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DEVELOPMENT OF A BRAZED-ALUMINUM-HONEYCOMB VACUUM VESSEL FOR A THIN SUPERCONDUCTING SOLENOID MAGNET

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ABSTRACT

In high-energy physics experiments, a thin superconducting solenoid magnet is required to provide for accurate momentum measurements of particles. In order to minimize the material in the cryostat as well as in the coil, itself, a brazed honeycomb vacuum vessel has been investigated. As a result, a weight reduction ratio of 1/4 has been achieved in comparison with a solid plate.

INTRODUCTION

The SDC detector[1], which is one of major colliding particle detectors to be built in the SSC acceralator, contains an SDC superconducting solenoid magnet[2] as a core component. Its purpose is to provide the momentum of charged particles produced by $\sqrt{s}$ = 40 TeV pp-collisions. The particle momenta are determined by measuring the curvature of these secondaries in a magnetic field. The SDC superconducting solenoid magnet expected to create an axial magnetic field of 2T. Figures 1-(a) and (b) show isometric views of the SDC detector and the end configurations of the SDC superconducting solenoid magnet, respectively.

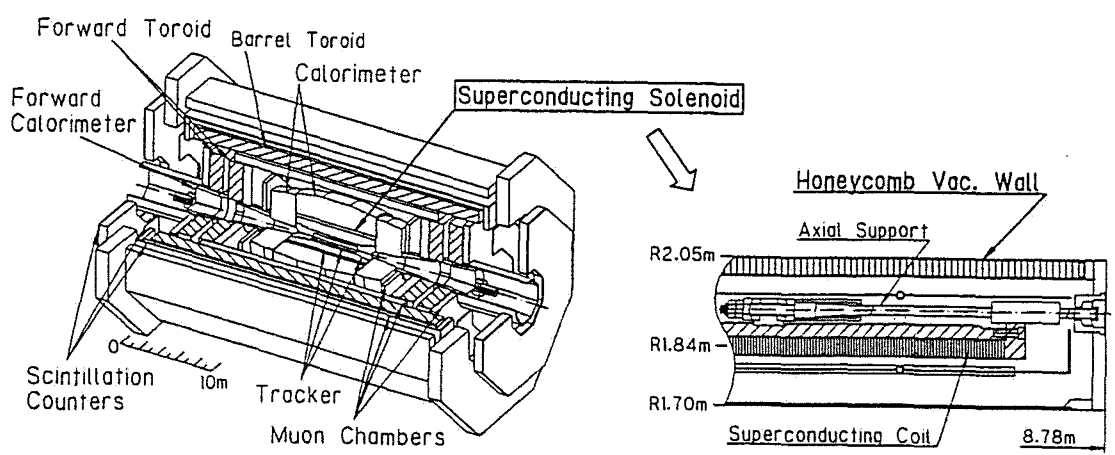


Figure 1. (a) Isometric view of the SDC detector and (b) the end configuration of the SDC solenoid magnet

Table 1. Design parameters of the SDC solenoid magnet.

Dimensions:		
Cryostat	Inner radius	1.70 m
	Outer radius	2.05 m
	Half length	4.39 m
Coil	Effective radius	1.84 m
	Half length	4.15 m
Electrical:		
Central field		2 T
Nominal current		8000 A
Inductance		4.6 H
Stored Energy		146 MJ
Stored energy/cold mass		7.4 kJ/kg
Mechanical:		
Effective cold mass		20 tonnes
Total weight		25 tonnes
Radial magnetic pressure		1.6 MPa
Axial compressive force		13 MN
Axial decentering force (at ΔZ = 2.5 cm)		0.4 MN

Table 2. Transparency of the SDC solenoid wall.

Element	X_0	λ_0
Superconducting coil:		
Outer sup. cylinder	0.348	0.0787
Superconductor		
Al stabilizer	0.438	0.0990
Nb.Ti/Cu	0.181	0.0167
GFRP	0.016	0.0058
Al strips	0.022	0.0051
Cryostat:		
Outer vac. wall (Honeycomb)	0.08	0.0180
Outer rad. shield	0.022	0.0051
Inner rad. shield.	0.022	0.0051
Inner vac. wall	0.067	0.0152
Super-insulation	0.007	0.0023
Total	1.20	0.251

The energy of particle was decreased when it pass through the superconducting magnet in front of a calorimeter. In order to minimize this effect, the radiation length(X_0) and absorption length(λ_0) must be as small as possible in the radial directions. The design parameters of the SDC solenoid magnet and the transparency of the SDC solenoid wall are given in tables 1 and 2, respectively.

