APAE MEMO NO. 136

ARF PROJECT M 526
SHOCK HYDRODYNAMICS OF AN
EXPLODING STEAM PRESSURE VESSEL

FINAL REPORT

CONTRACT NO. AT(11-1)-318

ORIGINALLY ISSUED NOVEMBER 25, 1955

RE-ISSUED AS AN APAE MEMO SEPTEMBER 4, 1958

Alco Products, Inc.
Post Office Box 414
Schenectady, New York

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FOR

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SHOCK HYDRODYNAMICS OF
AN EXPLODING STEAM PRESSURE VESSEL

I. INTRODUCTION

Under the sponsorship of Alco Products, Incorporated, Armour Research Foundation has undertaken a brief survey of the shock problems associated with an assumed failure of a pressurizer vessel in the vapor container of Alco's APPR-1 reactor plant. This project is an extension of consultant service by Mr. F.B. Porzel of the Armour Research Foundation to Alco on August 19, 1955 and has been designated as Armour Research Project No. M 526. The present paper contains more detailed calculations and considerations of the analysis presented at that time and later in a preliminary letter report from Mr. Porzel to Mr. Haines on August 29, 1955.

The purpose of these investigations is to determine, grossly and simply, if a shock hazard exists from an explosive failure of the pressurizer vessel within the vapor container. Because of the geographical location of the reactor, an explosive failure which ruptured the vapor container might release radioactive contamination from the reactor to nearby inhabited communities. The present study is considered more as a generic problem than of a specific design and is concerned with the shock hydrodynamics of an exploding steam vessel.

Substantial parts of the reactor are contained in a large vapor container, a vessel approximately 32 feet in diameter and 64 feet high, of which approximately 50 feet is above ground level; its net volume is approximately 34,000 cubic feet. The pressurizer vessel of special interest is approximately 7 feet high, made up by a cylinder 18 inches in diameter, upon which a sphere 3 feet in diameter is superposed. The pressurizer vessel weighs 4600 pounds and contains 36 cubic

feet of saturated water at 1500 psi pressure and 600°F. Other pressurized vessels of substantial large size and volume are in close proximity but for the present attention has been restricted to the pressurizer vessel. The hydrodynamic problem has been simplified by the assumption of an instantaneous failure of the bolting assembly by which the pressurizer vessel is attached to its base; it is this assumption, stringent from a practical point of view, which permits an explosive decompression of the pressurizer vessel.

The model assumed for these calculations is as follows. Upon rupture at the lower plate, water at the interface is violently accelerated by the pressure difference. The ambient pressure (approximately 15 psi absolute within the vapor container) is signaled within the pressurizer vessel as a rarefaction wave moving upward in the pressurizer vessel. The leading edge of this rarefaction must move toward the top of the pressurizer with the velocity of sound (like 5000 ft/sec) and traverse the pressurizer in slightly over 1 ms. It is this rarefaction which accomplishes most of the acceleration necessary to evacuate the water from the pressurizer vessel. At the base of the pressurizer, this rarefaction wave terminates in a shock wave which is generated in air at a strength determined in Section II which follows; it is this shock which lends meaning to the term explosive decompression, and accomplishes the necessary acceleration of air ahead of the evacuating water. Preliminary calculations indicated that in the absence of energy release by the water, (such as a case like 1500 psi water at room temperature) this shock would be small, like 0.5 psi overpressure. However, upon passage of the rarefaction through saturated water at 1500 psi, the pressure reduction leaves the water partly as superheated steam at conditions to be determined by the shock hydrodynamics. Qualita-

tively, this energy release on flashing results in an appreciable shock at the interface and generally raises the pressure-distance curve during the rarefaction phase.

Several other shock problems are presented. First, the change of state on flashing is an energy release equivalent to many pounds of TNT and constitutes explosion within the vapor container itself. Next, the shock wave generated by the rupture will be locally reflected at various surfaces, enhanced manyfold by the reflection process and present a possible danger to other pipelines and vessels in the near vicinity. Finally, the pressure difference across the pressurizer vessel itself makes it behave as a free missile released within the vapor container.

Preliminary calculations on this problem showed that a shock problem did indeed exist under the rather stringent assumptions assumed for the purpose of this hydrodynamic study. The calculations now confirm these conclusions and it is now suggested that Alco re-examine the practical probability that the pressurizer vessel could instantly and completely fail according to the assumptions initially used. Further study of this problem will be deferred until that re-examination is complete, and at a later time the problem may be reconsidered on the basis of less stringent assumptions, including whatever further considerations the present study suggests.

