ICSV13 - Vienna
The Thirteenth International Congress on Sound and Vibration
Vienna, Austria, July 2-6, 2006

DYNAMIC RESPONSE OF DUAL-LAYER SHIP STRUCTURES SUBJECTED TO UNDERWATER EXPLOSION

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ABSTRACT

In order to improve the survivability of warships to underwater explosion, a new type of shock absorption and isolation rubber structure is proposed. The structure uses the principle of energy absorption through structural deformation and shock wave reflection between the interfaces of materials with great impedance mismatch. Experiments were conducted to study the shock protection ability and dynamic responses of a model with the shock absorption and isolation structure. The shock protective layer (SPL) is stuck to the outer hull of the experimental model, which is a 13.2m×0.8m×0.45m steel box with stiffeners simulating ship structures. The box with its ballast weighs 3500 kg and has a 0.35m waterline. The first natural frequency is 5.5Hz, corresponding to a bend mode. Experiments were conducted in a semispherical water pond. The spherical explosion charge is 0.5kg TNT and located 5m under the bottom of the model. Shock responses of the model with and without shock protection layer are compared.

INTRODUCTION

Research on underwater explosion is very important to improve the safety and survivability of warships. Since the beginning of the 20^{th} century, the influence of underwater explosion on ship structures and equipments mounted on ships have been deeply studied. In general, warships may be subjected to contact or non-contact explosion from the air or underwater arms. Contact explosion usually damages warships and even destroys them. Non-contact explosion will cause the whole ship exposed to a strong impact load, results in permanent deformation in the hull body and leads to breakdown of equipments and casualty of personnel. Underwater explosion is a complicated problem since warships will be subjected successively to shock wave pressure and the pulsation pressure of gas bubble under non-contact explosion. Duration of shock waves is short but the pressure is high. Furthermore, shock waves attenuate quickly but can stir up each mode of ship structures. Gas bubble pressure is small but keeps a long time, which usually excites the low frequency modes of a ship body and makes the whole body deform significantly as well as serious damage to equipments of low vibration frequencies. In reality, ships might not be completely damaged after suffering the explosion shock wave but might be destroyed after experiencing bubble pressure. Therefore, gas bubble pulsation can not be neglected [1]. Hans [2] reviewed the applicability of various hydrocodes, such as Lagrangian, Eulerian, Coupled Eulerian-Lagrangian and Arbitrary Lagrangian-Eulerian, in the analysis of structural responses to underwater explosions. Nathan [3] used the LS-DYNA/USA coupled computer code to create a virtual underwater explosion environment, and analyze the response of a surface ship exposed to an underwater shock. Michael [4] presented a cavitating acoustic spectral element formulation and applied the method to the response of a surface ship excited by an underwater explosion.

Along with the fast development of weaponry, especially the significant improvement of the breakage of non-contact underwater explosion, it is very important to improve the anti-explosion ability of ships. In 1990s, USA navy researchers [5] studied shock resistant characteristics of ship structures with coatings. It was shown that the thickness, shear deformation modulus and sound impedance of coatings had a large effect on structural stress. In 2002, Roshdy [6] put forward a double layer shock resistant structure at the conference of USA navy warship structure. The proposed structure has a light alloy material in the core, which can absorb most shock energy by plastic deformation of the alloy core. David and Haydn [7] studied a sandwich plate with a metal pyramidal core. Xue and Hutchinson [8] used the finite element method to simulate blast resistance of clamped sandwich beams and monolithic beams of the same mass. Pyramidal truss, square honeycomb and folded plate core geometries were considered. Fleck and Dshpande [9] developed analytical formulae to analyze structural responses of metallic sandwich beams subjected to both air and water blasts. Analysis results show that an improvement of blast resistance is achieved by employing sandwich construction in the case of water blast. However, in air blast, sandwich construction gives only a moderate gain in blast resistance compared to monolithic construction. These analysis results cannot be validated by experiments. According to recent papers, coating [5] is either an elastic material or a nearly incompressible rubber, the rest [6–9] about shock resistant structures are only limited to metallic core sandwich beams or plates.

