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Development of Subminiature Type 3 Composite Pressure Vessel for Cooling Unit in Electric Appliances

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ABSTRACT

In this study, we have developed a composite pressure vessel that is compact and can store refrigerant at high pressure to increase the refrigerant volume. The composite pressure vessel is made of aluminum-based duralumin, which has high rigidity and excellent elongation in the inner liner, considering the characteristics of products in the aerospace and defense industry, where the safety of the applied product is considered as a priority. High strength carbon fiber was applied to the outside. In order to evaluate the performance of the developed product, burst test and cycling test were carried out. In burst test, an excellent safety margin equivalent to 2.7 times the operating pressure was obtained. In cycling test, a stable failure mode in which 'pre-burst leak' occurs is proved and the soundness of the product is proved.

Key Words : Composite(복합재), Pressure Vessel(압력용기), AL Liner(알루미늄 라이너), Cooling Unit(쿨링 유닛), Subminiature(초소형)

1. Introduction

Artificial intelligence products used in the aerospace and defense industries consist of several tens of millions of electronic components, and it costs several tens of thousands to several hundreds of thousands of dollars when failures occur due to poor heat source discharge. Consequently, these products remove heat of electronic components by cooling gases, which are stored in compressed condition to increase the storage amount. Many ultra-small pressure vessels made of metals have been used to compress these cooling gases[1]. However, these metal pressure vessels have the problem of using many vessels and cooling tubes because they have a pressure limitation for compressing refrigerants[2]. To address this problem, this study developed an ultra-small composite pressure vessel that can increase the amount of refrigerant by storing refrigerants at a high pressure. For this composite pressure vessel, aluminum-based duralumin was applied to the inner liner, which

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boasts high rigidity and excellent elongation, considering the characteristics of aerospace and defense products for which safety is the first priority. In addition, high-strength carbon fibers were applied to the outside to withstand a high internal pressure. To evaluate the performance of the developed product, a burst test and a repetition test were carried out. The burst test achieved an excellent safety margin that is equal to 2.7 times the applied pressure.

2. Design and Structural Analysis of Ultra-small Composite Pressure Vessel

2.1 Liner Design Theory

The factors that have the largest effect on performance in the design of a filament wound pressure vessel are the liner shape and the angle of the wound fibers. In fact, when a pressure vessel is manufactured, the mandrel is produced first in accordance with the design requirements (e.g., required vessel volume, external shape, and boss radius).

In this study, the semi-geodesic path equation was used to calculate the winding pattern in a curved shape of an anisotonic tension dome. The formulation of the semi-geodesic path equation applied to the filament wound pressure vessel is as follows[3-5]:

$$\frac{d\alpha}{dx} = \frac{\lambda(A^2\sin^2\alpha - rr''\cos^2\alpha) - r'A^2\sin\alpha}{rA^2\cos\alpha} \tag{1}$$

Since the dome is axially symmetric, it can be calculated by integration from the start of the dome or the initial value at the opening if the surface shape of the liner is given[6].

The dome shape designed based on this anisotonic tension dome theory forms lines that are different from those of an isotonic dome shape, which

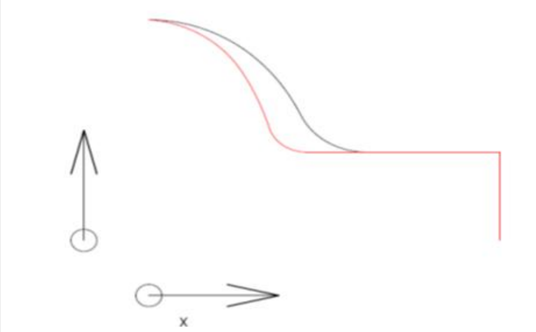

Fig. 1 Dome contour of iso-tensoid vs. non iso-tensoid

is generally used in composite pressure vessels, as shown in Fig. 1.

