OPTIMIZATION OF PROTECTIVE CONTAINER SUBJECTED TO THERMAL AND IMPACT LOADS

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ABSTRACT

Optimization technology based on numerical simulation is applied to minimize the weight of air transportation protective container. Thermal and structural numerical calculations of a spherical protective container subjected to high-speed impact accident simulations are was used to determine the optimal thickness of container's layers.

INTRODUCTION

In accordance with International Atomic Energy Agency (IAEA) regulations, a protective container for air transportation of radioactive materials (a Type C package), must meet certain strict requirements. The Type C package must be strong enough to withstand an impact on a hard surface at any angle and at a speed of at least 90 meters/second (m/s), and at the same time the package must withstand a 1000°C fire of at least one-hour duration. The problem to prepare an optimal design meeting minimum weight and strength requirements is significant problem for designers.

Thermal-resistant and impact mitigating layers of the container can be made of different materials such as aluminum honeycombs, polyurethane foam [1], Kevlar [2], etc. However, it is well known that natural wood (pine, birch, redwood, etc.) has excellent heat resistance and impact absorbing properties due to its cell structure. A small PAT-2 package for air transportation of Pu-234 samples designed and tested in the USA [3]. Maple and red wood were used to provide thermal and impact protection for this container.

PROBLEM STATEMENT

In an air disaster, the air transport container can be subjected to high-level thermal and impact loads. If the impact speed, temperature and duration of a fire are chosen as primary air crash parameters, then the optimization problem may be analyzed as presented below.

It is necessary to determine the optimal geometric parameter - thickness of the protective layers of the container that meets all requirements and a minimum weight. The optimization problem may be expressed as follows:

To find:

$$\min W = W(h_1, h_2, \ldots, h_n)$$

$$H(h_1, h_2, \ldots, h_n) \in D \qquad \text{(Eq. 1)}$$

$$D = \{H: g_\kappa(H) \le 1,\ \kappa = 1, 2, \ldots, m\}$$

Where:

- W – criterion function – weight of container,
- H – vector of n unknown parameters defining design space dimension - n,
- D - solution domain which is defined by functional constraints $g_k(H) \le 1$.

Under thermal and impact loads, functional constraints are defined as thermal (Eq. 2) and impact (Eq. 3) conditions:

$$g_1 = \frac{T_{max}}{T_{lim}} \le 1,$$
$$T_{max} = T(H, X, t). \qquad \text{(Eq. 2)}$$

Where:

- T_{max} – maximal temperature at a given container's element during and after the fire,
- T_{lim} – limiting value of temperature,
- X – geometry vector defining a point at given coordinate system,
- t – time.

$$g_\kappa = \frac{\varepsilon_{max}^{(i)}}{\varepsilon_{np}^{(i)}} \qquad k = 2, 3, \ldots, n,$$

$$\varepsilon_{max}^{(i)} = \varepsilon(H, X, t). \qquad \text{(Eq. 3)}$$

Where

- $\varepsilon_{max}^{(i)}$ – maximal strain in container element number i,
- $\varepsilon_{lim}^{(i)}$ – ultimate value of strain.

To define the maximal temperature T_{max}, a nonlinear thermal conductivity equation has to be solved:

$$\rho C \frac{\partial T}{\partial t} - \nabla\lambda\,\nabla T = Q \qquad \text{(Eq. 4)}$$

With initial and boundary conditions:

$$T\,(X, t_0) = T_0\,(X), X \in \Omega \qquad \text{(Eq. 5)}$$

$$-\lambda\left(\frac{\partial T}{\partial t}\right)_{\Gamma} = \alpha\left(T - T_{\alpha}\right), X \in \Gamma \qquad \text{(Eq. 6)}$$

Where

- Ω, Γ – volume and exterior surface of the container,
- $C = C\,(X, T, t)$– specific heat,
- $\lambda = \lambda\,(X, T, t)$ – thermal conductivity,
- $\rho = \rho\,(X, t)$ – density,
- α, T_0, T_a – heat-transfer factor, initial and ambient temperatures,
- $Q = Q\,(X, T, t)$ – internal heat source.

