Verification of a theoretical model of tensor skin under water impact by considering the fluid-structure interaction

YanTing REN^a, XinMing QIU^{a*}, T.X. YU^{b,c}

^aDepartment of Engineering Mechanics, Tsinghua University, Beijing 100084, China

^bDepartment of Mechanical Engineering, Hong Kong University of Science and Technology, Clear

Water Bay, Kowloon, Hong Kong

^c Y.K. Pao Chair Professor, Zhejiang University, Hangzhou, China

* Corresponding author, 8610- 62772329, e-mail: qxm@tsinghua.edu.cn

Abstract

Tensor skin is a kind of composite sandwich structure developed to improve the helicopter's crashworthiness when impact with water. In our previous theoretical studies, by employing the rigid, perfectly plastic idealization and the pulse approximation method, the static and dynamic responses of tensor skin under uniform pressure loading were obtained.

In this study, a fluid-structure interaction simulation model of tensor skin impinging with water is established by FE method, in order to **verify** the previous theoretical analyses. It is found that the interaction pressure between the skin plate and water varies with the configuration of the plate. Compared with a solid plate, the pulses generated by a honeycomb-core sandwich plate and a tensor skin are more moderate. By studying the effect of water depth, the theoretical predictions of the tensor skin, as developed in our previous papers, are found valid for the deep water case.

1 From the comparison among the theoretical predictions, pulse loading FE model, and the
2 fluid-structure interaction FE model, the deviation of the theoretical prediction in the final
3 deflection is found to be about 10% compared with that obtained from the fluid-structure
4 interaction FE model. Thereby, the theoretical model and the pulse approximation method are
5 confirmed to have sufficient accuracy for the design of tensor skins.

6 **Keywords**

7 Tensor skin, Verifications, Theoretical model, Water impact, Energy absorption

8

1

2 **Notations**

3 ρ is density

4 E_l, E_t are the elastic modulus in the longitudinal and transverse directions, respectively

5 G is the shear modulus

6

7 h_1, h_2, h_3 are thicknesses of cover, carrying and tensor ply, respectively

8 H is the depth of water

9 I is momentum

10 k is the geometric parameter of tensor skin

11 l is a half of the beam length

12 M_p is the fully plastic bend moment

13 q is pressure applied on the beam

14 q_s is the static critical pressure

15 $\bar{q}_s \equiv q_s l^2 / M_p$ is the normalized static critical pressure

16 $\tilde{q} \equiv \bar{q} - \bar{q}_s$ is the excess pressure

17 t_d is impulse duration

18 t_f is the total response time

19 u is the central beam deflection

20 $\bar{u} \equiv u/l$ is the normalized central deflection

21

22

1. Introduction

As a kind of sandwich structure comprised of cover plies, tensor plies and carrying plies, tensor skin is originally developed by NLR (European National Aerospace Laboratory) in order to improve the crashworthiness of composite helicopter structures subjected to water impact [1], as shown in Fig.1. The cover ply is the loading face; the carrying ply provides the structure stiffness; and the tensor ply provides the capability to unfold by forming *plastic hinges*, before it is stretched and fails, leading to an increase in the load bearing capability and energy absorption of the structure.

Fig.1 The schematic of a tensor skin

Under the loading of tension or bending, the deformation mechanism of the tensor skin strip was found to be unfolding by forming plastic hinges before the failure due to stretching [2, 3]. The static and dynamic responses of tensor skin panels were compared with the honeycomb-core sandwich plates [4-7]. In the static transverse loading tests, the fully clamped panels were pushed by a blunt indenter that was perpendicular to the surface. Only 3-layer tensor skin panel was able to transfer sufficient running load to the substructure to initiate crushing. In the dynamic tests performed by dropping an impactor, large damage was found in the honeycomb-core sandwich, while the tensor skin managed to stop the impactor successfully, with broken faces and unfolded core. The failure behavior of the tensor skin was also calculated by using PAM-CRASH [8]. Compared with the above mentioned experimental results [4, 7-11], good agreements between the

1 calculated and measured forces and displacement for tensor skin panels were found. Besides, three
2 types of innovative Leading Edge structures that made of tensor skin were designed, and also
3 tested by the impact of a 4-pound substitute bird [5]. The relative numerical studies by
4 PAM-CRASH were given by Mi[12]. Both results confirmed the good performance of tensor skins
5 in the anti-bird strike designs.

