Design of Detonation Chamber for Destructing Chemical Warfare Materials

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Abstract

A detonation chamber is a device for destroying chemical warfare materials by detonation, using tens of kilograms of explosives in the process. Special structural features are required to satisfy fragment resistance, operability and leak tightness. The normal design code for static pressure vessels cannot be applied to the basic structural design of the robust detonation chamber because the detonation shock wave causes instantaneous dynamic pressure. Hence, the Code Case "Impulsively loaded pressure vessels," published recently by the American Society of Mechanical Engineers (ASME), was used as the design guideline. It requires dynamic pressure analysis and dynamic stress and strain analysis, which are used for the evaluation of each mode of failure such as fatigue damage and local strain limit, to allow detailed design. This paper introduces the design features with examples.

Introduction

A detonation chamber is a device inside which a chemical weapon is detonated using an explosive of several tens of kilograms. There are structural limitations in order to satisfy application-specific performance such as fragment resistance, operability, and leak tightness. The normal design code for static pressure cannot be applied as-is to the structural design for robustness. This is because the shock wave that occurs during detonation causes an impulsive load.

Recently, special codes for impulsive load, ASME Section VIII Division 3[1)] (hereinafter referred to as "ASME") and Code Case 2564[2)] (hereinafter referred to as "Code Case"), have been published, and these are now used as design guidelines. Kobe Steel participates in the Task Group of these ASME special codes and cooperates in providing information, etc. These special codes provide techniques for performing dynamic pressure analysis and dynamic stress/strain analysis to evaluate each fracture mode, including fatigue damage and local strain limitation. Although the design procedure and evaluation criteria are complicated and stringent, the application of these codes has made it possible to design safer detonation chambers and to evaluate their safety.

This paper introduces the structural features of detonation chambers, their design technique, and examples of their design and evaluation.

1. Structure of detonation chamber

Detonation chambers are basically steel pressure vessels, and **Fig. 1** shows a model diagram of the basic structure. For detonation destruction of chemical munitions, it has structural features with the functions of fragment resistance, operability, leak tightness, and robustness to withstand detonation impact. These features are outlined below.

1.1 Fragment resistance

If the chemical weapon to be detonation-destructed is a piece of ammunition, the fragments of the ammunition shell will collide with the inner surface of the detonation chamber at high speed during the detonation and cause damage to the surface. Robustness decreases when the pressure-resistant part intended to withstand the detonation