To define maximal strain $\varepsilon_{max}^{(i)}$ the motion equation, based on Lagrange principle, has to be solved:

$$\int_{\Omega}\sigma_{ij}\,\delta\varepsilon_{ij}\,d\Omega - \int_{\Omega}\left(F_i - \rho\ddot{u}_i\right)\delta u\;d\;\Omega - \int_{\Gamma}P_i\,\delta u_i\,d\Gamma = 0\,, \qquad \text{(Eq. 7)}$$

Equation (7) together with kinematical equations, equation of state, initial and boundary conditions make a total system of equations. Certain symbols are assumed in equation (Eq. 7). They are:

- σ_{ij}, ε_{ij} – components of stress and strain tensors,
- u_i – displacements,
- F_i, P_i – volumetric and area forces acting in Ω volume and on Γ surface area.

To solve optimization problem (Eq. 1), it is necessary to move towards the point H^* in the design space. On each moving step $\Delta H = \Delta H\,(\Delta h_1, \Delta h_2, \ldots, \Delta h_n)$ in the design space, it is necessary to solve the total system of nonlinear equations (Eq. 4) – (Eq. 7), and this could be very expensive algorithm. The optimization problem can be divided into a structural optimization problem and a parametric analysis. Let us assume that the structural problem has already been solved. As a result, the number of protective layers, their shapes and materials are defined. Then the parametric analysis to optimize the thickness for each protective layer can be solved using parametric optimization methods [4] and modern thermal and dynamic stress analyses codes for the solution of a nonlinear thermal conductivity transient problem and dynamic elastic-plastic deformation problem. The solution of nonlinear problems at each optimization step takes significant computer power, and it can be expensive way to create an optimal design. Therefore, from the practical point of view, it is of interest and useful to develop a methodology for parametric optimization for a given container scheme. As an example, we consider spherical shape of protective container consisting of several protective layers.

PARAMETRIC OPTIMIZATION

It is assumed that container is of spherical shape and includes three layers. The first layer is the internal case of the container; the second one is a thermal resistant and impact absorbing layer; and the third is the outer case of the container. The thickness of each layer divided by radius of internal contents *R* would be a correspondent design parameter.

$$h_i = \frac{\delta_i}{R}, \qquad i = 1, 2, 3.$$

Where *R* – radius of container's content. The accident conditions are:

1) Fire including two stages:
   - During the fire: $\alpha = 145$ watt/m^2, $T_a = 1000$˚C, $t = 1$ hour;
   - After the fire: $\alpha = 25$ watt/m^2, $T_a = 20$˚C, $t = 30$ hour.

2) Impact with hard surface at a speed of 90 m/s.

It is assumed that outer and inner cases are made of stainless steel and wood is used for the thermal-resistant and impact absorbing layer. To solve the thermal and dynamic problems UPAKS-T [5] and Dinamika-2 [6] program facilities were used.

THERMAL ANALYSIS

As the first stage, a parametric analysis of the thermal state is performed to determine how the h_i design parameters impact the maximum temperature of the container's contents. The temperature curves versus time are presented in Fig. 1 for variant *H* (0.3, 0.9, 0.05). These results are typical for such problems. Curve 1 is the temperature on the wood layer outer surface, and Curve 2 is the temperature on the inner case. It can be seen from Fig. 1 that temperature on the outer surface of wood increases rapidly during the fire and reaches a level of 981˚C. After the fire, the temperature decreases quickly and returns to the pre-fire level. The temperature profile within the inner case has a different behavior. It reaches a maximum of 100˚C during the fire. After the fire, the temperature continues to increase up to 152°C (2.6 hours after the end of the fire) and then slowly decreases.

