Preliminary Study on Deformation During Hydrostatic Testing in a Deep Tank

Geun-Gon Kim\(^1\), Tae-Hyun An\(^2\) and Tak-Kee Lee\(^3\)

\(^1\)Graduate Student, Department of Ocean System Engineering, Graduate School, Gyeongsang National University, Tongyeong, Korea

\(^2\)Graduate Student, Department of Naval Architecture and Ocean Engineering, Graduate School, Gyeongsang National University, Tongyeong, Korea

\(^3\)Professor, Department of Naval Architecture & Ocean Engineering, Gyeongsang National University, Tongyeong, Korea

KEY WORDS: Deep tank, Hydrostatic test, Deformation, Structural analysis, Beam theory

ABSTRACT: There are many different types of tanks on ships that meet various requirements. Each tank is required to undergo hydrostatic testing according to the Ship Safety Act after being installed onboard. In some hydrostatic tests, excessive deformation may occur. The overpressure of the air in the tank generated during testing is one of the possible causes of deformation. Based on the dimensions of the tank, nozzle, and pipes installed, it was confirmed that the overpressure of the air can cause problems with the structure, according to the Bernoulli equation. Additionally, finite element analysis (FEA) was performed on the tank structure to confirm the deformation and the stress occurring in the structure. From the perspective of deformation, the maximum deflection limit was set based on the criteria provided by the Eurocode and DNV. From the perspective of stress, the structural safety assessment was performed by comparing the allowable stress and equivalent stress generated in the structure. To determine whether the behavior of the actual structure was well implemented via FEA, beam theory was applied to the tank structure and compared with the FEA results. As a result of the analysis, severe deformation was found in some cases. This means that the overpressure of the air may be the cause of actual deformation. It was also confirmed that permanent deformation may occur.

1. Introduction

Many types of equipment with various uses are installed during the construction of complex marine structures (i.e., ships). Among them, tanks, which are pressure vessels for storing fluids such as fuel oil for engine operation, urea water for post-treatment of exhaust gas, freshwater for use by sailors residing on the ship, as well as natural gas in both liquid and gaseous states, are installed in various locations inside ships. Most cargo ships have a deep tank over part or all of the width between the tank top and the lower or upper deck.

When the onboarding installation of every tank is completed, a hydrostatic test is conducted to verify the tightness and structural safety of the tanks. This hydrostatic test is applied to various places, especially to check water leakage of tanks and pipes. Because pressure vessels are used in most industries, studies on hydrostatic tests have been performed for various purposes in a variety of fields.

Kiefner and Maxey (2000) and Kiefner (2001) defined the pressure level and test time for effective hydrostatic tests of pipelines. Stephen et al. (2010) analyzed the cause of the expansion of the pipe diameter that occurs in specific pipelines during hydrostatic tests. Cameron and Pettinger (2010) investigated the effect of hydrostatic tests on the growth of existing cracks in pipe material after the hydrostatic testing of pipelines.

Furthermore, Krieg et al. (2018) pointed out a technical constraint of hydrostatic testing, such that it cannot check all defects as it can only examine cracks under test pressure. They also introduced a better method, in-line inspection (ILI), in pipe structures. A key point for performing the ILI method is to install a smart pipe inspection gauge (PIG) in the pipeline. The PIG inserted into a pipe can collect data of potential risks, including external and internal defects, corrosion, and the structural safety of pipe systems, and can prevent problems before they occur. Hydrostatic tests have a technical limitation in that they only identify cracks that occur during the test. However, the ILI method is a non-destructive test that has the advantage of identifying all external and internal defects without performance degradation. Moreover, because the pipeline can be operated continuously, tests for...
defects can be conducted without interrupting production. Notably, the ILI method has the disadvantage of only being applicable to pipelines. Hydrostatic tests have been also studied in the nuclear power generation field. Zhang et al. (2021) described hydrostatic tests for nuclear pressure vessels and optimized the hydrostatic test procedure, related equipment, and associated technology to prevent the enormous damage that can be caused by nuclear accidents compared with those in other industries. Meanwhile, improved methods other than hydrostatic testing have also been suggested owing to concerns regarding the expansion of existing defects that may arise from performing hydrostatic tests. Liao et al. (2019) studied a method of analyzing the damage mechanism of pressure vessels for hydrogen storage according to loads based on acoustic emission information generated during hydrostatic testing.