Because the environment of underwater explosion is complex, shock response simulation of warships subjected to underwater explosion is generally complicated by free surface effects. For example, surface reflection waves will result in bulk cavitation, hull cavitation, bubble oscillation and migration toward free surface as well as cavitation closure pulses. Furthermore, there are complex fluid–structure interaction and ship dynamics, especially the double layer ship body with complex shock resistant structures. It is difficult for the present simulation methods to get the same results as measured on a real ship subjected to underwater explosion. Experiment, an important way to study underwater explosions, is also the most effective method to examine the credibility of numerical methods. In general, underwater explosion experiments of surface ships and submarines cannot be carried out in a pond. The expense of underwater explosion tests for real warships is great, and safety is hard to control. Considering cost and safety issues, the scrutiny on shock testing environment is one of the most pressing issues of recent years. The strength of the environmental protection has led to a virtual cessation of open explosive water testing. So it is very important to use a reasonable model to conduct a pond explosion experiment, in which the comprehensive function of shock wave and bubble pulsation is considered.

In this paper, a new type of shock absorption and isolation rubber structure is presented. The shock protective layer, which can be stuck to the outer hull of ships, is stable under normal pressure but very flexible under the pressure of UNDEX. Two models, one with SPL and the other without SPL, were devised with finite element method. Vibration modes of the model in the air as well as water were measured and compared. The experimental results of underwater explosion were compared to analyze the protective ability of SPL.

EXPERIMENT

Experimental model design with finite element method

A reasonable model is very important to underwater explosion. In order to analyze the comprehensive function of shock wave and bubble pulsation, a scaled model is designed with finite element method according to a real warship in this paper. The scaled model has the same modal parameters with the real warship. The finite element motion equations of structures in the air can be given in the time domain:

$$\boldsymbol{M}_e^s\{\ddot{\boldsymbol{u}}\}+\boldsymbol{C}_e^s\{\dot{\boldsymbol{u}}\}+\boldsymbol{K}_e^s\{\boldsymbol{u}\}=\{\boldsymbol{F}_s\} \tag{1}$$

where $\boldsymbol{M}_e^s$ is the structural mass matrix, $\boldsymbol{C}_e^s$ the structural damping matrix, $\boldsymbol{K}_e^s$ the structural stiffness matrix, $\boldsymbol{F}_s$ the applied time-varying load, the superimposed dot represents the temporal derivative. Modal parameters such as eigenfrequencies and mode shapes of the scaled model in the air can be obtained from equation (1).

The pressure equation of an ideal incompressible, inviscid and irrotational fluid can be written

$$\frac{1}{\boldsymbol{c}^2}\frac{\partial^2 \boldsymbol{p}}{\partial t^2}-\nabla^2\boldsymbol{p}=0 \tag{2}$$

where $\boldsymbol{c}$ is the sound velocity in the fluid, $\boldsymbol{p}$ is the fluid pressure. At the fluid-structure interface, the compatibility condition between the normal displacement $\boldsymbol{u_n}$ of the outer shell structure and the pressure $\boldsymbol{p}$ can be expressed as

$$\frac{\partial \boldsymbol{p}}{\partial \boldsymbol{n}}=-\rho_f\frac{\partial^2 \boldsymbol{u_n}}{\partial t^2} \tag{3}$$

where $\boldsymbol{n}$ is the normal direction of the fluid-structure interface, ρ_f is the fluid density. Discretizing equation (2), the finite element equations of motion for the fluid can be expressed as

$$\boldsymbol{M}_e^f\{\ddot{\boldsymbol{p}}\}+\boldsymbol{C}_e^f\{\dot{\boldsymbol{p}}\}+\boldsymbol{K}_e^f\{\boldsymbol{p}\}+\rho_f\boldsymbol{R}_e^f\{\ddot{\boldsymbol{u}}\}=\{\boldsymbol{F}_f\} \tag{4}$$

where $\boldsymbol{M}_e^f$ is the fluid mass matrix, $\boldsymbol{C}_e^f$ the fluid damping matrix, $\boldsymbol{K}_e^f$ the structural stiffness matrix, $\boldsymbol{R}_e^f$ is the fluid-structure interaction matrix, $\boldsymbol{F}_f$ the applied time-varying load.

