# 7. THE BARREL YOKE

As explained in Chapt. 3, the CMS magnet consists mainly of three parts: a superconducting coil, a vacuum tank and the magnet yoke. The solenoid produces an axial field whereas the yoke is responsible for the return of the magnetic flux. Due to the general design of the CMS detector, the yoke is split into a cylindrical central part, the barrel, and at the extremities, two endcaps made of 600 mm thick disks. This chapter describes the barrel part of the yoke.

To understand the behaviour of the barrel a large effort has been made by modelling the barrel yoke in two and three dimensions with different finite element programs. In addition, several manufacturing schemes for the thick iron absorber plates have been considered. Not only forged monoblocks, formed from an ingot weighing roughly double that of the final product, have been considered, but also a sandwich-like build-up of the iron. This allows the use of 210 mm thick steel plates, produced by continuous casting, connected together by welding, shear pins or shear keys. Fig. 7.1, p. XX, shows the central barrel ring with the main dimensions. The most important structural elements are shown in Fig. 7.2, p. XX.

Having different design options for manufacturing the massif iron blocks of the barrel yoke increases considerably the number of potential firms able to reply to a call for tenders.

In the following sections the results for the central barrel ring analysis as well as for the outer barrel wheels are presented.

## 7.1 STRUCTURAL ANALYSIS OF BARREL YOKE

### 7.1.1 Dimensions and loads

The barrel yoke has a total length of 13320 mm. It is split into five independent rings, each one being 2536 mm long. The four gaps between the barrel rings (2 x 120 mm + 2 x 200 mm) foreseen to exit the cables and services from the central detectors, add up to 640 mm.

The central barrel ring supports the vacuum tank in which are housed the inner detectors, bringing an additional mass of 1020 tonnes. The total mass of the central barrel in running condition will thus be 3050 tonnes. Table 7.1 shows the detailed list of the mass distribution for the different rings.

The main geometric parameters used for the finite element analysis (FEA) are:

- Iron lay-up for FEA: 210/620/620 mm
- Diameter on flat: 13990 mm
- Width of each barrel ring: 2536 mm

Small disagreements between technical drawing data and the geometrical data mentioned above maybe noticed. This is due to the fact that for the finite element analysis minimum tolerances have been taken into account concerning the mechanical resistance, although the maximum tolerances have been considered to compute the dead weight. Moreover, non structural parts, serving for example only as magnetic return flux iron, have not been modelled, but their mass is taken into account elsewhere.

For the central barrel ring as well as for the outer barrels the only load is gravity. Magnetic field introduces only axial forces in the rings. To resist these forces, the 70 mm shear pins mentioned below will be tack welded and the final assembly may make use of industrial bonding to increase the friction between the brackets and the slabs. For more details see Chapt. 7.1.6.

Having shown in the preliminary design report [7-1] that there is only a small difference between the barrel at rest, supported at the inside of the supporting foot, and the barrel supported on the outer surface during transport in the surface building, only the results for the barrel at rest are presented here.

**Table 7.1**
Mass distribution in central and outer barrels.

| | **Central Barrel** | **Outer Barrels** |
|---|---|---|
| Barrel ring (slabs + brackets) | 1150 tonnes | 1174 tonnes |
| Vac Tank central part (60 mm) with connecting ribs | 90 tonnes | none |
| Vac Tank outer shell extension cylinder (30 mm) | 46 tonnes | none |
| Vac Tank inner shell | 114 tonnes | none |
| End flanges of Vac Tank | 14 tonnes | none |
| Cold mass (coil) | 234 tonnes | none |
| Support feet | 72 tonnes | 66 tonnes |
| Tail catcher in steel | 100 tonnes | none |
| Hadronic Barrel HB | 900 tonnes | none |
| Elect. barrel EB. + Tracker | 120 tonnes | none |
| Muon chambers | 50 tonnes | 50 tonnes |
| Cabling on Vac Tank and services | 150 tonnes | none |
| Support for racks and cables | 10 tonnes | 10 tonnes |
| **Total mass** | 3050 tonnes | 1300 tonnes |

### 7.1.2 Material properties and allowable stresses

The allowable stresses contained within a specification are to be compared with stresses determined by analysis of the effect of loading the structure. This automatically leads to the discussion of the factor of safety. This factor of safety is inherent to the allowable stresses and provides for uncertainties that are associated with typical simplifying assumptions and average calculated stresses. It is not intended that highly localised peak stresses that may occur in modern finite element analysis must be less than the stipulated allowable stresses [7-2]. Here again, exercise of engineering judgement is required.

