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Experimental and numerical investigation of stress in a large-scale steel tank with a floating roof

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ABSTRACT

Research on steel tanks, particularly large-scale oil storage tanks, has increased significantly with the rapid development of the petrochemical industry. The focus on these thin-walled structures is related not only to the cost of the infrastructure, but also because failures due to accidents may generate huge economic costs, environmental losses, and casualties. The aim of the present study is an extensive experimental and numerical investigation of stress in a large-scale oil tank with a floating roof. Stresses in the shell and bottom of a large-scale oil storage tank are measured by a resistance strain gauge technique. Meanwhile, an improved 3-D finite element model is developed to analyze stresses in the tank. By comparing calculated results with experimental data, the validity of the developed finite element model is verified. Both experimental and numerical results suggest that the stress on the fillet joint is relatively high in the tank. The finite element model is used to study the effects of three key steel tank parameters on the stress of the fillet joint in the oil storage tank: the thickness of the annular bottom plate at the base, the radial width of the projection of the annular bottom plate outside the shell, and the radial width of the concrete ringwall. In addition, a new method related to the design of the annular bottom plate and concrete ringwall is proposed. This method will be helpful in improving the design of large-scale oil tanks and assessing their structural integrity.

1. Introduction

Large unanchored open-topped steel tanks are commonly used to store large volumes of hazardous fluid in the petroleum and chemical industry, because they are ideal for cost-effective and convenient aboveground oil storage [1]. However, because of their geometric slenderness and operating characteristics, these cylinders (which are welded together by curved steel sheets), are the most vulnerable type of petrochemical equipment. Large-scale oil tanks are prone to failure caused by high stress or large deformations, which may generate severe accidents and bring about considerable economic and environmental losses and casualties. For example, shell buckling is a type of damage resulting from the interaction of both circumferential tensile stress and axial compressive stress [2–6]. In addition, cracks can occur at the welding edges of the fillet joint between the bottom plate and tank shell, caused by the excessive bending stress attributed to the uneven settlement of the foundation, or cyclic stress changes attributed to frequent undersurface oil charging and discharging [7,8].

Large unanchored oil storage tanks have been built with vertical cylindrical shells and sloped bottoms that often rest directly on simply prepared subgrades and are supported by a concrete ringwall beneath the tank shells [9]. The way in which the bottom plate and tank shell come together is significant with respect to the design of the bottom annular plate and concrete ringwall. Three parameters for the annular plate are provided according to tank design standard API 650 [10]. First, annular bottom plates shall have a radial width that provides at least 600 mm between the inside of the shell and any lap-welded joint in the remainder of the bottom. Second, the minimum width that projects either beyond the outside edge of the weld attached the bottom to the shell plate or outside the shell plates is 50 mm. The third parameter is the minimum thickness, which is determined by the nominal plate thickness of the bottom shell course and is also related to the hydrostatic test stress of the bottom shell course. The one-foot method or variable-design-point method can be used to calculate the thickness of the shell plate. Nevertheless, the bottom annular plate is normally designed empirically, because there is no recognized formula or model to calculate the stresses on the bottom plate and determine the three parameters mentioned above.

Generally, three methods can be used to determine stress in a structure. These are analytical methods, experimental methods, and

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Fig. 1. Selected unanchored open-top oil tank with stepped walls used in the test.

Table 1
Structural parameters for a 100,000 m^3 tank.

Name	Thickness/mm	Height/mm	Material
Bottom sketch plate	12	–	Q235B
Bottom annular plate	20	1980 (width)	12MnNiVR
1st shell course	32	2420	12MnNiVR
2nd shell course	27	2420	12MnNiVR
3rd shell course	21.5	2420	12MnNiVR
4th shell course	18.5	2420	12MnNiVR
5th shell course, with stiffening ring	15	2420	12MnNiVR
6th shell course, with stiffening ring	12	2420	12MnNiVR
7th shell course, with stiffening ring	12	2420	12MnNiVR
8th shell course, with wind girder	12	2380	16MnR
9th shell course, with wind girder	12	2380	Q235B

numerical modelling methods.

