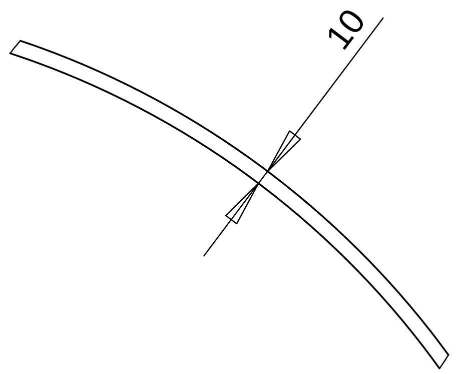




SECCIÓN K-K
ESCALA 1 : 10







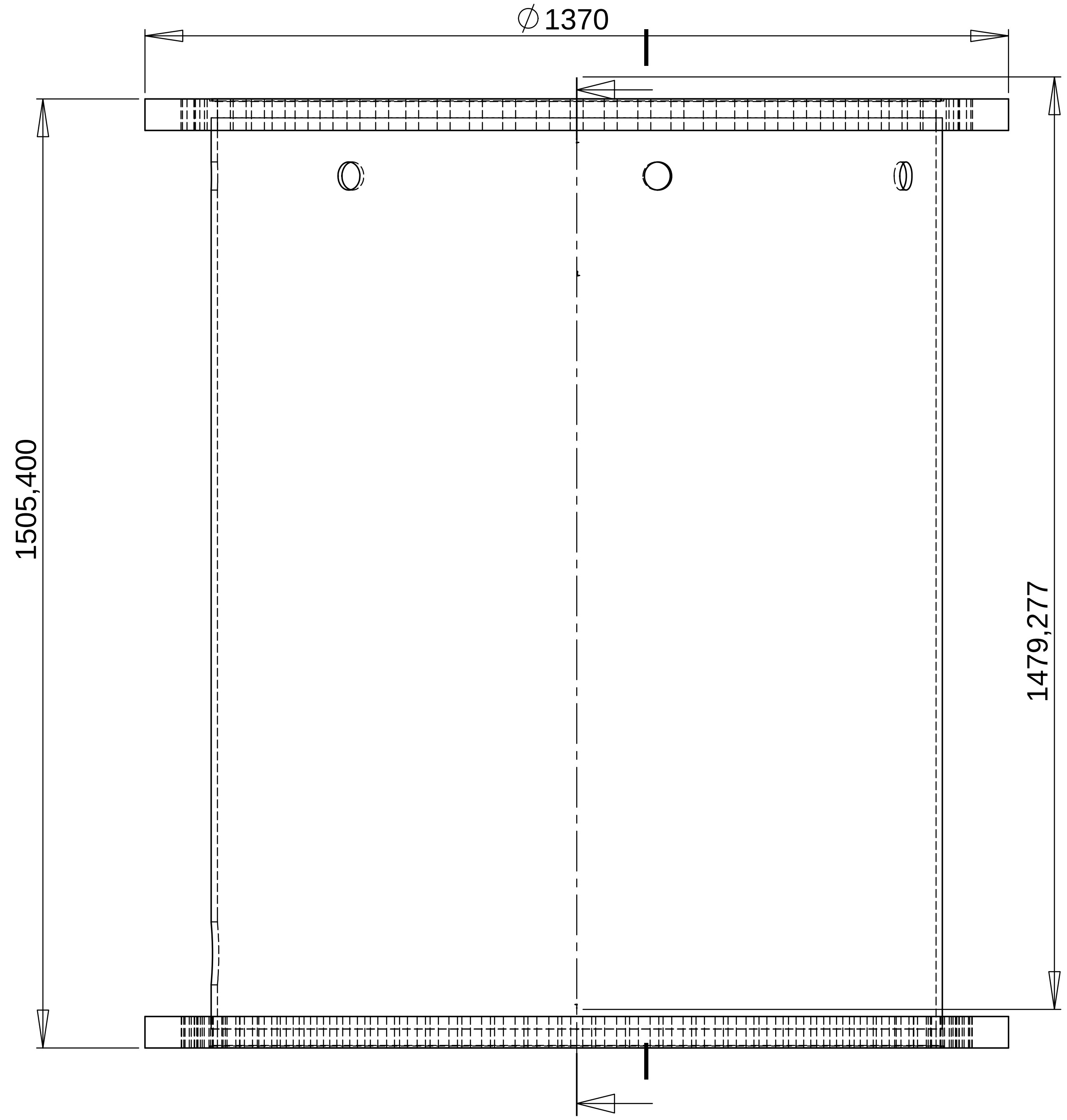
DETALLE M
ESCALA 1 : 5

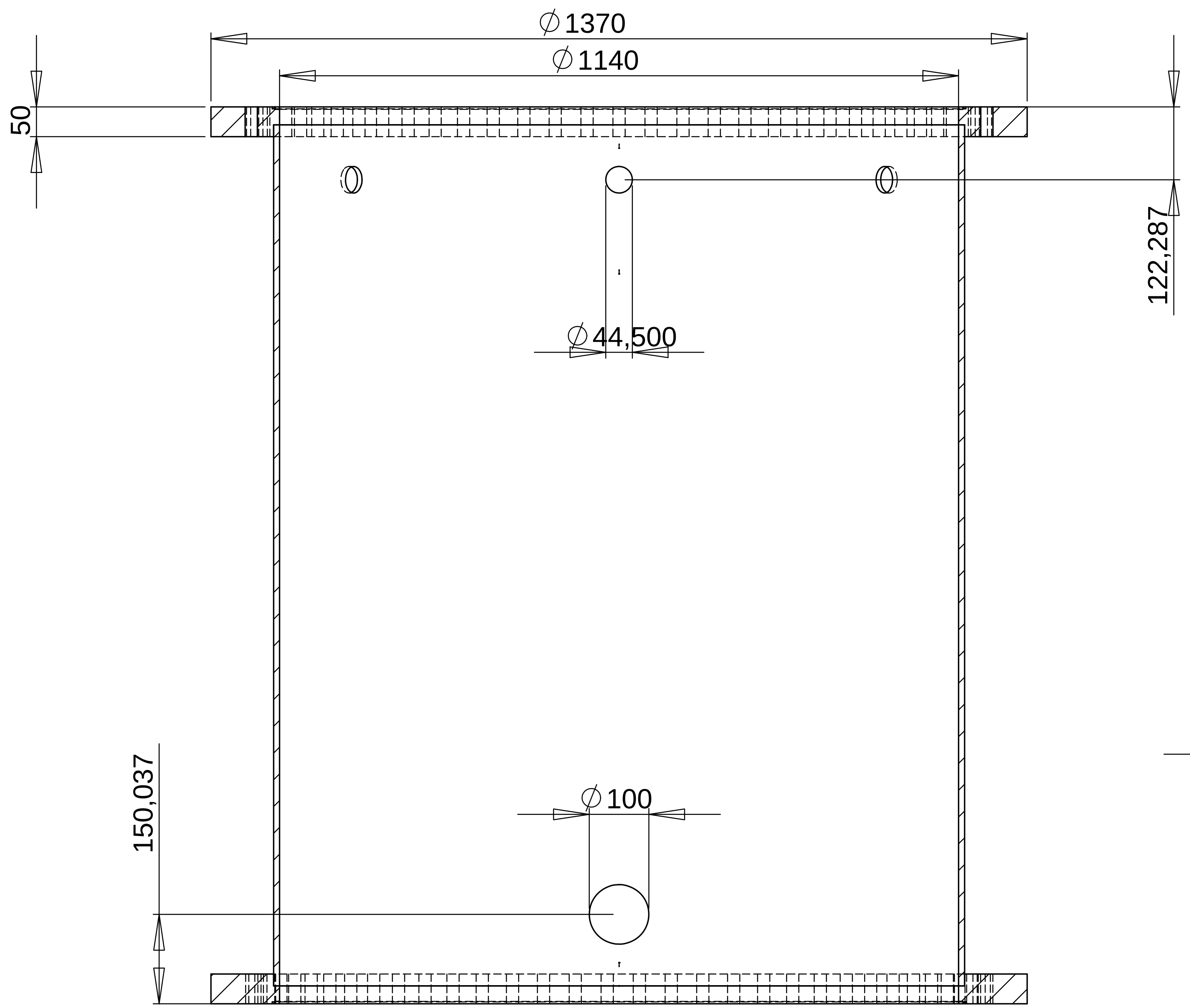
DETALLE L
ESCALA 1 : 5

TOLERANCIAS GENERALES: +- 0.1 (NO ACUMULABLES)
NO TOMAR MEDIDAS DIRECTAMENTE DEL DIBUJO
DIMENSIONES EN MILIMETROS

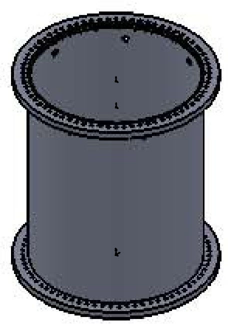
NEXT100	UNITS: mm			
PROYECT	NORM	MATERIAL	PROCESSING/FINISH	WEIGHT
DRAWN	CHECKED	APPROVED		SCALE 1:10
INSTITUTO DE FÍSICA CORPUSCULAR LAWRENCE BERKELEY NATIONAL LABORATORY	TITTLE Next100_Shell_no_Hub			DATE 01/10/2011
	DWG NO. DRWP0005	EDITION	FORMAT A3	SHEET SHEET 4 OF 7