Hydrostatic tests have been extensively studied in a variety of fields because the structural problems associated with pressure vessels gradually accumulate over time. Pressure vessels are highly vulnerable to various types of damage, such as wear, scratching, impacts, and aging. Such damage can reduce the load capacity and fatigue performance of pressure vessels (Liao et al., 2019). Therefore, optimization of the hydrostatic test procedure and technology has been conducted as a measure of regularly inspecting pressure vessels to ensure safety, and methods to minimize internal damage caused by hydrostatic tests have been studied.

In fact, many accidents occur during hydrostatic tests, and the causes of these accidents have been analyzed. In 2013, an accident occurred in South Korea when a leak ensued during the hydrostatic testing of a bolted-joint-type onshore storage tank; the test was continued, and the accident occurred because the water tank could not withstand the load and collapsed (Lee, 2013). In an overseas hydrostatic test accident, the top part exploded during a hydrostatic test of a newly produced vertical pressure vessel, which was caused by the temperature of freshwater used for the hydrostatic test. This case demonstrated that brittle fracture can occur when low-temperature freshwater is used (Parthiban, 2012). Furthermore, there was a case of explosion and collapse of a vertical pressure vessel that was caused by overpressure inside the container because the amount of air escaping was small compared with the speed of water being poured into the container during the hydrostatic test. It was found that the ventilation pipe at the top of the container was blocked by vinyl and other objects.

There are also accident cases showing that more than only pressure vessels should be considered when conducting hydrostatic tests. In an accident case for a spherical LPG tank, a problem occurred with a member supporting the tank. During the hydrostatic testing of the spherical tank, water was poured up to 80% of the height without problems, but the part supporting the spherical tank had aged and failed to support the weight of the tank as it filled with water, and the structure collapsed. This suggests that not only the pressure vessel but also the condition of the entire structure must be considered when conducting a hydrostatic test (Dey, 2017).

The above literature review reveals that, although many studies on hydrostatic tests have been conducted, most have focused on pipelines. Therefore, it is necessary to establish parameters that affect the degree of deformation and the structural safety of tanks according to the pressure conditions during hydrostatic tests for welded tanks in ships.

Onboard tanks are generally thin-walled structures with a relatively small thickness relative to the major dimensions. Because the thickness is small, changes in behavior in the thickness direction can be ignored and two-dimensional (2D) meshes can be applied. Hence, analysis can be performed effectively even with a relatively small number of elements (Cho, 2019).

In this study, finite element analysis (FEA) was conducted according to six analysis scenarios for a urea water storage tank, a deep tank welded to the transverse bulkhead in the stern. The accuracy of the results was verified by comparing the deformation results from the hydrostatic testing of the tank with those from the theoretical method applying beam theory.

2. Hydrostatic Test

2.1 Definition and Procedure

According to the Common Structure Rules for Bulk Carriers and Oil Tankers (IACS, 2021), the purpose of the hydrostatic test is to verify the water-tightness of the tank and the watertight boundary and the structural safety of the tank constituting the watertight compartment of the ship. The tanks of newly built ships or vessels undergoing major modifications or repairs must be tested for safety according to the hydrostatic test procedure before delivery of the ship. The hydrostatic test is conducted in a tank for storing liquids. The method of hydrostatic testing involves filling appropriate freshwater or seawater in the test area to the specified water level, unless there is another approved liquid. All the exterior surfaces of the test area are checked for structural deformation, expansion, buckling, and other related damage and leakage. Table 1 shows the test water head or pressure according to the subject of the test.

The hydrostatic test must be performed in accordance with the procedure specified in the Ship Safety Act (MOF, 2020). Hence, a ship survey officer must be present during a hydrostatic test, and the water head and pressure, as well as any water or air leakage, must be checked during the test. Furthermore, the existence of deformation of members under pressure must be checked, and if deformation is found, it must be corrected and reinforced.

The procedure of the hydrostatic test is as follows, for which the tank hydrostatic test procedure of Bay is referenced (Bay Tank and Vessel, 2016). Before and after filling operations, it must be checked that all vents are open and unobstructed. Then, the tank is filled with water up to the maximum water level, and tests are performed in stages by setting four water levels (i.e., 1/4, 1/2, 3/4, and full). Table 2 shows the filling rate criteria found in API 650, where it is recommended not to exceed the maximum water-filling rate.
### Table 1 Design testing load height $z_{ST}$ (IACS, 2021)