The finite element equations of motion for the wet structure can be expressed as

$$\boldsymbol{M}_e^s\{\ddot{\boldsymbol{u}}\}+\boldsymbol{C}_e^s\{\dot{\boldsymbol{u}}\}+\boldsymbol{K}_e^s\{\boldsymbol{u}\}=\{\boldsymbol{F}_s\}+\left(\boldsymbol{R}_e^f\right)^{\mathrm{T}}\{\boldsymbol{p}\} \tag{5}$$

Combining equation (4) and equation (5), the following matrix equation can be obtained.

$$\begin{bmatrix}\boldsymbol{M}_e^s & \boldsymbol{0}\\ \rho_f\boldsymbol{R}_e^f & \boldsymbol{M}_e^f\end{bmatrix}\begin{Bmatrix}\ddot{\boldsymbol{u}}\\ \ddot{\boldsymbol{p}}\end{Bmatrix}+\begin{bmatrix}\boldsymbol{C}_e^s & \boldsymbol{0}\\ \boldsymbol{0} & \boldsymbol{C}_e^f\end{bmatrix}\begin{Bmatrix}\dot{\boldsymbol{u}}\\ \dot{\boldsymbol{p}}\end{Bmatrix}+\begin{bmatrix}\boldsymbol{K}_e^s & -\left(\boldsymbol{R}_e^f\right)^{\mathrm{T}}\\ \boldsymbol{0} & \boldsymbol{K}_e^f\end{bmatrix}\begin{Bmatrix}\boldsymbol{u}\\ \boldsymbol{p}\end{Bmatrix}=\begin{Bmatrix}\boldsymbol{F}_s\\ \boldsymbol{F}_f\end{Bmatrix} \tag{6}$$

When the model is in water, its wet modal parameters can be predicted with equation (6).

In this paper, the experimental model is a 13.2m×0.8m×0.45m steel box with stiffeners simulating ship structures (Fig.2). The box with its ballast is 3500 kg and has a 0.35 m waterline. The bottom plates are 8mm thick and the side plates 5mm thick.

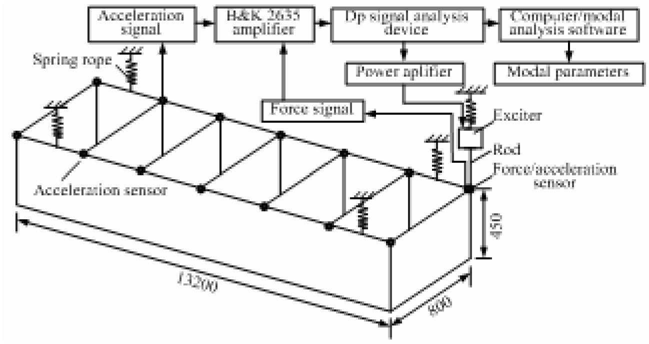


Fig.1 Configuration of the dry modal test

Fig.2 The setup of wetted modal test

Fig.3 Underwater explosion shock test

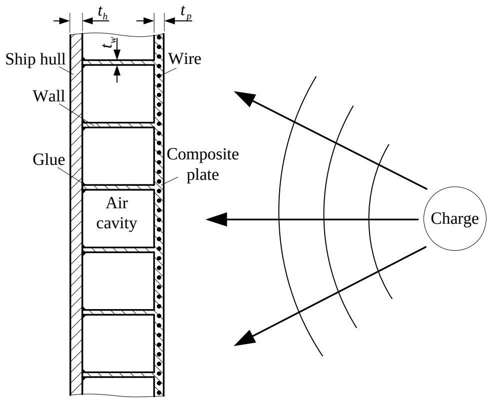
Modal test

In order to obtain dry and wet modal parameters of the model with and without SPL, two sets of experiments were conducted respectively. Fig.1 shows the configuration of the dry modal test of the model. The model is hanged by a steel hook and four elastic ropes so as to keep it in free. Fig.2 gives the experimental setup of the wetted modal test. Frequency response function (FRF) reflects the inherent structural dynamic characteristics. The experiment is to obtain an accurate FRF. Excitation source in the experiment was a B&K4809 electromagnetic shaker, which produces excitation on the shell body via a flexible rod. Excitation signal was generated by a signal analyzer (Data Physics) and was amplified by a B&K2607 power amplifier. Vibration signals were sensed by force and acceleration transducers and amplified by charge amplifiers BK2635. In the experiment, shaking was carried out in a sweep mode and Hanning windows were used in order to reduce leakage.