2.2 Winding Design Theory

The winding layer that supports most of the internal pressure of the composite vessel was designed based on the mesh theory. This theory starts from the assumption that fibers support the total internal pressure, and the pattern design of the winding layer is carried out through the following Eq. (2), which is a relationship equation between the winding angle and the curved shape of the dome [7,8]:

$$2 + \frac{rr''}{1+r'^2} = \tan^2\alpha \tag{2}$$

where r is the dome radius and α is the fiber angle. The winding design pattern of the developed product was derived by applying this mesh theory, and the result is shown in Table 1.

2.3 Structural Analysis

Finite element analysis was conducted to examine the performance of the proposed concept. This product has a 360-degree rotating shape around the

Table 1 Filament winding pattern

Pattern	Angle (deg.)	Quantity	Thickness (mm)
Hoop	88.18	1	0.35
Helical	28.42	1	0.46
Hoop	88.18	1	0.35

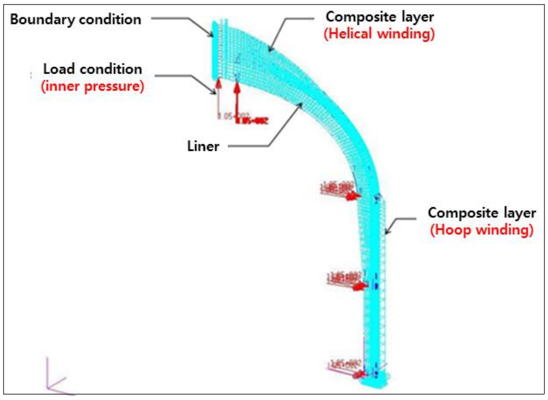

Fig. 2 Finite element modeling of composite pressure vessel

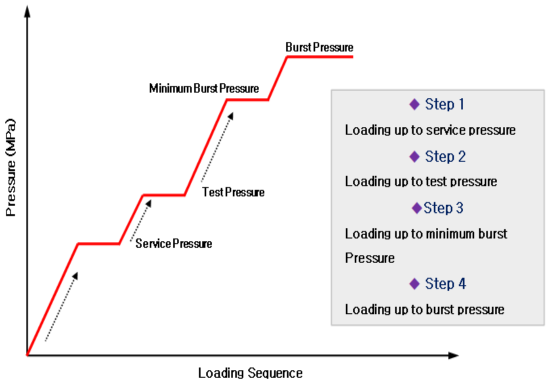

Fig. 3 Loading sequence of composite pressure vessel

central axis. Thus, as shown in Fig. 2, only some of the central angles were modeled, and periodic symmetry conditions were given to the section of the model in the circumferential direction due to the existence of a helical layer that has a repetitive shape in a fixed cycle. In addition, to accurately predict the anisotropy of the filament winding structure and the local stress distribution of the cylindrical-dome part, a second-order solid element of 20 nodes was applied, which was provided by the commercial finite element application ABAQUS.

For the load, the incremental path process was applied, which increased the internal pressure in steps, as shown in Fig. 3.

2.3.1 Stress Distribution with Autofrettage Pressure

Autofrettage is a pre-process that increases the fatigue characteristic of the liner when the applied pressure is charged by generating a compressive residual stress to the inner aluminum liner.

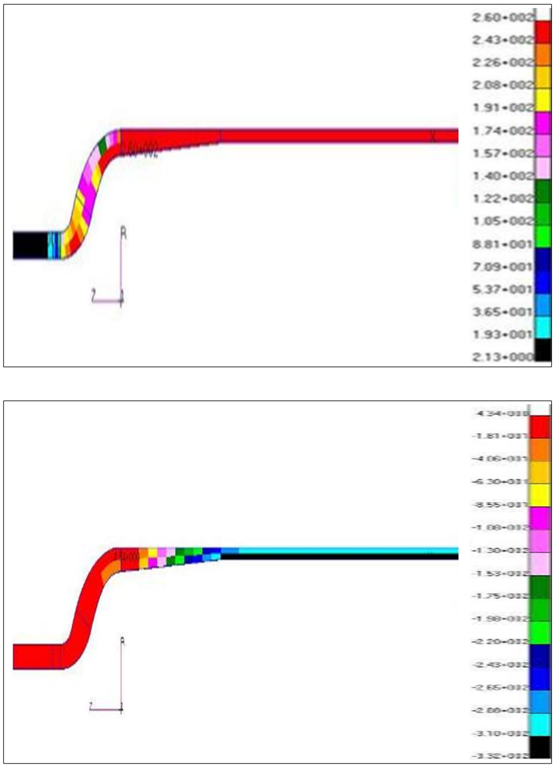

Fig. 4 Stress distribution of aluminum liner at autofrettage pressure; (upper) autofrettage pressure step, (lower) zero pressure step