| Compartment | $z_{ST}$ |
|-------------|-----------|
| Double bottom tanks | The greater of the following: $z_{ST} = z_{TOP} + h_{air}$, $z_{ST} = z_{md}$ |
| Hopper side tanks, topside tanks, double side tanks, fore and aft peaks used as tank | The greater of the following: $z_{ST} = z_{TOP} + h_{air}$, $z_{ST} = z_{TOP} + 2.4$ |
| Tank bulkheads, deep tanks, fuel oil bunkers | The greater of the following: $z_{ST} = z_{TOP} + h_{air}$, $z_{ST} = z_{TOP} + 0.1P_{PV}$ |
| Ballast hold | $z_{ST} = z_{k} + 0.9$ |
| Chain locker | $z_{ST} = z_{C}$ |
| Independent tanks | The greater of the following: $z_{ST} = z_{TOP} + h_{air}$, $z_{ST} = z_{TOP} + 0.9$ |
| Ballast ducts | Testing load height corresponding to ballast pump maximum pressure |

Note: $z_{ST} = z$ coordinate, in m, design testing load height. $z_{TOP} = z$ coordinate of the highest point of tank, excluding small hatchways, in m. $z_{md} = z$ coordinate, in m, of the bulkhead deck. $z_{k} = z$ coordinate, in m, of the top of hatch coaming. $z_{C} = z$ coordinate, in m, of the top of the chain pipe. $h_{air} = $ height of air pipe or overflow pipe above the top of the tank, in m. $P_{PV} = $ design vapour pressure, in kN/m², but not less than 25 kN/m².

### Table 2 Water-filling rate (API, 2020)

| Bottom course thickness | Tank portion | Maximum filling rate (mm/h) |
|-------------------------|--------------|-----------------------------|
| Less than 22 mm         | Top course   | 300                         |
|                         | Below top course | 460                      |
| 22 mm and thicker       | Top third of tank | 230               |
|                         | Middle third of tank | 300               |
|                         | Bottom third of tank | 460               |

### 2.2. Analysis of the Cause of Deformation and Establishment of Analysis Scenarios

Deformation can occur in the process of some hydrostatic testing. Conditions for the occurrence of deformation include the overpressure phenomenon found in tanks. This overpressure condition can cause structural damage to the outer wall of tank. However, in this study, it is assumed that the main dimensions of the tank satisfy relevant regulations, such as the Common Structural Regulations (IACS, 2021).

The cause of overpressure in a tank during the process of hydrostatic testing can be assumed as follows:

1. Reduced discharge performance of the air pipe
2. Difference in diameter between inlet pipe and air pipe
3. Water filling rate faster than the proper rate

First, overpressure can occur if the condition of the air pipe is poor. The air pipe discharges air from the inside to the outside of the tank during hydrostatic testing. If foreign substances accumulate inside the air pipe and interfere with or even block the air flow, causing reduced discharge performance, the air in the tank cannot be discharged normally through the air pipe and is compressed as the water fills in the tank, resulting in overpressure.

Second, if the difference in the diameter between the inlet pipe and air pipe is large, overpressure may occur if the ratio of the amount of air inside the tank going out through the air pipe to the flow rate of water entering the tank is out of the normal range. The regulations of the DNV require that the air pipe cross-section be 1.25 times that of the water inlet part (DNV, 2021).

Third, if the water-filling rate is increased to shorten the overall test time during the hydrostatic test, the air inside the tank can be compressed, which can cause overpressure. To prevent overpressure occurring for this reason, API 650 regulates the water filling rate for welded tanks, as shown in Table 2 (API, 2020).

In this study, rather than only considering the above three causes of overpressure, we examine the deformation of the tank outer wall based on the pressure state that can result from various causes. Therefore, the following three scenarios were established regarding the discharge of air from the tank:

1. When the performance of the air pipe is normal: the air in the tank is smoothly discharged during the water filling process
2. When the air pipe performance is reduced by 50%: only 50% of the air is discharged
3. When the air pipe performance is reduced by 100% blocked

In addition to the above scenarios for air pipe discharge performance, we decided to review the case where water was filled up to the 50% level and the case where water was filled up to the 90% level (i.e., just before reaching the full water level among the tank hydrostatic test procedures of Bay). Consequently, a total of six scenarios was defined, with air pipe performance and water filling level as variables, as listed in Table 3.