Underwater explosion

Underwater explosion was carried out in a semi-spherical water pond (Fig.3), which is 15 meters deep and has an area of 85 meters in diameter. The spherical explosion charge is 0.5kg TNT and located 5 meters under the model's bottom plate. The SPL is a honeycomb rubber layer of 38mm in thickness (Fig.4), stuck to the outer shells of the test model. Fig.5 is the test model coated with SPL.

Four accelerometers and four strainmeters were used to measure acceleration and strain responses of the models with and without SPL in the experiment. Fig.6 is the setup of the underwater explosion experiment. Fig.7 depicts the measurement points of acceleration and strain. A pressure gage was installed at a point 3.2 meters away from the explosion centre to measure the free field pressure (Fig.6).



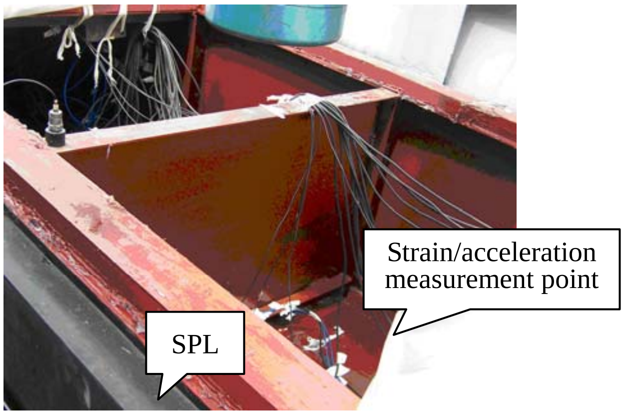
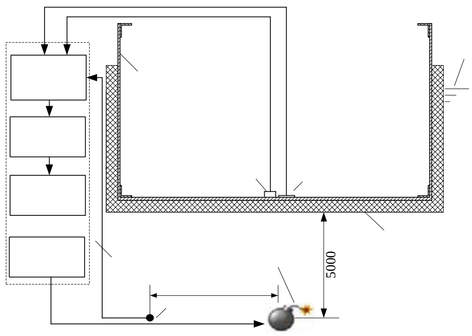


Fig.4 SPL structure *Fig.5 Experimental model with SPL*



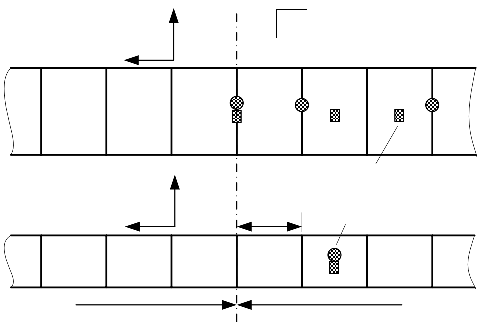


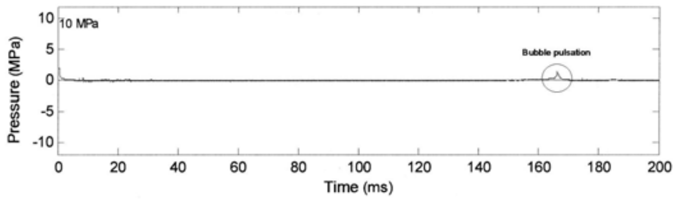
Fig.6 Test and Measurement setup *Fig.7 Placement of transducers*

RESULTS AND DISCUSSION

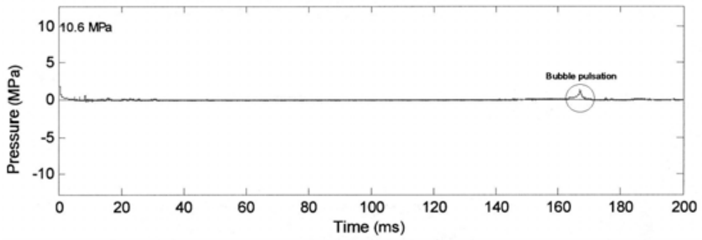
Tab.1 gives the first bending modal frequencies of the model with and without SPL obtained from the experiment and FEM, respectively. The results show that the wet modal frequencies of the model with SPL are decreased by about 10% compared with those of the model without SPL. For the dry and wet model, the decrease of first modal frequency is about 2.38Hz and 3.01Hz, respectively, in both cases with and without SPL. It is clear that the SPL coating causes natural frequencies of the model to decrease. The table also shows that the FEM results agree well with the experimental results, which indicates that the FEM model is correct. The model can be used to optimize SPL subjected to underwater explosion with ABAQUS code. In order to make the bubble pulse frequency close to the first bending frequency of the model, it is very important to choose a reasonable explosion heart position and explosion charge weight according to the modal frequencies, which can stir up the whipping motion of the model. Fig.8 is the time history of the free field pressure. The second bubble pulsation frequencies of the model without and with SPL are 6.02Hz and 6.04Hz, respectively. Both of them are close to the modal frequencies of the model.