Recently, brazed honeycomb panel are being used in building siding, and the bodies of transportation vehicles[3]. A major feature of honeycomb plate is the high stiffness, in spite of its light weight. Further, brazed honeycomb panel have the possibility for welding and high reliability due to the fact that no epoxy-resin is used. The design method of the honeycomb vacuum vessel and the test results of a prototype brazed honeycomb vacuum vessel are described in the following sections.

DESIGN OF A BRAZED HONEYCOMB VACUUM VESSEL

The major reasons of the destruction of the vacuum vessel are considered: the stress on the vacuum vessel is above the yield strength, and buckling takes place due to external pressure. These two causes must therefore be investigated concerning the vacuum vessel design.

The practical mechanical load on the vacuum vessel is a lateral external pressure of 0.1 MPa, an atmospheric pressure of 0.1 MPa, and a decentering force of 40 tonnes in the axial direction. The facing sheet consists of two layers. One is a skin sheet made of 6951-T6 and the other is a brazing layer made of 4043, which is melted during the brazing process. Its Young's modulus is 70000 MPa, Poisson's ratio is 0.3. The tensile strength is 250 MPa and the yield strength is 210 MPa. The allowable stress of 0.1 MPa is determined by the smaller value of each of the following equations:

$$\sigma_a = \frac{1}{4} \times \sigma_t = 62\text{MPa} \quad \text{or} \quad \frac{2}{3} \times \sigma_y = 140\text{MPa}, \tag{1}$$

where σ_a is the allowable stress, σ_t is the tensile strength, and σ_y is the yield strength. Therefore, σ_a is determined to be 62 MPa. The external pressure and axial compressive buckling stress are calculated based on a theory given in "Structural Analysis of Shells"[4].

In calculating the required minimum thickness of facing sheets (t), the stress in the vacuum vessel (σ_a) must be kept below the allowable stress.

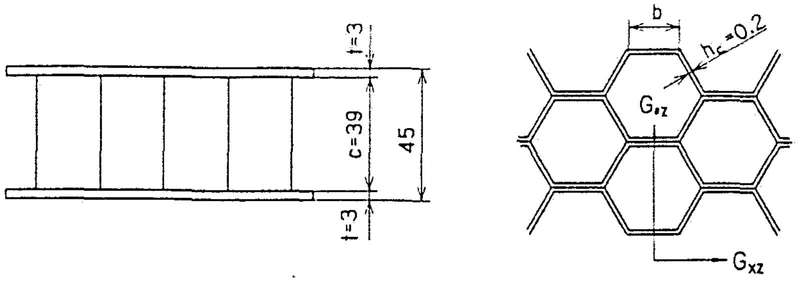


Figure 2. Honeycomb panel configuration in the preliminary design.

Its equation is given by

$$\sigma_\theta \leq \frac{p \bullet r}{2 \bullet t}, \tag{2}$$

where σ_θ is the stress in the circumferential direction. The calculated result is

$$t = \frac{p \bullet r}{2 \bullet \sigma_\theta} = 1.7 \text{ mm.}$$

The safety factor against buckling was assumed to be 2. So the design load therefore includes lateral external pressure of 0.2 MPa, an axial pressure of 0.2 MPa and a decentering force of 80 tonnes in the axial direction, respectively. The designed honeycomb panel configurations is preliminary as shown in Figure 2. In order to achieve the lightest weight possible while maintaining the rigidity of the honeycomb panel, the facing sheets are forced to be as thin as possible and the core thickness is increased. However, by considering the space in the radial directions of the superconducting solenoid magnet, the selected honeycomb panel configurations consists of facing sheets of 3 mm and a core thickness of 39 mm.

Calculations of the external pressure

The design steps to calculate the lateral external buckling pressure are as follows:

- Calculations of parameter: V_p, which is determined by the rigidity of the core in the circumferential direction.
- Calculations of the buckling coefficient: C_p, is determined by the honeycomb panel configurations.
- Calculations of compensation factor: γ, between theory and experiment.
- Calculations of the lateral buckling pressure with each value.