### Table 3 Considered scenarios with respect to air pipe performance and water filling level

| Case | Air pipe performance | Water filling level |
|------|----------------------|---------------------|
| Case 1 | 100 %                | 50 % of height      |
| Case 2 | 50 %                 | 50 % of height      |
| Case 3 | 0 %                  | 50 % of height      |
| Case 4 | 100 %                | 90 % of height      |
| Case 5 | 50 %                 | 90 % of height      |
| Case 6 | 0 %                  | 90 % of height      |
3. Possibility of Damage to Tank Considering Water Filling Situation

In this section, among the possibilities of occurrences of overpressure assumed above, the latter two causes are examined theoretically, considering the water filling situation. Before the theoretical review, the dimensions of the tank are presented in detail in Section 4.1. In the continuity equation for flow rate, the amount of fluid flow per unit time is the same for both fine and coarse streams (Cengel and Cimbala, 2014). This follows Eq. (1) below:

\[ Q_i = A_i \times V_i = Q_j = A_j \times V_j \]  

(1)

where \( Q \) = volume flow rate of member \( i \), \( A \) = cross-sectional area of member \( i \), and \( V \) = fluid velocity passing through member \( i \).

Because the flow rate, \( Q \), of member \( i \) per unit time is equal to the flow rate, \( Q_j \), of member \( j \) per unit time, the flow rate passing through the inlet pipe of the tank is equal to the flow rate passing through the air pipe. From this, the relationship for the fluid velocities of the inlet pipe (1) and air pipe (2) can be derived as follows:

\[ V_2 = \frac{A_i}{A_2} \times V_1 \]  

(2)

For example, if the diameter of the inlet pipe is 2.5 times the diameter of the air pipe, the fluid velocity relationship of the air pipe and inlet pipe is \( V_2 = 6.25 \times V_1 \). If we assume that it took time \( T \) for filling freshwater to a specific point of the tank after opening the inlet pipe 100%, the flow rate \( Q \) can be determined by multiplying Eq. (1) by the time \( T \). From this, the fluid velocity \( V_1 \) of the water inlet part can be expressed as Eq. (3):

\[ V_1 = \frac{Q}{A_1} T \]  

(3)

Meanwhile, the fluid velocity relationship for the inlet pipe and air pipe can be derived through Bernoulli’s law for fluid velocity and pressure, expressed as Eq. (4) (Cengel and Cimbala, 2014):

\[ \frac{P_i}{\rho} + \frac{1}{2} V_i^2 + gz_i = \frac{P_j}{\rho} + \frac{1}{2} V_j^2 + gz_j \]  

(4)

where \( P/\rho \) is the flow energy, \( V^2/2 \) is the kinetic energy, and \( gz \) is the potential energy.

In the abovementioned equation, the pressure inside the tank containing gas is too small to generate a meaningful difference resulting from the gas weight. Hence, it can be assumed that \( z_i = z_j \), and Eq. (4) can be simplified to Eq. (5):

\[ P_i + \frac{1}{2} \rho V_i^2 = P_j + \frac{1}{2} \rho V_j^2 \]  

(5)

As shown in Table 4, the additional pressure \( P_1 - P_2 \) caused by the flow rate change can be calculated. As a result, \( P_1 - P_2 \), which means differential pressure, can be calculated.

For example, if the total flow rate for 2 h when the diameter of the inlet pipe is 50 mm is 62.1 m³, the change in fluid velocity according to the degree of opening of the inlet pipe and the pressure difference between the two pipes can be calculated using the above process, and the results for this are presented in Table 4. If a hydrostatic test is performed assuming that the diameter difference between the inlet pipe and air pipe is 2.5 times, as in the above example, there is a possibility of structural deformation caused by the pressure difference resulting from the diameter difference between the inlet pipe and air pipe.

4. Deformation Analysis of Tank Outer Wall

4.1 Modeling and Boundary Conditions

In this study, static structural analysis was performed with respect to deformation scenarios through the commercial analysis program ANSYS ver. 21 R1 (DNDE, 2021), considering the structure of the urea water storage tank that is welded to the transverse bulkhead in the stern. For this analysis, an analysis model consisting of shell elements was created using the ANSYS/Spaceclaim program. The analysis model is shown in Fig. 1 below, and the dimensions of the model are summarized in Table 5. The structural steel used to build the ship was created using the ANSYS/Spaceclaim program. The analysis model is shown in Fig. 1 below, and the dimensions of the model are summarized in Table 5. The structural steel used to build the ship was reflected in the analysis model, and the mesh size is 150 × 150 mm. The material properties of the structural steel are summarized in Table 6.