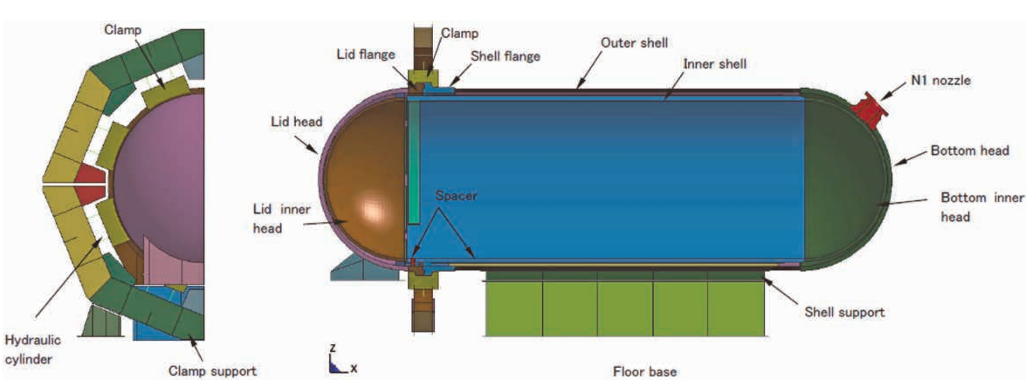

Fig. 1 Structural model of detonation chamber

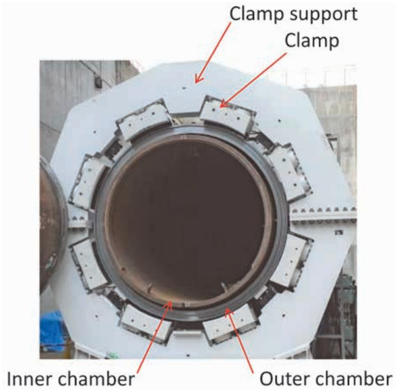

Fig. 2 Inside view of detonation chamber

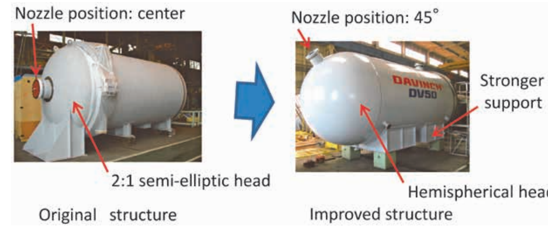

Fig. 3 Example of structural improvement

shock wave is thinned by damage. As shown in **Fig. 2**, the body and heads have double wall structures, each consisting of an outer shell and inner shell, in which the outer shell is responsible for the sealing performance of the pressure vessel, and the inner shell is responsible for fragment resistance. A gap is provided between the inner shell body and the inner shell lid head so that the inner shell does not become a closed vessel, and the shock wave pressure leaks from the inner shell to the inner surface of the outer shell.

The inner shell is made of high-strength steel of 800 MPa grade (JIS SHY685) with high fragment resistance. In case of damage, the inner shell alone can be pulled out from the outer shell and replaced.

1.2 Operability

It is necessary to carry in and place a chemical weapon inside the detonation chamber and collect the fragments after detonation. For this reason, the vessel is made horizontal, and the lids are hydraulically driven so that they can be fully opened. Fig. 2 shows the open state. For detonation destruction, the operator places each chemical weapon at the center of the axis of the detonation chamber. Therefore, the diameter is kept to 3m or less from the viewpoint of operability.

The horizontal vessel is subjected to a large, horizontal, impulsive load on the mount support during detonation. In early days, large strains and cracks occurred for this reason. In order to solve these problems, the support parts have been reviewed and improved in detail, including the size of the ribs, their combinations, and the welding method, on the basis of analysis and actual operation data. **Fig. 3** shows an example of the improved mount support structure.

The pressure caused by the shock wave is inversely proportional to the cube of the propagation distance. Therefore, in the case of a cylindrical vessel with a given capacity, the stress on the cylindrical body can be reduced by increasing the inner diameter and shortening the overall length. The above-mentioned operability problem, however, limits the diameter of the horizontal cylinder. Therefore, when increasing the processing capacity of explosives, the axial length of the cylinder is extended.

1.3 Leak tightness

The allowable concentration of the chemicals released from chemical weapons into the treatment environment is low, and the movable lids must have high leak tightness so that gas leakage during detonation can be prevented. In the original structure, the lid was fixed by a pair of split rings fastened by bolts, however, with the increase in explosive load, the structure was changed to a hydraulic clamping type with higher leak tightness (Fig. 2). The lid-side and body-side flanges are hydraulically tightened with eight clamps to maintain sealing performance between the two flanges. Their deformation behaviors have also been confirmed by dynamic analysis, and various mechanical improvements have been implemented, such as a fitting structure with a spigot joint on each flange.

1.4 Structural robustness against detonation shock waves

The greatest issue in the evaluation of structural robustness is brittle fracture due to explosion shock waves. To prevent this, the body has a multi-layered structure, in which thin plates are laminated to form a thick body structure as shown in **Fig. 4**. In this multi-layered structure, even if a crack occurs in the innermost layer (first layer), which receives the shock wave, the crack does not propagate into the second and outer layers due to discontinuity. Consequently, it avoids the occurrence of a brittle fracture that abruptly penetrates the entire thickness. In the event that a crack occurs in the first layer, as shown in the upper left of Fig. 4, the existence of the crack in the first layer can be confirmed

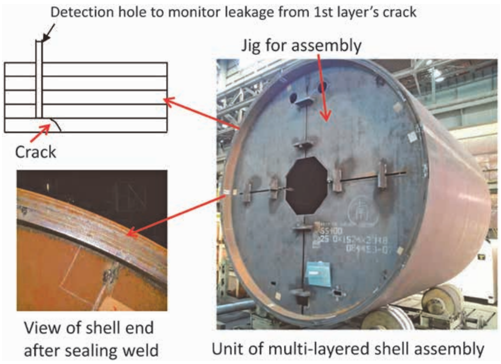

Fig. 4 Multi-layered outer shell

by detecting gas leakage from a detection hole. Meanwhile, the heads, nozzles, flanges, etc. have single-walled structures made of 3.5% Ni steel with high toughness, such as SA203Gr.E or SA350LF3, which is focused on brittle-fracture resistance rather than tensile strength.