The facing thickness used in the calculations was not 3.0 mm, but 2.69 mm. Since a thickness of 3.0 mm means a nominal thickness which contains a skin , thickness inaccuracy of 0.13 mm, and brazing layer thickness has a 0.6% error in the facing sheet's thickness.

The rigidity of the core is given by the following equation, which is considered in parameter V_p.

$$G_{\theta Z} = \frac{1}{\sqrt{3}} \bullet \frac{h_c}{b} G, \tag{3}$$

where $G_{\theta Z}$ is the transverse shear modulus of the core in the circumferential direction, h_c is the thickness of the core foil, b is the length of one side of a hexagon and G is the shear modulus of the core material, respectively. The core thickness contains an inaccuracy in the skin thickness of 0.03 mm, and a brazing layer thickness of 2% of the facing sheets thickness. h_c is therefore determined to be 0.146mm. The modulus (G) is calculated by following equation:

$$G = E/2(1+\mu). \tag{4}$$

The results of the calculations are $G_{\theta Z}$=203 MPa and G=27000 MPa, respectively. The results of these values are V_p=0.006232, C_p=0.001243 and γ=0.9, respectively.

The equation of the lateral buckling pressure is as follows[4]:

$$p_{cr} = \frac{\gamma C_p}{1-\mu^2} \bullet \frac{2Et}{R}. \tag{5}$$

The lateral buckling pressure based on these values was calculated to be

$$p_{cr} = 0.228\text{MPa}.$$

Calculations of the axial compressive buckling stress

The design steps to calculate the axial compressive buckling stress are as follows:

- Calculations of the parameter V_c, which is determined by the rigidity of the core in the axial directions.
- Calculations of the buckling coefficient C_c, which is determined by the honeycomb panel configurations.
- Calculations of the compensation factor γ, based on both the theory and the experiment results.
- Calculations of axial compressive buckling pressure σ_{cr}, with each value.

Where the G_{xz} used to calculate V_c, is the transverse shear modulus of the core in the longitudinal direction. G_{XZ} is given by

$$G_{XZ} = \frac{\sqrt{3}}{2} \bullet \frac{h_c}{b} G. \tag{6}$$

The result for the above equation is Gxz = 300 MPa. The calculated values are V_c = 0.1899, C_c=0.798 and γ=0.755, respectively.

The axial compressive buckling stress is calculated with an equation[4],

$$\sigma_{cr} = \eta \gamma C_c E \frac{c+t}{R\sqrt{1-\mu^2}}. \tag{7}$$

where η is the plasticity correction term whose value is typically 1 due to the defined in elastic range. The result of calculated axial compressive buckling stress by using these values is therefore σ_{cr}=923 MPa. This value is completely above the yield strength. The practical buckling strength is considered to be between the yield strength and the tensile strength. In fact, the plasticity correction term (η) should be modified using the stress-strain curve of this material and then calculated. Therefore, the following σ_{cr} is adopted, which is the yield strength for a conservative design,

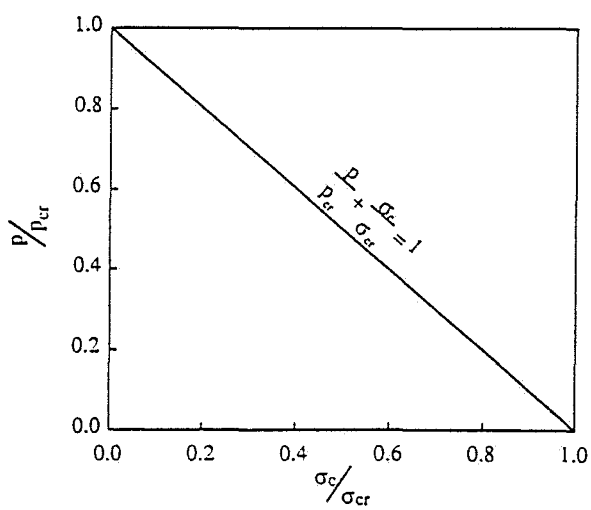


Figure 3. Buckling stress interaction curve for a honeycomb cylinder.