The urea water storage tank (the analysis model of this study) is welded to a part of the transverse bulkhead. The transverse bulkhead plate is externally reinforced, and another plate surrounding the tank is internally reinforced. There is a platform deck at the 50% height of the tank. In addition, the top and bottom of the tank are supported by welding through I-beams between the strength deck and the lower deck, respectively.

| Table 4 Estimation of additional pressure related to water velocity of inlet pipe |
|----------------|--------|--------|------------------|
| Inlet pipe condition | \( V_1 \) (m/s) | \( V_2 \) (m/s) | \( (P_1 - P_2) \) (kN/m²) |
| 100% open | 4.4 | 27.5 | 368.5 |
| 80% open | 3.5 | 22.0 | 232.8 |
| 60% open | 2.6 | 16.5 | 128.4 |
| 40% open | 1.8 | 11.0 | 61.6 |
| 20% open | 0.9 | 5.5 | 15.4 |
| 15% open | 0.7 | 4.1 | 9.3 |
| 10% open | 0.4 | 2.8 | 3.0 |
| 5% open | 0.2 | 1.4 | 0.8 |

Note: \( V_1 \) and \( P_1 \) = water velocity and pressure in inlet pipe

\( V_2 \) and \( P_2 \) = water velocity and pressure in air pipe
Before performing structural analysis of the tank, boundary conditions must be assigned for the support points. Fig. 2 shows the boundary conditions in the considered analysis. When the actual behavior of the urea water storage tank, which is the subject of analysis, is considered, the subject of analysis is welded as part of the transverse bulkhead, and it is supported by I-beams welded to the upper and lower parts of the tank. Moreover, if the platform deck inside the tank is considered, the actual behavior can be similarly depicted by constraining the degrees of freedom in all translation ($T_x, T_y, T_z$) and rotation ($R_x, R_y, R_z$) directions for the supports, as shown in Fig. 2.

4.2 Loading Conditions

The loading conditions applied to assess the structural strength in the hydrostatic testing of the urea water storage tank are summarized for each analysis scenario in Table 7 below. The self-weight of the urea water storage tank is 20.6 tons, and only the hydrostatic pressure was considered because the pressure of the portion occupied by air in the tank is equal to the atmospheric pressure if the air pipe performance is normal.

The density of the fluid used in the hydrostatic test was set to the density of freshwater, 999.9 kg/m$^3$. Regarding the hydrostatic pressure acting on the tank, in the 50% water filling scenario based on the tank height, a free water surface height of 4,500 mm was used, and in the 90% water filling scenario, a free water surface height of 8,100 mm was used. For example, in case 6, where the water level is 8,100 mm, pressures of 0 to 0.0794 MPa were applied to the inside of the tank filled with water for each tank height. The hydrostatic pressure was calculated by the gauge pressure, excluding the atmospheric pressure, and the gauge pressure is determined by Eq. (6):

$$P_{gauge} = \rho gh$$  (6)

where $\rho$ is the density of the fluid, $g$ is the gravitational acceleration, and $h$ is the height to the free water surface.
For the top part of the tank, which is not filled with freshwater, whether the gas in the existing tank is compressed or not should be considered. The performance of the air pipe was considered in each case, and the gauge pressure applied to the inside was determined by comparing the water filling level. The additional pressure generated was applied in the normal direction to the wall.

### 4.3 FEA Results and Discussion

The maximum deformation and the position where the maximum deformation occurred in each scenario are shown in Table 8. Safety assessment was performed based on the maximum allowable deflection of the structure to evaluate the risk of deformation that occurred in the structure. In the event of a deformation exceeding the maximum allowable deflection, it was determined that a critical degree of deformation occurred.

Regarding the criterion for the maximum allowable deflection, S/200 was set, where S denotes the span length, as the allowable deflection limit, considering the limit of deflection according to the serviceability limit state of steel structure design (Eurocode 3, 1993) and the limit of deflection according to the serviceability limit state of the offshore structure standard (DNV, 2018). The locations of the

| Case   | Max. deformation (mm) | Location                  |
|--------|-----------------------|---------------------------|
| Case 1 | 3.18                  | Side shell of lower tank  |
| Case 2 | 5.76                  | Side shell of upper tank  |
| Case 3 | 11.89                 | Side shell of upper tank  |
| Case 4 | 6.65                  | Side shell of lower tank  |
| Case 5 | 16.61                 | Tank top                  |
| Case 6 | 37.41                 | Tank top                  |

**Table 8 FEA results of maximum deformation**

*Fig. 3 Deformation analysis result*
members where the maximum deformation occurred are largely divided into three: the side shell of the lower tank, side shell of the upper tank, and the tank top. The maximum allowable deflections at these three locations are 25 mm, 20 mm, and 12 mm, respectively. The maximum deformations and the locations in each analysis case are summarized in Table 8. The analysis results are illustrated in Fig. 3. It was confirmed that dangerous levels of deformation occurred in cases 5 and 6 based on the maximum allowable deflection.