In mechanical terms, the analysis of the barrel is a purely static problem. Due to that fact, and the very good agreement of two independent FE analysis, we will apply a factor of safety of 1.5 to yield as foreseen by the American Institute of Steel Construction Specification for the Design, Fabrication and Erection of Structural Steel for Buildings (henceforth called “AISC Code”) and the Deutsches Institut für Normung (DIN 18800 T1).

For the steel grades foreseen for the construction of the CMS yoke the allowable stresses are listed in Table 7.2. The plates to be used in the barrel being relatively thick, between 75 and 220 mm, we have to further apply a reduction of 10 % on these values.

**Table 7.2**
Allowable stress for the steels foreseen for the construction.

| Type of Stress | Allowable Stress Fe 310 [MPa] | Allowable Stress Fe 360 [MPa] | Allowable Stress FeE 560 [MPa] |
|---|---|---|---|
| Von Mises | 120 | 160 | 290 |
| Shear | 70 | 95 | 160 |
| Tension/ Bending | 120 | 160 | 290 |

### 7.1.3 FEA model for central barrel

The CASTEM 2000 [7-3] finite element program was used to create the 2D-model of the barrel rings.

Figure. 7.3, p. XX, to Figure. 7.5, show the finite element model of the central barrel. It is made from 2-D plane stress 8-node elements. This type of element is chosen as it closely represents the stress distribution in which we are primarily interested. The brackets have at least 2 layers of these elements. This corresponds to a minimum of 5 nodes in the bending section.

The support feet for the central section are included in the model. This makes the model complete and shows also the influence of foot compliance.

The supports are fabricated by welding steel plates. As such, they cannot be truly represented by a 2-D model. The side plates, which are continuous, are modelled with real steel properties. The stiffening web plates, which are not continuous, are represented with diffuse properties. This gives correct displacement values for the support and thus for the entire yoke, but for stress related questions the foot has to be analysed separately. The entire structure is simply supported on the floor by constraining all the nodes of the floor interface in the negative vertical direction (i.e. downward) whilst upward movement is allowed. This technique represents well the stiffness of the foot. To keep the structure in place horizontally, one node has an horizontal boundary condition. No friction and no compliance of the floor have been taken into consideration.

The vacuum tank is represented by a single 60 mm thick steel cylinder which is connected at 12 points to the yoke. The weight of the inner detectors, the coil and the inner shell of the vacuum tank have been taken into account by giving an artificial density to the mesh representing the vacuum tank.

Between the slabs which are built up from three layers of iron (i.e. the middle and outer ones), 160 mm diameter shear pins are installed to keep the three layers together and provide bending stiffness. Washers which serve as spacers in between the compressing tie-rods are modelled with orthotropic elements. A standard steel modulus was used in the radial direction while only 1% of the Young’s and shear modulii were used in the perpendicular direction. This is low enough to transmit quasi no shear force and large enough to avoid numerical problems.

To resist the large shear forces occurring mainly around the foot region, a pin of

120 mm diameter is installed between YB 0/3/8 and 3/9 and also between 3/11 and 3/12. In the program this is achieved by nodal coupling. Fig. 7.6, p. XX, shows the location of these pins and the numbering scheme of the blocks.

Care has been taken to model fairly accurately the highly stressed regions which are believed to be (from previous analysis) the connection brackets and their fixations.

In this model only gravity acts on the self mass of the structure. No magnetic forces and neither friction nor any other contact problem has been introduced.

All material is steel with a Young's modulus of 210000 N/mm$^2$ except the cryostat and its radial connection to the yoke. These parts are in stainless steel with a Young's modulus of 190000 N/mm$^2$.

At a first glance the yoke seems to be symmetric but it is not, because of the asymmetry in phi. Therefore the whole ring must be modelled, giving a mesh with 12138 elements and 45698 nodes.