II. DISCHARGE OF PRESSURE VESSEL AFTER BLOWOUT

The particle velocity u produced by a rarefaction wave causing a pressure drop from p_0 to p_r is

$$u = \int_{p_r}^{p_0} \frac{1}{\rho} \sqrt{\frac{d\rho}{dp}} dp \quad (1)$$

where ρ is the density. The initial state is saturated water at

$$P_0 = 1500 \text{ lb/in}^2$$

The expansion is isentropic and results in a steam-water mixture.

From Keenan-Keyes Steam Tables, the following empirical formula can be obtained for the density along this particular expansion curve:

$$\rho = ap^b + 1 + kp \quad (2)$$

whence

$$\frac{d\rho}{dp} = a(b+1)p^b + k \quad (3)$$

Then the integrand in Eq. 1 becomes

$$I = \frac{1}{\sqrt{k}} \sqrt{\frac{\frac{a}{k}(b+1)p^b + 1}{p(\frac{a}{k}p^b + 1)}} \quad (4)$$

where the fitted empirical constants are:

$$a = 3.89 \times 10^{-14}$$

$$b = 1.86$$

$$k = 1.11 \times 10^{-8}$$

and the units for the hydrodynamic variables are:

$$\rho = \text{lb sec}^2/\text{in}^4 \quad p = \text{lb/in}^2$$

The integrand in Eq. 4 can also be expressed by the empirical equation

$$I = \frac{9500}{p} = 0.969 \times 10^{-7} p^{2.25} \quad (5)$$

From Eq. 1

$$u = \int_{P_0}^{P_F} I dp$$

or

$$u = 9500 \ln \frac{P_0}{P_r} = 626 \left[1 - \left(\frac{P_r}{P_0} \right)^{3.25} \right] \quad (6)$$

The particle velocity produced by a shock wave of strength

$$\xi = \frac{P_r}{P_a} \text{ advancing into atmospheric air } (P_a = 15) \text{ is}$$

$$\frac{u_s}{c_0} = \sqrt{\frac{2}{\gamma}} \frac{\xi - 1}{\sqrt{(\gamma + 1)\xi + (\gamma - 1)}} \quad (7)$$

where $\gamma = 1.4$, $c_0 = 13,700$ in/sec, the ambient sound velocity of air. The shock in air must be of such magnitude as to make the velocity continuous across the original interface; this means a quasi-stable configuration is reached when

$$u = u_s$$

This demand is consistent with shock tube theory for gases.

From Eqs. 6 and 7, we can then equate

$$9500 \ln \frac{P_0}{P_r} = 626 \left[1 - \left(\frac{P_r}{P_0} \right)^{3.25} \right] = 16,400 \frac{\xi - 1}{\sqrt{2.4\xi + 0.4}} \quad (8)$$

where

$$\frac{P_r}{P_0} = \frac{\xi}{\xi + 1} \quad (9)$$

with

$$\xi = \frac{P_0}{P_a} = 100$$

as the initial pressure ratio in the pressure vessel. By substituting

Eq. 9 in Eq. 8, the latter becomes an algebraic equation for ξ ,

which yields

$$\xi = 7.25$$

Thus, the shock advances with a pressure of

$$P_r = \zeta P_o = 110 \text{ lb/in}^2$$

and the discharge velocity of the jet issuing from the pressure vessel is, from either Eq. 6 or 7

$$u = 24,300 \text{ in/sec} \approx 2000 \text{ ft/sec}.$$

The volume of the pressure vessel is $V_o = 36 \text{ ft}^3$. The specific volume of the saturated water at pressure P_o is $v_o = 0.0235 \text{ ft}^3/\text{lb}$. The amount of water initially stored in the pressure vessel is

$$G_w = \frac{V_o}{v_o} = 1600 \text{ lbs}$$

$$\text{with mass } m_o = \frac{G_w}{g} = 4.14 \text{ lb sec}^2/\text{in.}$$

When the bottom of the pressure vessel blows out, a discharge opening of area $A = 254 \text{ in.}^2$ is created. The discharge rate of the steam-water mixture is

$$\frac{dm}{dt} = A \rho u. \quad (10)$$

After isentropic expansion to 110 lb/in.^2 , the composition of the steam-water mixture is 0.7076 parts water and 0.2924 parts steam; the corresponding density is $\rho = 1.26 \times 10^{-6} \text{ lb sec}^2/\text{in.}^4$ or $.835 \text{ lbs/ft}^3$.

