HSE
Health & Safety Executive

Pressure test safety

Prepared by the
Department of Chemical Engineering
& Chemical Technology
and BJS Research
for the Health and Safety Executive

CONTRACT RESEARCH REPORT

168/1998

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Health & Safety Executive

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A method has been devised for quantification of the hazards associated with the conduct of pressure tests and for the determination of protective barricades to prevent injury.

The hazard assessment involves the determination of the energy stored in a pressurised system and the destination of this energy following fracture of one or more components of the system. Computational methods are given for estimating this stored energy, the size and speed of missiles generated by such a failure and the magnitude of any accompanying blast.

Procedures are given for determining the loads which protective structures must resist and design information is provided for their construction.

General advice is provided on the construction and use of pressure test facilities, both fixed and mobile, and on the operational procedures which are required if the testing is to be carried out safely.

HSE BOOKS

First published 1998

ISBN 0 7176 1542 1

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ACKNOWLEDGEMENTS

We acknowledge the kind permission of the following publishers for permission to reproduce the following figures: Elsevier Science Ltd, The Boulevard, Langford Lane, Kidlington OX5 1GB, UK for figures 3.3, 3.4 and 3.5 [ref. 3.4], figures 3.1 and 3.6 [ref. 3.10] and figure 3.10 [ref 3.20]; The Institution of Chemical Engineers, 165–189 Railway Terrace, Rugby CV21 3HQ, UK for figure 1.5 [ref. 1.3]; The American Chemical Society, 2540 Olentangy River Road, PO Box 3330, Columbus OH 43210, USA for figure 5.1 [ref. 5.7]; Butterworth–Heinemann, Linacre House, Jordan Hill, Oxford OX2 9DP, UK for figures 4.7, 4.10, 4.17, 4.18, 4.19, 4.21, 4.22, 4.23, 4.24 and 4.28 to 4.36 [ref. 4.19]; McMillan Publishers Ltd, 4 Little Essex Street, London WC2R 3LF, UK for figures 4.2, 4.4, 4.5, 4.6 and 4.27 [ref.4.1, G.H. Kinney]; and TWI, Abington Hall, Abington, Cambridge CB1 6AL, UK for figures 1.3 and 1.4 [ref. 1.2].

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1 INTRODUCTION

1.1 Preliminaries

All items and equipment which operate at pressures above ambient need to be tested before use, to check that they function correctly, but more importantly, to check that, as far as this is possible, they are safe to use. We qualify the 'safe to use' statement because absolute safety is never fully attainable and this is particularly true of pressure testing of pressurised equipment. Indeed, there is one school of thought which maintains that a pressure test only demonstrates that the equipment passed the test on the day of the test and nothing more, rather like the disclaimer used on the MOT test for motor cars. We do not subscribe fully to this argument since we believe that very much more can be learnt from a test than this, however we, and the reader, should be alert to the limitated information provided by pressure testing as to the future behaviour of the pressurised equipment, particularly when it is used under conditions which differ substantially from those prevailing during the test.

Pressure testing can be divided into two types, those which are carried out at or below the design working pressure and those which are carried out at a substantially higher pressure. The former are generally concerned, in some way, with demonstrating that the equipment functions correctly. The latter are very much more concerned with seeking out serious flaws in the design or manufacture of the equipment and hence are inherently safety–related.

Figures 1.1 and 1.2 represent an analysis of HSE statistics for pressure related accidents. Many of these accidents occurred at quite low pressures, but this could be simply a reflection of the fact that more work takes place in this pressure range than at much higher ones. The analysis of failure types indicates that almost every conceivable failure has indeed happened, but with most failures, perhaps predictably, involving closures or welds.

This report is concerned with the procedures, other than those of general safety, necessary to carry out this testing safely, that is without causing injury, whether to the person carrying

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out the test or to anyone else in the vicinity. A secondary issue is the limitation of economic loss following failure of pressurised equipment during test. Further, we limit consideration to metallic pressure equipment, although such equipment may contain windows and sight glasses provided that this non–metallic part does not represent a major part of the equipment. Wholly non–metallic pressurised equipment can fail in ways that are quite different from metal equipment and are not considered in this report.

The necessity for consideration of safety during any pressurisation process is demonstrated by Figures 1.3 to 1.5. Figures 1.3 and 1.4 record the failure of a large reactor vessel in a brittle manner which distributed fragments over a wide area. Figure 1.5 shows a pipe which has split in a ductile manner along its length. Both of these failures result in the sudden release of a large amount of energy, partly as blast and partly as kinetic energy imparted to missiles. The situation is particularly acute during a pressure test since this is the time when one can be least sure of the ability the pressurised system to withstand the pressure without failure.

The hazard posed by a pressure test is a combination of the energy stored in the pressurised fluid in the equipment and the degree of ignorance about the suitability of the equipment to contain pressure. The range of variables is wide. The equipment could have an internal volume of a few cm^3 or several tens of m^3. The pressure could be as little as a fraction of a bar above ambient or as much as 10 kbar. It could be designed to work at 1 kbar and used at that pressure or it could be used at 2 bar; our understanding of the degree of suitability is very different for these two cases.

It is clear, therefore, that in some instances the hazard will be minimal, for example a vessel of a few cm^3 capacity at a pressure a little above ambient in a vessel with thick walls appropriate for use at much higher pressures, and the pressure test may pose minimal risk even when conducted on an open bench, except, of course, if the vessel is made of glass or contains a highly toxic material. Even the simplest case can be made risky!

This leads us to suggest that no pressure test should be conducted without first performing a hazard assessment. In the case of items which pose little hazard, this might be conducted very quickly and could indicate that very little protection is needed during the test. For others, very much more serious work may be required and it could point to the need for large fixed test facilities.

The object of this report is provide information relevant to the performance of the hazard assessment, both materially and operationally, and to indicate the measures which can be taken to reduce a hazardous operation to a low risk one. Issues which will be considered include the determination of the energy stored in a pressurised system, how that energy will be released in a failure and how it can be contained. Operationally we will look at how to conduct a test with a view to reducing the risk to personnel.

1.2 Test types

One of the first issues to be determined is the type or types of test to be conducted. Unfortunately there is considerable confusion as to the naming of the types of test which can be conducted. The naming which will be used here is as follows.

1.2.1 Research proof test

We group under this heading a number of highly hazardous pressure tests.

- A test carried out on a new design of vessel. One is here determining the quality of the design and not just the quality of the metallurgy and fabrication of the vessel. Some aspects of the design may not even have been quantified at the design stage and the test is required, in part, to yield measurements of, say, strain at critical points in the vessel. Some pressure vessel codes require that a prototype vessel is tested to destruction to verify the design.
- A test carried out where metallurgical data such as creep rates and fatigue life must be determined. These often involve tests of small pressurised cylinders to failure and it may be necessary to perform the tests under extreme (high and low) temperature conditions.

- An autofrettage operation carried out on a thick–wall vessel to increase its working pressure or to enhance its resistance to fatigue. During this operation the pressure is increased to a value in excess of the yield pressure of the main body of the vessel.

These tests should be conducted under conditions which anticipate total failure of the vessel, both by ductile failure and brittle fracture if the latter is remotely possible.

1.2.2 Proof test

A proof test is one carried out when a vessel is built to an established design and is, perhaps, part of a production run. It is applied when the vessel is first fabricated, significantly modified or repaired. The objective of the test is two–fold. The first is to check for serious manufacturing or metallurgical errors. The second is to produce a mild autofrettage of the material in the vicinity of stress concentrations, thereby reducing the risk of cracking in those regions, a process which is often called 'shakedown'. BS5500 specifies that, in order for this local autofrettage to take place, the proof–test pressure must take the membrane stress, in the case of a thin–wall vessel, to at least 85% of the yield stress of the material out of which the vessel is fabricated. In a thick–wall vessel it is reasonable to apply the 85% requirement to the bore of the vessel. In passing, we note that the maximum working pressure of a vessel so tested will normally be such as to limit the stress to 67% of yield. This test is a little less hazardous than the research proof test.

1.2.3 Leak test

A leak test is sometimes specified as low pressure (often meaning at 10% of working pressure) and sometimes as high pressure (usually meaning at working pressure), but clearly, leak tests can be carried out at any pressure. A leak test implies examining the vessel for leaks while at pressure and hence the operator may well need to approach the vessel to do this. The risk associated with this can be reduced by ensuring that the vessel undergoes a proof test before being tested for leaks. Indeed, BS5500 specifies that a vessel must be

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pressurised to a pressure in excess of the leak test pressure and then reduced to it before the vessel is approached for a visual examination. A figure of 10% of the working pressure would be a reasonable margin for this purpose.

1.2.4 Function test

In a function test, the various movable parts are actuated, so that for example valves are opened and closed, while the vessel is under pressure. The test would normally follow a leak test and could be carried out at any pressure up to the working pressure. The risk associated with a function test is very similar to that for a leak test, but with the possible added risk of actuator parts coming loose and forming missiles whilst being operated.

In some circumstances it may be necessary to conduct function tests with the fluids that the vessel may contain in service and this may increase the hazard associated with the test. If the working fluid is flammable, the added risk of a secondary explosion if the vessel fails must be considered.

1.3 Nature of the hazard

The two main hazards which one must contend with during pressure testing are the formation of missiles and the generation of a shock wave (a pressure wave which moves at supersonic speeds) following failure of some part of the system. The failure can be of the item being tested, clamping equipment holding it in place or the source of pressurising fluid. Experimental evidence, Esparza and Baker [1.1], indicates that shock waves are not formed when the pressurising medium is a compressed liquid, the disturbance created is only a sound wave – a loud bang – which does very little damage although the use of ear defenders may be indicated. Thus shock waves can largely be discounted when pressure testing using water at ambient temperature. On the other hand, shock waves are the norm when pressuring with a gas or a saturated volatile liquid. In contrast, missiles can be generated whatever the pressurising medium.

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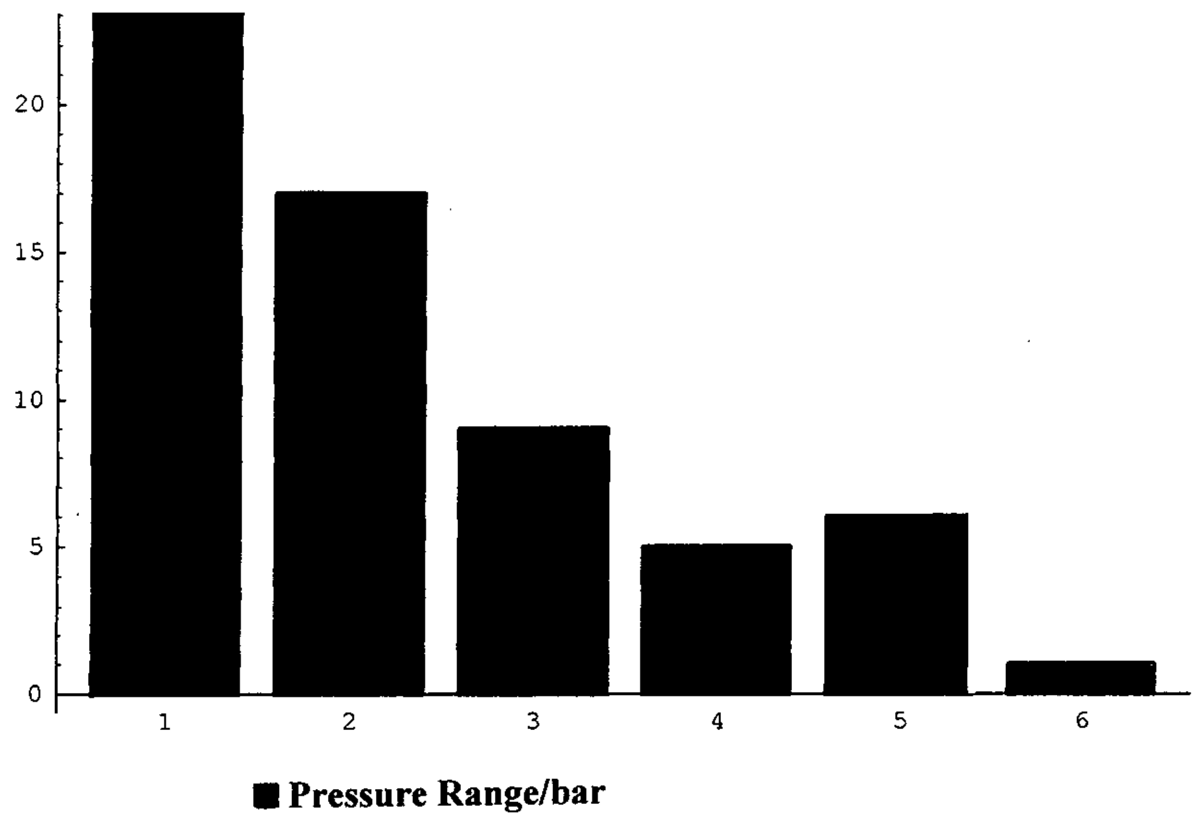
A quantitative determination of missile size and speed and shock strength is an important part of any hazard assessment and these issues are considered in depth in Sections 3 and 4 of this report. Prior to that, however, the available energy must be determined and this is considered in Section 2.

1.4 References

1.1 'Measurement of blast waves from bursting frangible spheres pressurised with flash evaporating vapour or liquid', E.D. Esparza, W.E. Baker, NASA Contractor Report 2811, Contract NSG–3005, NASA, (1977).

1.2 'Brittle fracture of a thick walled pressure vessel', British Welding Research Association, BWRA Bulletin, Cambridge, vol 7, 149–178, (1966).

1.3 'Blast Waves Generated by the Rupture of Gas–Pressurised Ductile Pipes', M.R. Baum, Trans. I. Chem. E., vol 57, 15–24, (1979).



1	1 - 10
2	10 - 100
3	100 - 200
4	200 - 500
5	500 - 1000
6	>1000

Figure 1.1. Statistics for pressure related accidents.

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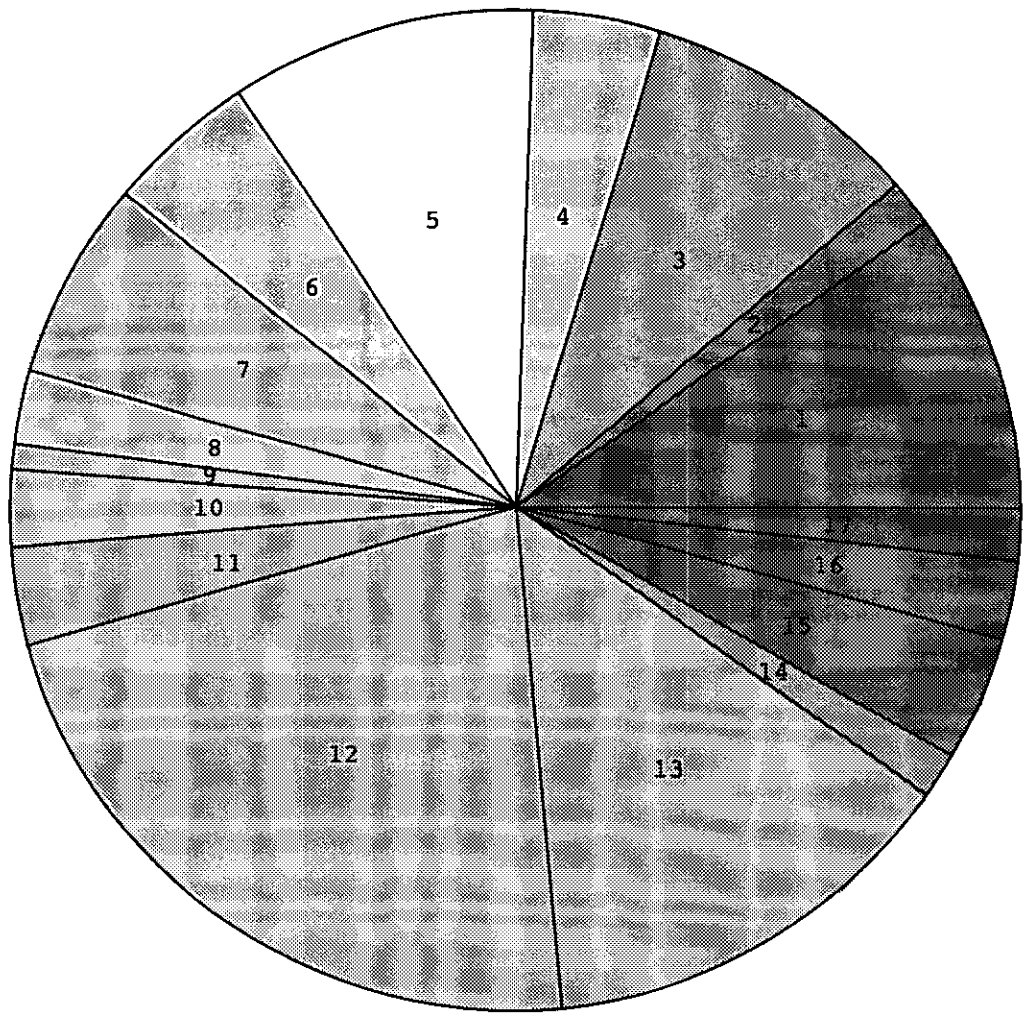

1 Pipework
2 Safety valve failure
3 Instrumentation & control failure
4 Assembly
5 Trapped pressure
6 Lost bolts
7 Fitting failure
8 Pipe whip
9 Clamp failure
10 Seal failure
11 Jets & sprays
12 Lost plugs & end caps
13 Vessel & weld failures
14 Vessel buckling
15 Secondary missiles
16 Cast component failure
17 Sight glass failure

Figure 1.2. Statistics for pressure related accidents.

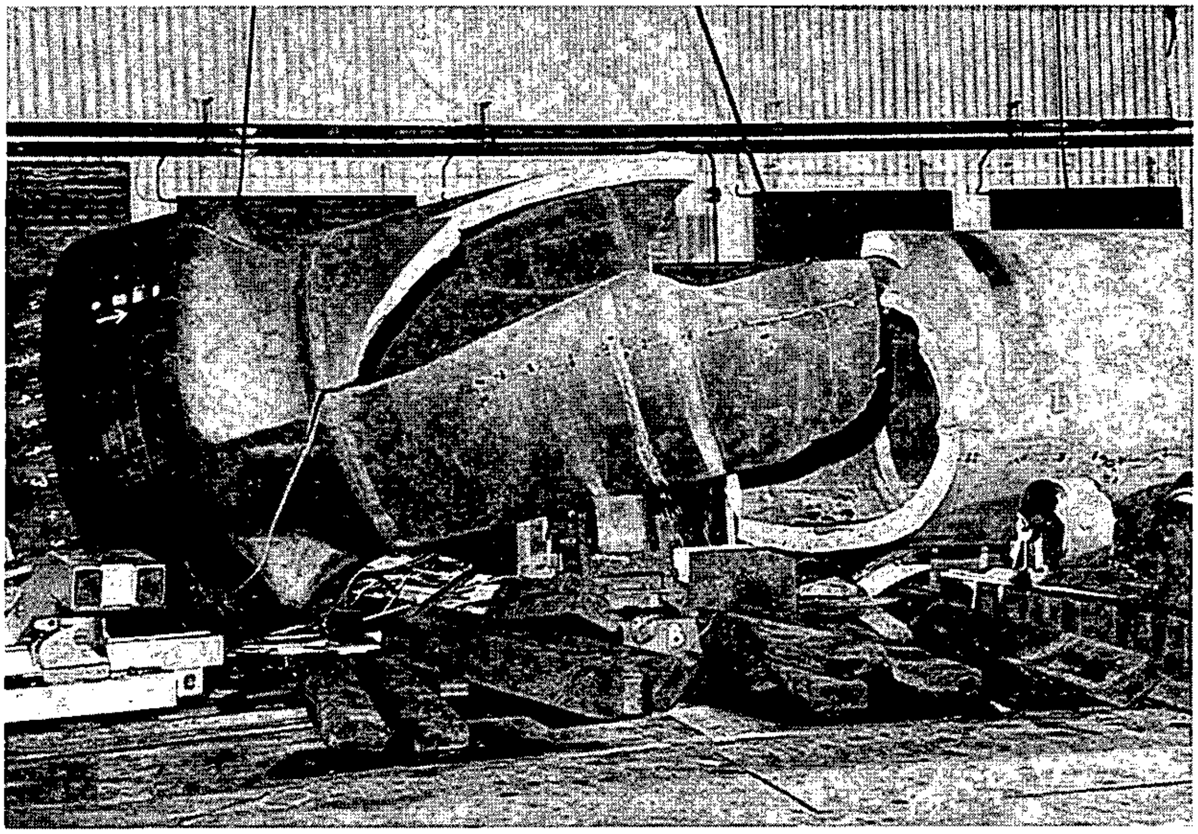

Figure 1.3. Failed reactor vessel.
(After BWRA [1.2])

Figure 1.4. 2000 kg piece of the vessel which was thrown a distance of 46 m by the explosion. (After BWRA [1.2])

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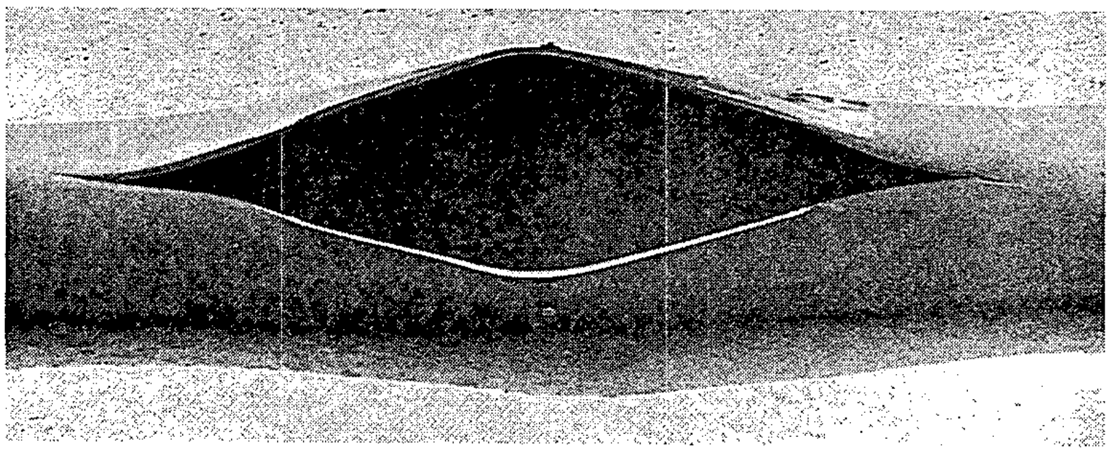

Figure 1.5. Ruptured pipe.
(After Baum [1.3])

2 STORED ENERGY

2.1 Energy sources

The extent to which a pressure vessel failing can lead to harmful effects depends on the amount of energy stored in the system at the time of failure. The total system energy E_t is defined as the sum of all of the energy available for dissipation in all possible forms. We can write:

$$E_t = E_x + E_c + E_s \tag{2.1}$$

Here: E_x is the fluid expansion energy;

E_c is the chemically released energy;

E_s is the strain energy.