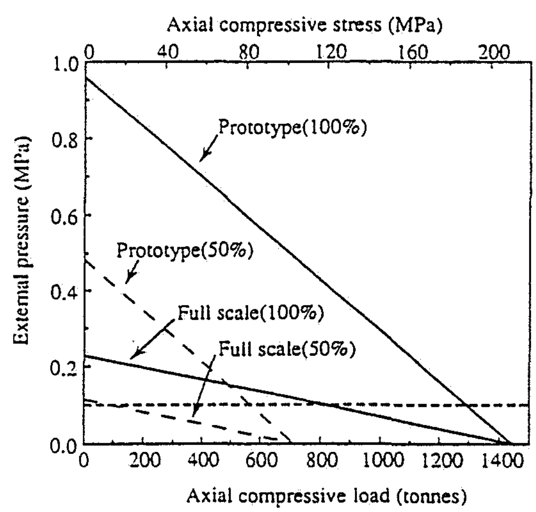


Figure 4. Buckling stress interaction curve.

$$\sigma_{cr} = 210 \text{ MPa}.$$

Calculations of the axial compressive stress

The axial compressive stress to be loaded is calculated by the axial load divided by the cross-sectional area of the facing sheets. Practically, the axial compressive load is shared by the honeycomb outer wall and the inner wall. By using a conservative design, the axial compressive load was assumed to be loaded only on the honeycomb skin layer. Thus, σ_c was calculated as

$$\sigma_c = 23.7\text{MPa}.$$

Buckling under a combined stress

The line guide for avoiding buckling against a combined external pressure and an axial compressive force is given by

$$\frac{p}{p_{cr}} + \frac{\sigma_c}{\sigma_{cr}} < 1. \tag{8}$$

Using each value mentioned so far, the present design can satisfy the above condition as follows:

$$\frac{0.2}{0.228} + \frac{23.7}{210} = 0.99 < 1.$$

The present configurations of the honeycomb panel can be just safe enough against buckling with a safety factor of 2. The buckling stress interaction curve for a honeycomb cylinder under combined external pressure and axial compression is shown in Figure 3. Figure 4 is a modification of Figure 3,indicating the compressive load versus the external pressure.

CALCULATIONS OF THE WEIGHT REDUCTION RATIO

In the case of an applied solid plate, the required thickness was calculated. The design steps are the same as those of the honeycomb vacuum vessel; the tentative plate thickness is selected and then evaluated against buckling using equation (8). The material assumed in this calculation is 5083-O.

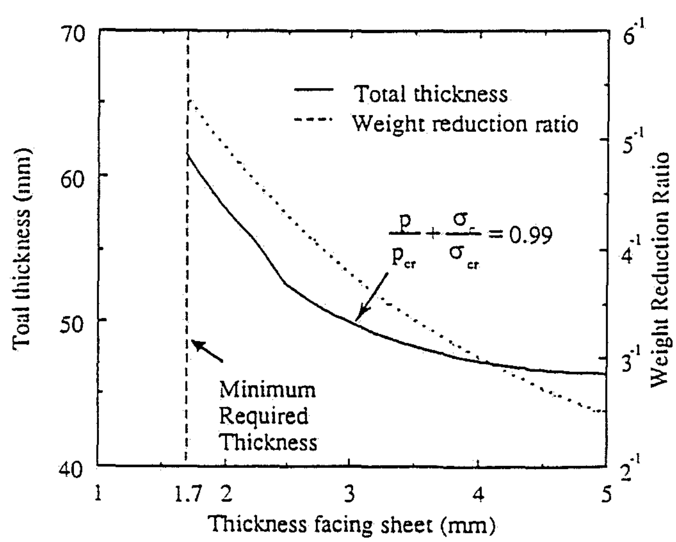


Figure 5. Correlation between the thickness facing sheet and the total thickness.

Table 3. Comparison of solid and honeycomb vacuum shells.