The inner shell body is installed on a spacer with its axis aligned with that of the outer shell body. The inner shell is not fixed to the outer shell so that it can easily be replaced, as mentioned above. Therefore, the inner shell moves in the axial and vertical directions due to the impact energy of detonation and collides with the outer shell. Thus, the impulsive load at the time of collision between the inner shell and the outer shell must be studied. Especially at the localized collision point, the stress generated in the outer shell is elevated. To alleviate this, measures have been taken such as changing the shape of the head from 2: 1 semi-elliptical to a hemispherical shape and changing the nozzle position 45° upward from the center of the head (original nozzle position), as shown in Fig. 3.

2. Structural design

When designing the thickness of an impulsively loaded pressure vessel, the simplest calculation method is to use the formula for a static pressure vessel and set a large safety factor (reduce the allowable stress) in consideration of impulsive load. Setting this safety factor, however, requires many actual values, which makes an appropriate setting difficult. In addition, the detonation chamber-specific structure itself, as mentioned above, plays a critical part in robustness, making the above formula inapplicable. Hence, Kobe Steel has established its own method of designing detonation chambers and evaluating their safety using the ASME special codes as design guidelines.

This section outlines this method of design and safety evaluation, while introducing an example of design and evaluation aiming at the largest class of detonation chamber to date, with a designed explosive load exceeding a 60 kg TNT equivalent.

2.1 Design Input

In the design of detonation chambers, the concept of "design pressure" as in the usual design code is inapplicable, and the detailed structure of each detonation chamber, the material used, and the explosive load (TNT equivalent) are used as design inputs. In addition, the detonation method, including the detonation position of the explosive and the detonation time difference, is also used as design inputs. Even with a given explosive load, the distribution of shock waves differs if detonated at a single position or divided into three positions. In addition, since the mechanical properties of the material used change in accordance with the strain rate under impulsive load, stress-strain curves based on high-speed tensile test results are used.

2.2 Explosion analysis

Fig. 5 shows the change with time of pressure distribution at the time of explosion, analyzed using the impact analysis software AUTODYN®. Note 1) This is an example of simultaneous detonation at three points, in which the inner diameter of the outer shell is 3.0 m, explosive load is 64.2 kg TNT (hereinafter, all the calculation examples in this paper use these conditions) and shows that the pressure wave repeats reflective diffusion with the passage of time.

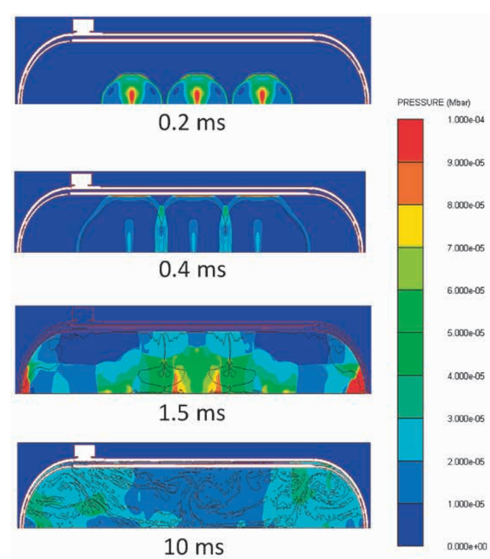

Fig. 5 Change with time of pressure distribution

Note 1) Analysis software based on explicit method designed for a wide range of impact problem analyses such as detonation. Provided by ANSYS1®.