Tab.1 Computed and measured natural frequencies

		FEM	**Experiment**
Dry mode	No SPL (Hz)	8.55	8.86
	With SPL (Hz)	7.82	7.63
Wet mode	No SPL (Hz)	6.01	5.85
	With SPL (Hz)	5.50	5.25



(a) Without SPL



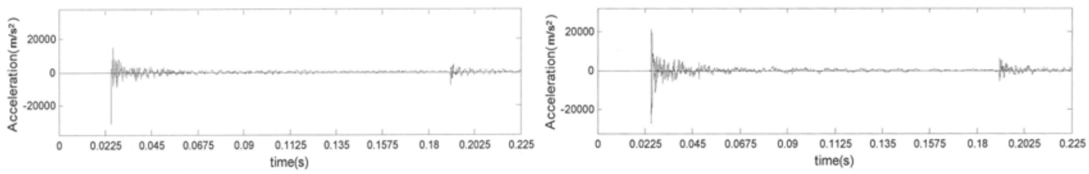
(b) With SPL

Fig.8 Free field pressure

Tab.2 gives the acceleration peaks of the four measurement points. Fig.9 and Fig.10 are the measured acceleration at A1 and A4. According to these data, a great reduction in acceleration responses can be achieved when using SPL. The average ratio of the acceleration peak with SPL to that without SPL is only 0.076. Therefore, about 92% shock acceleration can be cut off by SPL.

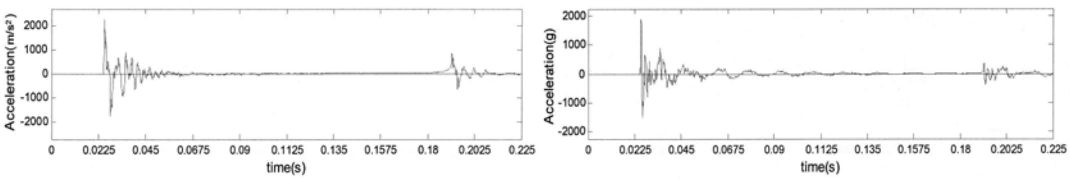
Tab.2 Acceleration response peaks

	A1	**A2**	**A3**	**A4**
No SPL (m/s^2)	30626.3	40694.1	19154.1	26491.6
With SPL (m/s^2)	2225.2	2657.7	1860.4	1829.0



(a) A1 *(b) A4*

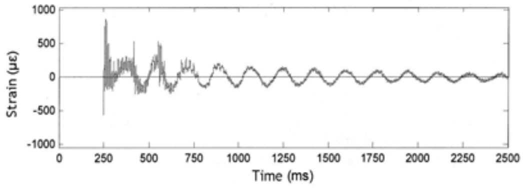
Fig.9 Acceleration histories of the model without SPL at A1 and A4 measurement points



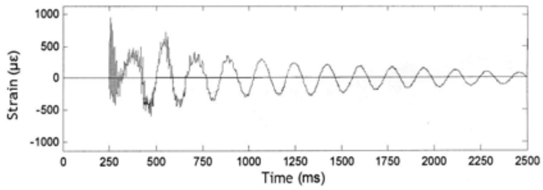
(a) A1 *(b) A4*

Fig.10 Acceleration histories of the model with SPL at A1 and A4 measurement points