6 A theoretical model was proposed by the present authors [13] to analyze the static and
7 dynamic responses of tensor skin, in which the rigid-perfectly plastic idealization was adopted. It
8 was found that the tensor skin exhibits **unique** properties in its load-displacement relationship.
9 That is, the load to initiate the plastic deformation is high, but the resistance decreases during the
10 structure's deformation process. Furthermore, the pulse approximation method, initially proposed
11 by Youngdahl, was confirmed to be applicable to this structure, tensor skin [13]. The main
12 conclusions from the theoretical analysis will be reviewed in Section 2.

13 As well known, the dynamic response of a structure is an interaction process between the
14 structure and the applied load pulse. However, in the theoretical approach, both the structure and
15 the applied load have to be idealized in order to make the analysis feasible. For the tensor skin, the
16 rigid-perfectly plastic material model was adopted in [14], and the effect of water impact on the
17 structure was simplified as a triangular pulse applied on the tensor skin [13]. In all the previous
18 studies of the tensor skin, including experimental studies, numerical simulations, and theoretical
19 modeling [2, 4, 5, 10, 13, 14], no fluid-structure interaction effect has been considered. Hereby, it
20 is necessary to **verify** the existing theoretical model of the pressure loaded tensor skin with the
21 fluid-structure interaction being taken into account.

22 In the present paper, in order to **verify** the previous theoretical model, the finite element

simulations of the tensor skin under water impact are described, with the consideration of the fluid-structure interaction during the impact process. First, the theoretical analysis on the tensor skin under uniformly distributed pressure is reviewed in Section 2; and then the fluid-structure interactive FE models are described in Section 3; followed by Section 4, in which the dynamic responses of solid plate, tensor skin and honeycomb-core sandwich under water impacts are compared with one another; after a discussion on the effects of water depth in Section 5, the theoretical predictions of tensor skin are finally **verified** in Section 6.

2. Review of the theoretical analysis of tensor skin

A theoretical model of tensor skin was proposed in our previous paper[14], in which the total response of tensor skin is divided into three stages: an elastic deformation stage of the whole structure; an unfolding stage of the tensor ply; and a stretching stage of the tensor ply. At the beginning of response, elastic deformation takes place in the entire structure, which will end by the fracture of the cover and carrying plies made of brittle composite. The second stage is the unfolding of the tensor ply, which is simplified as rigid segments connected by plastic hinges. The third stage is the stretching of the tensor ply, which may be terminated by tension failure. Among the three stages, the unfolding stage is the main contributor to the energy absorption. Therefore, the theoretical analysis of tensor ply will be reviewed in the following.

2.1 Static critical pressure

In the unfolding stage, by adopting the rigid, perfectly plastic material assumption, a deformation mechanism with stationary plastic hinges was proposed[14], supposing a uniformly distributed pressure q is applied on the tensor ply quasi-statically, as shown in Fig.2. Hereby, as

angle α increases, the quasi-static critical pressure $q_s(\alpha)$ will vary.

Fig.2 Tensor ply under uniformly distributed pressure in the unfolding stage [14]

From the energy conservation, the increase in the external work δW should be equal to the variation in the energy dissipation δD . That is,

$$\delta W = \delta D \quad (1)$$

Hence, the normalized static critical pressure is obtained as,

$$\bar{q}_s(\alpha) \equiv q_s(\alpha) l^2 / M_p \quad (2)$$

where l is a half of the beam length and M_p is the fully plastic bending moment.

Fig.3 The normalized static critical pressure versus the normalized central deflection $\bar{u}$ [14]

It is found that the static critical pressure $\bar{q}_s$ first decreases then increases with the increasing central deflection $\bar{u} \equiv u/l$, as demonstrated in Fig.3. The shape of the curve indicates a kind of *softening effect*; that is, the pressure to motivate the initial deformation is high, but the structure's resistance decreases as the deformation becomes larger. That is different from the

1 conventional structures which display the stable load-deflection characters.