The fluctuation status of the pressure distribution changes depending on detonation conditions, such as the detonation position and detonation time of the explosive. It should be noted that a static pressure vessel has a uniform pressure distribution in the vessel, whereas in a chamber that is subjected to a detonation shock wave, the received pressure differs depending on each position and time. Therefore, it is necessary to find the time history of pressure for each location. **Fig. 6** shows an example of the pressure time history of the inner surface of an outer shell head. The pressure does not decay monotonically but shows several peaks. This indicates that each shock wave repeats reflective diffusion to exert an impact force on the location.

2.3 Dynamic structure analysis

Dynamic stress and strain distributions are calculated by impact structure analysis software LS-DYNA®Note 2) on the basis of the time history of pressure at each position. Similar to pressure, stress and strain distribution that occur in the outer shell repeatedly show peaks. This also takes into account the impulsive load of the inner shell collision. Example analysis results will be introduced individually for each of the following evaluation items.

2.4 Evaluation based on ASME and Code Case

This section describes the basic concept of the evaluation method and evaluation examples for the four types of fracture modes.

2.4.1 Leak before burst mode of failure (LBBM)

The LBBM is specified in ASME KD-141 and indicates that no crack propagates at a high enough speed to cause catastrophic fractures. The multi-layered part of the body is structurally recognized as LBBM by ASME KD-810, while the single-walled part is judged to be LBBM if both of the following two conditions (I) and (II) are met.

(I) Assuming that a crack has reached 80% of the total thickness (t) of a material, the stress intensity factor, K_I, at the crack tip shall be smaller than the fracture toughness value, K_{Ic}, of the material.

(Ⅱ) The remaining ligament (distance from the crack tip to the free surface) shall be smaller than $(K_{Ic}/\sigma_{ys})^2$.

The following describes the detailed evaluation method. The crack shape is assumed to be semi-elliptical as shown in ASME KD-410, and the ratio of depth (a) to length ($2c$) to be 1:3. The K_I value at the crack tip is expressed by the following equation.[3)]

$$K_I = [\sigma_0 G_0 + \sigma_1 G_1 (a/t) + \sigma_2 G_2 (a/t)^2 + \sigma_3 G_3 (a/t)^3 + \sigma_4 G_4 (a/t)^4] \sqrt{\pi (a/Q)} \quad \cdots\cdots (1)$$

$$Q = 1 + 1.464 (a/c)^{1.65} \quad \cdots\cdots (2)$$

$$\sigma(x) = \sigma_0 + \sigma_1 (x/t) \quad \cdots\cdots (3)$$

$\sigma(x)$ is calculated by Eq. (3) on the basis of the instantaneous maximum stress:

wherein,

σ_0 to σ_4: coefficients when the stress $\sigma(x)$

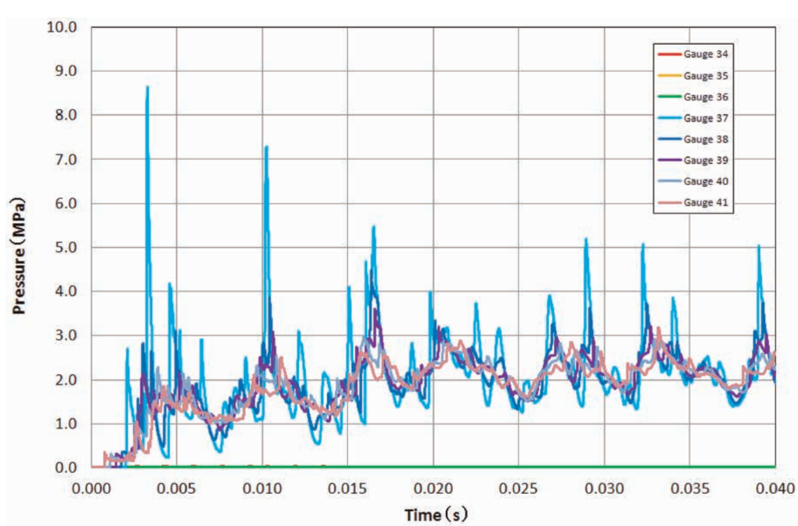

Fig. 6 Time history of pressure of inside surface of outer lid head

Note 2) High-speed nonlinear dynamic analysis software based on explicit method provided by Lawrence Software Technology Corporation.

orthogonal to the crack is expressed by a fourth-order polynomial

t: plate thickness,

x: distance (0 to t) from the edge,

Q: crack shape coefficient,

G_0 to G_4: influence coefficient, and

σ_{ys}: static yield strength.