The amount of energy released depends on:

- the mode of failure;
- the expansion path for compressed fluids;
- the type of fluid in the vessel;
- the final state of the system.

It is usual to distinguish between primary and secondary sources of energy. Primary sources are the expansion energy of the fluid in the vessel and the strain energy released during failure. Secondary sources are those due to chemical reactions which may occur once the fluid is released. For example, a cylinder containing hydrogen may fail (primary energy release) leading to an explosion of the hydrogen in the air (secondary energy release). For pressure testing it is often possible to choose a test fluid which will not lead to a chemical explosion once released. This may not be possible in the case of function testing.

After a vessel has failed, the law of energy conservation requires that the same amount of energy will be present although it is distributed in different forms. We may thus write the system energy after failure as:

$$E_t = E_k + E_b + E_g + E_h + E_l + E_a + E_r \tag{2.2}$$

Here: E_k is the kinetic energy of all the fragments;

E_b is the energy of the blast wave;

E_g is the energy of a ground shock;

E_h is the thermal energy (heat);

E_l is the radiant energy (light);

E_a is the sound energy;

E_r is the energy of the chemical reaction products.

Provided that no chemical explosion takes place, only the kinetic energy and blast energy are important. The remaining terms can be thought of as due to the irreversibility of the explosion process. Since it is the missiles and the blast wave that are the primary hazards, it is generally safe to assume that:

$$E_t = E_k + E_b \tag{2.3}$$

2.2 Fluid expansion energy

The energy stored in a system which will not form a chemical explosion is mainly in the form of energy due to compression which will be released when the fluid is allowed to expand. This is, therefore, the term which needs to be evaluated most carefully. In estimating this energy release, it is normal to assume that the expansion process is thermodynamically reversible and that it is sufficiently rapid for heat transfer to the surroundings to be negligible. The expansion process is, therefore, isentropic.

From the first law of thermodynamics, the change in internal energy U for a closed system for which there is a heat input Q and on which work W is done is given by:

$$\Delta U = Q + W \tag{2.4}$$

For a reversible process:

$$W = -\int_{p_i}^{p_f} p \, \mathrm{d}V \tag{2.5}$$

For adiabatic expanson $Q = 0$ and so:

$$\Delta U = W \tag{2.6}$$

For an expansion process, the work W is negative since it is defined as the work done by the surroundings on the system. Thus the expansion energy E_x in equation (2.1) is given by:

$$E_x = -W \tag{2.7}$$

or, if the expansion is adiabatic:

$$E_x = -\Delta U \tag{2.8}$$

In this section, a variety of estimation techniques is described based on these equations. In Section 2.8, example calculations are presented. These provide detailed guidance in the use of the techniques. They are all based on a cylindrical vessel of internal diameter 1.0 m and internal length 3.0 m. In the examples, different assumptions are made regarding the nature of the fluid in the vessel. A comparison of the calculated results is presented in Section 2.9.

2.3 Gas–filled systems

2.3.1 Pressure–volume product

The product of the system pressure p and the internal volume V of the system has the dimensions of energy and is used in the Pressure Systems Regulations [2.1] as a measure of the range of applicability of those regulations. It is tempting, therefore, to use this expression as the actual expansion energy:

$$E_x = p\,V \tag{2.9}$$

This method has the merit of extreme simplicity, the internal volume V can easily be measured or calculated and the test pressure will be known. It can, therefore, always be applied.

Unfortunately, it significantly under–estimates the expansion energy in most cases. This is because both the nature of the gas and the type of expansion process are ignored. We do not recommend this technique. If it must be used for lack of any other data, a safety factor η should be included:

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$$E_x = \eta \, p \, V \tag{2.10}$$

where $\eta = 1.5$ for $p \leq 50$ MPa (500 bar).

2.3.2 Perfect gas

Next in order of complexity is to assume that the vessel under test is filled with a perfect (ideal) gas for which the equation of state is:

$$p \, V = n \, R \, T \tag{2.11}$$

Here: p is the absolute pressure (Pa);

V is the internal volume of the system (m^3);

n is the amount of substance (mol);

R is the universal gas constant (8.314 J/K·mol);

T is the thermodynamic (absolute) temperature (K).

The amount of energy released on expansion depends on the thermodynamic path taken during the expansion. Two limiting cases are usually considered, isothermal and isentropic:

- isothermal expansion of perfect gas:

$$W = p_i \, V_i \ln(p_f/p_i) \tag{2.12}$$

- isentropic expansion of perfect gas:

$$W = \frac{1}{\gamma - 1} \, p_i \, V_i \left[(p_f/p_i)^{\frac{\gamma-1}{\gamma}} - 1 \right] \tag{2.13}$$

Here: p_f is the final (atmospheric) pressure (Pa);

p_i is the initial pressure (Pa);

V_i is the initial volume of the system (m^3);

γ is the ratio of heat capacities (c_p/c_v) (–), values of which may be found in Table 2.1 for several common gases.

The expansion energy E_x is then obtained from equation (2.7).

No real expansion is completely reversible due to dissipative processes such as viscosity and turbulence. Similarly, no real process is entirely adiabatic, although this is a reasonable

approximation for very rapid processes such as explosions. The thermodynamic pathway of a real explosion therefore lies between those for isothermal and adiabatic processes and is closest to the adiabatic shown in Figure 2.1. The isothermal expansion process always gives a larger value for the expansion energy and has been recommended by some authors as being conservative. However, no real gas obeys the perfect gas equation exactly and the effect of these gas imperfections is often much greater than the error involved in the assumptions made about the expansion process.

2.3.3 Real gases

The perfect gas model of gas behaviour makes the simplifying assumptions that the molecules of the gas occupy no space and that they do not attract one another. These assumptions are reasonable provided that the average distance between the molecules is large. This corresponds to high temperatures and low pressures. Neither of these requirements corresponds to the condition of fluids used in pressure tests.

At high pressures, the molecules of a gas are squashed together and the gas shows significant deviations from perfect gas behaviour. The best starting point for the calculation of the fluid expansion energy is experimental fluid property data available in the form of tables and charts. Table 2.2 provides a guide to some substances for which the data are available. It should be noted that most thermodynamic charts are restricted to pressures below a few hundred bar and that the number of substances for which data are available above 1 kilobar is very limited. The fluid expansion energy for an adiabatic process is the change in internal energy between the initial and final conditions. The examples in Section 2.8 show how this is evaluated. For more detailed discussion of the use of thermodynamic charts and tables, reference should be made to textbooks on engineering thermodynamics, e.g. [2.2].

It should be noted that the final state of the fluid after it has expanded is often at a very low temperature and may lie in the two–phase gas+liquid region. For a very rapid expansion, there is very little time for the liquid phase to be nucleated and for phase segregation to

occur. The final state may, therefore, be a thermodynamically unstable super–cooled gas. This form of instability is with respect to material diffusion. As the gas super–cools further, a point is reached at which it becomes mechanically unstable since further cooling would make the compressibility positive (that is the fluid would increase in volume on compression). There is thus a limit to the extent to which the gas can be super–cooled. As a rough rule of thumb, this is usually taken to be at the point at which the (mass) quality x is 0.9. From the pressure–enthalpy chart for nitrogen (Figure 2.2), it can be seen that, if the intial temperature is 300 K (near room temperature), a quality of 0.9 or less can only be obtained upon expansion if the initial pressure is above about 200 bar.

2.3.4 Empirical equations of state

For many substances, thermodynamic data can be represented by equations of state. These can be used with a variety of thermodynamic relationships to calculate the expansion energy. Equations of state are only accurate over a limited range of temperatures and pressures for which experimental data were available when they were devised. Generally, this means temperatures and pressures up to about the critical temperature and pressure of the substance in question. At higher pressures, the density of the substance may not be represented correctly.

There is a wide range of equations of state in current use; for reviews see [2.5–2.6]. They have a major advantage over tables and charts in that they can be used for mixtures as well as pure substances. Computer software is available from a number of vendors which will perform isothermal and isentropic expansion (flash) calculations for a wide range of substances and their mixtures. Their proper use does, however, require a knowledge of thermodynamic and fluid properties. The methods are particularly useful when the fluid is a mixture and may become partially liquefied during the expansion.

The most commonly used equations of state are the cubic equations of state such as the Peng–Robinson (PR) and the Soave–Redlich–Kwong (SRK) equations. A general failing of

these is that they under–estimate the density of the liquid phase. In many commercial computer packages, the option is available to calculate the liquid density using the corresponding states method, which is more accurate, and this option should generally be chosen.

Most software packages do not output the internal energy u and it is necessary to calculate the specific enthalpy h and specific volume v for the initial and final conditions. The internal energy can then be calculated thus:

$$u = h - p\,v \qquad (2.14)$$

in just the same way as in the use of charts and tables.

2.4 Liquid–filled systems

Calculation of the expansion energy for liquids is similar to that for gases. The system is closed in the thermodynamic sense and the expansion energy is therefore identified with the change in internal energy accompanying the expansion. Differences arise because liquids are very much less compressible than gases and this results in a whole new range of approximations being feasible.

The simplest formulation comes from recasting equation (2.5) in terms of the compressibility κ of the liquid. The isentropic compressibility κ_s is given by:

$$\kappa_s = -\frac{1}{V}\left.\frac{\partial V}{\partial p}\right|_S \qquad (2.15)$$

Thus $\mathrm{d}V = -V\kappa_s\,\mathrm{d}p$ and so, from equations (2.5) and (2.6):

$$\Delta U = \int_{p_i}^{p_f} p\,V\,\kappa_s\,\mathrm{d}p \qquad (2.16)$$

The isothermal compressibility κ_t is greater that the isentropic compressibility κ_s, so using κ_t in place of κ_s will produce a conservative value for ΔU. Thus for a liquid, for which the compressibility is small, and for small pressure changes, V and κ_t can be taken to be constant, giving:

$$\Delta U \simeq V \kappa_t \int_{p_i}^{p_f} p \, dp = \frac{1}{2} V \kappa_t \left(p_f^2 - p_i^2\right) \tag{2.17}$$

Note the quadratic dependence of ΔU on p.

The isothermal compressibility κ_t may be found from pVT data. Most tabulated values are for κ_t at low pressure, examples of which are given in Table 2.3 (from [2.4]). In reality, κ_t decreases with increase in pressure, that is it becomes more difficult to compress a liquid the more it is compressed. Thus, by using the low–pressure value of κ_t in equation (2.17), the expansion energy will be further over–estimated beyond that which results from replacement of κ_s by κ_t.

By way of example, the expansion energy for water, calculated from the low pressure isothermal compressibility, is shown in Figure 2.3.

Calculations which avoid these over–estimations of the expansion energy are rather more complex since the final temperature at the end of the expansion is not known and the calculation must be performed numerically, integrating along the expansion path. Thermodynamically, the changes in temperature T and internal energy U along the path come from:

$$\left.\frac{\partial T}{\partial p}\right|_S = \frac{T}{c_p} \left.\frac{\partial V}{\partial T}\right|_p \tag{2.18}$$

$$\left.\frac{\partial U}{\partial p}\right|_S = -p \left[\left.\frac{\partial V}{\partial p}\right|_T + \frac{T}{c_p} \left.\frac{\partial V}{\partial T}\right|_p^2 \right] \tag{2.19}$$

Generally, the specific heat at constant pressure c_p can be regarded as a constant but the derivatives $\left.\frac{\partial V}{\partial p}\right|_T$ and $\left.\frac{\partial V}{\partial T}\right|_p$ vary significantly with p and T. The one that is the more important in most cases is $\left.\frac{\partial V}{\partial p}\right|_T$

To calculate the derivatives, an equation of state is required to represent the pVT properties of the liquid. Very few substances have had their pVT properties measured to very high pressures. Fortunately, water, the most commonly used pressure test fluid, is one of them.

Its pVT properties may be represented by Huddlestone's equation:

$$\log_{10}\left[\frac{L^2\ p}{L_0 - L}\right] = A + B\,t + C\,(L_0 - L) \qquad (2.20)$$

where $L^3 = V(p,t)/V(p{=}0,t{=}0)$, $L_0{}^3 = V(p{=}0,t)/V(p{=}0,t{=}0)$ and t is the Celsius temperature. Table 2.4 gives values for the constants A, B and C in Huddlestone's equation for the limited number of substances for which they have been derived.

2.5 Fluid expansion energy chart

Figures 2.4 to 2.7 show the fluid expansion energy as a function of pressure for nitrogen and various liquids at 300 K (27 C).

In the case of nitrogen, the fluid expansion energy has been evaluated from the IUPAC equation of state [2.3]. In Figure 2.4, for initial pressures in the range 100 bar to 200 bar, the final state is assumed to be metastable gas (see Section 2.3.3), whereas, for initial pressures below 100 bar, the final state is equilibrium gas. In Figure 2.5, where values are given to 4 kbar, the final state is assumed to be two–phase. The difference between these two methods of estimation in the region of overlap is not large.

Figures 2.6 and 2.7 give the fluid expansion energy for various liquids. They have been calculated using Hudlestone's equation (2.20). Comparison of Figures 2.3 and 2.7 for water indicates that the approximate method (Figure 2.3) overestimates the expansion energy by a factor of about 2.

2.6 Strain energy

The elastic strain energy of a cylindrical vessel is usually small compared with the fluid expansion energy. However, there will be a few circumstances in which it needs to be taken into account, usually when the pressurising fluid is a low compressibility liquid. It may be

calculated from:

$$E_S = \frac{P_1{}^2\ V_1}{2\ E}\left[\frac{3\ (1\ -\ 2\ \nu)\ +\ 2\ K^2\ (1\ +\ \nu)}{K^2\ -\ 1}\right] \tag{2.21}$$

Here: P_1 is the initial pressure (Pa);

V_1 is the internal volume of the vessel (m^3);

E is the Young's modulus of the vessel material (Pa);

ν is Poisson's ratio for the vesel material (–);

K is the ratio of the outside to the inside diameter of the vessel (–).

2.7 TNT equivalent

In Sections 3 and 4, we make extensive use of experimental information from ordnance sources, usually involving the use of a high explosive such as TNT, in order to estimate the damage caused by a failing pressure vessel. The assumption is that the blast from a rupturing pressure vessel is the same as that caused by the detonation of such amount of TNT as will release the same energy. The link between TNT and stored energy is:

1 kg of TNT is equivalent to a stored energy of 4.5 MJ.

2.8 Examples

2.8.1 Perfect gas

We consider a vessel of internal diameter 1.0 m and internal length 3.0 m with flat ends and hence of internal volume 2.36 m^3 and take it to be filled with nitrogen at 10 MPa (100 bar) and 330 K (57 C). For a perfect gas of molar mass M, equation (2.11) can be rewritten:

$$p\,V\ =\ m\,R\,T\,/\,M \tag{2.21}$$

Thus the mass of gas $m = p\ V M/R\ T$ and, since $M = 0.028$ kg/mol for nitrogen, it follows that $m = 241$ kg. Now:

- for isothermal expansion to 0.1 MPa (1 bar): equation (2.12) yields $\Delta U = -109$ MJ and hence, $E_X = -\Delta U = 109$ MJ;

- for isentropic expansion to 0.1 MPa (1 bar): equation (2.13) yields $\Delta U = -43$ MJ, assuming that $\gamma = 1.4$, and hence $E_x = -\Delta U = 43$ MJ.

2.8.2 Use of pressure–enthalpy chart

We consider again a vessel of internal diameter 1.0 m and internal length 3.0 m and hence of internal volume 2.36 m^3 and take it to be filled with nitrogen at 10 MPa (100 bar) and 330 K (57 C). Use of the pressure–enthalpy chart in Figure 2.2 gives the initial enthalpy $h_i = 476$ kJ/kg and initial specific volume $v_i = 0.01$ m^3/kg. Assume that the gas expands isentropically to 0.1 MPa (1 bar):

- draw the isentrope from p = 10 MPa to p = 0.1 MPa;
- read off T_f = 88 K, $h_f(0.1\ \text{MPa}, T_f)$ = 235 kJ/kg, $v_f\ (0.1\ \text{MPa}, T_f)$ = 0.24 m^3/kg;
- thus $\Delta h = 235 - 476 = -241$ kJ/kg;
- now $\Delta u = \Delta h - \Delta(p\ V) = -241 - (-76) = -165$ kJ/kg;
- hence $\Delta U = V \cdot \Delta u / v_i = -39$ MJ;
- and $E_x = -\Delta U = 39$ MJ.

2.8.3 Use of fluid expansion energy chart

We consider a vessel of internal diameter 1.0 m and internal length 3.0 m and hence of internal volume 2.36 m^3 and take it to be filled with nitrogen at 10 MPa (100 bar) and 330 K (57 C). Use of the fluid expansion energy chart for 300 K in Figure 2.4 gives:

- the fluid expansion energy as 16.4 MJ/m^3:
- hence $E_x = -\Delta U = 39$ MJ.

The exact agreement with the value obtained in Section 2.8.2 is to some extent fortuitous in view of the different starting temperatures. However, it does indicate that the fluid expansion energy is relatively insensitive to the initial temperature due to the smaller mass present at higher temperatures being compensated by its higher specific energy, compensation which is exact for a perfect gas (to see this, equation (2.13) should be recast in terms of the energy in the volume of the vessel rather than the molar energy).

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2.8.4 Use of tables and temperature–entropy chart

We consider again a vessel of internal diameter 1.0 m and internal length 3.0 m and hence of internal volume 2.36 m^3 but now take it to be filled with hydrogen at 10 MPa (100 bar) and 330 K (57 C). Use of the table in Table 2.5 gives the following:

- at 300 K and 10 MPa: $v = 0.1312$ m^3/kg, $h = 4561.0$ kJ/kg, $s = 37.79$ kJ/kg·K;
- at 400 K and 10 MPa: $v = 0.1729$ m^3/kg, $h = 6045.1$ kJ/kg, $s = 42.11$ kJ/kg·K;
- by linear interpolation at 330 K and 10 MPa: $v_i = 0.1437$ m^3/kg, $h_i = 5006.2$ kJ/kg, $s_i = 39.10$kJ/kg·K;
- the mass of gas in the vessel $m = 2.36/0.1437 = 16.4$ kg.