The discharge rate is from Eq. 10

$$\frac{dm}{dt} = (254 \times 1.26 \times 10^{-6} \times 2.43 \times 10^4) = 7.75 \text{ lb sec/in.}$$

or alternately,

$$\frac{dm}{dt} = \frac{254}{144} \times \frac{24300}{12} \times .835 = 2950 \text{ lbs/sec.}$$

The time required to discharge the vessel completely is 0.54 sec.

If the pressure vessel tears loose, the thrust developed by the discharging steam-water mixture is

$$\ddot{x} = u \frac{dA}{dt} = A(p_1 - p_2) = 2.12 \times 10^5 \text{ lb}$$

The acceleration imparted to the vessel is

$$\frac{dV}{dt} = \frac{F}{M + m(t)} \quad (11)$$

where $M = 11.92 \text{ lb sec}^2/\text{in.}$ is the mass of the vessel, and $m(t)$ is the variable mass of water in the vessel. From Eq. 10,

$$m = m_0 - \int_0^t \frac{dm}{dt} dt = 4.14 - 7.75t \quad (12)$$

By substituting Eq. 12 in Eq. 11 and integrating, the velocity of the pressurizer vessel is given by

$$V(t) = 2.74 \times 10^4 \ln \frac{2.07}{2.07 - t} \quad (13)$$

At end of discharge ($t = 0.54 \text{ sec}$) the vessel would have attained a velocity of 700 ft/sec.

III. PRESSURE IN CONTAINER AFTER BLOWOUT

The volume of the container is $V_c = 34,000 \text{ ft}^3$ which corresponds to $G_a = 2660 \text{ lb}$ of dry air at NTP. After blowout, the steam-water mixture expands to the full volume of the container. The final specific volume of this mixture is

$$v_2 = \frac{34,000}{1600} = 21.2 \text{ ft}^3/\text{lb}$$

Thus, the final composition of the mixture is given by

$$x_2 v_2' + (1 - x_2) v_2'' = 21.2 \quad (14)$$

where

v' = specific volume of saturated water

v'' = specific volume of saturated steam

x = fraction of water in 1 lb steam-water mixture

and subscripts 1 and 2 refer to the initial and final states respectively;

The mixing of the steam-water mixture with the air in the container occurs at constant internal energy U . When G_a lb air and G_w lb steam-water mixture are mixed, the balance of internal energy (including the equivalent of the work of displacement and of the kinetic energy of the jet) can be written (h = enthalpy)

$$G_a U_a(T_1) + G_w \left[x_1 h_1'' + (1 - x_1) h_1'' + \frac{u^2}{2g} \right] = G_a U_a(T_2) + G_w \left[x_2 U_2'' + (1 - x_2) U_2'' \right] \quad (15)$$

where all terms on the left are known. Equation 15 can be written

$$G_a \left[U_a(T_1) - U_a(T_2) \right] + G_w \left[x_1 h_1'' + (1 - x_1) h_1'' + \frac{u^2}{2g} \right] = G_w \left[x_2 U_2'' + (1 - x_2) U_2'' \right] \quad (16)$$

Air and steam are under their respective partial pressures. The steam is saturated; its partial pressure is saturation pressure. The energy change for the air is

$$U_a(T_1) - U_a(T_2) = c_v(T_1 - T_2)$$

$$T_1 = \frac{V}{G_a} \frac{P_1}{R} \quad T_2 = \frac{V}{G_a} \frac{P_{2a}}{R} \quad \frac{c_v}{R} = \frac{1}{\gamma - 1}$$

$$G_a \left[U_a(T_1) - U_a(T_2) \right] = \frac{V}{\gamma - 1} (P_1 - P_{2a}) \quad (17)$$

Substituting Eq. 17 into Eq. 16 and dividing by G_w gives

$$x_2 U_2'' + (1 - x_2) U_2'' + 9.8 P_{2a} = 790 \quad (18)$$

with p in lb/in.² and U in Btu/lb.

