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THE PREDICTION OF PLASTIC DAMAGE DEGREE OF STIFFENED CYLINDRICAL SHELL CONSIDERING OF HYDROSTATIC PRESSURE

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Abstract. In underwater explosion, local representative compartment of submarine is simplified as a perfectly rigid-plastic stiffened cylindrical shell fixed on both ends. On the basis of the principle of momentum conservation, and ignoring the axial force of cylindrical shell, a new and fast theoretical calculation method for the plastic deformation in the typical location of stiffened cylindrical shell has been obtained, taking account of the hydrostatic pressure. At last, the dynamic plastic response of stiffened cylindrical shell under the load which is the combination of underwater explosion and hydrostatic pressure has been simulated, and the results of theoretical calculation agree well with the existing experiment data, which proves that the dynamic damage of the stiffened shell can be forecasted by the above dynamic mechanics model taking account of hydrostatic pressure.

Introduction

For the dynamic plastic damage of stiffened cylindrical shell, in theory, Hoo Fatt [1] made some improvements of the string model, which can provide transient deformation fast, and the cylindrical shell is supposed as a beam clamped at one end and simply supported at the other end by Lellep [2], which is suitable for most of material. Meanwhile, in the aspect of software, ABAQUS/Explicit is used by Cichocki [3] to get the plastic deformation, and ANSYS/LS-DYNA is used to do numerical experiments of cylindrical shell to study the artificial boundary by Xiongliang Yao [4], and Islam [5] uses ABAQUS to simulate the interaction between shock wave and cylindrical shell in the process of explosion, which embodies the geometric nonlinearity of material, and the influence of hydrostatic pressure effected on the cylindrical shell is studied through ABAQUS at the same time by Huifang Ma [6].

Viewed from the above research, in the first theoretical method the compressive stress cannot be considered in the mechanical model, and in the second theoretical method, the cylindrical shell is seen as a beam, of which the local damage cannot be obtained. The hydrostatic pressure is the main factor of the plastic damage problem for submarines, therefore it should be considered in the calculation method. In view of these, a typical compartment of the cylindrical shell is intercepted to be simplified to a dynamic model, of which the both ends are fixed. And in the premise of keeping the dynamics factor same, the hydrostatic pressure becomes the rectangle pulse through dynamic magnification coefficient. Then a fast dynamic plastic damage theoretical calculation method of the cylindrical shell is obtained. At last, the results of the above theoretical method are compared with software simulation's and the experiment's, which proves that the method can predict the dynamic plastic damage of submarines well.

1 DYNAMIC MECHANICS MODEL OF STIFFENED CYLINDRICAL SHELL

1.1 Load form on the stiffened cylindrical shell

During the research of the dynamic plastic damage of cylindrical shell in underwater explosion, the load acted on the cylindrical shell plays a key role in this process, therefore the hydrostatic pressure is taken into consideration associated with shock wave in this paper. The shock wave is thought as a form of exponential attenuation, which can be denoted as follows [7]:

$$\begin{aligned} &P_1 = P_m e^{-t/\theta} \\ &P_m = 52.4\times10^6 (W_{ep}{}^{1/3}/R)^{1.13} \\ &\theta = 0.084\times10^{-3}\times W_{ep}{}^{1/3}(W_{ep}{}^{1/3}/R)^{-0.23} \end{aligned} \tag{1}$$

where W_{ep} accounts for the loading dose (kg), and R is the critical distance (m).

In the research of the plastic damage of cylindrical shell under impact load, rectangular pulses should be adopted, but shock wave is an exponential pulse. In the paper[8], the final plastic deformation of structure has been related to these parameters:

$$w_m = I_i{}^2 \cdot G(P_i,\beta) \tag{2}$$

where G is a function determined by the structure dynamic characteristic, and I_i is the equivalent impulse acted on the cylindrical shell, and P_i is the equivalent load acted on the cylindrical shell.

Keeping the impulse I_i equal, the shock wave form is changed from exponential pulse to rectangle pulse. Therefore, the equivalent impulse acted on the cylindrical shell is:

$$I_i = \int_{t_y}^{t_f} P_1\, dt \tag{3}$$

The equivalent shock wave acted on cylindrical shell is:

$$P_i = \frac{I}{2\cdot t_{mean}} \tag{4}$$

where t_{mean} and I can be shown as $t_{mean} = \frac{1}{I}\int_{t_y}^{t_f}(t-t_y)\cdot P_1\, dt$ and $I = P_0(t_f - t_y)$, of which P_0 is the static limit load, and t_y is the moment that the structure begins to form plastic deformation, t_f is the moment that plastic deformation stops.