Assume that the gas expands isentropically to 0.1 MPa (1 bar). The temperature–entropy chart in Figure 2.8 gives:

- $T_f = 74$ K, $h_f(0.1 \text{ MPa}, T_f) = 1120$ kJ/kg, $v_f\ (0.1 \text{ MPa}, T_f) = 3$ m^3/kg;
- hence $\Delta h = 1120 - 5006.2 = -3886$ kJ/kg;
- now $\Delta u = \Delta h - \Delta(p\ V) = -3886 - (-1137) = -2750$ kJ/kg;
- hence $\Delta U = m \cdot \Delta u = -45$ MJ and $E_x = -\Delta U = 45$ MJ.

2.8.5 Use of temperature–entropy chart

We consider again a vessel of internal diameter 1.0 m and internal length 3.0 m and hence of internal volume 2.36 m^3 but now take it to be filled with carbon dioxide at 10 MPa (100 bar) and 330 K (57 C). The temperature–entropy chart in Figure 2.9 gives:

- at 330 K and 10 MPa: $v_i = 0.0032$ m^3/kg, $h_i = 347$ kJ/kg, $s_i = 1.15$ kJ/kg·K;
- the mass of gas in the vessel $m = 2.36/0.0032 = 737.5$ kg.

Assume that the gas expands isentropically to 0.1 MPa (1 bar). The temperature–entropy chart in Figure 2.9 gives:

- the final state is in the two–phase gas plus solid region: $T_f = 200$ K, $x_f = 0.74$;
- by interpolation: $h_f = 200$ kJ/kg, $v_f = 0.28$ m^3/kg;
- hence $\Delta h = 200 - 347 = -147$ kJ/kg;
- now $\Delta u = \Delta h - \Delta(p\ V) = -147 + 4 = -143$ kJ/kg;

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- hence $\Delta U = m \cdot \Delta u = -105$ MJ;
- and $E_x = -\Delta U = 105$ MJ.

2.8.6 Water–filled vessel

We consider a vessel of internal diameter 1.0 m and internal length 3.0 m and hence of internal volume 2.36 m^3 and take it to be filled with water at 10 MPa (100 bar) and 300 K (27 C). Use of the fluid expansion energy chart for 300 K in Figure 2.6 gives:

- the fluid expansion energy as 0.25 MJ/m^3:
- hence $E_x = -\Delta U = 0.59$ MJ.

This figure should be compared with the value of 0.50 MJ/m^3 obtained from the isothermal assumption used in Equation 2.17 and plotted in Figure 2.3. Clearly, the isothermal assumption seriously overestimates the stored energy.

2.9 Conclusions

Up to about 400 bar, the stored energy in a permananent gas is essentially independent of the molecular species and its temperature. The examples demonstrate that the nitrogen values given in Figures 2.4 and 2.5 can just as well be used for hydrogen and, by analogy, for any other permanent diatomic gas. Permanent gases of greater molecular complexity have slightly higher energy contents, 25% higher in the case of methane, and those of lower complexity such as argon (i.e. monatomic gases) have values 30% lower. So the use of nitrogen to represent all permanent gases will normally be adequate. More compressible gases such as carbon dioxide, which has a critical temperature close to ambient, have very much higher energy contents, almost three times in the example given, and this cannot be ignored. Relatively incompressible liquids, such as water, have very much lower energy contents, hence their attraction as pressure test fluids.

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Other liquids such as mineral oils and hydraulic fluids are often used as an alternative to water. These are not included in Figures 2.6 and 2.7 due to non–availablity of the necessary experimental information. However, liquid paraffin, which is included, should provide a reasonable approximation for their fluid expansion energy.

Above 400 bar, the compressibility of all gases steadily decreases with increasing pressure and by 4000 bar is comparable with that for most liquids. As a result the incremental increase in stored energy with increase in pressure decreases dramatically, see, e.g., Figure 2.5, and the stored energy in a gas approaches that for a liquid. Even so, the difference between nitrogen and water is still a factor of 5 at 4000 bar, but the difference is halved by about 10000 bar and, presumably, becomes even smaller at still higher pressures.

2.10 References

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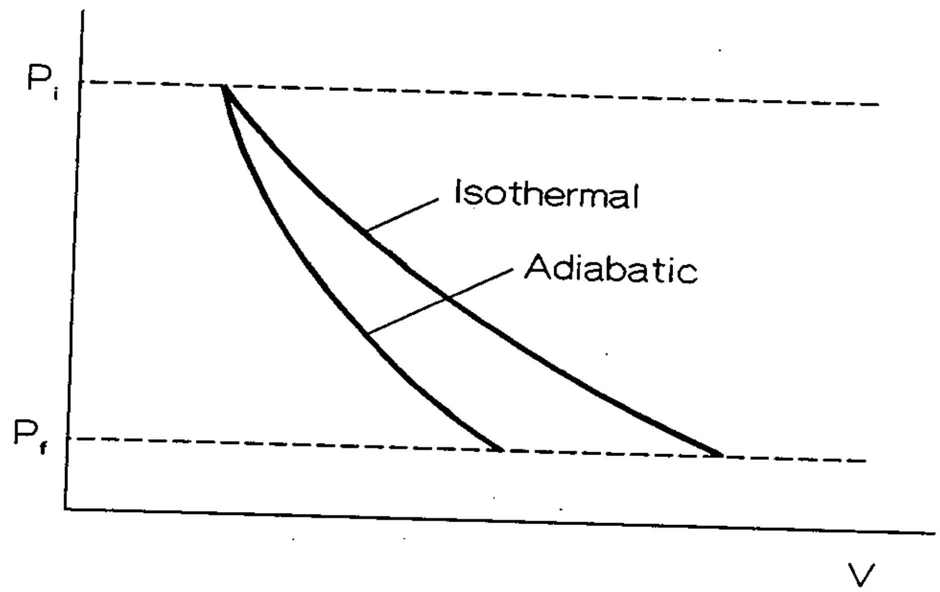

Figure 2.1. Isothermal and adiabatic processes

Contents

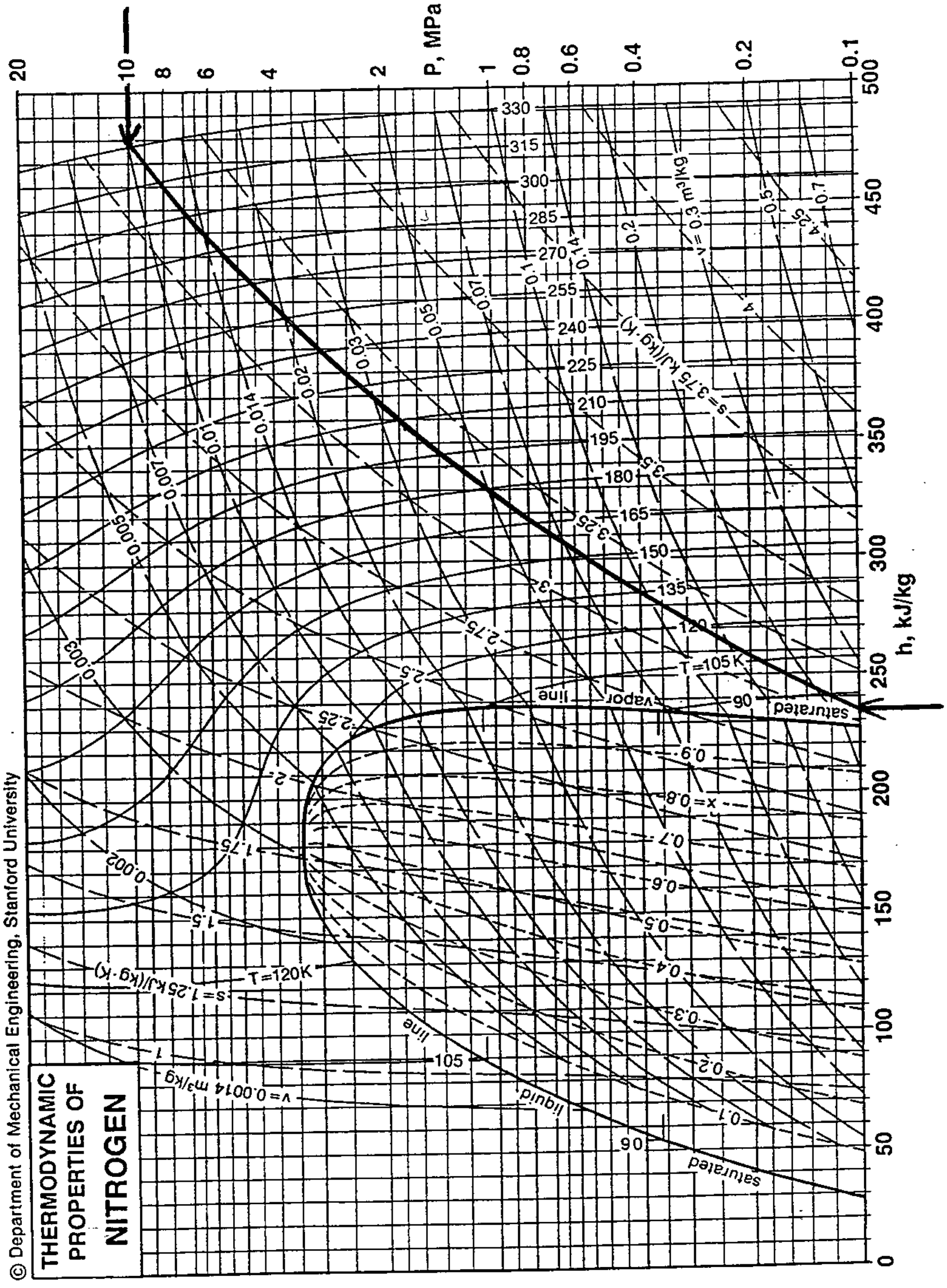

Figure 2.2. Thermodynamic properties of nitrogen
(After Reynolds [2.7])

Contents

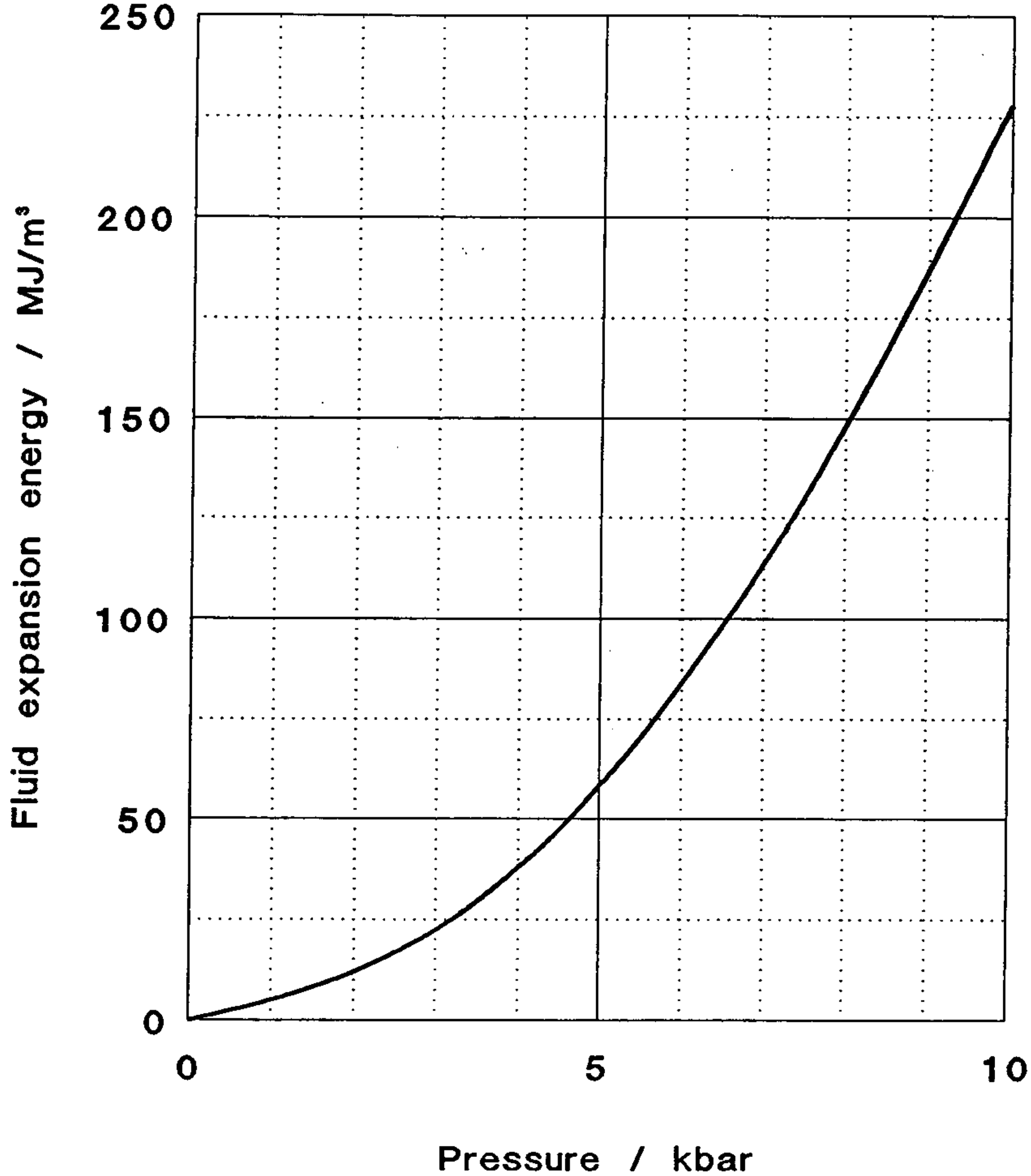

Figure 2.3. Approximate fluid expansion energy for water based on equation (2.17).

Contents

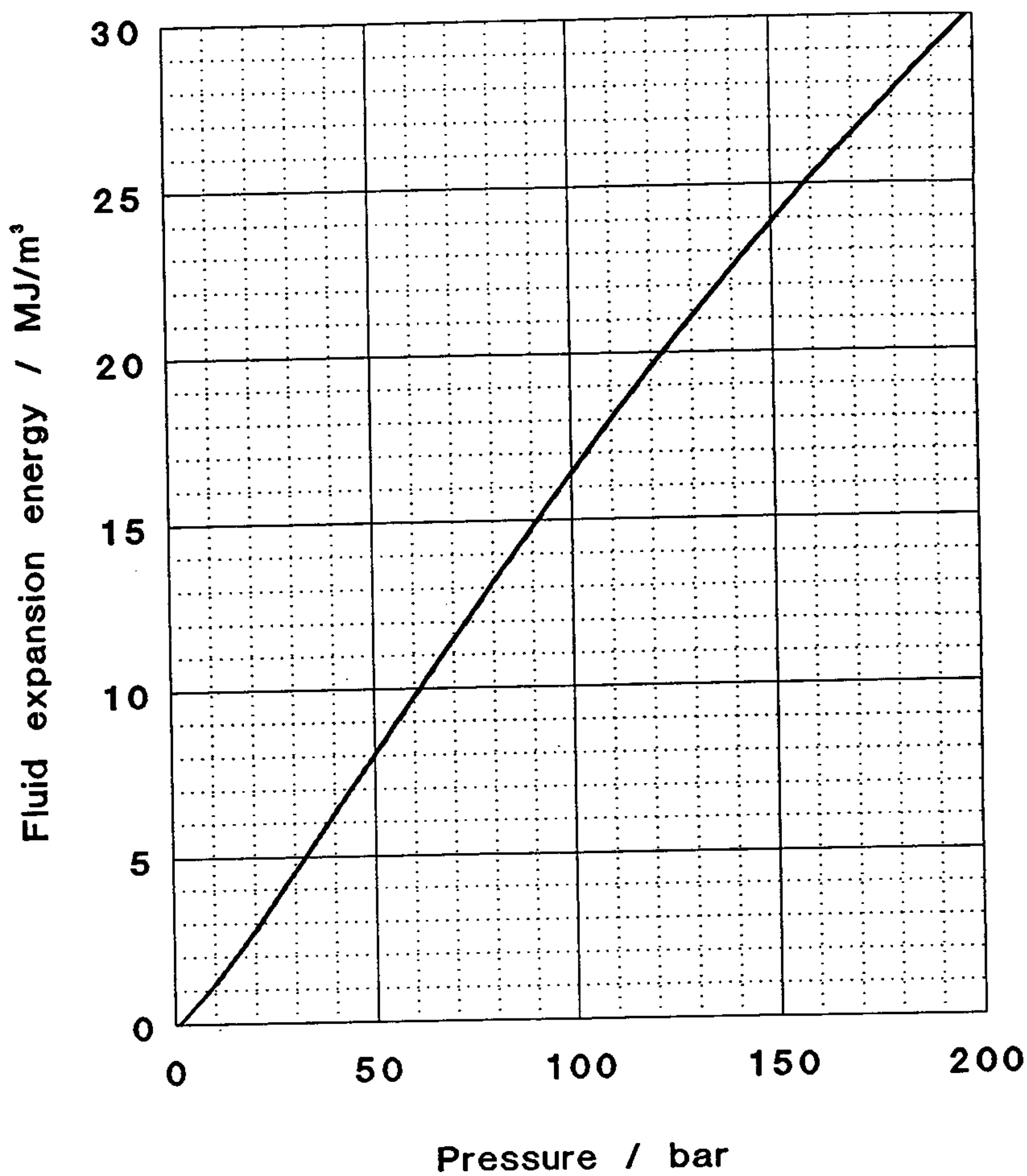

Figure 2.4. Fluid expansion energy for nitrogen up to 200 bar.

Contents

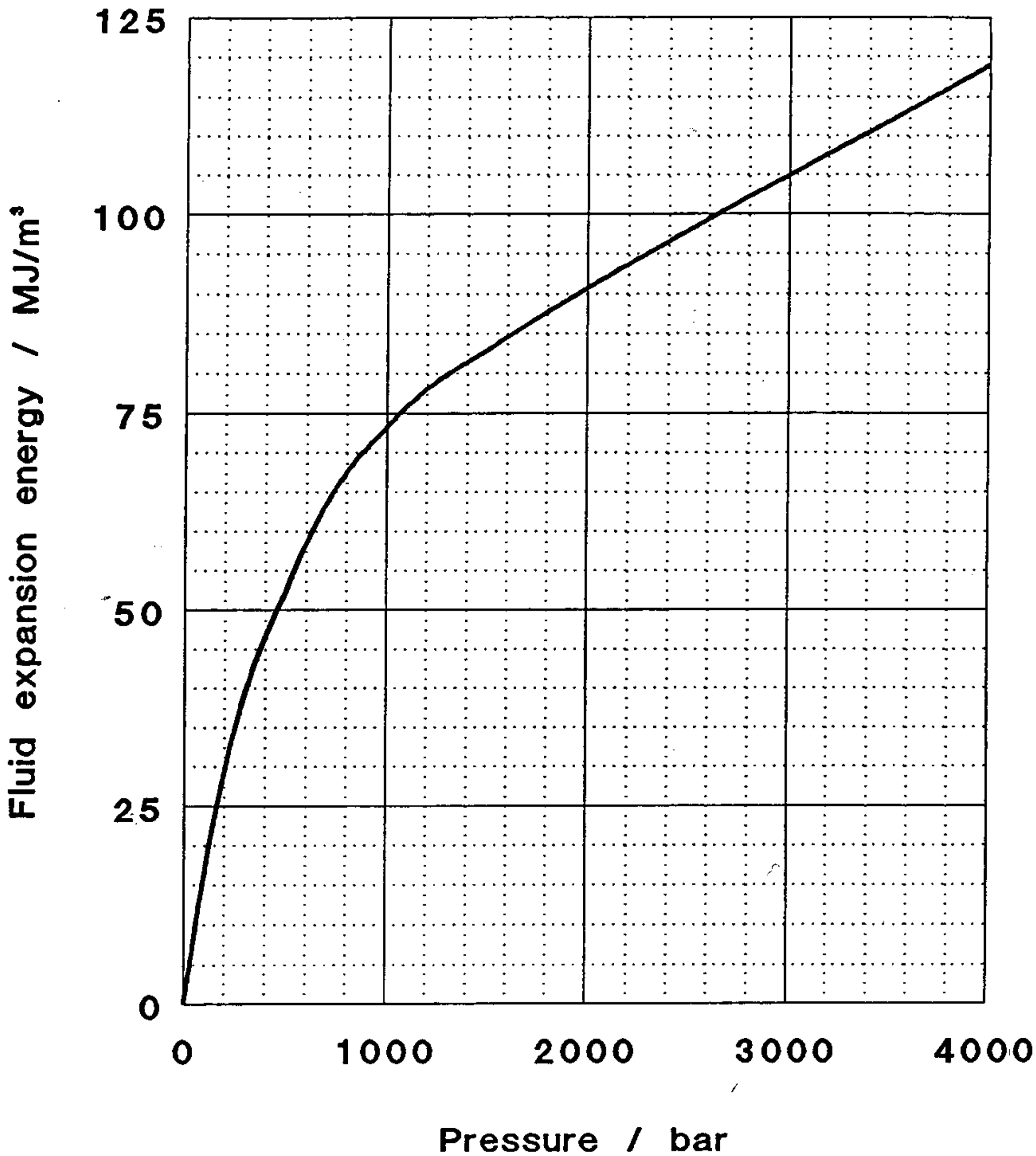

Figure 2.5. Fluid expansion energy for nitrogen up to 4 kbar.