	Solid	Honeycomb
Alminum alloy	5083-0	6951-T6/4045
Total thickness (mm)	27	45
Skin thickness (mm)	-	3.0 + 3.0
Effect. thick. (mm)	27	7.1
Weight reduction ratio	1	0.27
Radiation thick. (X_0)	0.303	0.08
Units to be assembled	12	32

The material properties are a Young's modulus of 70000 MPa, a Poisson's ratio of 0.3, tensile strength of 295 MPa and a yield strength of 150 MPa, respectively. The lateral external pressure was calculated by using NASA SP-8007.[5] The reason for this application is a good fit to the lower limit of the experimental buckling value. The calculations of the axial compressive buckling were carried out by using the theory given in "Structural Analysis of Shells". However, the axial compressive buckling stress was expected to be higher than the yield strength. Therefore, the axial compressive buckling stress was defined to be a yield strength of 150 MPa. After that, the required minimum thickness, which satisfies equation (8), was calculated. The results of the required minimum thickness is 25.7mm. By combining the inaccuracy of the plate thickness (0.9 mm), the solid plate nominal thickness was determined to be t_s = 26.6 mm. The effective thickness of the honeycomb panel was calculated using the following equation:

$$t_{ef} = \frac{A_{foil}}{A_{hexagon}} x t_{core} + 2t. \qquad (9)$$

The first term of the right-hand is the effective thickness of the honeycomb core. Where A_{foil} is cross-sectional area of the honeycomb core per unit and $A_{hexagon}$ is the area of the hexagon per unit, respectively. Therefore, the effective thickness of the honeycomb panel is calculated as follows:

$$t_{ef} = 7.1\ \text{mm}(0.0798 X_0).$$

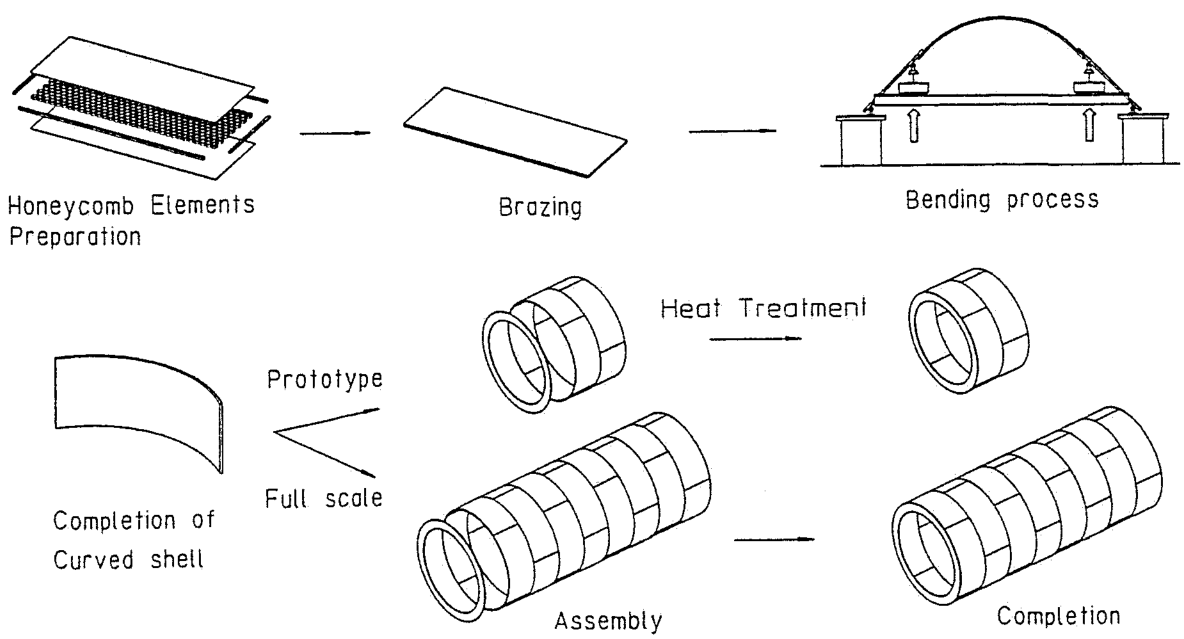


Figure 6. Fabrication steps of a brazed honeycomb vacuum vessel.