K_{Ic} is calculated using the equation below.

$$(K_{Ic}/\sigma_{ys})^2=5(CVN/\sigma_{ys}-0.05) \cdots\cdots (4)$$

wherein,

CVN: the absorbed energy value obtained by the Charpy impact test in relevance with the toughness value of the material.

Next, an evaluation example of the chamber, our subject this time, is shown. Regarding (I), K_I =107 MPa m$^{1/2}$, and K_{Ic} = 225 MPa m$^{1/2}$ which satisfies $K_I < K_{Ic}$. Regarding (II), the remaining ligament is 20% of the wall thickness of 75 mm, which is 15 mm, so $(K_{Ic}/\sigma_{ys})^2$ = 670 mm, which satisfies remaining ligament $<(K_{Ic}/\sigma_{ys})^2$. Therefore, since both conditions (I) and (II) are satisfied, it is judged to be LBBM.

2.4.2 Global plastic instability

The Code Case requires dynamic elastic plastic analysis to show that an impulsive load will not form a plastic instability state. For example, there shall be no complete plastic hinge formation around the opening. In the case of a detonation chamber, the exhaust gas nozzle part of the body-side head and the flanges on the lid side and body side are evaluated. The design margin of the impulsive load is specified as 1.732. Specifically, analysis evaluation is performed using a load of 175% of impulsive load (pressure), which corresponds to the designed explosive load.

Fig. 7 shows an example of the distribution of effective equivalent plastic strain viewed from the lid side of the body side flange. Although an effective equivalent plastic strain is observed on the flange member, an elastic region is confirmed to exist over the entire cross section. That is, no complete plastic hinge is confirmed to exist on the body side flange and no plastic instability state is judged to exist.

2.4.3 Local strain limit

The Code Case stipulates the following three conditions for the local plastic strain limit.

(1) The acceptable value of the maximum equivalent plastic strain shall be 0.2% on average for the thickness of the vessel.

(2) The acceptable value of the maximum plastic strain component, linearized through the thickness of the vessel, shall be 2% (1% at welds).

(3) The acceptable value of the maximum peak equivalent plastic strain at any point in the vessel shall be 5% (2.5% at welds).

These are the criteria for a one-time detonation, and the cumulative residual plastic strain caused by multiple detonations must be evaluated on the basis of the same criteria. It should be noted, however, that a shakedown has been experimentally confirmed in the high strain region even in the case of impulsive load deformation caused by detonation,[4)] and the so-called ratcheting phenomenon, in which the cumulative residual plastic strain increases with each detonation, does not occur. Thus, the evaluation should be performed on the basis of the amount of a single residual plastic strain caused by the maximum amount of detonation.

Table 1 shows an example of the analysis results of the maximum peak plastic strain component and the equivalent plastic strain at the maximum peak under the above condition (3) in the lid head in this case example. The example result is 50,000 (25,000 at welds) μ or less, which satisfies the criterion of condition (3). Although the calculated value is omitted, conditions (1) and (2) are also satisfied and it is judged to fall within the criteria.

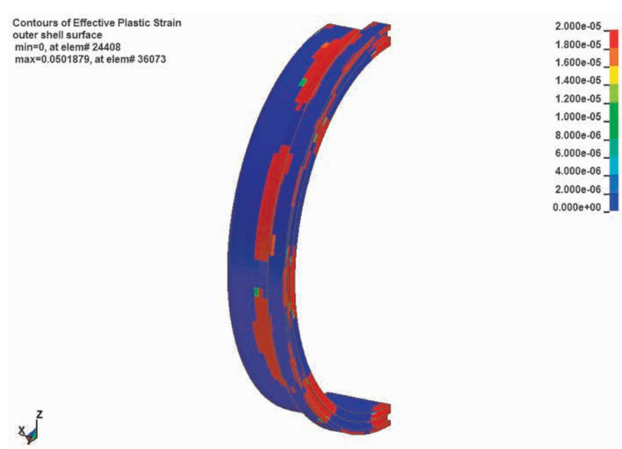

Fig. 7 Contours of effective equivalent plastic strain in shell flange (175% load)