To cross check the results, a 3-dimensional model has been made at Fermilab [7-4]. Fig. 7.7, p. XX, shows the 3-D mesh of that model including the cryostat and the support feet.

### 7.1.4 FEA Results for the central barrel

The colour plot in Fig. 7.8, p. XX, shows the undeformed structure in blue and the deformed in red. The amplification is 100. Due to the artificial density which was given to the cryostat, its deformed shape must not be considered. The maximum vertical displacement of the barrel ring is 10 mm. The widening of the ring in the horizontal direction is not symmetric because of the asymmetry of the structure and the boundary conditions fixing one node of the left foot in the x-direction.

In the horizontal plane of the beam axis the displacements are 6.4 mm to the left and only 4.5 mm to the right side. Horizontal and vertical displacements are shown in Fig. 7.9, p. XX, and Fig. 7.10 respectively. The deformation plots for the outer barrels are shown in Fig. 7.11, p. XX, to Fig. 7.13.

The verification of the stress distribution concentrates on the following parts of the structure: absorber slabs, brackets, shear keys between slabs, shear keys between slabs and brackets and the tie bars which are used to connect the brackets to the slabs.

Slabs and their shear pins:

The slabs are modelled as a 3 layer design. This is mechanically the most complicated way to build the magnet iron, but the fact that with such a design the maximum plate thickness needed is reduced to 210 mm, increases the number of potential firms and allows an economic use of steel plates. It is clear that if the slabs are manufactured as forged massif iron blocks there is no problem at all with the allowable stresses. To be sure that the barrel iron yoke could also be built with sandwiched iron plates we have taken this case as the reference study.

It turns out that the iron slabs with the following reference numbers, YB/0/2/08, YB/0/2/12, YB/0/3/09, YB/0/3/11, have to be reinforced in shear by adding 50000 mm$^2$ of effective shear section at the extremities:

This will be achieved by welding rectangular shear keys at the extremities of the above mentioned slabs. The weld will be 20 mm thick.

The shear stiffness is given by shrink fitting six pins between the layers. These pins have a diameter of 160 mm for the central barrel and 120 mm for the outer ones. The design of a typical absorber block is shown in Fig. 7.14, p. XX.

Brackets:

For the two brackets between the middle and outer slab at the 4 and 8 o'clock position, the maximum Von Mises stress was found to be 282 MPa, as shown in Fig. 7.15, p. XX. These are the most stressed parts of the whole structure. Due to the sharp corner, this value is certainly overestimated by the finite element analysis and in reality this corner will be rounded by the weld, but it remains nevertheless a highly stressed part and the four corresponding brackets will be manufactured in using an appropriate high grade steel or a fine grained structural steel.

The most inner and most outer brackets have maximum stresses of the order of 100 MPa and can be made of low carbon steel. The brackets from inner to middle slabs and from middle to outer slabs have mean stresses below 140 MPa except for those situated at the 4 and 8 o'clock position.

The calculation of the forces acting at the interface between the brackets and the slabs gives extremely important information. The radial contact is assured by tie-rods, 8 are presently foreseen in the central barrel (6 in the external ones), each giving a pre-load of 800 kN. This gives a total force of 6400 kN.

In the tangential direction, friction alone cannot retain the brackets. Therefore shear pins of 70 mm diameter and 800 mm length are inserted on both sides of the barrel.

The radial force values go up to 2500 kN except in the right and left support foot region where 4400 kN occurs but this is still well below the applied prestressing force.

The maximum tangential force transmitted is 7650 kN, which corresponds to a shear stress of 68 MPa for the most stressed pin.

### 7.1.5 FEA Results for the outer barrels

The outer barrels have only to support their own weight and the muon chambers. This totals approximately 1300 tonnes. Therefore less reinforcement is needed. The additional shear reinforcement above the support feet can be omitted and the shear pins between brackets and slabs can be reduced in length to 600 mm. Only 6 tie-bars are required, as opposed to 8 used in the central ring. Fig. 7.11, p. XX, to Fig. 7.13 show the deformation of the outer barrels.

The stresses are all in a range that allows the use of Fe 360 for all brackets and Fe 310 for the absorber blocks.