In addition, the damage occurrence possibility mentioned in section 3 was verified through FEA. The maximum additional pressure that can occur on the tank outer wall as a result of the diameter difference between the two pipes determined in section 3 was 0.369 MPa.

When the analysis was performed with the maximum additional pressure of 0.369 MPa at the 50% water filling level, which is a similar environment to the assumption in the damage possibility review, deformation at a dangerous level exceeding the maximum allowable deflection of 40.19 mm occurred on the side shell of the upper tank.

Linear analysis was conducted for a structure that uses structural steel, which is a representative material that has ductility. Then, the structural safety was assessed by comparing the relationship between equivalent stress and allowable stress that occurred on the structure after analysis. The allowable stress of the 3D flat plate structure using shell elements can be expressed as Eq. (7) (KR, 2021):

$$\sigma_e = 0.9 \beta \sigma_y / K$$  \hspace{1cm} (7)

where $\sigma_e$ is the equivalent stress, $\beta$ is the element division density coefficient, $\sigma_y$ is the yield stress, and $K$ is the material coefficient.

The element division density coefficient was calculated according to the guideline for direct strength analysis in the KR Rule Part 3 - Hull Structures of the KR Classification. The allowable stress calculated based on the properties of the mild steel is approximately 258.8 MPa. It was determined that the structural safety is satisfied if the equivalent stress generated at the maximum in the structure in each case for the six load scenarios does not exceed the allowable stress. The results are summarized in Table 9. Here, it should be considered that the stress analysis results for cases 3, 5, and 6 exceeding the yield stress of material are gradually amplified excessively after passing through the elastic region under the influence of the stress-strain curve slope for the linear material. In case 3, the deflection limit was not exceeded from the perspective of deformation, but it became plastic from the perspective of the material’s yield strength, confirming the possibility of permanent deformation.

In this section, linear elastic static analysis was performed for six load scenarios considering the air pipe performance and water filling level using the finite element method (FEM), and the results were analyzed from the perspectives of both deformation and stress. Furthermore, a critical level of deformation was confirmed by performing analysis on the theoretically calculated maximum pressure in the tank, and it was found that the early assumptions that were determined to cause overpressure inside the tank could be verified.

### Table 9 Structural safety assessment result (unit: MPa)

| Case | Equivalent stress result | Allowable stress | Remark |
|------|--------------------------|-----------------|--------|
| Case 1 | 101.7 | 258.8 | O.K. |
| Case 2 | 164.9 | O.K. |
| Case 3 | 294.2 | Not O.K. |
| Case 4 | 208.3 | O.K. |
| Case 5 | 531.4 | Not O.K. |
| Case 6 | 1196.9 | Not O.K. |
5. Theoretical Estimation of Deformation of the Tank Outer Wall

The thin-walled structure has a geometric feature with a beam or flat plate and a shell shape with a thickness smaller than the length. Each side of the tank has a flat plate shape with a very small thickness compared with the length. If the width is divided at regular intervals, it can be treated as a beam shape. Based on this, the tank outer wall can also be treated as a structure with continuous beams.

In this section, the maximum deflection was investigated through general beam theory. Deflection $\delta(x)$ is the displacement from a random dot on the axis of the beam (x-coordinate) in the y-direction. Beam theory and plate theory are commonly used to calculate deflection through integration of the bending moment equation (Ko and Jang, 2017). When performing calculations, the integral constant is expressed using boundary conditions, continuous conditions, and symmetric conditions. The boundary conditions refer to conditions regarding the deflection and slope at support points of the beam or plate.

The differential equation for deflection in beam theory is expressed as Eq. (8) below (Gere and Goodno, 2009):

$$Eh^2 = M, \quad Eh" = V, \quad Eh"" = -q$$

(8)

where $E$ is the elastic modulus of the beam material, $J$ is the second moment of area for the beam section, $M$ is the bending moment, $V$ is the shear force, and $q$ is the distributed load.

These equations are referred to as the bending moment equation, shear force equation, and load equation, respectively. The bending moment equation requires two integral constants because it is a quadratic differential equation. The slope $\nu^\prime$ of the deflection curve is obtained if it is integrated once, and the deflection $\nu$ is obtained if it is integrated twice. The deflection is 0 at the simply supported point, and both the deflection and the inclination become 0 at the fixed end support point. As a result, the deflection in the simply supported condition indicates the largest deflection compared with other boundary conditions. The equation for beam deflection in the simply supported condition at both ends is as follows:

$$\delta_{max} = \frac{5ql^4}{384EI}$$

(9)

where $L$ is the length of the beam.