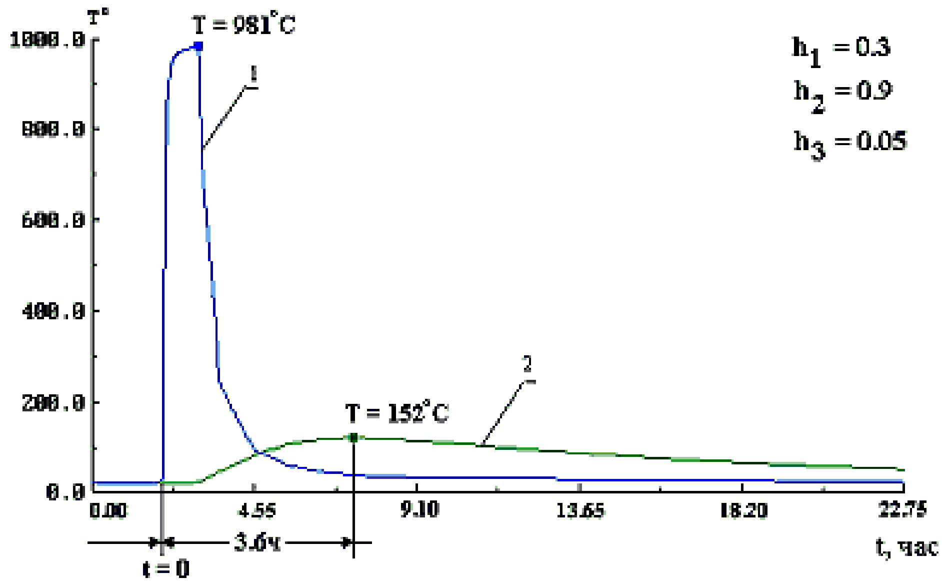

Fig. 1 Temperatures in the wood layer (1) and on the inner case (2)

To define a peak value of temperature T_{max} inside the inner case, it is necessary to evaluate the thermal lag of the container. It is evident that thickness of the thermal-resistant layer h_2 strongly impacts T_{max}. To determine impact, it is necessary to start the parametric analysis with the investigation of thefunctional dependence of $T_{max} = T_{max} (h_2)$.

As an initial estimate, we set the project H_0 with $h_1 = h_3 = 0.05$ and $h_2 = 0.60$ and investigate on how value T_{max} changes if h_2 increases. The functional dependence $T_{max} = T_{max} (h_2)$ has been determined by numerical simulations as shown in Fig. 2 (Curve 2). As it can be seen from Fig. 2, there is a range of h_2 where T_{max} strongly decreases. But for the value of h_2 greater then 1.6, T_{max} decreases slightly. If it is assumed that the temperature of 200˚C is an acceptable value, then the thickness of the thermal-resistant layer can be restricted by $h_2 = 0.90$.

The functional dependences $T_{max} = T_{max} (h_1)$, Curve 1 and $T_{max} = T_{max} (h_3)$, Curve 3, for $h_2 = 0.9$ = constant, are shown in Fig. 2. It can be seen from Fig. 2 that if the thickness of the inner case h_1 increases from 0.1 to 0.3, then temperature T_{max} decreases from 200˚C to 150˚C. The lowering of T_{max} can be explained by the inner case mass increasing. On other hand, if the thickness of the outer case h_3 increases, T_{max} increases from 200˚C to 260˚C. The phenomena can be explained by energy accumulation in the outer steel layer of container during the fire. The thicker outer case absorbs more energy during fire. After the fire, the absorbed heat energy accumulated on the outer case increases T_{max}. The parametric analysis shows that the project H^* (0.30, 0.9, 0.05) provides an optimal design for T_{lim} = 200˚C.

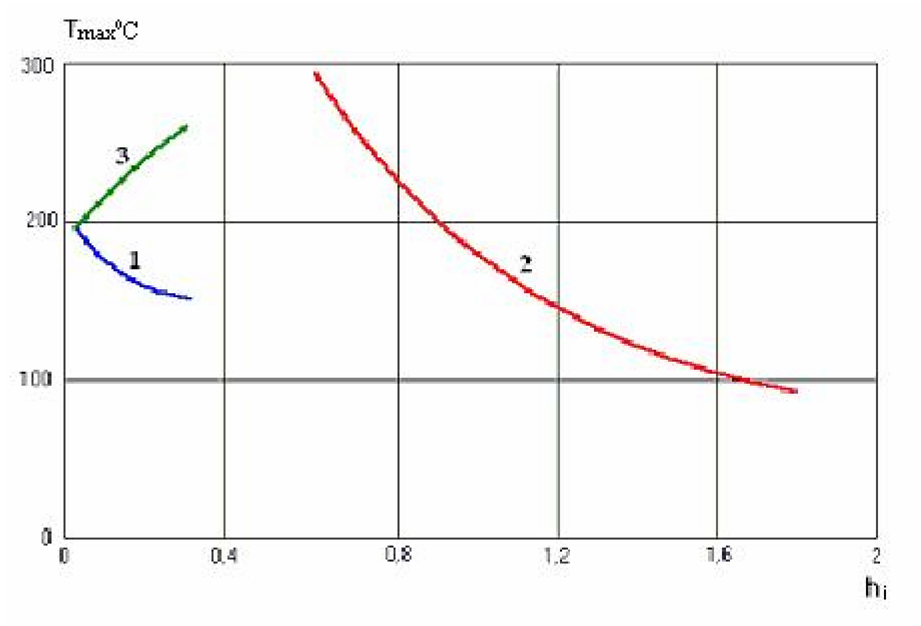

Fig. 2 Maximal temperature functional dependences on the thickness

DYNAMIC STRESS ANALYSIS

After the thermal analysis, it is necessary to investigate the dynamic deformations of the container over impact with hard surface at a speed of 90 m/s. In this case it is of greater interest to study the effect of varying the outer case thickness. Therefore, numerical calculations of container's deformation were carried out with values h_3 equal to 0.05, 0.1 and 0.2 and holding constant the values $h_1 = 0.3$ and $h_2 = 0.9$.