2 **2.2 Dynamic response of tensor ply**

3 The dynamic response of tensor ply was also analyzed in [14]. Here, the static critical
4 pressure $\bar{q}_s$ is regarded as a kind of *structural friction*. Therefore, the *excess pressure* defined as
5 $\tilde{q} \equiv \bar{q} - \bar{q}_s$ will produce the acceleration or deceleration of the structure.

6 Suppose a constant pressure $\bar{q} > \bar{q}_{s0}$ is applied on the tensor skin, here $\bar{q}_{s0}$ is the
7 normalized initial static critical pressure, which is given by Eq.(2) for $\alpha = 0$. According to energy
8 conservation,

$$9 \quad \dot{K} + \dot{D} = \dot{W}(q) \quad (3)$$

10 where $\dot{K}$ is the rate of kinetic energy, $\dot{D}$ is the rate of plastic dissipation, and $\dot{W}(q)$ is the
11 work rate of the pressure. Eq.(3) leads to a second order differential equation about α . Hereby, the
12 dynamic response could be obtained by solving this equation with specified initial conditions[14].

13 **2.3 Pulse approximation of water impact**

14 Youngdahl[15] proposed that the response of a general pulse could be approximately
15 equivalent to that of a rectangular pulse impulse I_e with an effective load P_e and pulse duration
16 $2t_{mean}$, which are defined by:

$$17 \quad I_e = \int_{t_y}^{t_f} P(t) \cdot dt, \quad P_e = I_e / 2t_{mean}, \quad t_{mean} = \frac{1}{I_e} \int_{t_y}^{t_f} (t - t_y) P(t) \cdot dt \quad (4)$$

18 where $P(t)$ is the non-negative pulse magnitude changing with time t , t_y and t_f are the time
19 instants when plastic deformation begins and ends, respectively. The final deflection of the
20 structure is determined by I_e and P_e for a general pulse load. Depended on the integral of the

1 general pulse, both I_e and P_e are not very sensitive to the pulse shape. The applicability of this
 2 approximation method for eliminating the pulse shape effects was validated for beams, circular
 3 plates and cylindrical shells, etc.[16].

4 In paper[13], the applicability of the above pulse approximation method was further
 5 examined for tensor skin. Meanwhile, as illustrated in [17], for a blunt or flat structure impinging
 6 on a water surface, the impact pulse only lasts for a very short time duration, which is very similar
 7 to a triangular pulse. It is found in [14], on the other hand, that the dynamic response of a tensor
 8 skin to a triangular pulse is quite close to that to an equivalent rectangular pulse. Fig.4 shows the
 9 triangular pulse and the equivalent rectangular pulse, and the difference between their final
 10 deflections is found to be smaller than 5%. Here $\bar{I} = \bar{q}\bar{t}_d$ and $\bar{q} = 1.3\bar{q}_{s0}$, $\bar{t}_d \equiv t_d/t_0 = 1$, where
 11 $t_0 = 0.2s$.

Fig.4 A triangular pulse and the equivalent rectangular pulse ($\bar{I} = 1.3\bar{q}_{s0}, \bar{t}_d = 1$)

3. The fluid-structure interaction simulation model

15 As mentioned in Introduction, none of the previous researchers has considered the effect of
 16 fluid-structure interaction, but this effect may provide a pressure pulse different from that adopted
 17 in the previous theoretical analysis. Therefore, by employing the commercial software
 18 PATRAN/DYTRAN, 3D simulation models are constructed in order to reveal the interaction effect

1 between water and tensor skin. Totally 7500 quadrilateral shell elements are divided for the
2 tension skin, and 9600 hexahedral solid elements (CHEXA in DYTRAN) are divided for the fluid
3 and air. Between different plies of tensor skin the surface to surface contact are defined, and the
4 fluid-structure interaction is defined between the plate and the Euler domain. It is confirmed that
5 the current mesh is fine enough to satisfy the convergence of the calculation, and the time step of
6 calculation is $10^{-8}s$.

7 3.1 Models of three kinds of plates

Fig.5 The FE models of (a) solid plate, (b) honeycomb-core sandwich, and (c) tensor skin.