Contents

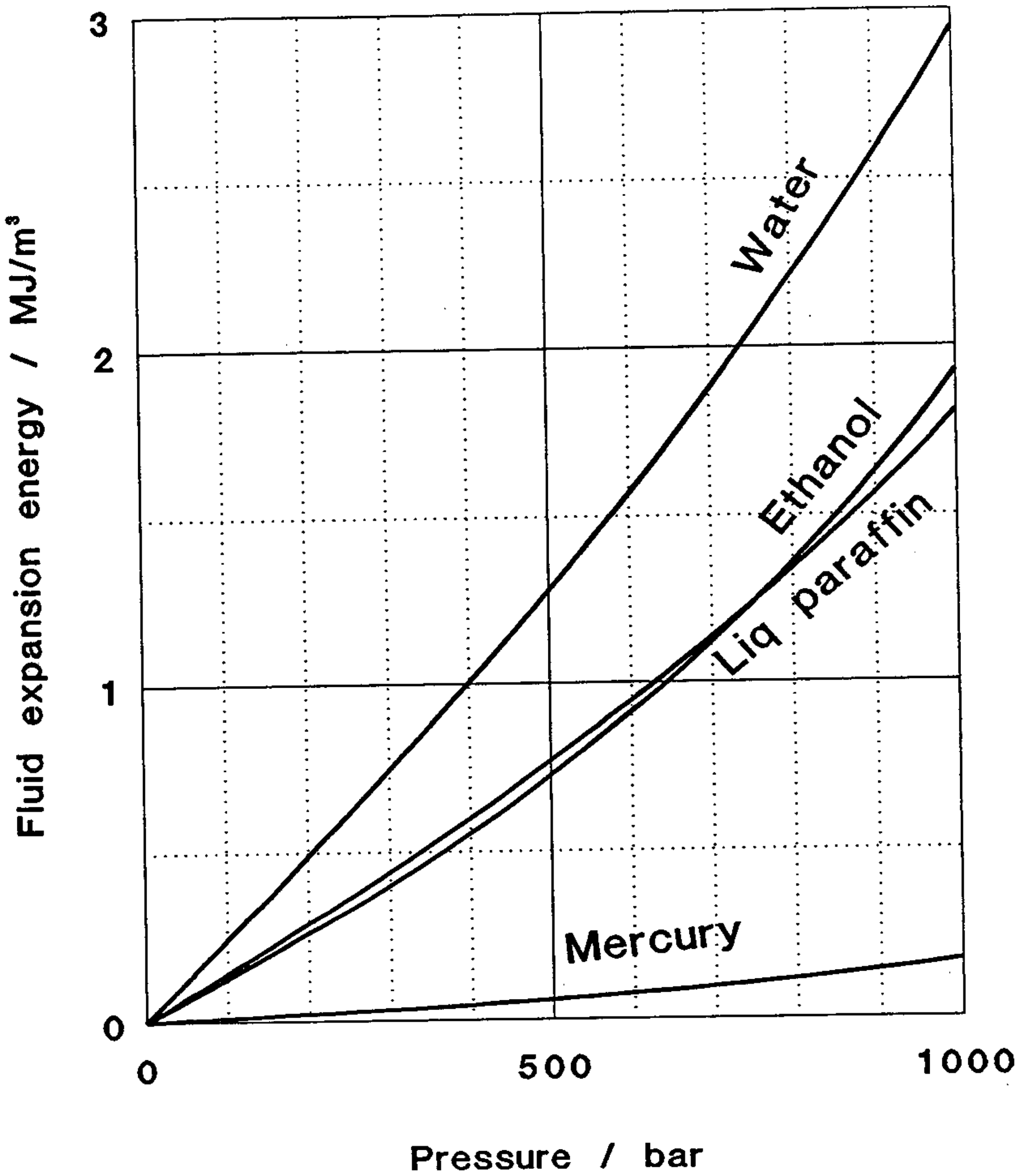

Figure 2.6. Fluid expansion energy for various liquids up to 1 kbar.

Contents

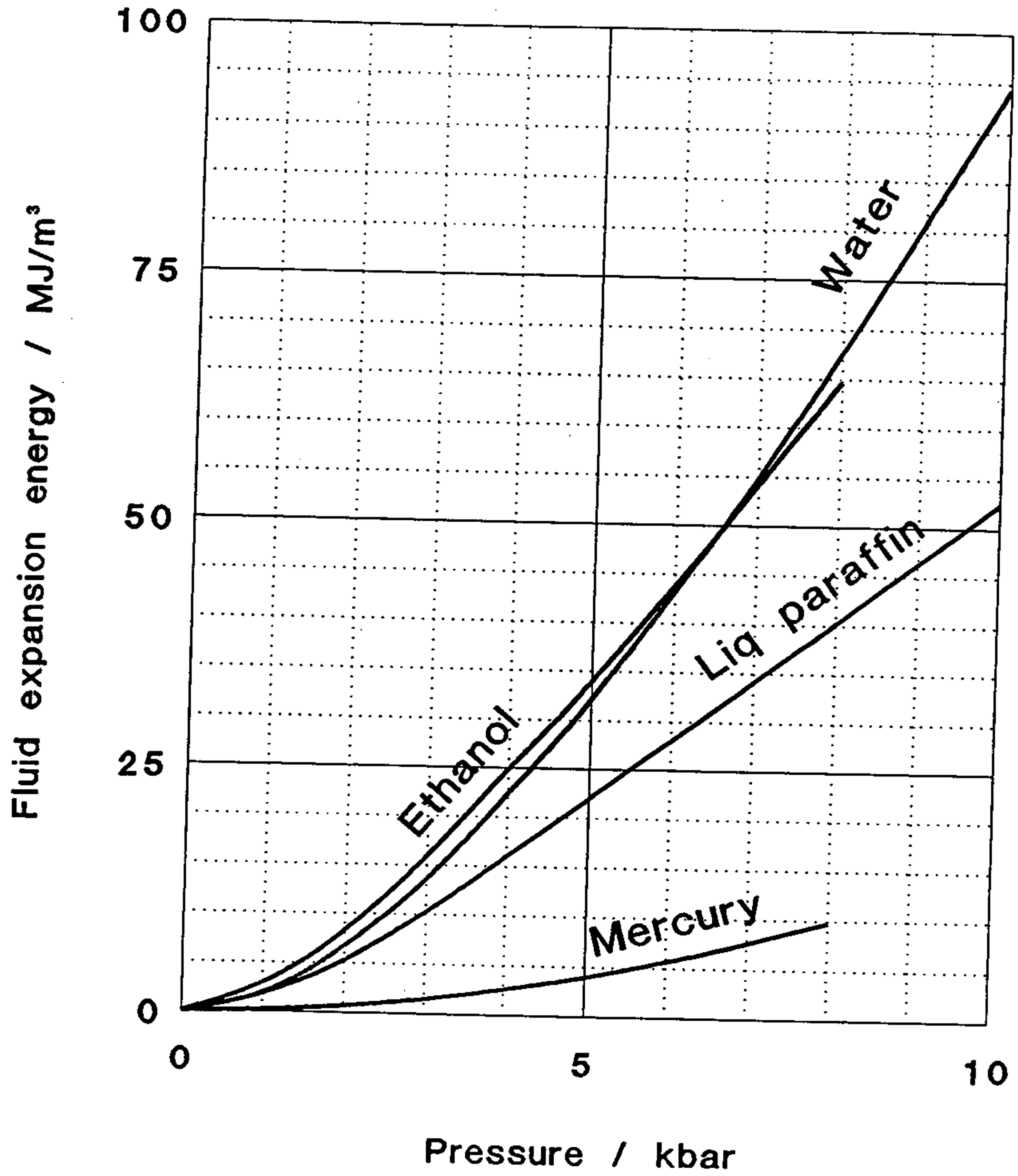

Figure 2.7. Fluid expansion energy for various liquids up to 10 kbar.

Contents

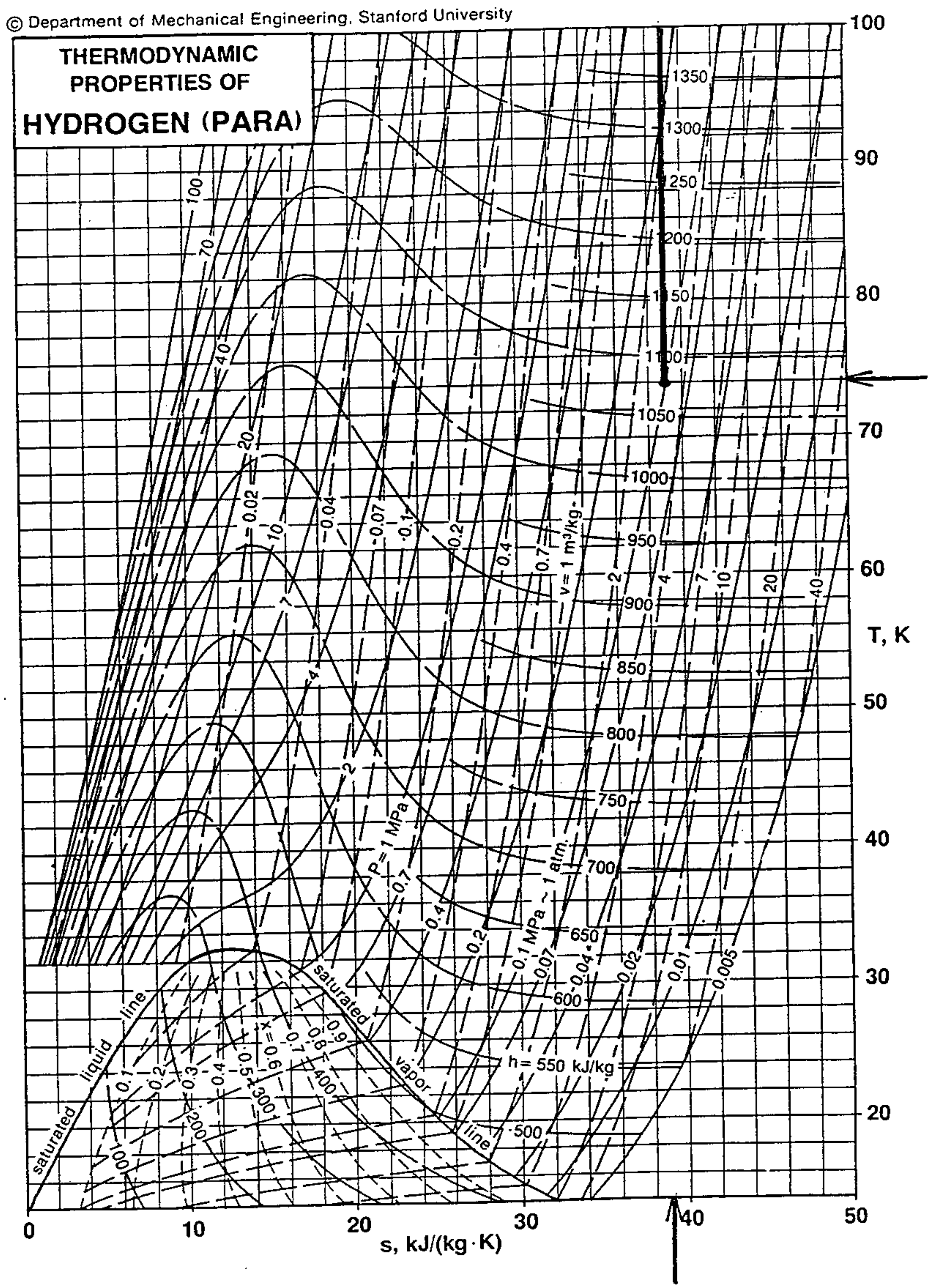

Figure 2.8. Thermodynamic properties of hydrogen. (After Reynolds [2.7])

Contents

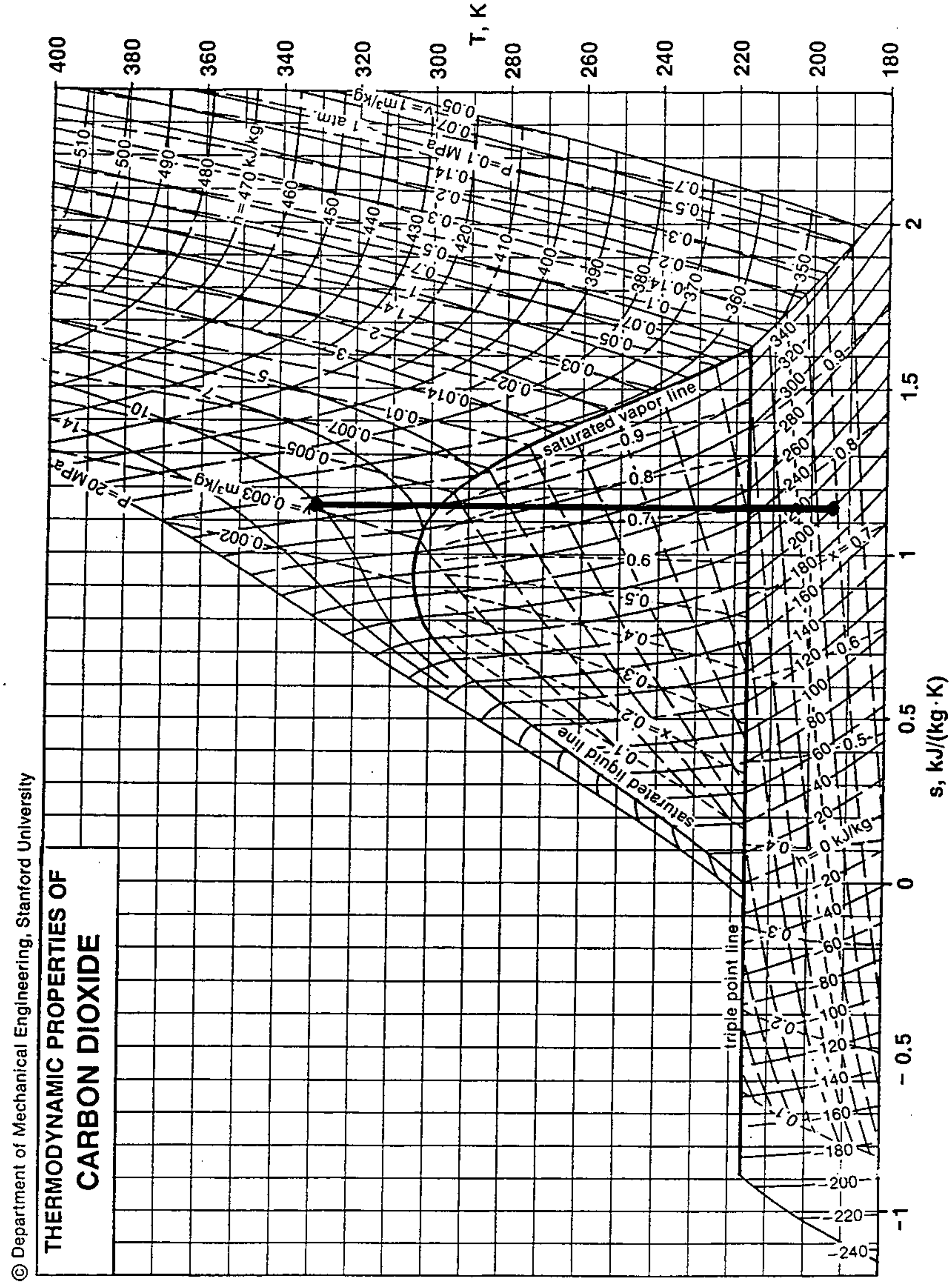

Figure 2.9. Thermodynamic properties of carbon dioxide.
(After Reynolds [2.7])

Contents

Gas	γ (300 K)
Ne	1.67
Ar	1.67
He	1.67
N_2	1.40
H_2	1.40
O_2	1.40
Air	1.40
H_2O	1.33
CO_2	1.30
SO_2	1.30
CH_4	1.31
C_2H_4	1.24
C_2H_6	1.22
C_3H_8	1.13

Table 2.1. Heat capacity ratio, γ, for various gases

Pure substances for which the thermodynamic properties are available in tabular or chart form

Substance	Reference No.
Acetylene	(6) (11k) (19)
Ammonia	(6) (17) (18a) (19)
Argon	(2) (6) (17) (21) (23) (30)
Benzene	(11l) (33)
i-Butane	(1) (11f) (19)
n-Butane	(1) (11d) (17) (19)
1-Butene	(5) (11j) (19)
Carbon Dioxide	(2) (4) (5) (6) (17) (18b) (19) (20)
Carbon Monoxide	(2) (6) (13) (18c) (19)
Carbon Tetrafluoride	(17)
Cyclopropane	(27)
n-Decane	(1)
Dichlorodifluoromethane	(17)
Dichloromonofluoromethane	(17)
Dichlorotetrafluoroethane	(17)
Ethane	(1) (6) (11b) (17) (19) (29)
Ethylene	(3) (6) (11h) (17) (19) (35)
Helium	(7) (10) (17) (32)
n-Hexane	(11g) (19)
n-Hydrogen	(2) (15) (17) (18d) (19)
p-Hydrogen	(12) (17)
Hydrogen sulphide	(5)
Methane	(1) (6)(8)(11a)(17)(19)(24)(28)
Methyl Chloride	(17)
Monochlorodifluoromethane	(17)
Monochlorotrifluoromethane	(17)
Neon	(16) (17)
Neopentane	(33)
Nitrogen	(1) (6) (9) (17) (18e) (19) (23)
Nitrous Oxide	(17)
n-Nonane	(5)
Oxygen	(2) (17) (18f) (19) (23) (25) (31)
n-Pentane	(1) (11e) (19)
Perfluorocyclobutane	(14)
Propane	(1) (6) (11c) (17) (19)
Propylene	(5) (11i) (17) (19)
Sulphur Dioxide	(17) (18g) (19)
Trichloromonofluoromethane	(17)
Trichlorotrifluoroethane	(17)
Water	(2) (18h) (19) (22) (26)

Table 2.2. Bibliography of thermodynamic charts and tables. (After Bett [2.2])

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Table 2.2 (cont). Bibliography of thermodynamic charts and tables.

Isothermal compressibilities. β_T, of some liquids at 1 bar, 20° (unless otherwise indicated)

	β_T (10^2 kbar^{-1})		β_T (10^2 kbar^{-1})
Hexane	15.40	Methanol	12.14
2,3-Dimethylbutane	17.97	Ethanol	11.19
Heptane	13.4	1-Propanol (0°)	8.43
Octane	11.3	1-Butanol (0°)	8.10
Cyclopentane	13.31	1-Pentanol (0°)	7.71
Cyclohexane	11.30	Ethanediol	3.64
Cycloheptane	9.22	1-Octanol (0°)	6.82
Cyclooctane	8.03		
Dodecane (37.8°)	9.9	Acetic acid	9.08
Pentadecane (37.8°)	9.1	Ethyl acetate	11.32
Octadecane (60°)	9.4	Acetone	12.62
Benzene	9.44	Diethyl ether	18.65
Toluene	8.96		
m-Xylene	8.46	Water	4.58
		Mercury	0.40
Chlorobenzene	7.45		
Bromobenzene	6.46		
Anisole	6.60		
Aniline	4.53		
Nitrobenzene	4.93		
Phenol (60°)	6.05	Molten salts:	
		$NaNO_3$, 623 K	1.99
Carbon disulphide	9.38	KNO_3, 657 K	2.19
Carbon tetrachloride	10.50	$RbNO_3$, 629 K	2.26
Chloroform	9.96	$CsNO_3$, 709 K	3.09
Bromoethane	12.94		
Iodoethane	9.82		
1,1-Dichloroethane	7.97		
Tetrachloroethylene (25°)	7.56		
Trichloroethylene (25°)	8.57		

Table 2.3. Isothermal compressibilities of liquids. (After Isaacs [2.4])

	A	*B*	*C*
Water	4.79	+0.0008	3.44
Benzene	4.5772	−0.00343	5.962
Carbon tetrachloride	4.5325	−0.00343	6.099
Ethanol	4.5000	−0.00267	5.79
Liquid paraffin	4.710	−0.0016	7.235
Mercury	5.8837	−0.000488	4.95

Table 2.4. Constants for use in Huddlestone's equation, for various liquids.