The same size of the pulse load and static load are done on the same multiple freedom system, but the dynamic response will be different. Therefore, the static load can be transformed into dynamic load through using dynamic magnification coefficient, and it can be obtained from the following method.

The hydrostatic pressure on the cylindrical shell is

$$P_2 = \gamma h_d \tag{5}$$

where γ is the proportion of water, and h_d is the distance from the center of cylindrical shell to the surface of it.

Because the stiffened cylindrical shell is long enough in this paper, it can be transformed into a beam model, of which both ends are fixed, to get the dynamic magnification coefficient, which is

$$D = \frac{w_{dy}}{w_{st}} \tag{6}$$

where w_{dy} is the midpoint displacement of the beam under dynamic load, and w_{st} is the midpoint displacement of the beam under static load, which is

$$w_{st} = \frac{P_2 l^4}{384EI} \tag{7}$$

Meanwhile, w_{dy} can be obtained by modal superposition method [9].

The motion equation of beam is

$$EI\frac{\partial^4 w(x,t)}{\partial x^4} + m\frac{\partial^2 w(x,t)}{\partial x^2} = P_2 \tag{8}$$

where EI is the beam rigidity, and m is the quality of the unit length of beam.

Therefore, w_{dy} can be written as

$$w_{dy} = \sum_{n=1}^{\infty} W_n(l/2) q_n(t) \tag{9}$$

where $W_n(l/2)$ is the vibration shape function of n order natural frequency, and $q_n(t)$ is the time function of n order natural frequency.

Now Matlab is used to solve the above equation. When n is 10000, the error that the results have been compared with the finite element software’s meets the engineering accuracy. Therefore, the dynamic amplification coefficient will be obtained by using the modal superposition method, when n take 10000eventually.

Then, the static load on the stiffened cylindrical shell will be turned into pulse load, which is

$$P_{2i} = \frac{P_2}{D} \tag{10}$$

At last, the ultimate pulse pressure on the stiffened cylindrical shell can be represented as

$$P_i = P_{1i} + P_{2i} \tag{11}$$

1.2 Dynamic model of mechanics stiffened cylindrical shell

The shock wave is assumed as the plane wave, and the length of the dynamic mechanics model shown in Fig.1 can be got according to the following inequality based on shock wave propagation law.

$$l = n \cdot l_n < \sqrt{(R_{\min} + 2a)^2 - R_{\min}{}^2} \tag{12}$$

where $R_{\min}$ is the shortest distance from the critical point to the surface of cylindrical shell, and a is the radius of cylindrical shell, and l_n is length between the two transverse bulkhead, and n is the number of tanks to meet the above inequality, and l is the length of dynamic mechanics model.

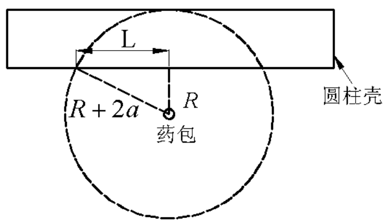

Fig.1 The length of typical compartment of the cylindrical shell

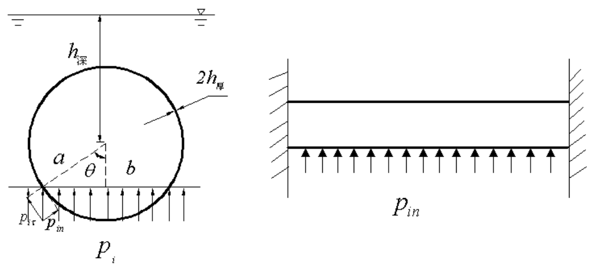

Fig.2 The calculation model of cylindrical shell

Because the stiffened cylindrical shell is long enough, and bulkheads are strong artifacts, therefore boundary conditions of dynamic mechanics model can be supposed clamped on both ends, shown in Fig.2. In order to simplify the calculation, the cylindrical shell should be rigid-plastic, and the axial force should be neglected.