### 7.1.6 Barrel Materials

The following table shows the grades of steel for the different parts of the barrel.

**Table 7.3**
Characteristics of different grades of steel to be used for the barrel.

| | Absorber steel<br>Fe 310 | Low carbon structural steel<br>Fe 360 D | Fine grained structural steel<br>FeE 560 | Stainless steel<br>SS 304 |
|---|---|---|---|---|
| Yield strength | > 175 MPa | > 235 MPa | > 450 MPa | > 195 MPa |
| Tensile strength | > 310 MPa | > 360 MPa | > 600 MPa | > 500 MPa |
| Elongation after fracture | > 18 % | > 20 % | > 14 % | > 40 % |
| Impact value (ISO-V/20 °C) | > 18 J | > 25 J | > 14 J | > 70 J |
| Used for | All 180 barrel iron blocks | 1440 pins +720 shear pins in blocks + 236 brackets + 10 support feet + 2 Ferris Wheels | 4 highly stressed brackets in the central wheel | complete vacuum tank |
| Mass [t] | **5200 t** | **1500 t** | **15 t** | **285 t** |
| TOTAL [t] | **7000 tonnes** | | | |

For the tie-bars which connect the brackets with the iron blocks we have foreseen a diameter of M36. They are made of St 50.2 and are widely used in civil engineering constructions. At a working stress of 800 MPa, they guarantee a prestress load of 800 kN per bar.

### 7.1.7 Axial forces on barrel rings

According to different computations made with the programs “Poisson” and “ANSYS” there are relatively strong magnetic axial forces between the different barrel rings. Integrated over the whole ring with 3 layers, the forces are of the order of 18000 to 24000 kN. Per sector there are 2000 kN to withstand. This axial force must be resisted by the construction.

The z-stops are blocking the most inner iron ring and thus the bracket connecting to the middle slabs. It is on the interface of this bracket with the middle slab where it is necessary to verify the friction and tack welded shear pins resistance against slipping. Looking at one sector only:

In the outer barrel there are 6 tie-bars, each of them prestressed to a nominal value of 800 kN. The slab is retained by 2 rows of 6 tie-bars thus 12 x 800 kN giving 9600 kN.

For bolted connections with measured bolt preload as it is the case here, a typical coefficient of friction value for mechanical pieces is 0.33. In other words, one third of the measured preload acts as friction. In this case 9600/3 kN are acting as friction, thus 3200 kN.

In addition, the doweling pins which have a diameter of 70 mm will be tack welded. This gives an additional 400 kN retaining force.

In conclusion there are 3600 kN to resist 2000 kN magnetic force acting to move the barrel towards the interaction point, i.e. there is a factor of safety against slipping of 1.8.

Tests are being carried out to see the impact of a “Loctite ®” type of product being applied at the time of assembly between slabs and brackets.

## 7.2 FERRIS WHEEL

From the very beginning structural welding has been avoided in the barrel yoke to allow a trial assembly at the factory thus permitting verification of the geometry before delivery to CERN.

Since the publishing of the Technical Proposal, a lot of work has been done to optimise the assembly process. At the same time, the design of the experimental area and the surface buildings has been finalised, giving more input. It turned out that a rotating jig with specially designed mobile pivoting support brackets was the best compromise between accessibility to the wheel, drilling needs and insertion and fastening of the heavy barrel slabs.

The Ferris Wheel has been studied in detail to prove that the design is sound [7-5, 7-6].

### 7.2.1 Assembly principle

The Ferris Wheel is used twice: first for trial assembly at the factory and then at CERN for final assembly.

To maintain static equilibrium, the barrel iron slabs will be inserted in pairs. The first piece will roll on a trolley which sits on rails just under the Ferris Wheel. It then will be lifted by hydraulic jacks and brought in the exact horizontal position by sliding pads. Once in position, it will be fixed to the pivoting support brackets which are locked in position during the whole assembly.

Once this is done, the second piece will be brought by the crane and positioned on top of the wheel. After the assembly of two slabs (and thus no torque reaction on the system) a hydraulic gear will rotate the entire wheel by 30 degrees.