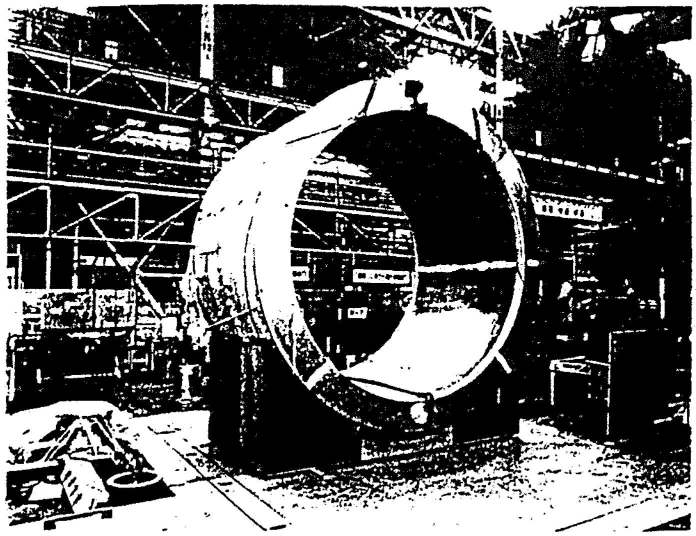


Figure 7. Picture of the honeycomb vacuum vessel.

The weight reduction ratio was calculated based on the effective thickness of the honeycomb panel divided by required solid plate thickness:

$$\frac{t_{ef-honeycomb}}{t_{ef-solid}}=\frac{1}{3.75}.$$

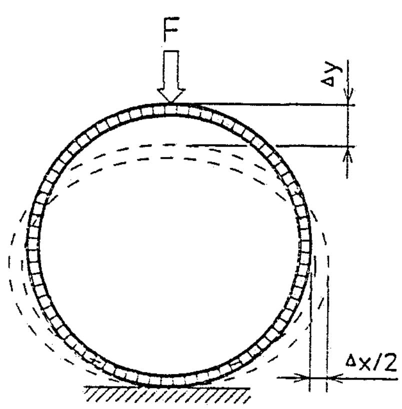
Figure 5 shows the correlation between the thickness of the facing sheet and the core thickness. Table 3 shows a comparison of the solid plate and honeycomb vacuum shells.

DEVELOPMENT OF A PROTOTYPE

A prototype brazed honeycomb vacuum vessel was fabricated in order to ensure the honeycomb vacuum vessel design and to establish the fabrication method. The prototype has a quarter length in the axial directions; a full-size in diameter was made. The fabrication process is shown in Figure 6, and a picture of the completed vacuum vessel is shown in Figure 7.

Several tests were carried out in order to verify the characteristics of this honeycomb vessel,[7] i.e., a stiffness confirmation test and buckling strength test.

A stiffness confirmation test was carried out in order to verify the cylindrical rigidity of the honeycomb cylinder. The measured value were almost the same as theoretical values shown in Figure 8. In the calculation method of the theoretical values, the stiffness of the honeycomb panel was assumed to be equivalent to that of the solid plate.



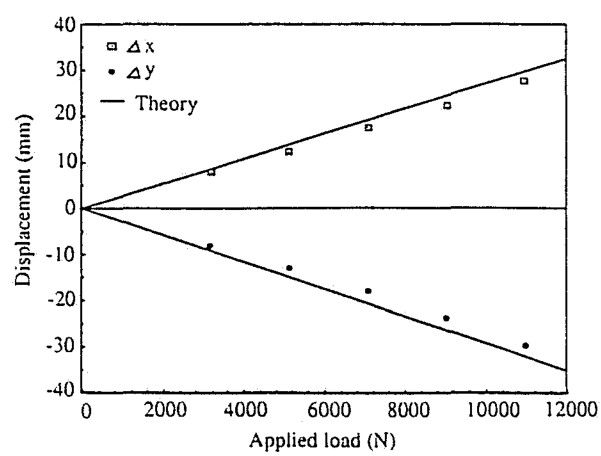


Figure 8. Results concerning the rigidity of a honeycomb cylinder test.

