Fuel Tank Wall Response to Hydraulic Ram during the Shock Phase

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Introduction

THIS Note outlines some preliminary work accomplished in a research program at the Naval Postgraduate School to study the effects of hydraulic ram in fuel tanks upon aircraft survivability and vulnerability. The problem of predicting the interaction of a high speed projectile with a liquid filled tank (hydraulic ram) is a complex but necessary study if kill probabilities are to be predicted for this type of event. Furthermore, to design survivable fuel tanks, it is necessary to understand hydraulic ram in detail. The hydraulic ram phenomenon may be divided into two phases: 1) the shock phase where the projectile exchanges some of its energy with the fluid by the production of a strong, nearly hemispherical shock wave originating from the entry wall, and 2) the cavity phase where further energy is imparted to the fluid by the production and collapse of a vapor filled cavity formed by separation of the fluid from the surface of the projectile as it moves through the tank. This Note considers the shock phase and examines the structural response of the entry wall during this phase.

Analysis

The shock phase fluid motion was determined by using the method of Yurkovich.[1] This method assumes a hemispherical, strong shock produced by a point energy release. The shock moves radially from the projectile impact point and loads the entry wall with a pressure pulse that can cause catastropic failure. The pressures resulting from such a flow are determined by using the Rankine-Hugoniot equations and the Tait equation of state for liquids to determine the shock front properties as a function of shock Mach number. Once the shock front properties are known, the pressure variation behind the shock is determined by assuming a power law density profile behind the shock so that the equations of motion may be integrated assuming a constant shock decay coefficient. The initial shock radius and Mach number are then calculated by an integration of the energy density behind the shock wave. These calculations were used to determine rigid wall pressure loadings as a function of time. It was assumed that the resulting pressure loadings could be approximated by:

$$p(r,t) = p^* C_1^{\left(\frac{m+q-1}{n}\right)} (r/C_2)(t_1/t) \qquad 0 < t < t_1 C_1^{-1/n}$$

and

$$p(r,t) = p^* \left(\frac{t}{t_1}\right)^{-m} \left\{1 - C_1 \left(\frac{t}{t_2}\right)^{n} \left[1 - \frac{r}{C_2 (t/t_3)^q}\right]\right\} \qquad t_1 C_1^{-1/n} \le t \le \infty$$

where $1 \ge \{\ \} \ge 0$ and the constants depend upon the projectile initial energy, the percentage of this energy lost during the shock phase, and the tank liquid. This form corresponds to a peak pressure that varies as $p^* (t/t_1)^{-m}$ with a linear decay behind the front to zero pressure. The location of the front is given by $C_2(t/t_3)^q$. Figure 1 shows a

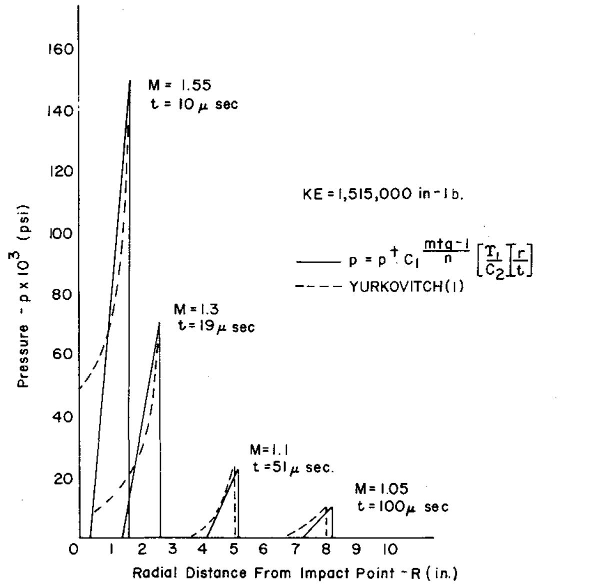

Fig. 1 Shock pressure distribution for 0.50 caliber projectile impacting H_2O.

plot of the pressure as a function of radius at several times for a 50 caliber bullet with 151,500 in. lb energy impacting a water filled tank. Also shown is the approximate pressure distribution that was used for the wall loading.

The wall response to the shock wave shown in Fig. 1 was computed using the digital computer program SATANS.[3] SATANS is an algorithm for the geometrically linear and nonlinear static and dynamic response of arbitrarily loaded shells of revolution. The governing equations are based upon Sanders' nonlinear thin shell theory for the condition of small strains and moderately small rotations. The program's applicability to the present problem is limited due to the neglect of shear effects and material nonlinearities. However, it does provide an estimate of the general nature of the response.

The wall selected for analysis is a circular aluminum plate with a radius of 10 in. and a thickness of 0.1 in. The period of time examined was $0 \le t \le 50$ μsec. Plots of the deflection of the plate at several times are given in Fig. 2. Also shown in Fig. 2 are the corresponding locations of the shock wave front. Note that the bending deformation of the plate does not precede the shock front. However, due to the large displacement the middle surface of the plate stretches, and a much faster traveling membrane force

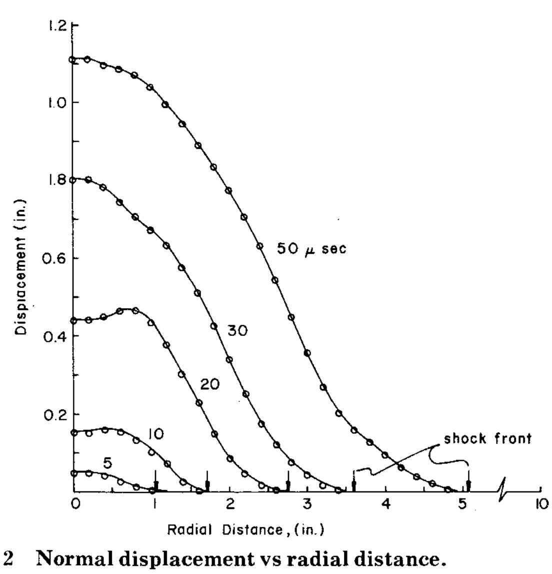

Fig. 2 Normal displacement vs radial distance.