11 The FE models of three different plates: solid plate, honeycomb-core sandwich plate and the
12 tensor skin, are depicted in Fig.5. The tensor skin comprised of cover ply, tensor ply and carrying
13 ply, has the equal thicknesses of three plies, $h_1 = h_2 = h_3 = 1mm$. The material of structure is
14 assumed to be elastic-plastic material, and the Euler domain is assumed to be ideal fluid. The
15 material properties of tensor skin are listed in Table 1.

Table 1 The material properties of tensor skin [2, 12]

Ply	Material	$\rho(Kg/m^3)$	$E_l(MPa)$	$E_t(MPa)$	$G(MPa)$	$\sigma_y(MPa)$	ε_c
Cover	carbon fiber composite	1840	37049	37049	$G_{12} = G_{13} = 2141$	345	1.05%
Carrying	carbon fiber composite	1840	37049	37049	$G_{12} = G_{13} = 2141$	345	1.05%
Tensor	Dyneema/Epoxy resin composite	1400	15405	15405	$G_{12} = G_{13} = 1409$	425	3.2%

17

18 In the Table, ρ is density; E_l and E_t are the elastic modulus in the longitudinal and

1 transverse directions, respectively; G is the shear modulus, $\mu_{12} = \mu_{13} = 0.34$ are Poisson ratios;
2 ε_c is the fracture strain.

3 The total mass of the tensor skin, solid plate and honeycomb-core sandwich are adjusted to
4 be the same. The solid plate that has thickness of 7mm, is made of the same material as the tensor
5 ply listed in Table 1. The top and bottom surfaces of the honeycomb-core sandwich plate are the
6 same material as the cover and carrying plies of the tensor skin, as listed in Table 1 and both have
7 thickness 2.5mm. The thickness of the honeycomb core is 25mm, and core density is $184 \text{Kg} \cdot \text{m}^{-3}$
8 [18-20].

9 Around each plate, the same frame made of mild steel with density $7800 \text{Kg}/\text{m}^3$ is built,
10 which constrains the displacement in the x and y direction, and the rotations in all the x , y , z
11 directions.

12 **3.2 Model in the Euler domain**

13 The Euler domain can be divided into two parts, *i.e.*, air and water. The density of air is
14 $1.24 \text{Kg}/\text{m}^3$ with specific internal energy $1.9 \times 10^5 \text{kcal/kg}$ and $\gamma = 1.4$. The volume of air is
15 assumed to be $10 \times 10 \times 2 \text{m}$ as shown in Fig.6(a). The volume of the water being impinged is
16 $10 \times 10 \times H$, where H is the depth of water, which varies in computing cases. The water is
17 assumed to be ideal fluid with density $1000 \text{Kg}/\text{m}^3$. The region of water is discretized by solid
18 elements (CHEXA in DYTRAN) as shown in Fig.6(a). On all the 4 side surfaces parallel to the
19 impact direction, **non-reflective** free boundary conditions are applied; while the bottom surface is
20 fully constrained.

(a) The whole calculation region and (b) the water region

Fig.6 The FE model of Euler domain

3.3 Definition of fluid-structure interaction

The contact between the impinging plates and the water is defined according to the coupling surfaces with interactive failure [21]. The aim of considering the fluid-structure interaction by using the coupling algorithm is to enable the objects modeled in Eulerian and Lagrangian meshes to interact. Here, the solid structure is discretized by Lagrangian meshes and the water is discretized by Eulerian meshes. The first task in coupling is used to transfer the forces between the two solver domains. In the same time, the stresses in the Eulerian elements produce forces to act on the coupling surface and distort the Lagrangian elements.

The coupling surface must form a closed and positive volume to meet DYTRAN's internal requirements [21]. In order to create a closed volume, it is necessary to add dummy shell elements. Since the additional grid points created for the dummy elements do not move as they are not connected to any structural elements, it should be large enough to cover the whole deformation range of the structure; otherwise the negative volume produced by large deformation would terminate the simulation.

It is not necessary to build additional coupling surface for the honeycomb-core sandwich plate, because its' top surface, bottom surface and cell wall of the core have already formed a

1 closed volume. However, for the tensor skin and the solid plate, dummy shell elements need to be
 2 created to get a closed volume. By taking the tensor skin as an example, the definition of the
 3 coupling surface is illustrated as follows. The height of the coupling surface adopted is larger than
 4 the maximum deflection of tensor ply, as shown in Fig.7. Meanwhile, at least one element must
 5 intersect with the Eulerian elements to transfer the forces between the two domains.