The same procedure is repeated with another pair of slabs. At the same time, the peripheral brackets which will form the “base” for the second layer are fastened between adjacent slabs. The second and third layer of iron slabs are assembled in the same manner.

With the barrel ring assembly complete, the support feet are attached. Hydraulic jacks provide height adjustment so that when the pivoting support brackets are released, the barrel ring stands on its support feet. The final problem is to remove the barrel ring from the Ferris Wheel structure. This is achieved by extending the central shaft of the Ferris Wheel by 5m, and removing the foot that stands in the way. The barrel ring is then put on the transporting beam and moved out with a system of high pressure air pads installed under the transporting beam, (Chapt. 10, and Fig. 10.1-A and 10.1-B).

The Barrel ring will be stored in the hall, the Ferris Wheel being ready for the subsequent assembly.

A special Ferris Wheel is foreseen for the central barrel, because the pivoting support brackets are not required. This is due to the fact that the central part of the vacuum tank outer shell acts as the support structure for slab assembly, and can be incorporated in the Ferris

Wheel (see Fig. 26.13-B, p. XX).

Figure. 26.13-A, p. XX, shows the Ferris Wheel for the outer rings. The pivoting support brackets which allow the re-use for several assemblies, as needed for the 4 outer barrels, can be seen.

### 7.2.2 Doweling and machining of z-stop surfaces

Due to gravity, the barrel rings deform to an ellipse. This deformation creates large forces in the circumferential direction which cannot be resisted by friction alone. Shear pins of 800 mm depth and 70 mm diameter will be needed in the central barrel to hold the ring together. In the outer barrels, the shear pin diameter of 70 mm will be retained, but the length will be reduced to 600 mm.

To drill the dowel holes is not an easy task. They have to be drilled between two plates of different steel qualities and have to be straight. Test drillings of 70 mm diameter and 800 mm depth have been performed with a special drilling head to prove the feasibility of the technique, see Chapt. 10, and Fig. 10.2-A and 10.2-B, p.XX. The holes drilled at the interface of two bolted plates had a quality of H8 and were straight within 0.2 mm [7-7].

Another important and difficult machining operation is the preparation of the z-stop seats. In the most inner corners of the 12 sided barrel we have a total of 24 z-stops transmitting the axial force of 100 MN coming from the end caps.

After assembly of the first layer of iron slabs on the Ferris Wheel, the whole unit will be taken by the crane (lifting capacity at least 400 tonnes) and positioned on the revolving table of a vertical lathe equipped with a milling head. The doweling of the first layer and the machining of the z-stop seats will be done on this lathe. This can be seen on Fig. 7.16, p. XX.

With the machining completed, the Ferris Wheel will be lifted back on its bearings, to continue the assembly of the remaining layers of iron slabs. A drilling machine, equipped with a dedicated drilling head, installed on a height adjustable platform which is attached to the support members of the Ferris Wheel will be used for the doweling of the second and third layers.

## 7.3 SUMMARY

The various barrel yoke rings of the CMS detector have been analysed with the finite element program CASTEM 2000.

The analysis has shown, for the central barrel ring, that the brackets between middle and outer slabs in the 4 and 8 o'clock position have a maximum Von Mises stress in the corners of the order of 280 MPa. This needs a higher grade alloy steel or fine grained structural steel.

At the same angular location but one step radially inwards, the brackets show a maximum stress of 181 MPa. These should be manufactured in fine grained structural steel.

All remaining brackets in the central barrel and all brackets in the outer barrel rings have stress values lower than 140 MPa and they will be manufactured with a low carbon structural steel such as Fe 360.

To prove the security of the barrel yoke design, an analysis has been made [7-1] with one of the most stressed brackets (4 o'clock) removed. Like spokes in a wheel, the loss of this bracket is taken by a slight increase in stress in all other brackets but does not lead to failure. The remaining brackets show stresses still well below the yield strength. In other words the CMS barrel yoke is a fail safe structure. The maximum stress that occurs at the remaining 8 o'clock bracket is 318 MPa. General deflections increase by only 1 mm in this case.

The main assembly principles as well as the doweling and machining techniques of the most important components have been described. Visits to potential manufacturers have proven the total feasibility of the project in terms of maximum dimensions, weights and required tolerances for the machining.