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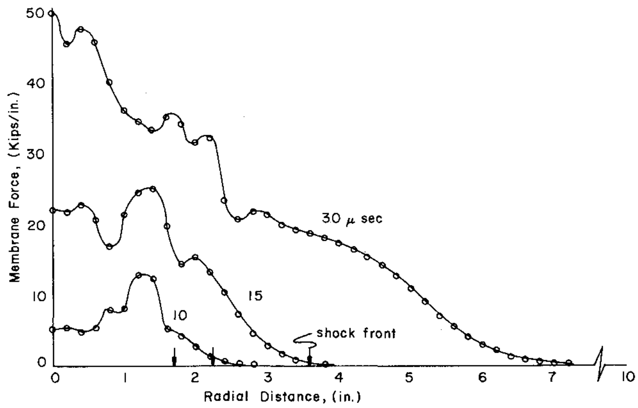

Fig. 3 Membrane force vs radial distance.

propagates along the plate ahead of the front as shown in Fig. 3. Due to the very high energy loading condition and the thinness of the plate, the maximum membrane and bending stresses were approximately 600 and 720 ksi, respectively, and occurred at the center of the plate between 30 and 35 μsec. These stresses are considerably larger than the aluminum can withstand, and considerable cracking and yielding will occur in this situation.

In this analysis the fluid-wall interaction has been neglected entirely, and the response of the wall was computed using the in vacuo structural response equations and the incident pressure. The interaction of the wall and the fluid is a complex phenomenon in which the fluid pressure acting on the wall is a function of the wall motion. There have been many studies of this interaction process for the condition of a flat plate vibrating against a fluid and a submerged shell engulfed by a pressure pulse, where it is assumed that the fluid satisfies the linearized wave equation. A separation of the structure equations from the fluid equations, a considerable simplification of the problem, can be accomplished by applying the so-called piston theory where the fluid response is assumed to be one-dimensional in space normal to the wall. Several studies have been conducted that evaluate the piston theory in specific fluid-structure interaction problems. A summary of these analyses is given in Ref. 2. The results presented in Ref. 2 indicate that the accuracy of the piston theory approximation depends upon the wave lengths involved; hence it depends upon the particular structure and incident pressure arrangement.

Further studies are planned to consider the effects of the fluid-wall interaction. In particular, damping effects will be put in the SATANS code and the applicability of the piston theory will be determined for both the shock phase and the cavity phase of hydraulic ram. Experiments also are being conducted to determine the fluid shock wave position as a function of time using shadowgraph techniques. The percentage of the initial projectile energy lost during the shock phase and the shock wave decay coefficient also will be determined for various entry wall-projectile combinations. These data will be used to substantiate the entry wall incident pressure loadings. Follow-on experiments are being planned to simultaneously obtain holographic interferograms of the fluid and wall to determine wall motion. These studies will yield an accurate theory which predicts entry wall motion and stresses. This information will enable the engineer to predict the vulnerability of various tank designs to the shock phase of hydraulic ram.

References

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Optimal Skin Thickness Variations for Active Cooled Panels

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Nomenclature

C = constant in equation for skin thickness
k = material thermal conductivity
N = exponent in equation for skin thickness
$\dot{q}$ = aerodynamic heat transfer rate
S = half-width of panel section between coolant passages
T = skin temperature
V = volume of skin between coolant passages
ΔT = skin temperature change from panel section midpoint to coolant passage
τ = skin thickness

Subscripts

o = constant skin thickness value
N = panel section midpoint value for τ, or panel section midpoint temperature minus coolant passage temperature for ΔT

ACTIVE cooling of the airframe using the hydrogen fuel heat sink is a promising innovation for practical, long-lived structures for hypersonic aircraft. An attractive active cooling system is an internal convective cooling system which uses a secondary fluid circulated through panel passages to transfer the structure heat load to a centrally located hydrogen fuel heat exchanger.[1] The unit weight of this system structure depends in part upon the desired mean structure temperature and the allowable maximum skin temperature which occurs midway between coolant passages. The purpose of this Note is to show that a potential weight reduction can be achieved by varying the skin thickness and thereby minimizing the skin temperature change between coolant passages.

For the section of a panel illustrated in Fig. 1, if one assumes one-dimensional heat flow with constant thermal properties and sections that are long compared to their width, a simple heat balance for a uniform aerodynamic heat load, $\dot{q}$, gives,

$$\Delta T = T(0) - T(S) = \int_0^S \frac{\dot{q}}{k}\frac{x}{\tau}dx \tag{1}$$

For a panel with constant thickness, τ_0, there results $\Delta T_0 = \dot{q}S_0^2/(2k\tau_0)$. We now ask the question: what thickness variation will yeild a minimum ΔT for a constant skin weight (or volume)? This is a trivial problem in the calculus of variations with the results, $\tau/\tau_0 = (3/2)(S_0/S)(x/S)^{1/2}$ and $\Delta T_{\min} = (4/9)(\dot{q}/k)(S^3/V)$. For $S = S_0$, $\Delta T_{\min}/\Delta T_0$ is then equal to 0.889. Hence, an 11% reduction in ΔT is achieved, but this result is impractical since the skin thickness must be zero at $x = 0$. To determine if a comparable reduction in the temperature change across the panel section can be obtained for other τ variations with nonzero values of τ, the integral, Eq. (1) was evaluated for a class of thickness variations, $\tau = \tau_N[1 + C(x/S)^N]$ with

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