6
 7 Fig.7 The definition of the fluid-structure interaction for the tensor skin

8 In the previous theoretical analysis, the central deflection of the tensor skin, which is fully
 9 clamped and subjected to pressure, is predicted. In the current FE model, the plate is supposed to
 10 impinge the water with a given initial velocity v_0 . In order to make the results of FEM and
 11 theoretical predictions comparable, the relative deflection is defined as $u_r = u_m - u_b$, and the
 12 corresponding normalized deflection is defined as $\bar{u} = u_r / l$, where u_m is the displacement of the
 13 plate center and u_b is the displacement of the boundary, as shown in Fig.8.

14
 15 Fig.8 The definition of the normalized central deflection

4. Comparison between three kinds of plates

The dynamic responses of three kinds of plates, *i.e.*, solid plate, honeycomb-core sandwich plate and tensor skin with the same length and width 1.5m, are compared with each other. Since all the three plates have the same total mass, 18.45Kg, the same initial velocity will result in the same initial momentum. The ratio of the water depth to the plate length is H/l . Here $H/l = 5$ is firstly adopted and the tensor skin with $k = 0.5$ is taken as an example in calculation. Three different initial velocities, $v_0 = 10, 15, 20$ m/s, are taken into account. DYTRAN can export the interaction pressure, which acts on the Eulerian elements intersecting with coupling surface. It is an average value considering the overall effect.

The deformation profiles of three different plates with same initial momentum, $I = 276.75$ Ns, are depicted in Fig.9. As seen from this figure, the solid plate is damaged shortly after impact, whilst no damage takes place in the tensor ply of the tensor skin.

Fig.9 The deformation profiles of three plates at different time instants ($I = 276.75$ Ns)

The normalized central deflection for different plates are shown in Fig.10, for the same initial momentum $I = 184.5$ Ns. As demonstrated in this figure, the tensor skin has the largest central

1 deflection, i.e., $\bar{u} = 0.65$ due to the unfolding process of the tensor ply. The core compression of
 2 the sandwich plate leads to a small central deflection, $\bar{u} = 0.033$. The solid plate has the smallest
 3 one, $\bar{u} = 0.004$, since its deformability is limited before the rapid breakage after the water impact.
 4 The solid plate is safe for the applied momentum smaller than 138.38Ns, while for the
 5 honeycomb-core sandwich plate and tensor skin, the critical momenta are 369Ns and 664.2Ns,
 6 respectively.

Fig.10 The normalized central deflection ($I = 184.5\text{Ns}$.)

9 The interaction pressure between water and plates are shown in Fig.11. Clearly, the intensity
 10 of the impact increases with the initial impact velocity for all three kinds of plates. For the solid
 11 plate, as illustrated in Fig.11(a), after the instant of the first contact to water, the interaction
 12 pressure first increases and then decreases rapidly. The pulse shape is very close to a triangular
 13 one. For the honeycomb-core sandwich plate, as shown in Fig.11(b), a short plateau of interaction
 14 pressure appears after the contact, and the pulse shape is not far from a rectangular one. Since the
 15 honeycomb-core sandwich absorbs energy mainly by the core's compression, the peak value of
 16 interaction pressure is clearly reduced. The plateau of the interaction pressure is also found in the
 17 pulse generated by the tensor skin, as shown in Fig.11(c), which is much lower and longer though.
 18 The pulse shape for the tensor skin is more like a trapezium pulse. This phenomenon can be

1 explained as follows. The cover ply and the carrying ply break immediately after impact with water,
 2 and then the unfolding of tensor ply significantly reduces the peak of the contact pressure, while
 3 extends the pulse duration in the same time.

Fig.11 The interaction pressure between the plate and water

9 The interaction pressure of three kinds of plates are compared in Fig.11(d), for
 10 $I = 184.5\text{Ns}$. The peak pressures of the honeycomb-core sandwich plate and tensor skin are only
 11 33% and 25%, respectively, of that of the solid plate, whilst the pulse duration varies just in the
 12 reversed manner.