The following dimensionless quantity have been introduced during the analysis of the plastic dynamic response of stiffened cylindrical shell:

$$\left.\begin{array}{lll} n = \dfrac{N_\varphi}{N_0}, & m = \dfrac{M_x}{M_0}, & p_i = \dfrac{P_i a}{N_0} \\ y = x / L, & W = \dfrac{w\mu a}{N_0 t^2}, & \tau = t / t_{mean} \\ c = L / \sqrt{ah}, & N_0 = 2\sigma_0 h, & M_0 = \sigma_0 h^2 \end{array}\right\} \tag{13}$$

where t_{mean} is the time that the pulse load have an effect on the structure, and μ is the quality of the unit area of the shell and the added mass of the unit area of the shell, and N_0 is membrane limit force, and M_0 is the ultimate bending moment, and L is the half length of the dynamic mechanics model, and x tis the location which is needed to know the plastic damage, and h is half shell thickness when the small bars on the cylindrical shell are turned into shell thickness according to the area equivalent principle, and σ_0 is yield limit.

The motion equation of stiffened cylindrical shell can be written as:

$$\frac{m''}{2c^2}+n-p_i\cos\theta+\ddot{W}=0 \tag{14}$$

where θ is the angle between the incident shock wave direction and the normal direction of cylindrical shell, which can be written as $\cos\theta = b/a$.

Due to the nonlinear material, the simplified Tresca yielding condition [10] should be adopted during the analysis of the plastic dynamic response. When the shell is in the first phase of motion, there are two different plastic formats in different section, where $y=v_0$ is set as its boundary circle, shown in Fig.4(a).

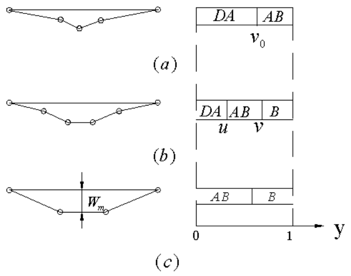

Fig.4 Plastic movement process of cylindrical shell

At $0 \le y \le v_0$, $n=-1, -1 \le m \le 1$.

At $v_0 \le y \le 1$, $m=1, n=-1$.

According to the moment, velocity and displacement continuous conditions in the plastic hinge, $\dot{W}$ can be obtained as

$$\dot{W}=(p-1)\tau \tag{15}$$

At $\tau = 1$, the pulse load falls to zero, the motion of the shell comes into the second phase. At this time, three plastic sections will come out in the half length of the shell shown in Fig.4 (b), u and v is assumed as the boundary of three sections.

At $\tau \ge \tau_1$, the motion of the shell comes into the third phase. At this time, two plastic sections will come out in the half length of shell shown in Fig.4(c). W_m is the largest plastic deformation in the central of the shell, which can be obtained through continuous conditions[2].

$$W_m=\int_{\tau_1}^{\tau_f}\frac{6}{c^2\dot{u}}\left[\frac{c^2}{6}-(1-u)^{-2}\right](1-u)d\tau \tag{16}$$

where τ_1 is the end of the second phase, and τ_f is the end of the whole motion, u and z are intermediate variables, which are obtained as

$$u=v_0-z-\frac{\sqrt{3}}{2c}\ln\left(\frac{z+2\sqrt{3}/c}{z-2\sqrt{3}/c}\right)\left(\frac{v_0-2\sqrt{3}/c}{v_0+2\sqrt{3}/c}\right) \tag{17}$$

$$z^2 = \frac{6}{c^2}[2-(2p_i\cos\beta-3)(p_i\cos\theta-\tau)^2(p_i\cos\theta-1)^{-3}] \tag{18}$$

$$v_0 = \frac{1}{c}\sqrt{6/(p_i\cos\theta-1)} \tag{19}$$

At $v=1$, this is the time at the end of the second stage, and the time is τ_1.Therefore, substituting $u=u_1$ and $z_1=1-u_1$ into Eq.(19), τ_1 can be obtained, meanwhile the end time τ_f is obtained as

$$\tau_f = \tau_1 + \frac{\dfrac{(-2\sqrt{3})sh[c(1-v_0)/\sqrt{3}]+v_0ch[c(1-v_0)/\sqrt{3}]}{ch[c(1-v_0)/\sqrt{3}]-v_0sh[c(1-v_0)/\sqrt{3}]}(p_i\cos\theta-\tau_1)}{\dfrac{(-2\sqrt{3})sh[c(1-v_0)/\sqrt{3}]+v_0ch[c(1-v_0)/\sqrt{3}]}{ch[c(1-v_0)/\sqrt{3}]-v_0sh[c(1-v_0)/\sqrt{3}]}+\sqrt{6}} \tag{20}$$