Table 1 Maximum peak equivalent plastic strain of lid head

Plastic strain component						Equivalent plastic strain	Acceptable range
$\varepsilon_{xx,p}$	$\varepsilon_{yy,p}$	$\varepsilon_{zz,p}$	$\varepsilon_{xy,p}$	$\varepsilon_{yz,p}$	$\varepsilon_{zx,p}$	$\varepsilon_{eq,p}$	
-589	75	210	23	1	521	792	<50,000 <25,000 weld

unit ; μ

2.4.4 Fatigue fracture evaluation

When the LBBM is proved, the fatigue fracture evaluation permits the use of ASME KD-3 to obtain the number of cycles until a crack occurs. If the LBBM is not proved, fracture mechanical evaluation must be performed on the basis of ASME KD-4, which evaluates the propagation of cracks that have occurred.

As mentioned above, the latest model of the detonation chamber uses a material with high fracture toughness and the LBBM has been proved, so evaluation based on ASME KD-3 has been carried out. The following describes the outline.

(1) Strain amplitude

From the time history of strain amplitude, the equivalent strain amplitude is calculated for each strain peak group for each time via the following equation using the rainflow counting algorithm from ASME KD-353.

$$ {}^{mn}\Delta\varepsilon_{ij} = {}^{m}\varepsilon_{ij} - {}^{n}\varepsilon_{ij} \quad \cdots\cdots \quad (5) $$

$$ {}^{mn}\Delta\varepsilon_{range} = \sqrt{2}\sqrt{\left\{\left({}^{mn}\Delta\varepsilon_{11}-{}^{mn}\Delta\varepsilon_{22}\right)^2+\left({}^{mn}\Delta\varepsilon_{22}-{}^{mn}\Delta\varepsilon_{33}\right)^2+\left({}^{mn}\Delta\varepsilon_{33}-{}^{mn}\Delta\varepsilon_{11}\right)^2+6\left({}^{mn}\Delta\varepsilon_{12}^2+{}^{mn}\Delta\varepsilon_{23}^2+{}^{mn}\Delta\varepsilon_{31}^2\right)\right\}/3} \quad \cdots\cdots \quad (6) $$

wherein,

${}^{m}\varepsilon_{ij}$: strain component at time m

${}^{mn}\Delta\varepsilon_{range}$: equivalent strain amplitude

(2) Fatigue strength reduction factor

Because the fatigue strength of local shape discontinuities, such as the base of the nozzle, is decreased, the strain amplitude is multiplied by a fatigue strength reduction factor for the evaluation. The fatigue strength reduction factor is basically a factor related to the stress concentration coefficient calculated from the shape.

(3) Fatigue curve

Being a low cycle fatigue due to plastic strain, it is evaluated using a strain-life curve instead of a stress-life curve. The evaluation is performed on the curve shown in **Fig. 8** with a margin of three times the life of the best fit curve estimated from the ASME design fatigue curve. The value obtained by multiplying the strain amplitude at each time by the fatigue strength reduction factor and the corresponding lifetime N_i on the fatigue curve are obtained, and the fatigue damage level $1/N_i$ of one strain peak group is calculated. Next, the total $\Sigma 1/N_i$ is calculated for all strain peak groups generated by one detonation. This yields the total fatigue damage level from one detonation.