PROPERTIES OF SATURATED HYDROGEN (PARA)

T K	P MPa	volume, m³/kg v_f	v_g	enthalpy, kJ/kg h_f	h_{fg}	h_g	entropy, kJ/(kg·K) s_f	s_{fg}	s_g
13.80	0.007042	0.012983	7.952	0.0	447.2	447.2	0.0	32.408	32.408
14	0.007896	0.013011	7.185	1.2	447.9	449.1	0.082	31.994	32.076
15	0.01343	0.013158	4.491	7.4	450.6	458.0	0.508	30.042	30.550
16	0.02153	0.013314	2.960	14.4	452.2	466.6	0.950	28.264	29.214
17	0.03284	0.013481	2.038	21.9	452.7	474.6	1.400	26.628	28.028
18	0.04807	0.013660	1.454	30.1	452.0	482.1	1.856	25.110	26.966
19	0.06796	0.013854	1.068	38.9	450.0	488.9	2.316	23.686	26.002
20	0.09326	0.014065	0.8045	48.3	446.8	495.1	2.781	22.337	25.118
20.28	0.101325	0.014127	0.7466	51.0	445.6	496.6	2.910	21.975	24.885
21	0.1247	0.014296	0.6185	58.4	442.0	500.4	3.251	21.047	24.298
22	0.1632	0.014550	0.4837	69.2	435.7	504.9	3.728	19.801	23.529
23	0.2094	0.014831	0.3837	80.8	427.5	508.3	4.214	18.586	22.800
24	0.2642	0.015147	0.3081	93.3	417.4	510.7	4.709	17.391	22.100
25	0.3284	0.015503	0.2496	106.7	405.1	511.8	5.216	16.202	21.418
26	0.4029	0.015911	0.2038	121.2	390.2	511.4	5.740	15.006	20.746
27	0.4885	0.016386	0.1672	137.0	372.3	509.3	6.284	13.787	20.071
28	0.5861	0.016951	0.1374	154.4	350.7	505.1	6.855	12.525	19.380
29	0.6967	0.017644	0.1129	173.7	324.4	498.1	7.467	11.186	18.653
30	0.8214	0.018532	0.09207	195.8	291.6	487.4	8.140	9.719	17.859
31	0.9615	0.019760	0.07387	222.2	248.6	470.8	8.916	8.020	16.936
32.94	1.284	0.031888	0.03189	346.5	0.0	346.5	12.536	0.0	12.536

PROPERTIES OF GASEOUS HYDROGEN (PARA)

P, MPa (T_{sat}, K)		T, K sat	100	200	300	400	600	800	1200	1500
0.101325	v, m³/kg	0.7466	4.070	8.147	12.22	16.29	24.43	32.57	48.85	61.06
(20.28)	h, kJ/kg	496.6	1399.8	2971.3	4509.6	5976.6	8880.9	11806.0	17833.5	22547.2
	s, kJ/(kg·K)	24.885	42.689	53.475	59.729	63.952	69.840	74.046	80.146	83.650
0.20	v, m³/kg	0.4006	2.061	4.130	6.194	8.257	12.38	16.51	24.75	30.94
(22.81)	h, kJ/kg	507.7	1398.3	2971.3	4510.1	5977.3	8881.7	11806.9	17834.3	22548.0
	s, kJ/(kg·K)	22.936	39.869	50.667	56.924	61.147	67.035	71.242	77.342	80.846
0.50	v, m³/kg	0.1631	0.8241	1.656	2.482	3.307	4.957	6.607	9.906	12.38
(27.12)	h, kJ/kg	508.9	1393.6	2971.5	4511.6	5979.4	8884.2	11809.5	17836.9	22550.6
	s, kJ/(kg·K)	19.986	36.046	46.880	53.143	57.367	63.257	67.463	73.563	77.067
1.0	v, m³/kg	0.06946	0.4119	0.8308	1.245	1.658	2.483	3.307	4.957	6.194
(31.26)	h, kJ/kg	465.1	1386.0	2971.7	4514.1	5982.8	8888.4	11813.8	17841.3	22554.9
	s, kJ/(kg·K)	16.664	33.114	44.008	50.280	54.507	60.398	64.605	70.705	74.209
2.0	v, m³/kg		0.2059	0.4184	0.6261	0.8328	1.245	1.658	2.482	3.101
	h, kJ/kg		1371.4	2972.3	4519.1	5989.7	8896.7	11822.6	17850.1	22563.5
	s, kJ/(kg·K)		30.114	41.122	47.413	51.645	57.539	61.747	67.847	71.351
4.0	v, m³/kg		0.1034	0.2123	0.3168	0.4204	0.6268	0.8329	1.245	1.554
	h, kJ/kg		1345.2	2974.1	4529.3	6003.5	8913.3	11840.0	17867.6	22580.8
	s, kJ/(kg·K)		26.993	38.213	44.538	48.781	54.681	58.890	64.990	68.494
7.0	v, m³/kg		0.06019	0.1240	0.1843	0.2436	0.3616	0.4793	0.7146	0.8911
	h, kJ/kg		1313.9	2978.3	4544.9	6024.2	8938.1	11866.0	17893.8	22606.6
	s, kJ/(kg·K)		24.349	35.834	42.207	46.465	52.374	56.584	62.685	66.188
10.	v, m³/kg		0.04345	0.08877	0.1312	0.1729	0.2556	0.3379	0.5025	0.6260
	h, kJ/kg		1292.9	2984.2	4561.0	6045.1	8962.9	11892.0	17920.0	22632.3
	s, kJ/(kg·K)		22.615	34.299	40.715	44.987	50.904	55.116	61.217	64.720
20.	v, m³/kg		0.02528	0.04795	0.06945	0.09043	0.1318	0.1729	0.2551	0.3167
	h, kJ/kg		1281.7	3015.9	4617.9	6115.4	9045.0	11978.2	18007.0	22718.0
	s, kJ/(kg·K)		19.291	31.274	37.794	42.105	48.047	52.265	58.367	61.869
50.	v, m³/kg		0.01551	0.02401	0.03257	0.04097	0.05751	0.07391	0.1066	0.1311
	h, kJ/kg		1456.2	3185.6	4816.6	6336.4	9290.7	12233.9	18265.8	22973.9
	s, kJ/(kg·K)		15.362	27.268	33.906	38.282	44.275	48.508	54.614	58.113
100.	v, m³/kg		0.01192	0.01608	0.02031	0.02448	0.03272	0.04086	0.05708	0.06923
	h, kJ/kg		1850.1	3553.2	5192.3	6725.3	9700.6	12655.0	18692.0	23396.7
	s, kJ/(kg·K)		12.618	24.314	30.983	35.397	41.433	45.682	51.794	55.291

Table 2.5. Thermodynamic properties of hydrogen. (After Reynolds [2.7])

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3 FRAGMENTATION

3.1 Fragment Generation

3.1.1 Introduction

The size and initial speed of fragments formed by a pressure vessel when it fails depend on the mode of failure. We can classify the relevant modes of failure as:

- complete fragmentation due to brittle fracture, usually into a large number of fragments;
- complete destruction of the vessel in a ductile manner, usually into a small number of rather large fragments: a typical example would be a large welded vessel where failure begins in a longitudinal seam, extends from end to end of the vessel and then runs around the circumferential seams joining the cylinder to the domed end–caps;
- loss of a major section of the vessel in a ductile manner: a typical example would be failure of the circumferential weld between the cylinder and one end cap;
- loss of a closure, such as a manhole in a large welded vessel or an end closure in a vessel where the closure gives essentially full–bore access to the vessel;
- loss of a plug or other small closure, which gives only limited access to the vessel.

In the last three cases, failure leads to the production of two fragments, one small and one large. If the large fragment is sufficiently massive or is adequately anchored, it may not move at all, so that only one genuine fragment is produced. On the other hand, the larger fragment may have the characteristics of a rocket, particularly if it is not adequately anchored, and may have the capability of doing considerable damage.

The object of this section is to survey the means available for the estimation of fragment masses and initial speeds. This information will be needed when assessing the protection needed to prevent injury and damage by such fragments.

3.1.2 Nature of pressurising fluid

During pressure–testing, the three most likely fluids to be used are water, hydraulic oil or a gas such as air or nitrogen. In particular circumstances, other fluids might be used but, as far as fragmentation is concerned, the range of properties associated with the ones selected here will cover almost all others.

The nature of the pressurising fluid is important in that it determines the total amount of energy available at a given test pressure and also the way in which all or part of this energy is converted to kinetic energy of the fragments. Figures 2.4 to 2.7 show the total amount of energy available for these three fluids as a function of pressure. Note that very much more energy is contained in a vessel pressurised with gas than for one pressurised with liquid. Other things being equal, this means that fragment velocities will be largest when a gas is used as pressurising medium. However, one cannot conclude that in all circumstances the velocities when a liquid is used will be very much lower. In some circumstances they will, in others not. These issues will be discussed further below.

The procedure adopted in this section will be first to consider fragmentation when gas is used as the pressurising medium and then to examine by how much these velocities should be changed when a liquid is used. Implicit in the argument is that the mode of failure is independent of the nature of the fluid. This has never been proved, but it appears to be a reasonable assumption.

3.1.3 Total failure due to brittle fracture

Although pressure vessels should never be made of a brittle material, mistakes can be made, whether during stock control or fabrication. It could also be that a vessel intended for high temperature use is made from a material with poor low temperature properties and is tested on a cold day when the temperature is below the brittle–ductile transition temperature.

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Experimental evidence [3.21, 3.27] points to all fragments having the same initial speed when a brittle vessel breaks up into a large number of fragments. An upper bound to fragment speed can therefore be obtained by setting total available energy equal to the kinetic energy of the whole vessel, that is:

$$E = \frac{1}{2} m V^2 \tag{3.1}$$

where E denotes available energy, m the mass of the vessel and V the initial speed of each fragment. Equation (3.1) over–estimates fragment speed, however, since as the fragments move apart, following fragmentation, gaps develop between the fragments through which fluid under pressure can escape and in some cases form an external shock wave or blast. Information on the extent of this over–estimation is patchy and incomplete, but that which is available will be examined here.

The failure of a pressurised vessel in a brittle manner is very similar to that of a shell full of explosive when it is detonated. One source of information on fragment size and speed is therefore the results of military tests on armaments. Moore [3.24], basing his studies on the experimental data of Gurney [3.25] and Sterne [3.26] for the detonation of explosives packed inside a casing, proposes that for a gas–filled vessel only 60% of the available energy goes into the kinetic energy of the fragments, a figure which he believes, on the basis of the very limited validatory evidence available to him from accidents involving pressure vessels, may well be an overestimate by as much as a factor of two.

Baum [3.27] treats the flow of gas between the fragments of an exploding vessel as undergoing choked flow (when the flow becomes unchoked towards the end of the gas expansion phase, acceleration of the fragments is essentially complete) and, by treating the gas as perfect, obtains a differential equation for the outwards force experienced by each fragment and, by integrating with respect to time, he derives an equation for the fragment speed at the end of the acceleration phase. In addition, he performed a number of experiments in which glass spheres and cylinders were pressurised with air (pressures of up to 28 bar were used) and then caused to fragment by impact. Fragment positions were recorded by high–speed cine photography, from which velocities were calculated.

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Baum's theory and experiments were in essential agreement and his results can be summarised very briefly as: for spherical vessels, 30% of the available energy was transformed into kinetic energy of fragments, for cylindrical vessels, this figure increased to 40%. However, due to his use of a different method of computation of the available energy, these figures need to be reduced by some 5–10% to put them on the same basis as are used in this report.

There is thus good agreement between the proposals of Moore and Baum provided we accept Moore's assessment of the magnitude of the over–estimate which results from using the high explosive results directly. We suggest, therefore, that 35% of available energy be used in estimating fragment speeds, based on the more conservative of the two figures proposed by Baum, since cylindrical vessels are more common than spherical. The balance of the available energy, 65% of the total, is presumed to be the amount which goes into the shock wave or blast associated with the vessel failure.

The situation with regard to liquid–filled vessels is less satisfactory in that no experimental information is available. Undoubtedly, a similar situation exists in that not all of the available energy is translated into kinetic energy of the fragments; some of the pressurised fluid will escape past the fragments as they fly apart during the explosion phase. However, the fluid mechanics associated with this are very different to those applying to the release of a gas and it is unwise to assume that similar quantitative provisions apply. Accordingly, the only conservative solution is to assume that for the purposes of estimating fragment speeds, all available energy is communicated to the fragments. As before, we assume that all fragments have the same speed.

All of these tests, and the theories behind them, apply to vessels with essentially uniform wall thickness. Real vessels have features such as manholes, nozzles, flanges and screwed closures where the mass per unit area of vessel surface is significantly larger than in the main vessel shell. We suggest that the influence of these features should be handled in the following way:

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- determine the mass of a vessel of the same overall shape and wall thickness as the actual vessel but without the massive features;
- use this value of the mass to determine the speed of the main vessel fragments;
- determine the ratio of the mass of each massive feature to the mass of the part of the main vessel shell which it replaces. If we assume that the force on each massive feature is the same as that acting on the part of the main vessel shell which it replaces and that it acts for the same period of time, the kinetic energy acquired by the massive feature will be the same as that part of the main shell. Thus, the speed of the massive feature is then the speed of the main vessel fragment divided by the square root of the ratio of the masses.

The number and size of fragments produced during brittle failure is in general unknown, apart from the general observation that the number of fragments produced is usually large. We postulate, as a working procedure when assessing the requirements for protection against failure by brittle fracture that one assumes the largest fragment is 20% of the shell, the smallest 1% of the shell. In addition, there are the identifiable items which may be ejected intact: end caps, end closures, manhole covers, nozzles and so on. Each of these items should be identified and the mass, size and shape and speed of each determined prior to the assessment of protective requirements.

3.1.4 Total failure due to ductile fracture

As indicated above, total failure due to ductile failure is likely to result in the formation of a small number of rather large fragments and one should be able to identify these by inspection since it is most probable that the fractures will develop along lines of weakness, such as welds.

The assumption that all fragments will acquire the same initial speed is less likely in this situation than it was in the case of brittle fracture, but no experiments have been carried out to determine whether or not this is the case. Moreover, there is no reason to suppose that

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the same fraction of the available energy is translated into kinetic energy of the fragments, but again there is no evidence with which to test this hypothesis.

We suggest that a conservative solution to this problem be obtained by treating the total failure of a vessel as a series of failures in which only one part of the vessel is ejected at a time, that is, it is subsumed within the following case.

3.1.5 Loss of a major section

The most likely major section to be lost in a ductile failure is a closure. This could be a domed end in a welded vessel, a flanged closure or a screwed end plug or cap. The assumed failure modes include failure of the circumferential weld between a domed end and the cylindrical part of the vessel, failure of the bolts holding a blind flange on to a nozzle, failure of the nozzle to shell or nozzle to flange weld, failure of the threads in the end plug or cap.

In all cases, the expected scenario for a gas–filled vessel involves acceleration of the closure in a two–stage process. In stage 1 gas escapes from the vessel through the developing circumferential gap between the closure and the vessel, with choking taking place on the cylindrical front thus created. This stage persists until the fragment displacement reaches $\frac{1}{2}R$, where R is the radius of the breach, at which point, the limiting flow area becomes the circular hole in the vessel itself and choking takes place on this plane. To a first approximation, therefore, one can treat the force on the fragment as constant during stage 1 and equal to the initial pressure in the vessel acting over the area of the vessel opening (this assumes that the volume of gas in the vessel is sufficiently large for this to be reasonable). During the second stage, the situation is more complex since the jet leaving the choked vessel opening will undergo expansion downstream and exert a varying force on the fragment which will depend on the relative dimensions of the vessel opening and the fragment as well as properties of the fluid.

Moore [3.24] proposed that the initial closure acceleration should be treated as being maintained until the fragment was clear of the vessel by an amount equal to the diameter of the hole left in the vessel, that is, the initial pressure within the vessel is maintained on the underside of the closure fragment until this clearance is reached. The total distance over which this force would operate would therefore be the diameter of the hole plus any additional travel which the closure would have to make before a clear leakage gap was formed (this latter would thus be very important for screwed plug closures but irrelevant for domed ends on a welded vessel).

Baum [3.28] has conducted a series of tests in which a flat end cap was rested on the end of a cylindrical pressurised vessel (but held in place). When released, high speed cine films were taken of the ejected closure so that speeds could be determined. His results show that Moore's procedure is essentially correct for the sort of closure used on welded vessels (that is domed ends, flanges and the like), but could significantly over–estimate the speeds when the mass of the closure was high relative to the minimum thickness required of the closure. This situation is most likely to arise with the screwed–plug or end–cap closure where thread–strength limitations lead to the use of over–long plugs.

Although it would be nice to take advantage of this overestimate of fragment speed, which could amount to as much as a factor of two in some cases, the complications involved are such that it may well be better to use the Moore formula in all routine cases, but if that leads to unacceptably high speeds, one could accept the more detailed analysis provided that one can be sure that it is done properly.

In the case of a liquid–filled vessel, the fluid pressure within the vessel will decay very much more rapidly than when it is gas–filled. Therefore, the assumption that the acceleration of the closure remains constant until a gap equal to one diameter has opened up may no longer be true. The assumption will be closest to being valid when the volume of the vessel is very large compared with the volume swept out by the closure during its escape. This suggests the following procedure to evaluate the speed of the closure:

- evaluate the speed assuming the material within the vessel is a gas;
- assume that the whole of the stored energy within the vessel (based on a liquid–filled vessel) is converted into kinetic energy of the closure and hence determine a speed for this case;
- take the actual speed as the smaller of the above two speeds.

3.1.6 Speed of a rocketing fragment

An analysis of the rocketing motion of a vessel which has lost its end can be undertaken by assuming a two–stage process:

- in stage I, which incorporates the period when expansion waves move up and down the vessel: assume that conditions in the vessel do not differ significantly from the initial conditions;
- stage I lasts until the vessel has lifted by a height equal to a quarter of its diameter: thereafter the area for efflux becomes constant at the cross–sectional area of the vessel;
- in stage II, assume that there is reversible adiabatic (isentropic) decay of pressure of a perfect gas inside the vessel;
- stage II lasts until the acceleration of the vessel reaches zero and its velocity is a maximum.

If in addition it is assumed that flow out of the vessel is choked and that gravitational effects are negligible, it can be shown that the maximum velocity $V_{\max}$ of the vessel is given by:

$$\begin{aligned} V_{\max} &= \left[\alpha\,(\beta-1)\,\frac{D}{2\,\beta}\right]^{\frac{1}{2}} + \frac{\alpha}{\kappa}\left[\frac{2}{\gamma+1}\right]\left[1-\beta^{-(\gamma+1)/2\gamma}\right] \\ &+ \frac{\alpha}{\beta}\left[\left[\frac{\beta\,D}{2\,(\beta-1)\,\alpha}\right]^{\frac{1}{2}} - \left[\beta^{(\gamma-1)/2\gamma}-1\right]\frac{1}{\kappa}\left[\frac{2}{\gamma-1}\right]\right] \end{aligned} \tag{3.2}$$

Here: $\alpha = p_0 \frac{\pi}{4}\frac{D^2}{m}\ (\mathrm{m/s^2})$;

$\beta = \frac{p_0}{p_a}$;

$\kappa = \left[\frac{\gamma\ R\ T_0}{M}\right]^{\frac{1}{2}} \frac{C_d}{L}\left[\frac{2}{\gamma+1}\right]^{(\gamma+1)/2(\gamma-1)}\ (1/\mathrm{s})$;

p_0 is the initial pressure in the vessel (Pa);

p_a is the ambient pressure (Pa);

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T_0 is the initial temperature (K);

L is the inside length of the vessel (m);

D is the inside diameter of the vessel (m);

m is the mass of the vessel (kg);

C_d is the discharge coefficient (–): a value of unity is suggested;

γ is the specific heat ratio of the gas (–);

M is the molar mass of the gas (kg/kmole);

R is the universal gas constant (8.3143 K/mole·K).

3.1.7 Loss of a plug or small closure

The situation here is not dissimilar to that for loss of a major section (Section 3.1.5) except that the volume of the vessel is assumed to be so large that there is negligible loss of pressure until the plug is well removed from the vessel. The force on the plug is therefore maintained at a higher level for a rather longer period of time. The only experiments conducted into this situation are those of Baum [3.29] who investigated the ejection of a small plug by a gas–filled vessel. These are consistent with the assumption that the full system pressure remains acting on the plug until the clearance between the tail end of the plug and the breach in the vessel reaches twice the diameter of the plug (it was only half this in Section 3.1.5). In the absence of any information to the contrary, we propose using this same procedure for liquid filled vessels as well but with the proviso that the total kinetic energy of the missile should not exceed the stored energy of the fluid in the vessel.