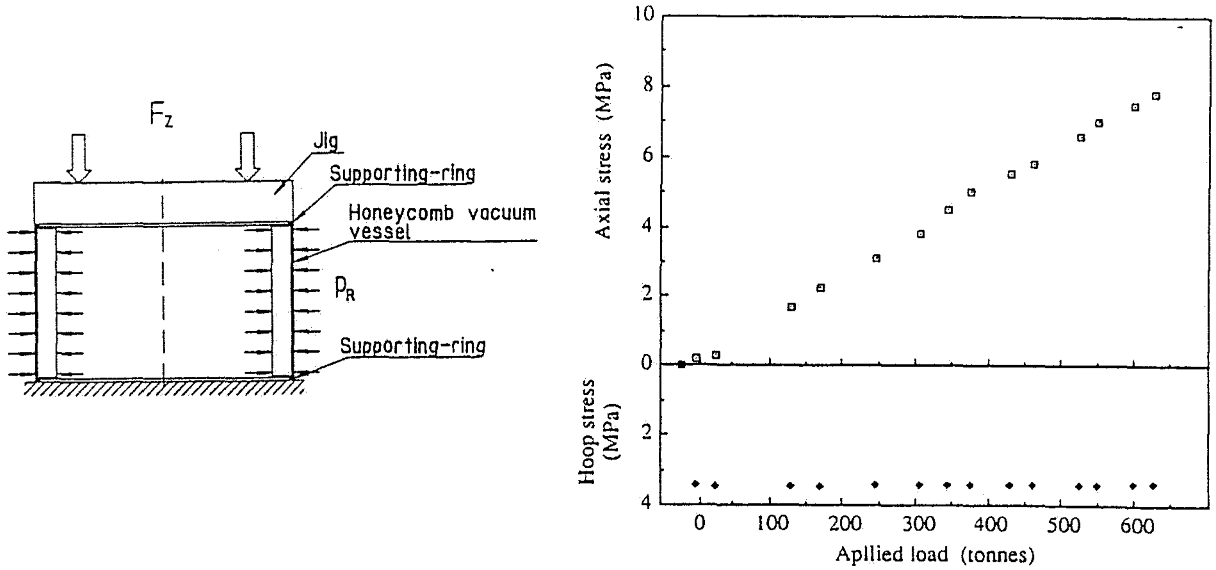


Figure 9. Results of the buckling strength test.

Since the axial compressive buckling force under an atmospheric pressure must be increased in the prototype vacuum vessel to satisfy the safety factor in Eq.(1), the applied load was one atmospheric pressure plus a 600 tonne(max.) axial compressive load, which is the critical buckling condition corresponding to a 50% load, i.e., the practical applied load due to the safety factor was 2 against buckling. In the buckling test, based on the width(75mm) of the supporting-ring, the outer vacuum wall was loaded by 540 tonnes, which is 90% of the overall load , in order to avoid any possibility to the buckle inner vacuum wall before buckling the outer vacuum wall.

The buckling test results are given in Figure 9. The stress on the outer vacuum wall was increased proportionally. In general, although unstable displacement and stress distributions appear when buckling occurs, these phenomena haven't been observed.

CONCLUSION

The design method for a light-weight, thin characteristic vacuum vessel using a brazed honeycomb panel was established. The fabrication technique of a curved honeycomb panel from a flat panel was considered without miss the own rigidity. A vacuum vessel that was applied these curved panel was successfully developed. The reliability and effectiveness of the brazed honeycomb vacuum vessel have been ensured by several test results since the prototype vacuum vessel. It has been understood that the brazed honeycomb vacuum vessel is very effective for the SDC superconducting solenoid magnet. By using the brazed honeycomb vacuum vessel for the SDC superconducting solenoid magnet, light-weight and transparence are excepted to be greatly improved.

REFERENCES

1. G.Trilling et. al., " Solenoidal Detector Collaboration: Technical Design Report," SDC-92-201, 1992.
2. A.Yamamoto et al., " Design study of a thin superconducting solenoid magnet for SDC detector," to be published in Proc. of Applied Superconductivity Conf. (ASX-92), Chicago, IEEE, Trans, Applied Superconductivity.
3. K.Okuto et al., "The analysis and design of honeycomb welded structures," Keikinzoku-yosetsu, Vol.29 No.8 (1991), p13 (in Japanese).
4. E.H.Baker et. al.,"Structural Analysis of Shells," McGraw-Hill Book Company, 1972.
5. " Buckling of thin walled circular cylinders," NASA SP-8007, 1968.
6. T.Tsumura et. al.,"Kyodo Sekkei Data Book," Syokabou, 1973. (in Japanese)
7. I.Ohno et. al., "Brazed Honeycomb Vessel R&D for the SDC Solenoid Magnet," Proc. of IISSC, San Francisco, 1993.