The hydrostatic pressure associated with water must be applied up to the water level in the tank, and the compressed air pressure must be applied to the tank space above the water level. Therefore, the maximum deflection of the tank according to beam theory can be calculated by applying the uniformly distributed air pressure, which is calculated in each scenario defined in Table 3, to Eq. (9).

The equation for the deflection of the beam with respect to the fixed support condition at both ends, which is the support condition with the smallest maximum deflection, is as follows:

$$\delta_{max} = \frac{ql^4}{8EI}$$

(10)

The maximum deflection of tank in the above two boundary conditions calculated by applying Eqs. (9), (10) is summarized in Table 10 along with the results of the finite element analysis. In the beam theory, the maximum deflection can occur greatly in the simply supported conditions, and the maximum deflection can occur less in the fixed end condition. Except Case 5 and 6, it can be seen that the approximate value of FEA falls within the min-max value obtained through the beam theory. Through this, It was revealed that the deformation result of FEA was a valid approximate value.

Case 5 and 6 appear to be out of the linear elasticity range of the material assumed in general solid mechanics because of the significant deformation. In this case, the value cannot be estimated by calculating the deflection by the beam theory.

6. Summary and Conclusion

In this study, deformation scenarios were assumed considering the cause of overpressure in the hydrostatic test process for deep tank structures. Then, analysis was conducted based on the resulting pressure state, and the structural safety assessment was examined from the perspectives of deformation and stress that occurred in the structure. Furthermore, the theoretical solution was obtained for deformation through mathematical expressions by applying general beam theory to evaluate the validity of the FEA results. Comparing the approximation obtained through FEA and the theoretical solution, the boundary conditions of the actual structure were generally found between the simply supported condition and fixed support condition. The main findings of this study are as follows.

1. The focus was placed on the problem of occasional deformations during the process of hydrostatic testing after installation of a tank. It was assumed that such deformation is caused by overpressure resulting from the air inside the tank, which may occur during a hydrostatic test.
2. It was inferred that the causes of overpressure during a hydrostatic test were low air pipe performance, diameter difference between the inlet pipe and air pipe, and fast water filling rate above the appropriate level.
(3) To confirm that deformation occurs because of the overpressure of the internal air during hydrostatic testing of the tank, the difference in diameter between the inlet pipe and the air pipe was considered using the Bernoulli equilibrium equation in terms of flow rate. Using this example, the possibility of a pressure rise was mathematically verified.

(4) Six deformation analysis scenarios were established using the water filling level and the air pipe performance as the variables, and FEA was conducted for the urea water storage tank. The analysis showed a large deformation of up to 37 mm.

(5) As a result of FEA, the stress results exceeded the allowable stress of the material in the deformation analysis scenarios of cases 3, 5, and 6. In these cases, a fracture of the structure can occur. In case 3, the deflection limit was not exceeded in terms of deformation. However, it was confirmed that permanent deformation can occur as a result of plasticity in terms of the yield strength of the material.

(6) The maximum deflection that can occur was calculated using beam theory. It was demonstrated that overpressure may actually be a cause of deformation by verifying that the approximation of the deformation obtained through FEA is included in the resulting solutions of the maximum deflection calculation.

This study demonstrated that overpressure resulting from the compression of internal air during a hydrostatic test can cause deformation. However, analysis was conducted by applying the resulting pressure in a specific state as a loading condition, without considering the variable for time when assuming the deformation scenarios. In future studies, it will be necessary to additionally examine the occurrence and accumulation of deformation by conducting transient analysis using the water filling rate and the diameter difference between the inlet pipe and air pipe as variables.

Conflict of Interest

Tak-Kee Lee serves as an editor of Journal of Ocean Engineering and Technology, but has no role in the decision to publish this article. No potential conflict of interest relevant to this article was reported.

References

American Petroleum Institute (API). (2020). Welded Tanks For Oil Storage. Retrieved August 2021 from https://mycommittees.api.org/standards/cre/scast/Documents/Std%20650/650_13th%20Edition.pdf

Bay Tank and Vessel. (2016). Hydro Testing Procedure. Retrieved August 2021 from https://docs2.cer-rec.gc.ca/l/eng/lisapi.dll/fech/2000/90464/90552/534348/2837345/2857911/3274807/A83910-14_Attachment_6.6_Tank_Hydrostatic_Test_Procedure_A5Q4Q5.pdf?nodeid=3275253&vernum=2

Cameron, K., & Pettinger, A.M. (2010). Effectiveness of Hydrostatic Testing for High Strength Pipe Material. Proceedings of the 2010 8th International Pipeline Conference, Calgary, Alberta, Canada, 647-651. https://doi.org/10.1115/IPC2010-31426

Cengel, Y.A., & Cimbala, J. (2014). Fluid Mechanics Fundamentals and Applications (3rd ed.). Seoul, Korea: Kyobo Publisher.