Eliminate x_2 :

from Equation 14

$$x_2 = \frac{v_2'' - 21.2}{v_2'' - v_2'} \quad (19)$$

from Eq. 18

$$x_2 = \frac{U_2'' - 790 + 9.8 p_{2a}}{U_2'' - U_2^0}$$

Equate

$$\frac{v_2'' - 21.2}{v_2'' - v_2^0} = \frac{U_2'' - 790 + 9.8 p_{2a}}{U_2'' - U_2^0} \quad (20)$$

Eliminate p_{2a} (the partial pressure of the air): the ratio of the partial pressures is equal to the ratio of the number of moles of the two components:

$$\frac{p_{2a}}{p_{2w}} = \frac{\frac{G_a}{29}}{(1 - x_2) \frac{G_w}{18}}$$

or

$$p_{2a} = \frac{1.032}{1 - x_2} p_{2w} \quad (21)$$

From Eq. 19

$$1 - x_2 = \frac{21.2 - v_2''}{v_2'' - v_2^0}$$

whence

$$p_{2a} = 1.032 \frac{v_2'' - v_2^0}{21.2 - v_2''} p_{2w} \quad (22)$$

Substituting Eq. 22 into Eq. 20

$$\frac{v_2'' - 21.2}{v_2'' - v_2^0} = \frac{U_2'' - 790 + 10.114 p_{2w}}{U_2'' - U_2^0} \frac{v_2'' - v_2^0}{21.2 - v_2''} \quad (23)$$

Equation 23 can be solved from steam tables; the result is

$$p_{2w} = 9 \text{ lb/in}^2$$

From Eq. 19

$$x_2 = 0.5$$

From Eq. 21

$$p_{2a} = 18.5 \text{ lb/in}^2$$

The total absolute pressure after blowout is

$$p_2 = 27.5 \text{ lb/in}^2$$

IV. SHOCK LOADING OF THE VAPOR CONTAINER

The pressure p_2 of 27.5 lb/in^2 just calculated represents the long-term static pressure which the steam-air mixture within the vapor container will eventually reach, provided the container is not ruptured. A detailed study of the transient pressures on the vapor container walls is beyond the scope of this contract but a reasonable estimate of their magnitude may readily be made on the basis of experience with similar processes in explosions.

It is a characteristic of strong spherical shocks that they propagate with a peak pressure at the shock front which is about double the pressure on their interior. This pressure is of short space and time durations and at high shock strengths does not contribute preponderantly to the over-all energy within the wave. As a consequence, it may be reasonably estimated that the explosion of the pressurizer vessel results in a shock wave of approximately 55 psi absolute as an average over the entire vapor container just before the shock wave reaches the container walls. As mentioned, the pressures would rapidly decay behind such a shock. This incident shock strength has an overpressure of approximately 40 psi and upon collision of this shock wave with the vapor container walls, reflected pressures are achieved on the walls with a pressure multiplication of the incident

shock strength which is a function of both the shock strength and the angle of incidence. However, for angles of incidence up to 40° , this pressure multiplication is not far different from the value calculated for "head-on" incidence. The incident pressure ratio is

$$\xi_1 = \frac{\text{Incident pressure}}{\text{Ambient pressure}} = \frac{55}{15} = 3.67$$

The pressure ratio across the reflected shock at normal incidence is

ξ_2 , and given by

$$\xi_2 = \frac{8\xi_1 - 1}{\xi_1 + 6} = \frac{28.4}{9.7} = 2.9 \quad (24)$$

Hence, the overpressure behind the reflected shock is

$$P_R = (\xi_2 \xi_1 - 1) \cdot 15 = 160 \text{ psi}$$

As a lower estimate, the peak transient pressure on the walls of the vapor container will be in the order of 160 psi. Because of their short duration, they may not result in failure of the upper parts of the vapor container walls. The lower parts of the walls are subject to substantially higher pressures, but are further protected by the secondary shield.

More serious reflected pressures will also occur because of the complex geometry involved, especially near the converging dome at the top of the vapor container. It is possible for local multiple reflections to occur with pressure multiplication several times that at normal incidence. Further estimates can be made theoretically or on the basis of shock tube studies available at Armour Research Foundation.

Near the vicinity of the pressurizer vessel, the reflected pressures are considerably higher both because of the higher incident shock strength and because of the increase in pressure multipli-

cation at the high pressures involved. Using the equation for reflected pressure at normal incidence as an estimate, we obtain

$$\begin{aligned}\xi_1 &= \frac{110}{15} = 7.4 \\ \xi_2 &= \frac{58.1}{13.4} = 4.34\end{aligned}\tag{25}$$

The reflected overpressure is

$$P_r = (\xi_2 \xi_1 - 1) \cdot 15 = (7.4 \times 4.34 - 1) \cdot 15 = 465 \text{ psi.}$$

Because of complex reflections at corners and in turn due to the complex geometries involved, it is entirely possible that local peak transient pressures on the order of 1000 psi may be reached near the pressurizer vessel.