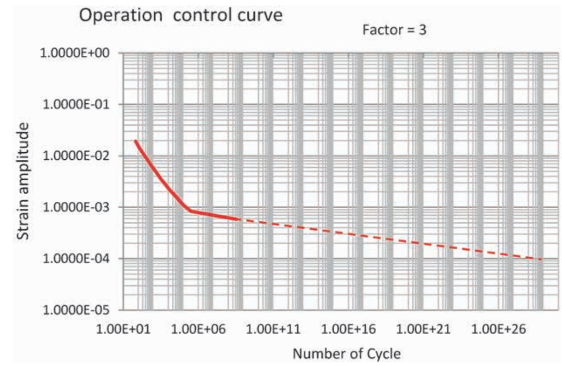

Fig. 8 Fatigue curve

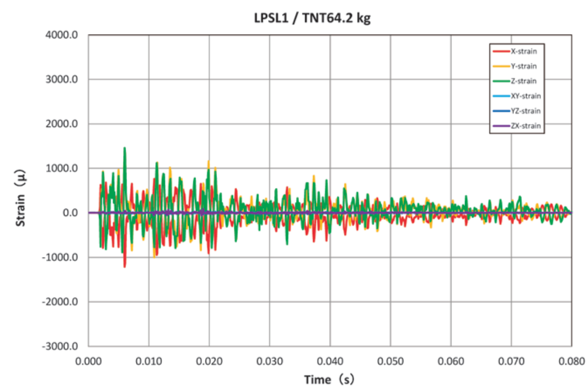

Fig. 9 Time history of dynamic strain of top of lid head

(4) Fatigue damage level during multiple detonations

For example, if the planned number of detonation cycles is n and the explosive conditions, such as explosive load, are the same for all detonations, the cumulative fatigue damage level is given by $n \times \Sigma 1/N_i$. If the cumulative fatigue damage level reaches 1, it is judged to have reached the end of life, so the tolerable detonation cycle in the design calculation is given by $n = 1/(\Sigma 1/N_i)$.

Fig. 9 shows the dynamic strain time history of this example (lid head part). The fatigue damage level per detonation cycle, obtained by the above procedure, is 7.8E-05, and the tolerated number of detonation cycles is 12,800 cycles. In this example, the tolerated number of detonation cycles by design is approximately 1,700 cycles, and the design criteria is satisfied.

In the case of this example, all four fracture modes satisfy the criteria, and safety is guaranteed by the design.

3. Design verification and operation management

ASME KD-12 stipulates design verification by testing, and design verification has been performed on detonation chambers by actually measuring the strain in detonation testing. There is also a technique called fracture testing, however, in the

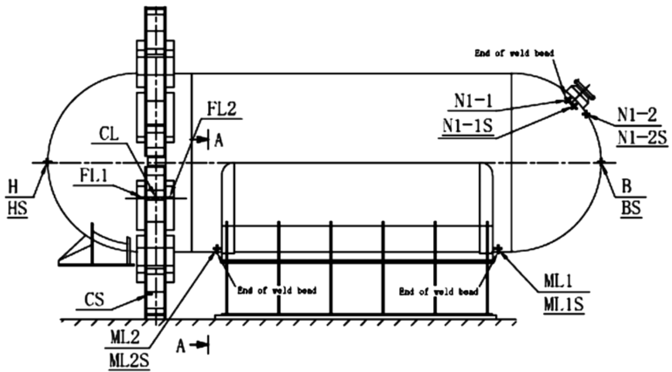

Fig.10 Positions of strain gauge

case of detonation chambers, the strain generated by the designed explosive load is measured. The critical points shown in **Fig.10** are selected as the measurement positions. The static strain gauge confirms the cumulative residual plastic strain (Section 2.4.3), and the dynamic strain gauge confirms the fatigue damage from each detonation (Section 2.4.4). The cumulative calculation of fatigue damage can be performed during the test using a proprietary calculation program.

In addition, strain is measured during actual operation, and cumulative residual plastic strain and cumulative fatigue damage are calculated for each detonation, which is used for maintenance by predicting repairs and replacement timings.

Conclusions

More than ten years have passed since a detonation chamber was introduced into a real-world chemical weapons destruction project. During that period, structural design techniques have been established on the basis of the ASME's special design codes for impulsively loaded pressure vessels. Kobe Steel will strive to further improve the design and reflect it in maintenance, improvement, and development for further enhancement of its safety.

Finally, we would like to express our sincere gratitude to the late Dr. Robert E Nickell, the former president of ASME, for his cooperation and guidance in design.

References

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2) Code Case 2564-4. Impulsively Loaded Pressure Vessels, Section VIII Division 3.
3) WRC Bulletin 471 May. 2002.
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