Cho, J.R. (2019). Theory and Applications of Finite Element Method (1st ed.). Paju, Korea: Donghwa Publisher.

Det Norske Veritas (DNV). (2018). Design of Offshore Steel Structures (DNVGL-OS-C101). Retrieved from https://rules.dnv.com/docs/pdf/DNV/OS/2018-07/DNVGL-OS-C101.pdf

Det Norske Veritas (DNV). (2021). Rules For Classification : Ships (RU-SHIP). Retrieved August 2021 from https://www.dnv.com/news/rules-for-classification-of-ships-july-2021-edition-203529

Dey, A.K. (2017). 5 Examples of Hazards of Pressure Testing. Retrieved from https://whatispiping.com/hazards-of-pressure-testing/

DNDE. (2021). Ansys Mechanical Basic. 221-266.

Eurocode 3. (1993). Design of Steel Structures - Part 1-1: General Rules and Rules for Buildings. Retrieved from https://www.phd.eng.br/wp-content/uploads/2015/12/en.1993.1.1.2005.pdf

Gere, J.M., & Goodno, B.J. (2009). Mechanics of Materials (7th ed.). Seoul: Cengage Learning.

International Association of Classification Societies (IACS). (2021). Common Structural Rules for Bulk Carriers and Oil Tankers. Retrieved from https://www.iacs.org.uk/publications/common-structural-rules/csr-for-bulk-carriers-and-oil-tankers/

Kiefner, J.F., & Maxey, W.A. (2000). The Benefits and Limitations of Hydrostatic Testing. In API’s 51st Annual Pipeline Conference & Cybernetics Symposium, New Orleans Louisiana.

Kiefner, J.F. (2001). Role of Hydrostatic Testing in Pipeline Integrity Assessment. In Northeast Pipeline Integrity Workshop, Albany, New York.

Ko, D.E., & Jang B.S. (2017). Ship and Offshore Structural Mechanics (1st ed.). Paju, Korea: Textbooks Publisher.

Korean Register (KR). (2021). Part 3 - Hull Structures.

Krieg, M., Nestleroth, J.B., Henning, T., & Haines, H. (2018). In-Line Inspection In Lieu of Hydrostatic Testing for Low Frequency Electric Resistance Welded Pipe. Proceedings of the 2018 12th International Pipeline Conference, Alberta Canada.

Lee, S.H. (2013). Three Months before the Samsung’s Water Tank Disaster, there was a ‘Preview’. Retrieved from http://www.yna.co.kr/view/AKR20130806141500057

Liao, B.B., Wang, D.L., Hamdi, M., Zheng, J.Y., Jiang, P., Gu, C.H., & Hong, W.R. (2019). Acoustic Emission-based Damage Characterization of 70 MPa Type IV Hydrogen Composite Pressure Vessels During Hydraulic Tests. International Journal of Hydrogen Energy, 44(40), 22494-22506. https://doi.org/10.1016/j.ijhydene.2019.02.217

Ministry of Oceans and Fisheries (MOF). (2020). Ship Safety Act. Additional Shipbuilding Inspection, Article 11, Clause 2.
Parthiban, K.K. (2012). An Accident with Brittle Fracture during Hydrotest. Retrieved from http://www.venus-boiler.com/technical_papers.php

Stephen, C.R., Dimitris, D., Vassilis, G., William, L., and Rosenfeld, M.J. (2010). Analysis of Pipe Expansion Associated with Field Hydrostatic Testing. Proceedings of the 8th International Pipeline Conference, Alberta Canada. 179-185. https://doi.org/10.1115/IPC2010-31666

Zhang, Z., Feng, H., Zhao, W., Li, M., Xu, X., & Li, J. (2021). The Technical Scheme Optimization of Nuclear Vessel Hydrostatic Test. International Conference on Environment Science and Advanced Energy Technologies, 709, 042051. https://doi.org/10.1088/1755-1315/769/4/042051

**Author ORCIDs**

| Author name   | ORCID           |
|---------------|-----------------|
| Kim, Geun-Gon  | 0000-0001-9617-1450 |
| An, Tae-Hyun   | 0000-0002-7799-1872 |
| Lee, Tak-Kee   | 0000-0002-5944-156X |