3.2 Protection against fragment perforation

3.2.1 Introduction

Most work on missile penetration of targets is of ordnance origin, although in recent years, the importance of 'industrial missiles' and the damage which they can cause to plant has been recognised and some research has been directed to this aspect, mostly by the nuclear

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industry, which is concerned with the damage to other parts of the plant caused by missiles which result from failure of another, pressurised, part of the plant. This distinction is important since, as was stated above, fragments (missiles) produced by failure of pressurised equipment tend to have speeds which rarely exceed 500 m/s, whereas ordnance missiles tend to have speeds in excess of 500 m/s and the mechanism of penetration at lower speeds (a few hundreds of m/s) is not the same as at nearer 1000 m/s, thus making the extrapolation of ordnance results to lower speeds a somewhat uncertain operation.

3.2.2 Terminology

In this section the following terms will have the meaning indicated.

- Penetration: this occurs when the missile embeds itself within the target but no evidence of this can be seen on the non–impact side.
- Perforation: this occurs when the missile, or debris from the target in front of the missile, appears on the non–impact side of the target.
- Complete perforation: this occurs when the missile emerges from the non–impact side of the target and continues on its way, although with a reduced speed when compared with the initial impact speed.

Experiments show that the damage caused by a missile on a given target can be very variable. This is reflected in the fact that all quantitative statements of target response are based on the 50% measure. Thus, for example, if a given missile is said to just perforate a given target, the actual situation is that 50% of the missile tests resulted in complete perforation and 50% resulted in penetration or perforation. Unfortunately, the degree of variability in a series of experimental measurements is rarely reported.

3.2.3 Shielding materials

Although all materials provide some protection against missiles, the objective is simply one of absorbing the kinetic energy of the missile by something of very much larger mass, not

very many have been experimentally investigated. Most work has concentrated on ductile metals (usually steel or aluminium) and brittle structural materials such as concrete and, to a lesser extent, brickwork. Reports on other materials, such as polycarbonate, fibrous composites and textile–type materials are found infrequently. A reasonable body of knowledge also exists for penetration into soil. Much as one might like to discuss the use of all of these materials together as a single class, this is simply not possible as the ways in which they are penetrated by missiles differ considerably. We must therefore look at each class separately.

3.2.4 Protection provided by ductile shields

The most commonly used, and investigated, metallic shielding material is mild steel. Most fragments from a failing pressure vessel will be significantly harder than mild steel, so it is safe to assume that we are primarily concerned with the case in which the projectile retains its essential shape during impact with the shield. This situation will also apply when an aluminium alloy is used for shielding.

When a fairly slow–moving blunt missile approaches a thin ductile shield in a normal (perpendicular) direction (but the energy is such that perforation will just take place), the missile first causes an indentation in the shield (see Figure 3.1) before shearing a plug of the shield material, essentially of the same diameter as that of the missile (see Figure 3.2) and allowing the missile through, preceded by the plug. The kinetic energy of the missile is therefore absorbed in the initially elastic, but then plastic, bending and stretching of the shield material as the indentation is formed, followed by the absorption of the remainder of the energy during the shearing of the plug. However, if that same blunt missile, but at a higher speed, impacts a thicker shield, again such that perforation just takes place, the rigidity of the shield reduces the extent of the indentation and hence the amount of energy absorbed by this mechanism, leaving more of it to be absorbed during shearing of the shield to form a plug. In contrast, missiles at higher, ordnance, speeds produce very little plastic deformation prior to perforation.

This competition between the energy absorbing mechanisms is well illustrated by the results obtained by Corran [3.4] for 12.5 mm cylindrical missiles impacting mild steel plates at 50–250 m/s. Figure 3.3 shows the distribution of energy between the various modes of energy absorption. Most notable is the way in which the energy absorbed by shearing a plug of metal from the shield in front of the missile (the plugging work) increases from essentially zero for a thin shield to 20% when the shield is 6 mm thick. Similarly notable is the fall of the energy of stretching the metal of the shield, the membrane plastic work, from more than 50% for a thin shield to essentially zero for a thick one. That the stretching is greatly reduced for a thick plate is easily seen in Figure 3.4 for stainless steel shields of thickness 1.3 mm to 6.5 mm (the speeds given are the perforation speeds).

Another complication arises from the shape of the face of the missile which impacts with the shield. Corran investigated the effect of the curvature of the front surface of the 12.5 mm missile impacting on 1.3 mm mild steel shields. The results are summarised in Figure 3.5. For nose radii below 11 mm, the perforation was purely tensile, often with petalling (see Figure 3.6) with no plugging whatsoever. For radii in excess of this, a plug of shield was created by shearing. For all radii, the deformation was large, so a large portion of the kinetic energy was still absorbed by plastic deformation.

Not unrelated to the issues raised in the preceding paragraph is the situation where the missile impacts the shield obliquely. Here, even a flat–ended cylindrical missile may perforate the shield without forming a plug.

These results indicate that perforation, and by implication, penetration which stops short of full perforation, is a complex balance of phenomena involving both the missile and the shield or target. It indicates that a simple all–embracing theory to cover all sizes and shapes of missile and all thicknesses and choice of material for the target is unlikely to be achieved. One is therefore limited to consideration of empirical correlations. Generally these are developed to consider a limited range of variables and it is clear from the above discussions that the use of the correlations outside the range of fitting is dangerous in the extreme.

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However, given sufficient of these correlations, it might be possible to develop an overall safety envelope around them, but it must be realised that such an envelope may well be very conservative.

The actual development of such an envelope is complicated by the incomplete knowledge of the range of applicability of the correlations available. Very rarely are the actual experimental results published by the ordnance organisations and only rarely do they include the full range of limits of applicability.

3.2.5 Correlations for ductile materials

<u>Recht</u>

Recht appears to be responsible for a number of similar but semi–empirical models of the penetration process. Unfortunately, the original literature is not accessible to us (it is mostly American military research) and one is reduced to working with correlations which have been published. In many cases it is not clear whether the correlations are Recht's own or are extensions to it. For our purposes, the most useful version is that of Smith and Hetherington [3.17] since it expresses penetration and, in the limit perforation, in terms of readily available properties of the shield material. The perforation version of this equation is, after making a number of approximations and simplifications to make it more appropriate to present needs:

$$t = 1.61 \frac{M}{b\ A} \left[V - \frac{a}{b} \ln(1 + [b\ V/a]) \right] \qquad (3.3)$$

Here: t is the thickness of the target to resist perforation by 50% of missiles (m);

$a = 2\ \tau \ln(2\ z)$;

$b = 0.25\ (K\ \rho)^{0.5}$;

$z = (E/\sigma_y)/(1 + 2\ E/\sigma_y)^{0.5}$;

M is the mass of missile (kg);

A is the presented area of the fragment (m²);

V is the speed of the missile (m/s);

ρ is the density of the target material (kg/m³);

K is the bulk modulus of the target material (Pa);

E is the Young's modulus of the target material (Pa);

σ_y is the yield strength of the material in tension (Pa);

τ is the compressive shear strength of the material at failure (Pa).

Equation (3.3) holds for blunt cylindrical missiles impacting normally on the target. (It would appear that the original work of Recht was on conically–nosed missiles, so the extension to blunt missiles may be unproven.) Other versions of this equation may be found in equation 22 of Recht [3.12] and equation 7.7 of Backman [3.10].

Smith and Hetherington state that equation (3.3) is limited to missiles with a length/diameter ratio of less than 10 and for missile speeds of less than 1000 m/s. They also state that this equation 'has been shown to provide good estimates of penetration for a wide range of materials including plastics, ceramics, metals and concrete'. This makes it a particularly attractive equation for predicting the protective power of materials for which penetration and perforation experiments have not been performed.

The range of missile and target sizes and missile speeds over which equation (3.3) has been validated is unknown, but we suspect that only small missiles (bullets) have been used. Recht [3.12] refers to experiments with missiles of 6.25 mm and 10.8 mm diameter and masses of 5.3 g and 26.6 g, respectively. He also refers to experiments with shields of 3.2 mm, 6.4 mm and 9.6 mm mild steel plate and missiles 21 mm long, leading to perforation speeds of 141 m/s, 290 m/s and 470 m/s. Unfortunately, insufficient information is given to enable one to check whether any of these measurements are consistent with the equation.

<u>DeMarre</u>

This equation was produced by the US Naval Ordnance Laboratory in 1955. It is quoted by Recht [3.12] and when converted into SI units gives:

$$t = 4.6\times10^{-6}\, M^{0.333}\, V^{1.333} \qquad (3.4)$$

Here: t is the thickness to prevent perforation by 50% of missiles (m);

M is the mass of the missile (kg);

V is the speed of the missile (m/s).

The text implies that equation (3.4) applies to mild steel shields. Other than this, no information is available as to the limits of applicability of the equation.

Stanford Research Institute (SRI)

The SRI correlating equation appears to be based on tests conducted with steel missiles 7.5 mm in diameter with lengths in the range 75–375 mm fired at rigidly–supported steel targets with speeds in the range 20–120 m/s (Tulacz [3.13]). The steel targets were up to 7 mm in thickness (Nielson [3.15]). The targets were normally clamped at 100 mm centres, a spacing which was sufficiently close to influence the perforation speeds, so some measurements were carried out at other spacings to quantify the effect; however, the range of spacings used is not known. The steel targets used had a nominal ultimate tensile strength of 321 MPa (Nielson [3.15]). The correlating equation may be expressed as:

$$E/d = \sigma_u (4.15\, t^2 + 0.01\, t) \qquad (3.5)$$

Here: E is the kinetic energy of the missile (J);

d is the diameter of the missile (m);

t is the thickness of the target (m);

σ_u is the ultimate tensile strength of the target (Pa).

Although equation (3.5) has ultimate tensile strength as an explicit parameter, it should be remembered that its value was not varied during the tests, other than for the natural variability in the properties of the target material, so its status in the position indicated is not validated.

Ballistics Research Laboratory (BRL)

Very little is known about the experiments behind the BRL correlation but the obvious suspicion is that they involved the impact of high velocity bullets and shells on steel targets. Converting the equation given in Brown [3.19] into SI units gives:

$$t = 4.9{\times}10^{-7} (M\, V^2)^{0.667} / d \qquad (3.6)$$

Here: t is the thickness of the shield for perforation by 50% of missiles (m);

M is the mass of the missile (kg);

V is the speed of the missile (m/s);

d is the diameter of the missile (m).

NDRC

NDRC, in 1946, surveyed the information available at that time for the impact of hard metal missiles in steel plate. This information is not available to us, only reports derived from it. However, these secondary reports are not themselves consistent, so there is some uncertainty as to exactly what was reported in the NDRC document. We have, therefore, chosen to work with the formulation given by Christopherson [3.21] and which is identical to the one given in Tulacz [3.13]. This formulation is largely verifiable by comparison with the examples given in the latter paper. We note in passing that Christopherson states that the formulation is for the perforation of steel armour plate rather than mild steel, a fact which is not picked up in any subsequent publication. Converted into SI units, the correlation is:

$$t = 1.5\times10^{-7} M^{0.71} V^{1.42} / d^{1.13} \tag{3.7}$$

Here: t is the thickness of the shield for perforation by 50% of missiles (m);

M is the mass of the missile (kg);

V is the speed of the missile (m/s);

d is the diameter of the missile (m).

The ranges of variables over which equation (3.7) is valid are known only imperfectly. Brown [3.19] suggests that it is likely to have been concerned only with speeds in excess of 300 m/s. He also suggests that missiles of up to 80 mm diameter may well have been involved. Christopherson [3.21] states that the correlation was based in part on missiles in the 100–900 kg range with diameters of up to 330 mm and with targets of thickness of up to 60% of the missile diameter. The speeds of these large missiles (actually armour–piercing bombs) is not specified other than that they would not exceed 300 m/s.

3.2.6 Comparison with experimental data

In Table 3.1, we compare the various empirical correlations with the rather limited amount of published experimental data for perforation speeds of hard steel missiles in softer steel targets. We have limited the comparison to those cases where all relevant information is available, namely, diameter and length of missile, speed of impact and thickness of target. The actual steel used for the target was usually referred to as mild steel but rarely were actual material properties listed. It is probable therefore that the actual target properties vary from test to test. The best overall performance is provided by the BRL correlation. In most cases it is slightly conservative, predicting the need for a slightly thicker shield than is actually required. The exceptions to this conclusion all lie in the region of very thin shields. Second best are the Recht and NDRC correlations, predicting slightly under–conservative thicknesses and again failing for the thinnest shields. The DeMarre and SRI correlations are unacceptable.

Table 3.1

Ref	d mm	M kg	V m/s	t mm	Recht mm	DeMarre mm	SRI mm	BRL mm	NDRC mm
15	32	4.3	11	1.2	0.3	0.2	0.4	1.0	0.6
2	32	4.3	12.5	1.0	0.4	0.2	0.4	1.2	0.7
2	32	4.3	9.5	1.0	0.2	0.1	0.3	0.8	0.5
2	63	19.8	80	15.0	14.8	4.2	7.7	19.3	14.3
2	63	19.8	102	25.0	22.7	5.8	9.9	26.7	20.2
2	85	9.7	79	6.0	3.9	3.3	4.4	8.8	6.0
2	85	9.7	99	12.0	5.8	4.4	5.6	11.8	8.3
2	43	1.22	98	6.0	2.8	2.2	2.5	5.8	4.1
2	40	1.0	43	1.0	0.6	0.7	0.8	1.8	1.2
2	40	3.0	83	3.0	2.0	1.6	1.9	4.4	3.0
22	6.23	.006	298	4.8	4.1	1.6	1.2	5.1	4.0
22	6.23	.006	79	1.6	0.4	0.3	0.2	0.9	0.6
23	25.4	0.45	300	25.0	18.7	6.9	6.8	22.3	17.8
23	12.5	.035	67	1.3	0.5	0.4	0.3	1.1	0.8
23	12.5	.035	90	1.9	0.8	0.6	0.5	1.7	1.2
23	12.5	.035	135	2.7	1.7	1.0	0.9	2.9	2.1
23	12.5	.035	155	3.0	2.1	1.2	1.0	3.4	2.5
23	12.5	.035	186	5.0	2.8	1.6	1.3	4.4	3.3
23	12.5	.035	188	6.0	2.9	1.6	1.3	4.4	3.3

Note that the BRL correlation predicts the thickest shielding requirement for all of the cases listed in Table 3.1. Figures 3.7 to 3.9 show that this is also true for a very much wider range of variables. Thus, if we presume that the correlations were all a satisfactory representation of the perforation speeds of the measurements used in performing the correlations, then by choosing the BRL correlation for the design of protective shields in high pressure work, one can be sure that one embraces, in a conservative manner, all known measurements. We propose, therefore, to select the BRL equation as the design equation for steel protective shields for use in high pressure work. Equation 3.6 gives the thickness of steel required to prevent perforation by 50% of the missiles. BRL suggest increasing this thickness by 25% in order to stop perforation by all missiles, an adjustment which we believe should be made. In addition, in order to avoid the poor performance for thin shields we propose putting a lower limit of 3 mm on all steel shield thicknesses.

3.2.7 Protection with reinforced concrete

Missile impact on a reinforced concrete wall will result in local damage, provided the strain energy capacity of the latter is larger than the kinetic energy of the missile. This consists of spalling of concrete from the front (impacted) face and scabbing from the rear together with missile penetration into the wall. If damage is sufficiently great, perforation will result. Figure 3.10 illustrates this behaviour for increasingly energetic missiles. A low energy missile penetrates the concrete and causes spalling (a). A more energetic missile penetrates further, without perforating, but causes scabbing and hence secondary missiles (b). One more energetic still perforates the wall (c); (d) illustrates the situation when the wall is insufficiently anchored to absorb the kinetic energy of the missile and the whole wall moves.

Protective shield design with concrete must be such as to prevent situations (c) and (d). Situation (b) should also be avoided where possible, but if this is not feasible, the hazard caused by scabbing can be removed by attaching a properly designed 'scab' plate, usually of steel, to the rear face of the wall. If a scab plate is used, the wall should be sufficiently thick to prevent perforation. Christopherson [3.21] suggests that this 'scab' plate should be 5 mm

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in thickness and he draws attention to the necessity for it to be securely attached to the concrete since it has to resist the impact of the concrete 'scab' without itself being dislodged.

In his review, Kennedy [3.20] demonstrates that although several correlations have been put forward for the penetration and perforation of concrete, only one of them, the NDRC correlation, fits all of the experimental data, but it is important to recognise that this does mean all available data. The NDRC correlation is based on an approximate theory of penetration, rather than being purely empirical; hence one can have greater confidence in the extrapolation of its results. We therefore concentrate on this correlation to the exclusion of all others, particulary as all subsequent work demonstrates that the NDRC correlation continues to be accurate. It is generally stated that the NDRC correlation predicts wall thickness to within 20% for the range over which it has been validated. Experimental evidence [3.21] points to the fact that penetration and perforation of concrete is independent of the amount of reinforcing present, provided that there is sufficient for the wall not to disintegrate totally.

The original version of the NDRC correlation dates from 1946 and drew heavily on the work on bombs for penetrating concrete during the second world war. However, crucial parameters in the correlation were classified and it was not until very much later (1966) that theoretical and experimental considerations enabled the missing information to be made good. At this point the correlation was valid for the calculation of penetration, perforation and scabbing thicknesses of concrete for missile diameters up to 400 mm with speeds in the range 170–900 m/s and for wall thicknesses/missile diameter in the range 3–18. Tulacz [3.13] believes that the correlation was validated for missiles with masses in the range 0.2–1200 kg.

The final step was to extend the validity of the correlation for wall thickness/missile diameter ratios below 3. In 1972, it was extended all the way down to zero by Kennedy.

To use the NDRC correlation, one must first calculate the depth of penetration into an infinitely thick slab of concrete from:

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$$G_{(x/d)} = 2.55\times10^{-9}\, K\, N\, M\, V^{1.80}/\, d^{2.80} \qquad (3.8)$$

where: $G_{(x/d)} = (x/2d)^2$ if $G_{(x/d)} \leq 1$ (that is $x/d \leq 2.0$)

and: $G_{(x/d)} = (x/d) - 1$ if $G_{(x/d)} \geq 1$ (that is $x/d \geq 2.0$)

Here: $K = 15000/\sigma_c^{0.5}$;

σ_c is the ultimate compressive strength of concrete (Pa);

N varies from 0.72 for a flat–nosed missile to 1.14 for sharp–nosed missile;

M is the mass of the missile (kg);

V is the speed of the missile (m/s);

d is the diameter of the missile (m);

x is the depth of penetration into an infinitely thick block of concrete (m).

The perforation thickness t (m) is then calculated from:

$$t/d = 3.19\,(x/d) - 0.718\,(x/d)^2 \qquad \text{for } x/d \leq 1.35$$

$$t/d = 1.32 + 1.24\,(x/d) \qquad \text{for } x/d \geq 1.35$$

The scabbing thickness s (m), the thickness which just resists scabbing, is calculated from:

$$s/d = 7.91\,(x/d) - 5.06\,(x/d)^2 \qquad \text{for } x/d \leq 0.65$$

$$s/d = 2.12 + 1.36\,(x/d) \qquad \text{for } x/d \geq 0.65$$

This thickness s is the required thickness for a plain reinforced concrete wall to resist attack by the specified missile. If a scab plate is used, the thickness of concrete can be reduced to t.

3.2.8 Protection with polycarbonate

Polycarbonate sheet or plate is used as a lightweight transparent armour for protection against missiles at sub–ordnance speeds in applications which include safety goggles, machine guards, aircraft windscreens and police riot shields. It is therefore worth considering for the provision of protection during pressure testing. Polycarbonate has superior impact and perforation resistance compared with other polymers and it is surprising that very little work has been published on the penetration and perforation resistance of the material. We have been able to find only two significant studies, [3.30] and [3.31].

In order to assess this experimental information, we have used Recht's formulation

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(equation 3.3) as a basis since Smith and Hetherington [3.17] consider it valid for a wide range of materials, including plastics. Recht's equation, however, requires values for a number of mechanical properties of the material and not all are readily available. Additionally, plastics can have a wider variability in properties than, say, mild steels, so there will be more uncertainty in the actual properties of the polycarbonate used than there would be for mild steel. Furthermore, the mechanical properties of plastics show a greater dependence on strain rate than do metals and missile impingement is essentially a high rate process. We have selected the following values for the relevant mechanical properties as being representative of an ordinary polycarbonate under high rates of strain:

- K (bulk modulus) 4×10^9 Pa;
- E (Young's modulus) 2.4×10^9 Pa;
- σ_y (yield strength) 100×10^6 Pa;
- τ_u (ultimate shear strength) 65×10^6 Pa.