13 From the above comparison, it is concluded that the interaction pressure between the water
 14 and the plate depends on the structure configuration due to the effect of the fluid-structure

1 interaction. Therefore, the fluid-structure interaction has to be adequately considered in the
2 subsequent analysis of the dynamic response of various kinds of structures to water impact.
3 Compared with the solid plate, the impact pulses applied to the honeycomb-core sandwich and the
4 tensor skin are more moderate, as shown in Fig.11(d), which is good for their energy absorption.
5 Obviously, the energy absorbing capacities of the tensor skin and the honeycomb-core sandwich
6 plate are both superior to the solid plate of the same mass, especially when the medium interaction
7 is considered. It should be noted that there is almost no air existing between the plate structure and
8 water during the impact process, as discussed in paper [17].

9 **5. Effects of the water depth**

10 Previous studies indicated that an impact on water would produce almost the same effect to
11 an ordinary structure as an impact on a hard surface if the water is deep enough [22]. So the effect
12 of water depth should be investigated. In this section, first the distribution of particle velocity
13 along the water depth direction is investigated for a rigid plate impinging on water. Secondly, the
14 dynamic responses of the three kinds of plates will be compared with each other when they
15 impinge water of different depth.

16 **5.1 Rigid plate impinging on water**

17 Suppose a rigid plate with size $1 \times 1m$ and material density $\rho = 7800 Kg/m^3$ impinges on
18 water at the initial velocity $20m/s$.

Fig.12 The interaction pressure of a rigid plate impinging on water with different depth

As shown in Fig.12, the peak of the interaction pressure increases with water depth and the time duration decreases with water depth. No reflection from bottom surface has been observed during the time stage of 0~0.02ms. The particle velocities at different distance below the water surface h' are plotted in Fig.13. Clearly, after impact, a disturbance is propagating along the depth direction in the water. In the same time, the intensity of the disturbance is decreasing since the peak of the particle velocity decreases with h' . At the end of the impact pulse, i.e., 0.01ms for $H=5m$ as shown in Fig.12, the disturbed water depth is about 0.7m. The disturbance will reach the bottom at about 0.05ms.

Fig.13 The particle velocity at different distance below the water surface ($H = 5m$)

5.2 Solid plate, honeycomb-core sandwich and tensor skin impinging on water

Suppose all three different plates have the same length and width $1m$, the same mass and initial velocity $20m/s$, thus the same initial momentum $195.2Ns$. The central deflections of different plates impinging on water of different depth H/l are plotted in Fig.14.

Fig.14 The effect of water depth on the normalized center deflection

As shown in Fig.14, the solid plate and the honeycomb-core sandwich are all damaged after impinging on water of the current depth. Their final central deflection is independent from the water depth, and the total response time decreases with water depth. In comparison, the tensor skin is not damaged after impact, and its final deflection increases with the water depth. For water of low depth, that is low H/l , curves are fluctuating with time which is estimated to be caused by

1 the reflection waves from the bottom. While the responses of these plates after impinging on deep
 2 water of $H/l \geq 5$ tend to agree with the respective theoretical predictions. Compared with the
 3 FE simulation, the errors in the central deflection predicted by the theoretical models decrease
 4 with the water depth, as shown in Fig.14(d). It is smaller than 8% for $H/l = 5$.

Fig.15 The energy transferred to water affected by water depth

7 Fig.15 shows that the proportion of the total energy transferred to the water is affected by
 8 water depth. Here the energy transferred to the water is calculated by initial kinetic energy K_0
 9 minus the plastic dissipation of the plate D . As demonstrated in this figure, the energy transferred
 10 into water by the tensor skin is smaller than the other two plates. The proportion of the energy
 11 acquired by deep water is found to be 46.2%, 28.9% and 19.9% for the solid plate, the
 12 honeycomb-core sandwich and the tensor skin, respectively. Again the plastic energies dissipated
 13 by the tensor skin and the honeycomb-core sandwich are larger than that of the solid plate.

14 6. Verification of the theoretical predictions for tensor skin

15 In the theoretical analysis of tensor skin[13], the water impact pulse was simplified as a
 16 rectangular pulse or a triangular pulse. In order to verify the theoretical predictions, the dynamic
 17 response of the tensor skin impinging on water is simulated by the fluid-structure interaction FE
 18 model, as well as the pulse loading FE models, which do not consider the fluid-structure

1 interaction.