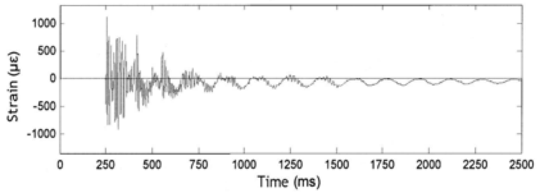
Fig.11 and Fig.12 are the strain responses of the hull with and without SPL measured at S1 and S4. Tab.3 gives the strain peaks in the stages of shock wave and bubble pulsation, respectively. According to these figures and the table, the hull strain is decreased greatly when coated with SPL. In shock wave stage, the average strain of the four measurement points is reduced by more than 51%, and in bubble pulsation stage, the average strain is reduced by about 35%. Furthermore, in the bubble pulsation stage, the transverse strain is decreased more than the longitudinal strain. Especially at S2, the longitudinal strain of the hull with SPL is almost similar to that of the hull without SPL. The average transverse strain of the four points is reduced by about 48%, and the average longitudinal strain is reduced by about 21%. Therefore, SPL can protect the hull structure from shock wave and bubble pulsation, and its protective ability is better for shock wave.



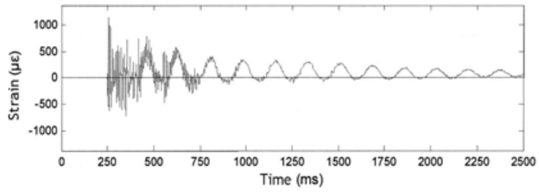
(a) *S1-1*



(b) *S1-2*

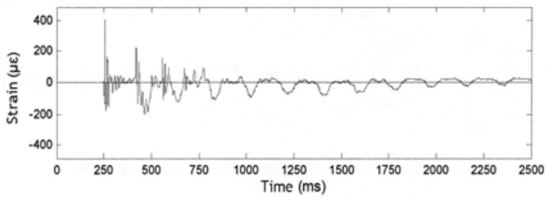


(c) *S4-1*

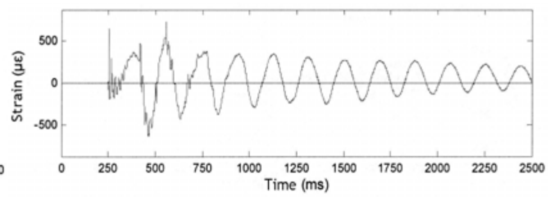


(d) *S4-2*

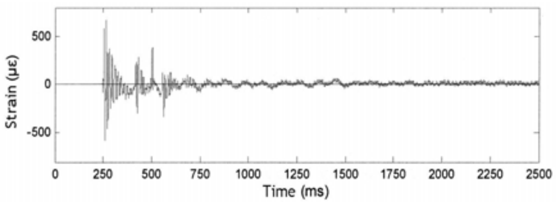
Fig.11 Strain histories of the model without SPL at S1 and S4



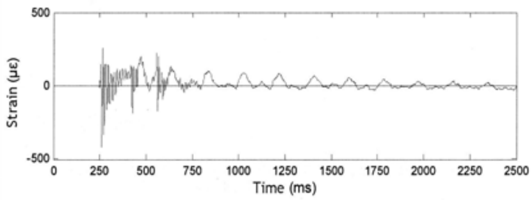
(a) *S1-1*



(b) *S1-2*



(c) *S4-1*



(d) *S4-2*

Fig.12 Strain histories of the model with SPL at S1 and S4

Tab.3 Strain peaks at the shock wave stage and the bubble pulsation stage

	No SPL ($\mu\varepsilon$)		**With SPL ($\mu\varepsilon$)**	
	Shock wave	Bubble pulsation	Shock Wave	Bubble pulsation
S1-1	954.8	592.1	442.5	224.5
S1-2	1044.6	814.8	699.6	793.1
S2-1	898.4	657.3	438.9	268.7
S2-2	1127.3	587.1	515.1	586.4
S3-1	748.8	406.9	154.3	249.8

S3-2	720.0	222.5	467.4	197.1
S4-1	1227.5	638.2	748.3	427.8
S4-2	1269.2	866.1	461.8	250.8

CONCLUSIONS

The shock protective rubber structure was studied with experiments. Dynamic responses of the model with and without SPL have shown a high efficiency of shock isolation of the novel shock protective layer structure. The experimental and numerical results agree well with each other and the finite element model can be used to optimize SPL in further research.

ACKNOWLEDGEMENTS

The authors would like to thank the Navy of China for financially supporting this research under the contract No. 6140625, and CSSRC for providing help in the experiments.

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