Substituting Eqs.(17)-(20) into (16),the maximum deflection in the centre of shell is obtained as

$$w_m = \frac{W_m N_0 t^2}{\mu a} \tag{21}$$

2 Validation of dynamic mechanics model

In this part, the experimental data [11] is adopted to verify the forecast method of the plastic damage of cylindrical shell mentioned in the second part. The cylindrical shell in this experiment is 1.2m long, and the diameter of it is0.275m, at the same time the thickness of it is 2 mm. In order to do the experiment, the equipment is shown in Fig.5, and the ends of the cylindrical shell are welded in the two ends of the cage. And the displacements of A1 and A2 have been measured during the process. At last, the theoretical results can be got by substituting the working conditions into Eq.(21), and Table 1 can be obtained through comparing the theoretical results with the experiments data.

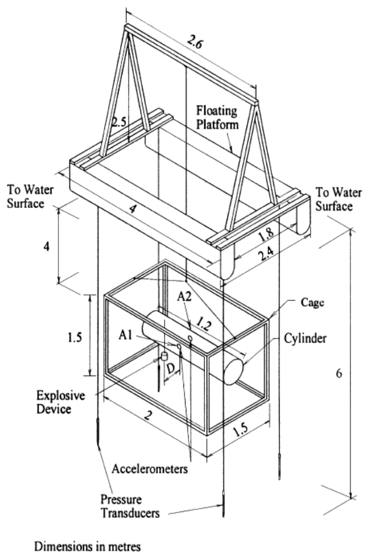

Figure5. Schematic of the floating support platform, cage, cylinder and charge

Table 1Thecomparisonbetweeentheoretical value and experimental value of the deflection in the midpoint of stiffened cylindrical pressure shell

Condition(Quantity-Charge Distance-Depth)	Theoretical value(mm)	Experimental value(mm)	Error
5Kg-300m-4.75m	3.15	4	21.3%
5Kg-150m-4.75m	8.34	10	16.6%
10Kg-300m4.75m	10.28	12	14.3%

The reasons causing these errors are: (1) The dynamic mechanics model is rigid-plastic, but the real stiffened cylindrical shell is elastic-plastic material. (2) The load on the dynamic model is a rectangular pulse load, but the actual effect is exponential pulse load on the structure. (3) The assumed boundary condition is both ends clamped, but the actual one is not completely clamped. Because the errors are in the permission scope of the engineering precision, that the dynamic model can forecast the dynamic plastic damage of stiffened cylindrical shells considering of the hydrostatic pressure well can be verified.

3 Conclusion

Based on the principle of the plane wave, the length of the dynamic mechanics model can be obtained, and its ends are clamped. Then a calculation method of the deformation of stiffened cylindrical shell considering of the hydrostatic pressure will be obtained. At last, the following conclusions can be obtained through comparing the experimental results with the theoretical’s.

(1) To keep the corresponding parameter the same, the hydrostatic pressure is to be converted into the dynamic load through the dynamic magnification coefficient. Meanwhile shock wave can be converted into rectangular pulse using the principle that the impulse is equal at the same time. Then these two kinds of loads are on the dynamic model of stiffened cylindrical shell at the same time, and this method has the feasibility through comparing the calculation results with the experiments’.

(2) Based on the plane wave assumption, the selection method of local typical cabins are obtained. At the same time, the selected length is compared with the plastic damage zone in the experiment, which proves that the interception method of stiffened cylindrical shell is practical in engineering.

(3) Based on the plane wave, the radial force acted on cylindrical shell will be multiplied by an incident shock wave pressure angle. Then a calculation method of the deformation of a typical location of the cylindrical shell can be obtained using the kinetics principle. At last, the calculation results compared with the experiments’, the error is within the scope of the project approval, then the dynamic model can predict the plastic damage of stiffened cylindrical shell considering of hydrostatic pressure well.

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