The stress distribution when the autofrettage pressure is applied in Fig. 4 shows that the aluminum liner went beyond the yield stress (260 MPa) and underwent full plastic deformation. After that, the liner under zero pressure was subjected to compressive stress and had a negative stress. At this moment, the liner had a pressure of approximately 96.6% of the compressive residual stress (based on Von Mises).

2.3.2 Stress Distribution with Applied Pressure

Fig. 5 shows the stress distribution when the applied pressure is charged. The circumferential stress in the vessel is 41.4% of the minimum burst pressure. This means that the stress generated by the applied pressure is less than half of the stress generated in the fibers when the minimum burst pressure is applied, and it implies that the safety margin of the design is at least twice as high.

2.3.3 Stress Distribution with Internal Pressure

Fig. 6 shows the stress distribution during internal pressure charging, and the circumferential stress is 55.4% of the minimum burst pressure.

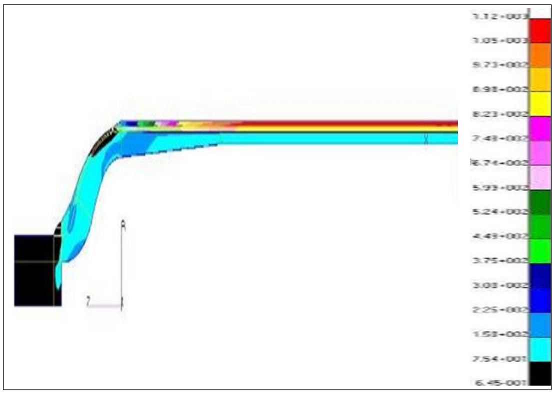

Fig. 5 Stress distribution of composite pressure vessel at service pressure

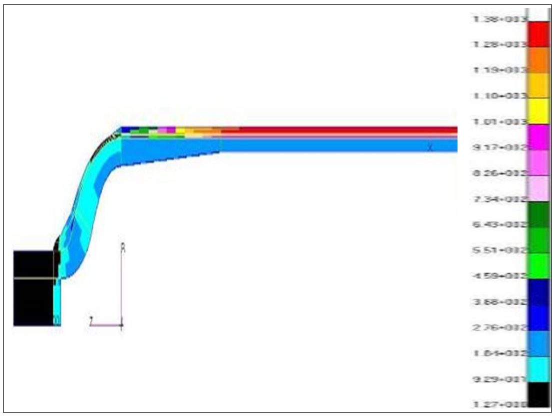

Fig. 6 Stress distribution of composite pressure vessel at test pressure

2.3.4 Stress Distribution with Minimum Burst Pressure

Fig. 7 shows the internal stress of the composite under the required minimum burst pressure. By the definition of the minimum burst pressure, the stress of the composite must be equal to or higher than the fiber burst pressure. Thus, the predicted fiber burst pressure is 3,030 MPa, but the maximum stress of the composite under the minimum burst pressure is 1,890 MPa, which is smaller than the fiber burst pressure. Therefore, it can be determined

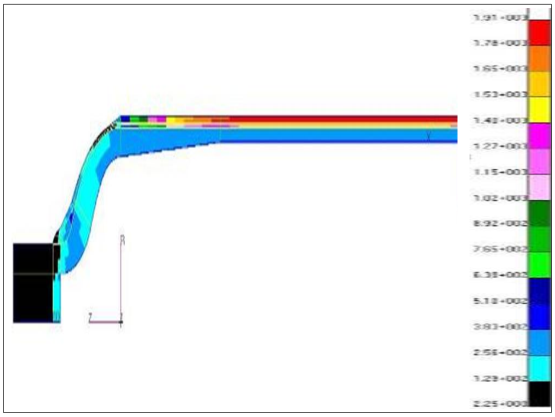

Fig. 7 Stress distribution of composite pressure vessel at minimum burst pressure