Some computational results are shown in Fig. 3. The results show that container deceleration time Δt weakly depends on the outer case thickness and equals to Δt ~1.2…1.4 ms. Maximum deceleration values are similar for different values of thickness h_3 and are about ~10^4g.

Deformed configurations of different variations of container are presented in Fig. 3. In each variation, the container changes from an initial spherical shape to a flat deformed shape in the impact area and results in large local distortion of the heat-resistant wood layer. Also, there are no plastic deformations in the inner case.

As the thickness h_3 goes up from 0.05 to 0.2, the kinetic energy of the inner case of container also increases (~4 times). At the same time, the deformation of the container Δ increases from 0.51 to 0.74. The maximal level of the outer case strain is located in the flex point area. In addition, tensile circumferential strains dominate on the outer surface. Also, the compressive longitudinal strains dominate on the inner surface.

The numerical results show that if the outer case thickness goes up from 0.05 to 0.2, tensile circumferential strains increases from 9% to 11% and the outer casing weight doubles. The outer case with $h_3 = 0.05$ has the least kinetic energy, and the impact absorbing wood layer deforms less then in the other case. The results indicate that an outer case thickness $h_3 = 0.05$ is the most optimal.

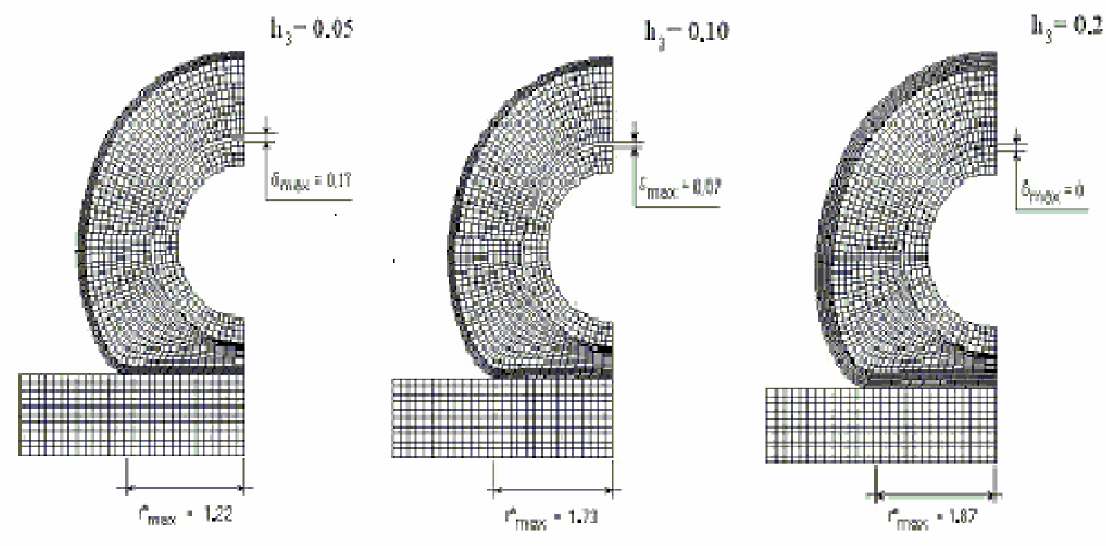

Fig. 3 Deformed configurations of different variations of container

CONCLUSIONS

On the basis of implemented numerical investigations, the following conclusions can be made:

1. The variant *H** (0.30, 0.9, 0.05) is an optimal design for fire and impact with a hard surface. This design results in a maximum of temperature of inner case of 150˚C and is less than design temperature of 200˚C. The outer case with $h_3 = 0.05$ has the least kinetic energy and the impact absorbing wood layer deforms less then when h_3 is larger.

2. Maximum temperature of deformed state of the optimal container increases by only 5 to 6% as compared to analysis of un-deformed configuration of the container.

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