Using these values in equation (3.3), we obtain Figure 3.11 giving the thickness of polycarbonate as a function of missile speed for a range of values of missile mass (kg) divided by fragment presented area (m^2).

Wright [3.30] reports results for the impact of 7 mm diameter steel spheres and cylinders of length 9 mm impacting on plain polycarbonate sheet and Ciolek [3.31] for the impact of missiles of mass 0.31 kg with a fragment presented diameter of 6.4 mm on laminated polycarbonate sheet. A comparison of these results with those predicted by the Recht equation is given in Table 3.2

Table 3.2

Ref	d mm	M g	V m/s	t(expt) mm	t(pred) mm
30	7	1.4	215	2	4
30	7	1.4	320	5	8
30	7	1.4	510	12	18
30	7	2.7	125	2	3
30	7	2.7	210	5	7
30	7	2.7	350	12	18
31	6.4	310	32	9.5	29
31	6.4	310	38	12.7	41

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In general, the polycarbonate performed better than the predictions of the Recht equation, not excessively so in the case of the small missiles, but by a factor of 3 for the 1.2 m long rods considered in [3.31]. This poor performance in the case of long rods could be due to the range of validity of the Recht equation being limited to a length/diameter ratio of 10 or to the fact that the target was of laminated construction, presumably to increase its ability to resist impact.

Overall, we believe the Recht equation appears to provide reasonable predictions and on the limited amount of evidence available, these predictions are conservative. We suggest, therefore, that this equation be used for the determination of shield thicknesses for polycarbonate although recognising that its use involves greater uncertainty than is the case for concrete and mild steel.

3.2.9 Penetration into soil

The process of penetration of missiles into soil differs considerably from all of the other cases considered so far in that soil is a soft material, often not very homogeneous. As a result, small deflections of a compact missile (that is with a length–to–diameter ratio of order one) are not resisted and the missile tends to execute a tumbling motion through the soil. On the other hand, a long rod–shaped missile entering the soil end on will find it rather less easy to tumble. In addition, the path taken by the missile through the soil is rarely straight and typically, the depth of penetration is only 80% of the path length taken by the missile. One consequence of these factors is that penetration depth is very variable.

Most experimental information on penetration into soil dates from the second world war or earlier, but with some additional experimental work carried out at Sandia Laboratories (USA) in the 1960s. Again, much of this information is not available to us and the results presented here are culled, second–hand, from other sources.

For compact missiles with a length/diameter ratio of order one, i.e. those which tumble

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through the soil, one might expect that the fragment presented area is of little consequence and that the mass of the missile is the main determinant of the depth of penetration (as well as the speed of course). This is the logic behind the NDRC formula, which has the form:

$$x = f(V)\, M^{0\cdot333} \tag{3.9}$$

where x is the penetration depth (straight) (m);

M is the mass of the missile (kg).

However, no directly useable expression for $f(V)$ appears to have been published.

Christopherson [3.21] presents some of the results, for bullets of up to 20 mm diameter, on which this equation was based and gives graphical representations of $f(V)$ for various soils. He then uses these representations of $f(V)$ to determine the predictive power of the equation for bombs of up to 1 m diameter dropped during the second world war. The comparison is necessarily rough, but nevertheless, remarkably good. We have, therefore, determined an analytic expression for $f(V)$ which gives a reasonable fit to the Christopherson data:

$$x = 0.40\, S\, M^{0\cdot333} \ln(1 + 5.4\times10^{-4}\, V^2) \tag{3.10}$$

where x is the penetration depth (straight) (m);

M is the mass of the missile (kg);

V is the speed of the missile (m/s);

S is the soil parameter: 0.5 for sand,
1.0 for average soil,
2.0 for soft soil.

For long missiles (length/diameter > 10), Smith and Hetherington [3.17] recommend the use of the equation developed by the Sandia Laboratories:

for speeds < 61 m/s:

$$x = 6.06\times10^{-3}\, S\, N\, (M/A)^{0\cdot5} \ln(1 + 2.15\times10^{-4}\, V^2) \tag{3.11}$$

for speeds > 61 m/s:

$$x = 1.16\times10^{-4}\, S\, N\, (M/A)^{0\cdot5}\, (V - 30.5) \tag{3.12}$$

where the soil parameter S = 4–6 for densely packed sand,
8–12 for stiff clay,
10–15 for topsoil;

and the nose shape factor $N = 0.56$ for blunt–nosed missiles.

All other variables are as for equation (3.10). These equations are stated to be valid for M in the range 27–2600 kg, but this must be an error.

These equations for penetration into soil must be treated as subject to a great deal more uncertainty than those presented earlier for mild steel and concrete. There is an equally large uncertainty in the amount by which these penetration depths should be multiplied in order to determine the thickness of a free standing earth barrier which will not be perforated by the missile. Brown [3.19] suggests that it should be a factor of 3.

3.2.10 Multilayer shield construction

Several attempts have been made to determine whether these is any advantage in the use of multilayer construction of shields as regards perforation by missiles. The results are inconclusive in the case of shields made from steel, whether the separate layers form a close sandwich or whether they are separated by an air gap. In some tests, a small improvement and in others a small degradation was found in comparing a single layer construction with a multilayer one of the same overall thickness [3.23]. We believe, therefore, that there is no case for treating a multilayer steel shield as having a superior performance.

The case with respect to multilayer concrete shields is more clear cut: sub–dividing a shield into more than one layer severely impairs the stopping power of the shield. This is because there are now two or more surfaces on which scabbing can occur and this reduces the effective thickness of the shield when compared with a shield with a single such surface.

3.3 Examples

3.3.1 Brittle failure of gas–filled vessel

A vessel of internal diameter 1.0 m and internal length 3.0 m with a cylindrical wall thickness of 20 mm and flat ends of thickness 80 mm is tested at 100 bar with nitrogen.

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The fragments formed during brittle failure of this vessel may include:

• fragments of the cylindrical part of the vessel ranging from 1 to 20% of the mass of the vessel (as suggested in Section 3.1.3);

• ends of the vessel which remain intact (particularly if initiation of the brittle failure is in the cylindrical part of the vessel);

• fragments caused by disintegration of the ends of the vessel.

As a first pass in the analysis, we will consider fragments which represent 1 and 20% of the mass of a vessel with a uniform wall thickness of 20 mm and ends which remain intact. Once this analysis is complete one can determine by inspection whether consideration of further fragments needs to be made.

• Mass of cylindrical part of vessel = 1470 kg.

• Mass of each end of vessel = 490 kg.

• Internal surface area of vessel = 11.0 m^2.

• The stored energy for this vessel was determined in Section 2.8.3 as 39 MJ.

• Section 3.1.3 indicates that 35% of this energy is converted into kinetic energy of the fragments; thus total kinetic energy of fragments = $0.35 \times 39 = 13.6$ MJ.

• Mass of a vessel of uniform wall thickness of 20 mm = $11.0 \times 0.020 \times 7800 = 1716$ kg.

• Speed of fragments following brittle failure = $(2 \times 13.6{\times}10^6 / 1716)^{0.5} = 126$ m/s.

• Hence, from Section 3.1.3:

speed of fragments from cylindrical part of vessel = 126 m/s;

speed of fragments from ends of vessel = $126 \times (20 / 80)^{0.5} = 63$ m/s;

smallest expected fragment from cylindrical part of vessel = 17 kg;

largest expected fragment from cylindrical part of vessel = 340 kg;

• Assuming a circular shape for fragments from cylindrical part of vessel:

area of 17 kg fragment = 17/7800/0.020 = 0.11 m^2, corresponding to a disc of 0.37 m diameter;

area of 340 kg fragment = 340/7800/0.020 = 2.2 m^2, corresponding to a disc of 3.3 m diameter;

Take the design basis for protection against fragment perforation as the need to protect against:

- a 17 kg mass in the form of a disk 0.37 m diameter hitting the wall end on with a 'fragment presented area' of $0.37 \times 0.020 = 0.0074$ m^2 (treat this as a missile with an effective impact diameter of $2 \times (0.0074 / \pi)^{0.5} = 0.097$ m) and a speed of 126 m/s;
- a 340 kg mass in the form of a disk 3.3 m diameter hitting the wall end on with a fragment presented area of $3.3 \times 0.020 = 0.066$ m^2 (effective impact diameter = 0.290 m) and a speed of 126 m/s;
- a 490 kg vessel end in the form of a disk 1.00 m diameter hitting the wall end on with a fragment presented area of $1.00 \times 0.080 = 0.080$ m^2 (effective impact diameter = 0.320 m) and a speed of 63 m/s;

From Equation 3.6, thickness of mild steel sheet required to resist perforation by 50% of the missiles is:

- for 17 kg missile: 21 mm;
- for 340 kg missile: 52 mm;
- for 490 kg missile: 24 mm.

Thus, from Section 3.2.6, the thickness of steel to resist all missiles during a brittle failure is $52 \times 1.25 = 65$ mm.

If the protection is to be provided by reinforced concrete of compressive strength 20 MPa, the thickness required can be calculated from Equation 3.8. In this equation:

- $K = 15000 / (20 \times 10^6)^{0.5} = 3.35$.
- Take $N = 1.0$.

Then,

- for the 17 kg fragment, $G_{(x/d)} = 0.60$, $x/d = 1.55$, $t = 0.31$ m, $s = 0.41$ m;
- for the 340 kg fragment, $G_{(x/d)} = 0.56$, $x/d = 1.50$, $t = 0.92$ m, $s = 1.21$ m;
- for the 490 kg fragment, $G_{(x/d)} = 0.18$, $x/d = 0.85$, $t = 0.70$ m, $s = 1.05$ m.

Hence the thickness of concrete required to resist perforation by all of the missiles and not to

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produce scabbing on the other side of the wall is 1.21 m. If scabbing can be tolerated, e.g. because scab plates are fitted, this can be reduced to 0.92 m.

If soil protection is used, Equation 3.10 is probably the most appropriate equation to use. If we assume average soil, the depth of penetration is:

- for the 17 kg fragment: 2.3 m;
- for the 340 kg fragment: 6.3 m;
- for the 490 kg fragment: 3.6 m.

If however, the fragments were to impact end–on and not tumble within the soil, the depth of penetration is given by Equation 3.12 and the depth of penetration is:

- for the 17 kg fragment: 4.5 m;
- for the 340 kg fragment: 6.7 m;
- for the 490 kg fragment: 2.5 m.

Thus penetration of up to 7 m into soil which is essentially infinite in extent is possible. On the basis of the multiplier given in Section 3.2.9, this latter translates into a requirement for 21 m of soil to be sure of total protection.

By inspection, we see that small fragments formed by disintegration of the ends of the vessel will result in missiles with less penetrating power than does an intact end. No further investigation of fragments is therefore needed.

It is seen that brittle failure of a large vessel, when pressure tested with gas, results in highly energetic fragments which are difficult to stop. Very large thicknesses of steel or concrete are required if the impact with the wall is end–on. Examination of the perforation equations indicates that for the 340 kg fragment, a face–on impact would reduce the wall thickness requirement by a factor of about 10. In reality, most impacts will lie somewhere between these two extremes and it is a question of engineering judgement as to how close to the upper limit one needs to go in a particular set of circumstances. The authors would prefer to err on the side of caution and use the end–on figures wherever possible.

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3.3.2 Brittle failure of water–filled vessel

In this section we consider the same vessel as in Section 3.3.1 but when undergoing an hydraulic test using water as the pressurising medium. The fragments formed will be the same as before, but the lower energy content of water means that the fragment speeds will be lower.

• From Figure 2.6, stored energy = 0.25 MJ/m^3, and since volume of vessel = 2.36 m^3, total stored energy = 0.59 MJ.

• Section 3.1.3 suggests assuming total kinetic energy of fragments equals total stored energy available. i.e., 0.59 MJ.

• Speed of fragments from cylindrical part of vessel following brittle failure $= (2 \times 0.59{\times}10^6 / 1716)^{0{\cdot}5} = 26$ m/s.

• Speed of end of vessel $= 26 \times (80 / 20)^{0{\cdot}5} = 13$ m/s.

From Equation 3.6, thickness of mild steel sheet required to resist perforation by 50% of the missiles is:

• for 17 kg missile: 2.5 mm;

• for 340 kg missile: 6.4 mm;

• for 490 kg missile: 2.9 mm.

Thus, from Section 3.2.6, the thickness of steel to resist all missiles during a brittle failure is $6.4 \times 1.25 = 8.0$ mm.

If the protection is to be provided by reinforced concrete of compressive strength 20 MPa, the thickness required can be calculated from Equation 3.8, again using $K = 3.35$ and $N = 1.0$.

• For the 17 kg fragment, $G_{(x/d)} = 0.035$, $x/d = 0.37$, $t = 0.10$ m, $s = 0.22$ m.

• For the 340 kg fragment, $G_{(x/d)} = 0.033$, $x/d = 0.36$, $t = 0.31$ m, $s = 0.64$ m.

• For the 490 kg fragment, $G_{(x/d)} = 0.010$, $x/d = 0.20$, $t = 0.20$ m, $s = 0.44$ m.

Hence the thickness of concrete required to resist perforation by all of the missiles and not to

produce scabbing on the other side of the wall is 0.64 m. If scabbing can be tolerated, e.g. because scab plates are fitted, this can be reduced to 0.31 m.

We find a very large reduction in the thickness of steel plate required to resist the vessel fragments when compared with the gas–filled case and a somewhat smaller, although still large reduction when reinforced concrete is used.

3.3.3 Failure of a ductile vessel

The vessel is assumed to be the same as in Section 3.3.1 but to fail in a ductile manner. Identifiable failures are:

- loss of one end by failure of a circumferential weld immediately adjacent to the end resulting in two missiles, the end itself and the remainder of the vessel which may act as a rocket (if it is inadequately tied down to resist the rocketing forces);
- loss of a small closure, 50 mm in diameter and 100 mm long, which is screwed into the end of the vessel for a depth of 50 mm, by a shear failure of the threads.

Failure is assumed to take place during a gas pressure test to 100 bar.

Following Section 3.1.5, a steady pressure of 100 bar is assumed to be applied to the underside of the end of the vessel following the end–cylinder failure until the end has been displaced a distance equal to the diameter, 1 m, from the cylinder. The energy aquired by the end during this process is (force × distance moved):

- $E = 10^7 \times \pi \times 0.5^2 \times 1.0 = 7.8\times10^6$ J = 7.8 MJ.
- Hence speed of ejected end of vessel = $(2 \times 7.8\times10^6 / 490)^{0.5} = 178$ m/s.

If the cylindrical part of the vessel, plus its intact end, acts as a rocket, following the weld failure, its speed can be evaluated using the method of Section 3.1.6. The mass of the rocketing fragment will be 1470 + 490 = 1960 kg. We will assume the pressurising gas is air

with a value for γ of 1.40. Then:

$\alpha = 10^7 \times \pi \times 0.5^2 / 1960 = 4000$ m/s^2;

$\beta = 100$

$\kappa = 67$

and maximum speed of rocketing fragment, from Equation 3.2, is 90 m/s. The corresponding kinetic energy is 8 MJ.

From Section 3.1.7, the pressure on the underside of the small plug ejected from the end of the vessel acts until the plug has been displaced a distance of 150 mm (50 mm until the bottom end of the plug clears the outer surface of the vessel and then two diameters more, i.e., 100 mm). Thus the energy given to the plug is:

$E = 10^7 \times \pi \times 0.025^2 \times 0.150 = 2950$ J.

The mass of the plug is:

$m = \pi \times 0.025^2 \times 0.100 \times 7800 = 1.53$ kg.

Hence the speed of the plug is:

$V = (2 \times 2950 / 1.53)^{0.5} = 62$ m/s.

For the calculation of the amount of protection required, we assume the rocket and the plug impact blunt end forwards and the end impacts end–on. Then from Equation 3.6 we obtain the following thicknesses of steel to resist perforation by 50% of the impacts:

for the vessel end: 96 mm;

for the rocketing vessel: 31 mm;

for the plug: 3 mm.

These figures must be increased by 25% if protection is to be against all missile impacts.

If these same failures were to take place during a hydraulic test with water then it is clear that the speeds of the ejected end of the vessel and the rocketing fragment will be very much less since the total stored energy available is only 0.59 MJ (see Section 3.3.2) whereas the air driven case requires essentially 8 MJ for both the ejected end and the rocketing vessel. Thus the worst case scenario is to assume that all 0.59 MJ all goes into the kinetic energy of the

end or into the kinetic energy of the rocket. The plug, however, requires only 2.95 kJ, which is very much less than the energy available so, following Section 3.1.7, we assume the energy imparted hydraulically is the same as that imparted pneumatically. Thus we obtain for the missile speeds:

for the vessel end: 49 m/s;

for the rocketing vessel: 25 m/s;

for the plug: 62 m/s.

and from Equation 3.6, the amount of steel to resist perforation by 50% of the missiles:

for the vessel end: 17 mm;

for the rocketing vessel: 6 mm;

for the plug: 3 mm.

Again, if the protection is to resist perforation by all missiles, and not just 50% of them, these thicknesses must be increased by 25%.

3.4 References and bibiliography

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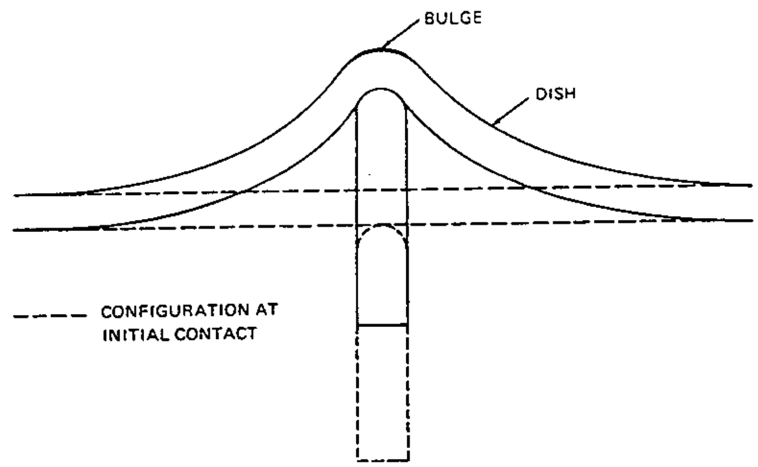

Figure 3.1. Permanent deformation of a thin target element showing bulging and dishing. (After Backman [3.10])

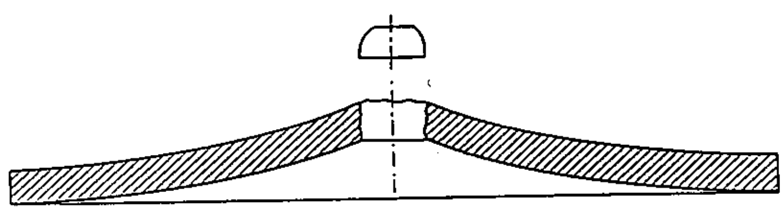

Figure 3.2. Plug detatched from thin target element following impact by missile.