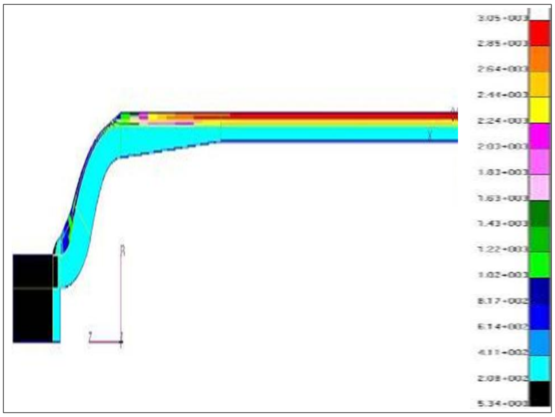

Fig. 8 Stress distribution and deformation of composite pressure vessel at design burst pressure

that the fibers will not burst at least under the minimum burst pressure. Furthermore, this proves that the present design has an actual burst pressure above the minimum burst pressure

2.3.5 Stress Distribution with Design Burst Pressure

For the prototype of this developed product, vessel burst was induced in the hoop layer (circumferential fiber-reinforced layer) of the body. The vessel burst begins when the reinforcement fibers are burst, and a stable burst mode can be obtained as the fibers start to burst in the hoop layer of the body and spread along the length of the body.

Considering the safety margin and the strength reduction due to the volume ratio with the resin when fibers are input and formed into a composite, it is expected that the fiber-reinforced composite layer of this vessel will burst at 3,030 MPa. This pressure was calculated as 1,391 bar by finite element analysis. When it was compared with the measured value through an actual burst test, the error was 5%, proving the reliability of the theoretical solution.

3. Tests of Ultra-small Composite Pressure Vessel

3.1 Burst Test

The vessel burst test was conducted with prototypes. After filling the test apparatus with a liquid so that no air would remain in the vessel, it was slowly pressured at a fixed rate. To determine the minimum burst pressure, the vessel was pressured to 966 bar (14,000 psi), which is twice the applied pressure, and then kept for 60 s. Then, the vessel was slowly pressured at a fixed rate until it burst.

As a result of the burst test, the vessel burst at 1,329 bar, which is lower than the design burst pressure (1,391 bar), as shown in Table 2.

This value is less than 5% of the standard error for the design burst pressure, verifying the excellent reliability of the theoretical solution.

Table 2 Burst test result of composite pressure vessel

	Bust test
Volume	200cc
Diameter	39.9mm NOM.
Weight	0.19kg
Burst pressure	1,329 bar
Burst mode	

Table 3 Cycling test result of composite pressure vessel

	Cycling test
Volume	200cc
Diameter	39.9mm NOM.
Weight	0.19kg
Cycling No.	32,634 Cycling
Leak mode	

3.2 Repetition Test

A hydraulic repetition test was performed at constant temperature for the composite pressure vessel sample. The test procedure and method followed the ISO11119-2, which is the European industrial composite pressure vessel certification standard (0–60 MPa, 15 times or less per min, repetition until leakage).

As a result of the repetition test, which is shown in Table 3, a safe burst mode was obtained, as a leakage before burst occurred in the body of the vessel, and a safety margin that was six times higher than the certification standard (5,000 repetitions) was achieved.

4. Conclusions

This study developed an ultra-small/ultra-high-pressure composite pressure vessel for high compression of a pressure vessel used to cool the heat source in cooling units for electronic equipment. For this composite pressure vessel, aluminum-based duralumin was applied to the inner liner, which boasts high rigidity and excellent elongation, considering the characteristics of the aerospace and defense industries that place the highest priority on product safety. In addition, high-strength carbon fibers were applied to the outside to withstand a high internal pressure. The performance of the developed product was evaluated through a burst test and a repetition test. The burst test results showed an excellent safety margin that was 2.7 times the applied pressure, and the repetition test derived a stable burst mode in which a leakage before burst occurred, proving the integrity of the product.

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