To obtain some feeling for the pressure durations, one can estimate the equivalent energy release upon rupture of the pressurizer vessel from the results in Section III above. The final absolute pressure after blowout of 27.5 psi absolute is an overpressure of approximately 12.5 psi. At the time when the shock strikes the wall, complete mixing of steam and air has not yet occurred; a contact surface probably exists and the equation of state of air applies at the walls. Assuming the ratio of specific heats to be $\delta = 1.4$, corresponding to air alone, the final energy density per unit volume E is

$$E = \frac{P}{\delta - 1} = \frac{P_0}{\delta - 1} = \frac{5}{2} \left(\frac{12.5 \times 10^6 \text{ ergs cal}}{15 \cdot 4.2 \times 10^7 \text{ ergs}} \right) = .05 \frac{\text{cal}}{\text{cm}^3}\tag{26}$$

So far as the wall is concerned, the total volume might as well be air, and

$$V = 3.4 \times 10^4 \text{ ft}^3 = 10^9 \text{ cm}^3$$

results in a corresponding total apparent energy

$$E_T = .05 \times 10^9 \text{ cal} = 5 \times 10^7 \text{ cal.}$$

This energy release is approximately equal to the energy release from 5×10^4 grams or 100 pounds of TNT.

The positive duration of the pressure wave of one pound of TNT in air is of the order of 1.5 milliseconds. This duration scales according to the cube root of the energy release and the incident wave would have a duration like 7 ms. Because of the reflection process, the positive duration of the reflected wave will be further increased by roughly 1.25, so the transient pressure at the walls of the vapor container will last for durations like 10 milliseconds. Despite their high values, these transients may not be serious with regard to impulse loading of the vapor container walls but might prove serious by locally raising the pressure momentarily above the yield strength. The final quiescent pressure of 27.5 psi is, of course, indefinitely long according to the assumptions made for the purpose of this problem.

The determination of the ultimate load on a reinforced concrete and steel shell subjected to dynamic loading is an extremely complex problem which cannot be solved by conventional design methods. However, it may be possible to categorize the problem as serious or not serious by a comparison of the periods of the elastic modes of vibration of the shell with the positive phase duration of 10 milliseconds. If such shell periods are long compared with 10 milliseconds, then the response can be less serious than the static application of 160 psi would indicate. If the periods are short compared with 10 milliseconds, the response based on elastic theory can be more serious than static considerations would indicate. One would then either require an analysis incorporating plastic deformation of the shell, or suggest strengthening of the shell to the point where it

is safe based on a legitimate use of an elastic theory.

V. CONCLUSIONS

On the basis of this cursory study it is concluded that:

1. The shock problem which is associated with the explosive decompression of the pressurizer vessel cannot be dismissed on the basis of the present assumptions and is sufficiently serious to warrant re-examination of the assumptions made in the present study to determine their practical probability.
2. The discharge velocity of the steam for the pressurizer vessel will be in the order of 2000 feet per second.
3. Pressures in the order of 500 psi can be expected in the vicinity of the pressurizer vessel when the shock wave preceeding the steam jet strikes surrounding solid walls.
4. The pressurizer vessel will behave like a free rocket which will discharge the pressurizer vessel in approximately 0.5 seconds, at which time it will have obtained a velocity of approximately 700 feet per second.
5. The long duration pressure within the vapor container will be approximately 27.5 psi which is probably safe according to the specifications understood for the vapor container.
6. The explosive decompression of the pressurizer vessel will give rise to a shock wave within the vapor container of an incident strength of approximately 40 psi overpressure after full expansion; such a pressure will produce transient reflected overpressures of approximately 160 psi on the vapor container walls. This may not be serious from the standpoint of impulse damage, but the problem is considered sufficiently serious to warrant further examination of the reflection process and the response of the shell to dynamic loading. It is

suggested that such an analysis also be deferred pending consideration of the initial assumptions.

VI. PERSONNEL

Personnel from Armour Research Foundation who participated in this project were Mr. F.B. Porzel, Senior Science Advisor, Dr. Eric Lype, Senior Research Engineer, and Mr. Roman Gostkiewicz, Research Engineer.

As of November 1, 1955 the finance of this project was approximately as follows:

Net appropriation	\$1,100.00
Total expenditures, including staff salaries travel, and overhead	880.41
	<hr/>
Net available	\$ 219.59

Expenditures for the current month are estimated at \$200.00

Respectively submitted,

ARMOUR RESEARCH FOUNDATION OF
ILLINOIS INSTITUTE OF TECHNOLOGY

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APPROVED:

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