2 **Theoretical prediction**: The tensor skin with $k = 0.5$, width and half length $l = 1$, total
3 mass $18.45Kg$ is studied here. With a triangular pulse and a rectangular pulse ($\bar{q} = 1.3\bar{q}_{s0}$ and
4 $\bar{t}_d = 1$) are respectively applied, the results are shown in Fig.4. The corresponding value of this
5 impulse is $I = 240.1Ns$.

6 **FE with Pulse loading**: The same tensor skin is full clamped at the both ends, subjected
7 to the above mentioned rectangular pulse and triangular pulse in the theoretical predictions.

8 **FE with Fluid-Structure interaction**: The same tensor skin impinges water with initial
9 velocity of $v_0 = 13m/s$, where the initial momentum is equal to the applied impulse of the above
10 two cases, $mv_0 = I$. Here the water depth is taken as $H/l = 5$.

(a) Normalized central deflection

(b) Normalized kinetic energy

Fig.16 The dynamic response of tensor skin ($k=0.5$) under impact of $\bar{I} = 1.3\bar{q}_{s0}, \bar{t}_d = 1$

The dynamic responses predicted by above three models are compared in Fig.16. The normalized central deflection and the kinetic energy $\bar{K} \equiv K/M_p$ obtained by FEM simulations are all lower than that by the theoretical predictions, but the discrepancies are all small. In the theoretical predictions, the total impulse applied is taken as $I = mv_0$, that is, it is equal to the initial momentum of the plate in the interaction FE model. In fact, the total initial momentum is transferred to the plate as well as the water, which could be confirmed by the energy conservation, as shown in Fig.15. However, the deviation of about 10% indicates the simplified theoretical model is acceptable.

7. Conclusions

In this paper, in order to verify the previous theoretical prediction of tensor skin under water impact, the fluid-structure interaction models are constructed for a solid plate, a honeycomb-core sandwich plate and a tensor skin.

It is found that the interaction pressure varies with the configuration of the plates. The interaction pressure generated by the solid plate is recognized as a triangular pulse; and those generated by the honeycomb-core sandwich and the tensor skin are close to a rectangular pulse and a trapezium pulse, respectively. Compared with the solid plate, the loading pulses on the honeycomb-core sandwich plate and the tensor skin are more moderate, and their energy absorbing capacities are confirmed to be superior to the solid plate of the same mass.

From a study of the water depth effect, it is found that the theoretical predictions for the tensor skin are appropriate for deep water case. The final deflection obtained by the FE model

with fluid-structure interaction is smaller than those given by the pulse loading FE model and the theoretical prediction, while the difference is estimated as 10%. Thereby, the theoretical model and the pulse approximation method are **verified** by the FE simulation with a consideration of the fluid-structure interaction.

Acknowledgements

The authors gratefully acknowledge the financial support from the National Science Foundation of China No.11372163, the National Fundamental Research Program of China through Grant No.2010CB832701 and No.2011CB610305. The 3rd author (T.X. Yu) would like to thank the support from the NSFC Key Project No. 11032001.

References

1. Thuis, H.G.S.J. and Wiggenraad, J.F.M., *A tensor skin concept for crashworthiness of helicopters in case of water impact*. 1994, National Aerospace Laboratory NLR.
2. Labeas, G. and Kermanidis, T.H., *Crushing behavior of the energy absorbing 'tensor skin' panels*. Fatigue Fract Engng Mater struct 2003. **26**: p. 449-457.
3. Thuis, H., Vries, H. and Wiggenraad, J., *Subfloor skin panels for Improved Crashworthiness of Helicopters*. 1999, National Aerospace Laboratory NLR. p. 1-16.
4. Michielsen, A.L.P.J., Wiggenraad, J.F.M., Ubels, L.C., Frijns, R.H.W.M., Kohlgrüber, D., Labeas, G. and McCarthy, M.A., *Design, Test and Analysis of Tensor Skin Panels for Improved Crashworthiness in Case of Water Impact*, in the *AHS Crashworthiness Specialists' Meeting "Crash Safety Challenges and Innovative Solutions"*. 1998, National Aerospace Laboratory: Phoenix, Arizona. p. 1-10.
5. Ubels, L.C., Johnson, A.F., Gallard, J.P. and Sunaric, M., *Design and testing of a composite bird strike resistant leading edge*, in *The SAMPE Europe Conference & Exhibition*. 2003, National Aerospace Laboratory Paris, France. p. 1-16.
6. Huis, H.G.S.J., Vries, H.P.J.D., and Wiggenraad, J.F.M., *Sub-floor skin panels for improved crashworthiness of helicopters in case of water impact*. in *Annual Forum Proceedings - American Helicopter Society*, v 1. 1995.
7. McCarthy, M.A. and Wiggenraad, J.F.M., *Numerical investigation of a crash test of a composite helicopter sub floor structure*. Composite Structures, 2001. **51**: p. 345-349.
8. Kermanidis, Th., Labeas, G., Apostolopoulos, C. and Louis Michielsen, *Numerical simulation of composite structures under impact*. International Conference on Structures Under Shock