Three analytical methods are available for calculating the lower nodal point of a large oil tank: the rigid foundation beam method by API (Denham) [11], the Li method from the Chinese Academy of Sciences [12], and the Wu method from the China University of Petroleum [13]. Compared to conservative analytical methods, numerical methods have been widely recognized as accurate, because the rapid evolution of computer technology has facilitated the handling of complex problems [14–16]. Chen et al. [17] carried out a finite element analysis of liquid storage tank foundations using settlement differences as a boundary condition, from which the stresses on the bottom plate were obtained. Kim et al. [18] performed a failure analysis of fillet joint cracking in an oil storage tank using chemical analysis and stress corrosion tests based on slow strain rate tests and mechanical tests. The researchers indicated that although the cracks were initiated by corrosion, failure was caused by the propagation of cracks caused by stress concentrations.

In summary, stress analysis of thin walled short cantilever shells is very important for evaluating the safety of oil tanks. Particularly for large unanchored crude oil tanks with floating roofs, this stress analysis plays a significant role in the structural damage analysis and design of tanks. Previous research on finite element tank models has focused on axisymmetric geometric models involving uniform tank shells, anchored tanks, settlement of the tank shell or bottom (but not the foundation), and isotropic ideal elastic-plastic materials, for which hardening is not considered [19–25]. In reality, large oil tanks have non-axisymmetric structures, because of both the different numbers and distributions of supports for wind girders and the stiffening ring that is welded to the outside of the wall around the circumference. These are applied to improve the ability of structures to maintain integrity. Thus, it is necessary to develop an improved model for investigating stresses in oil tanks accurately. Furthermore, there is limited research relating to stress testing of the shells and bottoms of large crude oil storage tanks.

This study aims to conduct a stress analysis of a large-scale oil tank with a floating roof through experiments and numerical modelling. The resistance strain gauge technique is adopted for the stress test. Various tank geometry features are thoroughly taken into account in the non-symmetrical, full finite element model: tank foundations with concrete ringwalls, unanchored sloping bottoms, annular plates and sketch plates, tapered shell thicknesses, wind girders and supports, top angles, stiffening rings, and support plates. From the observation and analysis of experimental and numerical modelling results, the stress distributions and properties of tank shells and bottoms are revealed, and a new method associated with stresses on the shell-to-bottom connection is proposed. Then, recommendations for the design of the annular bottom plate and concrete ringwall are given.

2. Experimental procedure

2.1. Strain measurement

Using a resistance strain measurement method, a large oil storage tank with a 629,000 barrels capacity was hydrotested in China's Huangdao oil depot, which belongs to SINOPEC Engineering Incorporation. The unanchored open-top oil tank with stepped walls was designed according to API650, as shown in Fig. 1. The tank geometry is presented in Table 1.

As shown in Fig. 2, to investigate stresses in different zones of the tank, experiments were performed on two important tank components: 1) the tank shell under the stiffening ring and 2) the tank bottom adjacent to the fillet joint. It is necessary to clarify that the two independent measurement profiles for the tank shell and tank bottom shown in Fig. 2 were chosen to guarantee the validity of the test data. Both cross sections are 60 m apart in the circumferential direction.

Strain was measured in one direction with an ordinary waterproof gauge (WFLA-6–11-3LT) attached to the outside surface of the tank shell, as well as a professional waterproof gauge (T J120-4AA) attached to the bottom of the tank (in consideration of hydraulic pressure), as shown in Fig. 3. There were 120 strain measuring points and 240 strain gauges in total. At each point, the strain gauges were fixed using an adhesion agent along two principal stress directions. To reduce test tolerance, mechanical prevention was carried out with silastic and adhesives such as AZ 709, Three Bond 1521B, and SB silastic. The static resistance strain indicators and laptop computers were placed in a container near the oil tank, as shown in Fig. 4. The four strain instruments were manipulated by four laptops. It is important that the test gear reads zero when testing and inspecting the equipment. Strain test data were collected by the computers every morning and night until the tank was fully filled with water.

2.2. Summary of experimental result

Stress can be directly calculated using Hooke's Law after setting the Young's modulus and Poisson ratio. The stresses for the tank shell in the longitudinal and circumferential direction are shown in Fig. 5. The absolute values of the axial and circumferential stresses increase as the water level increases. Near the shell-to-bottom fillet weld, the tank shell is compressed in the axial direction, and the maximal compressive stress can reach −208.16 MPa at 0.06 m above the top surface of the tank bottom plate, as shown in Fig. 5(a). The axial stress rapidly shifts from a negative value to a positive value as the measuring point rises, then fluctuates in a relatively narrow band around zero because of the stepped shell courses.

The circumferential stress of the tank shell is negative near the shell-to-bottom fillet weld because of the constraint of the tank bottom and