Revision	Description	Date	Approved
0	Se lanza el plano		





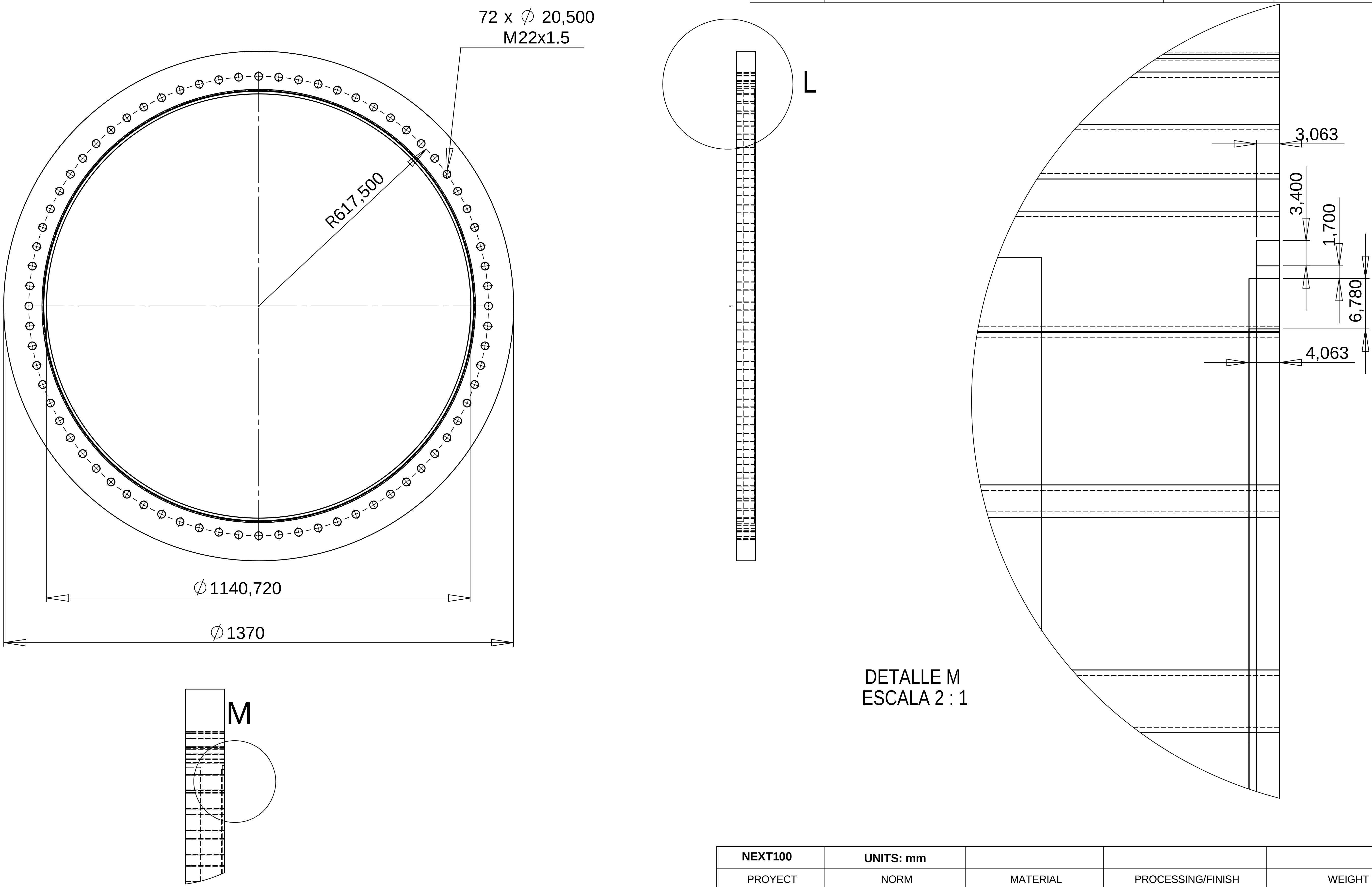
SECCIÓN I-I
ESCALA 1 : 10



TOLERANCIAS GENERALES: +- 0.1 (NO ACUMULABLES)
NO TOMAR MEDIDAS DIRECTAMENTE DEL DIBUJO
DIMENSIONES EN MILIMETROS

NEXT100	UNITS: mm			
PROYECT	NORM	MATERIAL	PROCESSING/FINISH	WEIGHT
DRAWN	CHECKED	APPROVED		SCALE 1:10
INSTITUTO DE FÍSICA CORPUSCULAR LAWRENCE BERKELEY NATIONAL LABORATORY	TITTLE	Next100_Cylinder_no_Hub		DATE 01/10/2011
	DWG NO. DRWP0006	EDITION	FORMAT A3	SHEET SHEET 6 OF 6

Revision	Description	Date	Approved
0	Se lanza el plano		



DETALLE L

TOLERANCIAS GENERALES: +- 0.1 (NO ACUMULABLES)
NO TOMAR MEDIDAS DIRECTAMENTE DEL DIBUJO
DIMENSIONES EN MILIMETROS

NEXT100	UNITS: mm			
PROYECT	NORM	MATERIAL	PROCESSING/FINISH	WEIGHT
DRAWN	CHECKED	APPROVED		SCALE 1:10
INSTITUTO DE FÍSICA CORPUSCULAR LAWRENCE BERKELEY NATIONAL LABORATORY	TITTLE Next100_FlangeCylinder_no_Hub			DATE 01/10/2011
	DWG NO. DRWP0007	EDITION	FORMAT A3	SHEET SHEET 7 OF 7