- 1 and Impact, 1998. **32**: p. 591-600.
- 2 9. McCarthy, M.A., Harte, C.G., Wiggenraad, J.F.M., Michielsen, A.L.P.J., Kohlgrüber, D. and
3 Kamoulakos, A., *Finite element modeling of crash response of composite aerospace subfloor*
4 *structures*. Computational Mechanics, 2000. **26**: p. 250-258.
- 5 10. Labeas, T.K.G. and Ubels, M.S.L., *Development and Validation of a Novel Bird Strike*
6 *Resistant Composite Leading Edge Structure*. Applied Composite Materials, 2005. **12**: p.
7 327-353.
- 8 11. Kermanidis, Th, Labeas, G., Sunaric, M., Johnson, A.F. and Holzapfel, M., *Bird strike*
9 *simulation on a novel composite leading edge design*. International Journal of
10 Crashworthiness, 2006. **11**(3): p. 189-202.
- 11 12. Mi, B.W. and Zhao, M.Y., *Analysis of anti-bird strike for composit material tensor skin wing*
12 *structures*. Computer Simulation, 2010. **27**(2).
- 13 13. Ren, Y.T., Qiu, X.M. and Yu, T.X., *The sensitivity analysis of a geometrically unstable*
14 *structure under various pulse loading*. International Journal of Impact Engineering, 2014. **70**:
15 p. 62-72.
- 16 14. Ren, Y.T., Qiu, X.M. and Yu, T.X., *Theoretical analysis of the static and dynamic response of*
17 *tensor skin*. International Journal of Impact Engineering, 2014. **64**: p. 75-90.
- 18 15. Youngdahl, C.K., *Influence of pulse shape on the final plastic deformation of a circular plate*.
19 International Jounal of Solids Structure, 1971. **7**: p. 1127-1142.
- 20 16. Li, Q.M. and Jones, N., *Foundation of correlation parameters for eliminating pulse shape*
21 *effects on dynamic plastic response of structures*. American Society of Mechanical Engineers,
22 2005. **72**: p. 172-176.
- 23 17. Bin, X., Zhen, C. and Xi, X., *The simulation analysis of the slamming of elastic flat-bottom marine*
24 *structure*. China Offshore Platform, 2005. **20**(1): p. 22-28.
- 25 18. Yi, T. and Chen, C.Q., *The impact response of clamped sandwich beams with ordinary and*
26 *hierarchical cellular cores*. International Journal of Impact Engineering, 2012. **47**: p. 14-23.
- 27 19. Qiu, X., Deshpande, V., and Fleck, N., *Finite element analysis of the dynamic response of*
28 *clamped sandwich beams subject to shock loading*. European Journal of Mechanics-A/Solids,
29 2003. **22**(6): p. 801-814.
- 30 20. Tang, Y. and Xue, M., *Thermo-mechanical characteristics analysis of sandwich panel with*
31 *honeycomb core*. Acta Materiae Compositae Sinica, 2005. **22**(2): p. 130-136.
- 32 21. *Dytran™ 2012 User's Guide*. 2012.
- 33 22. Korobkin, A., *Analytical models of water impact*. European Journal of Applied Mathematics,
34 2004. **15**(06): p. 821-838.