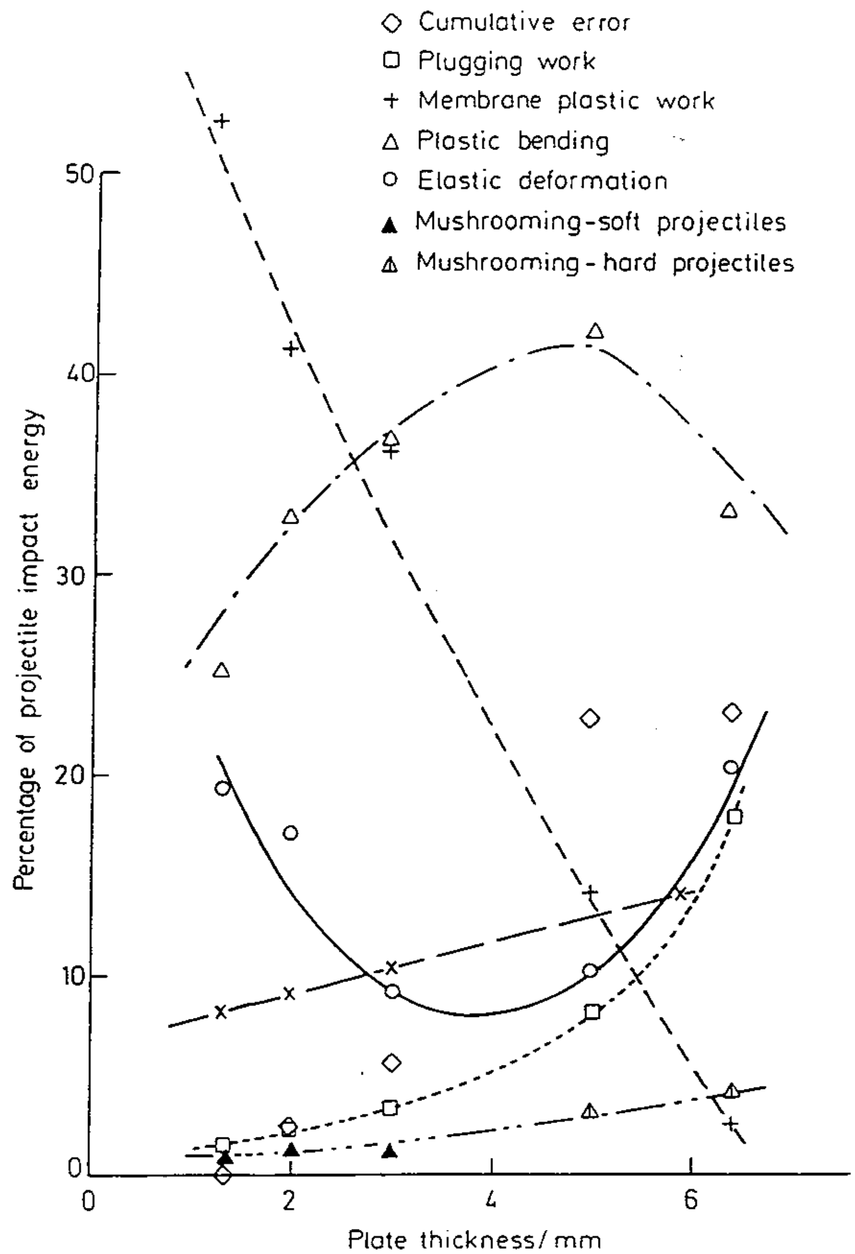

Figure 3.3. Breakdown of energy absorbed from impacting projectile.
(After Corran [3.4])

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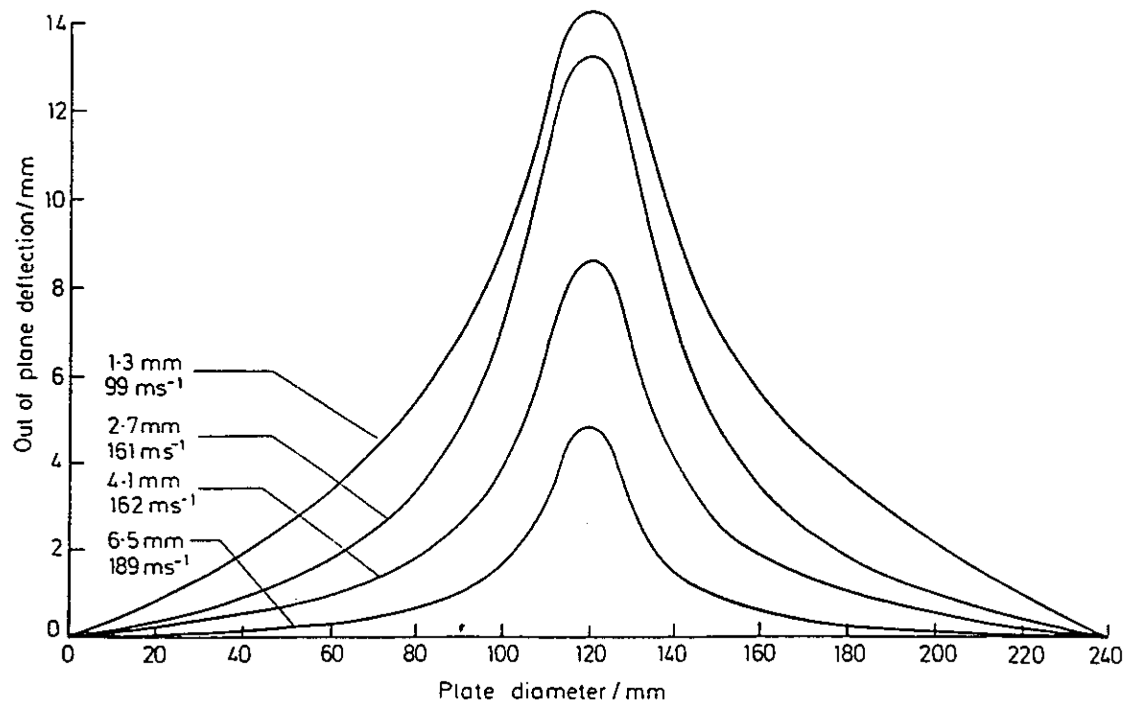

Figure 3.4. Deformation profiles of stainless steel plates.
(After Corran [3.4])

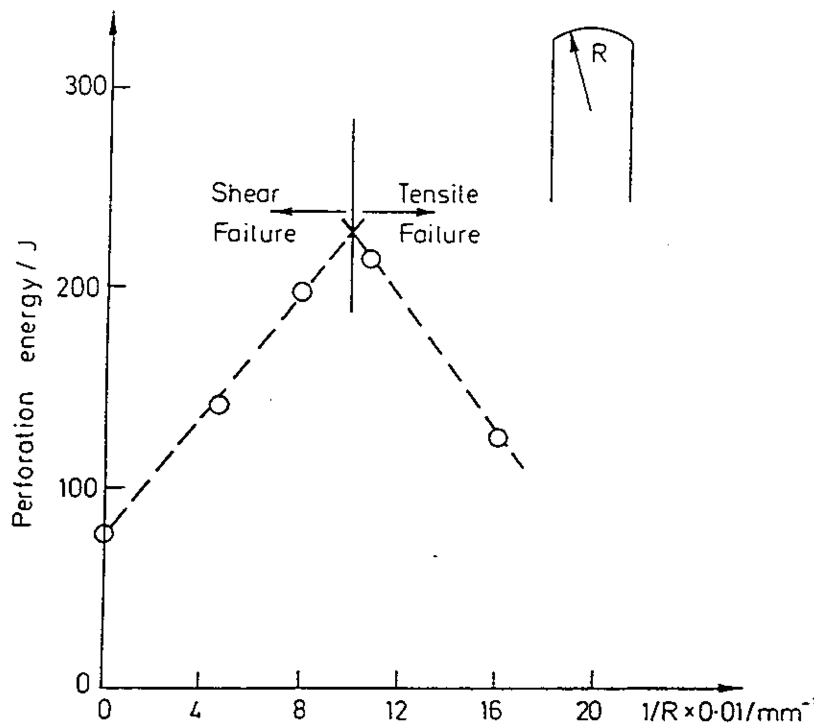

Figure 3.5. Variation of perforation energy with projectile nose radius in 1.3 mm targets. (After Corran [3.4])

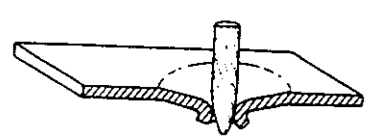

Figure 3.6. Petaling during perforation of a plate. (After Backman [3.10])

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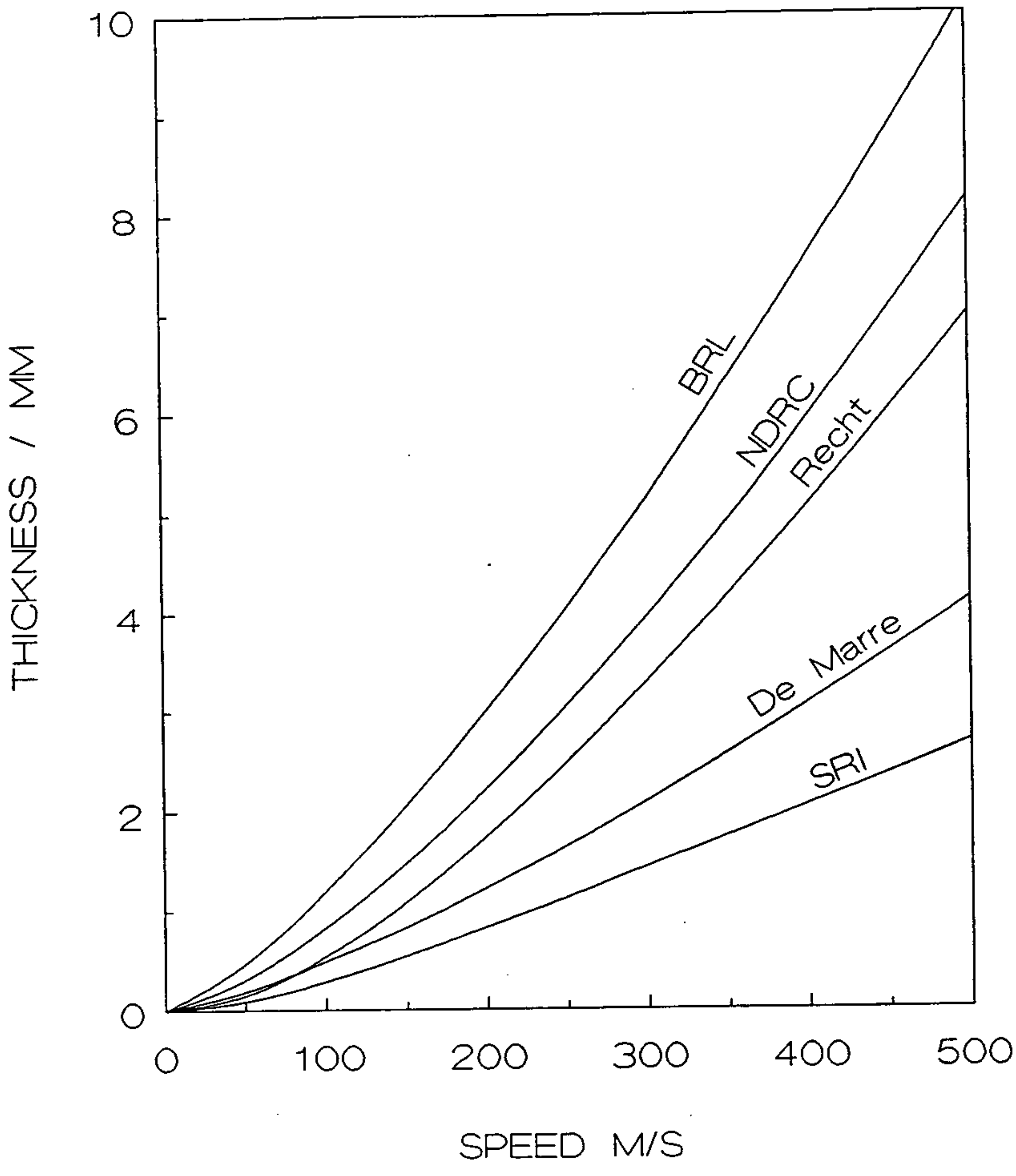

Figure 3.7. Perforation of steel plates by steel missile 20 mm long × 10 mm diameter – comparison of methods of prediction.

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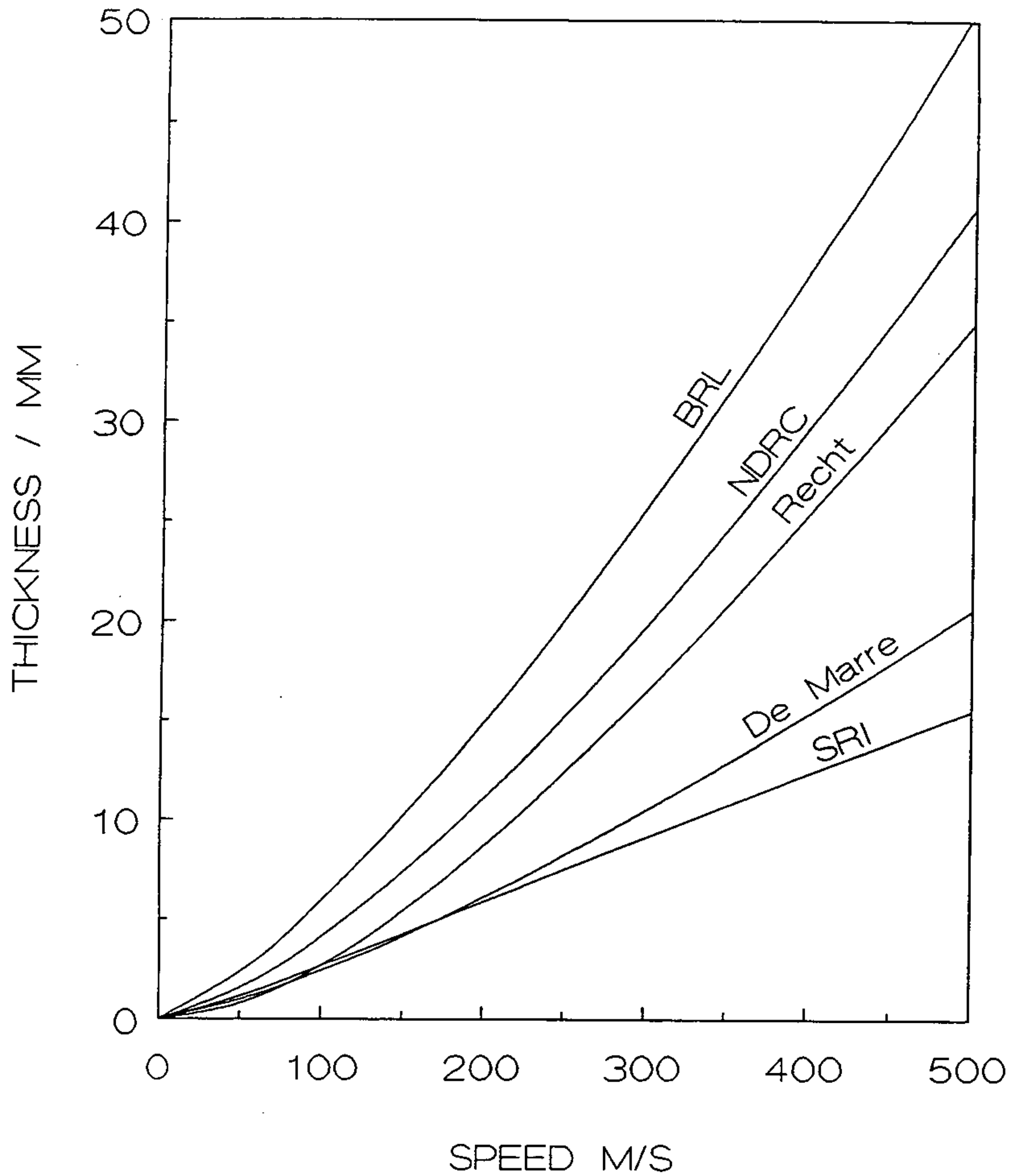

Figure 3.8. Perforation of steel plates by steel missile 100 mm long × 50 mm diameter – comparison of methods of prediction.

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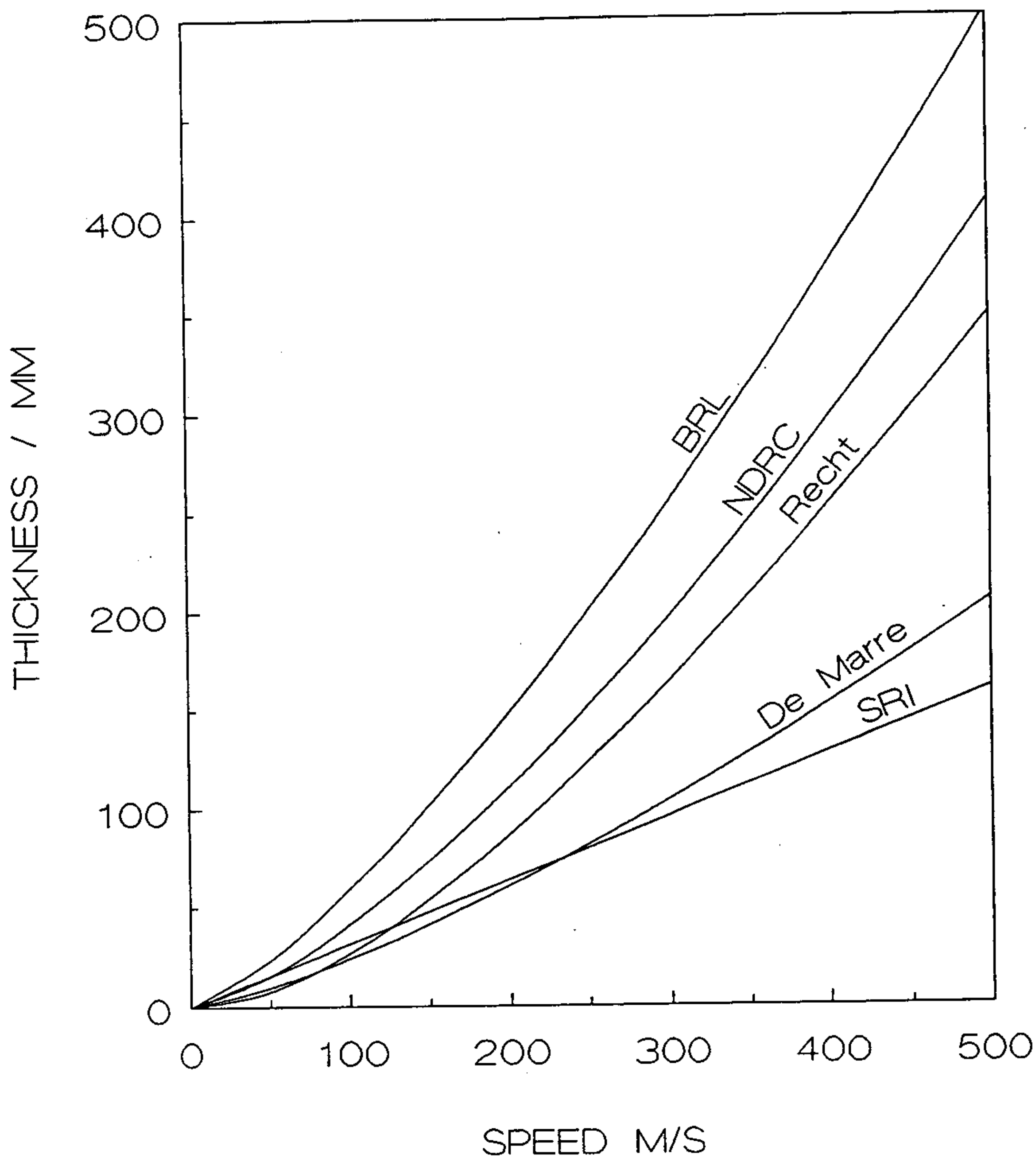

Figure 3.9. Perforation of steel plates by steel missile 1000 mm long × 500 mm diameter – comparison of methods of prediction.

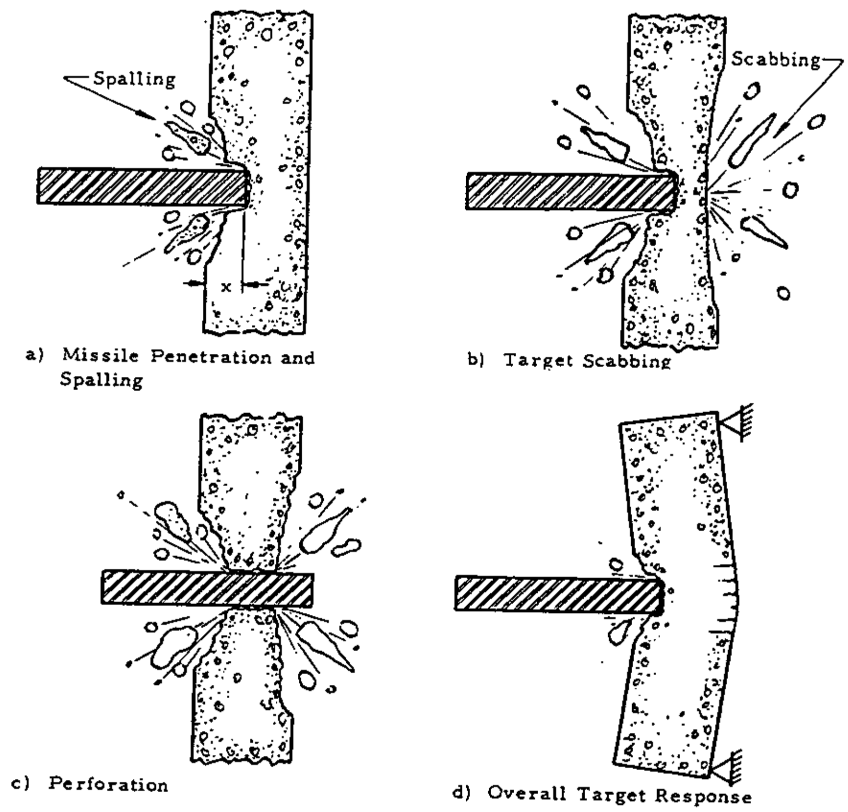

Figure 3.10. Missile impact phenomena in concrete.
(After Kennedy [3.20])

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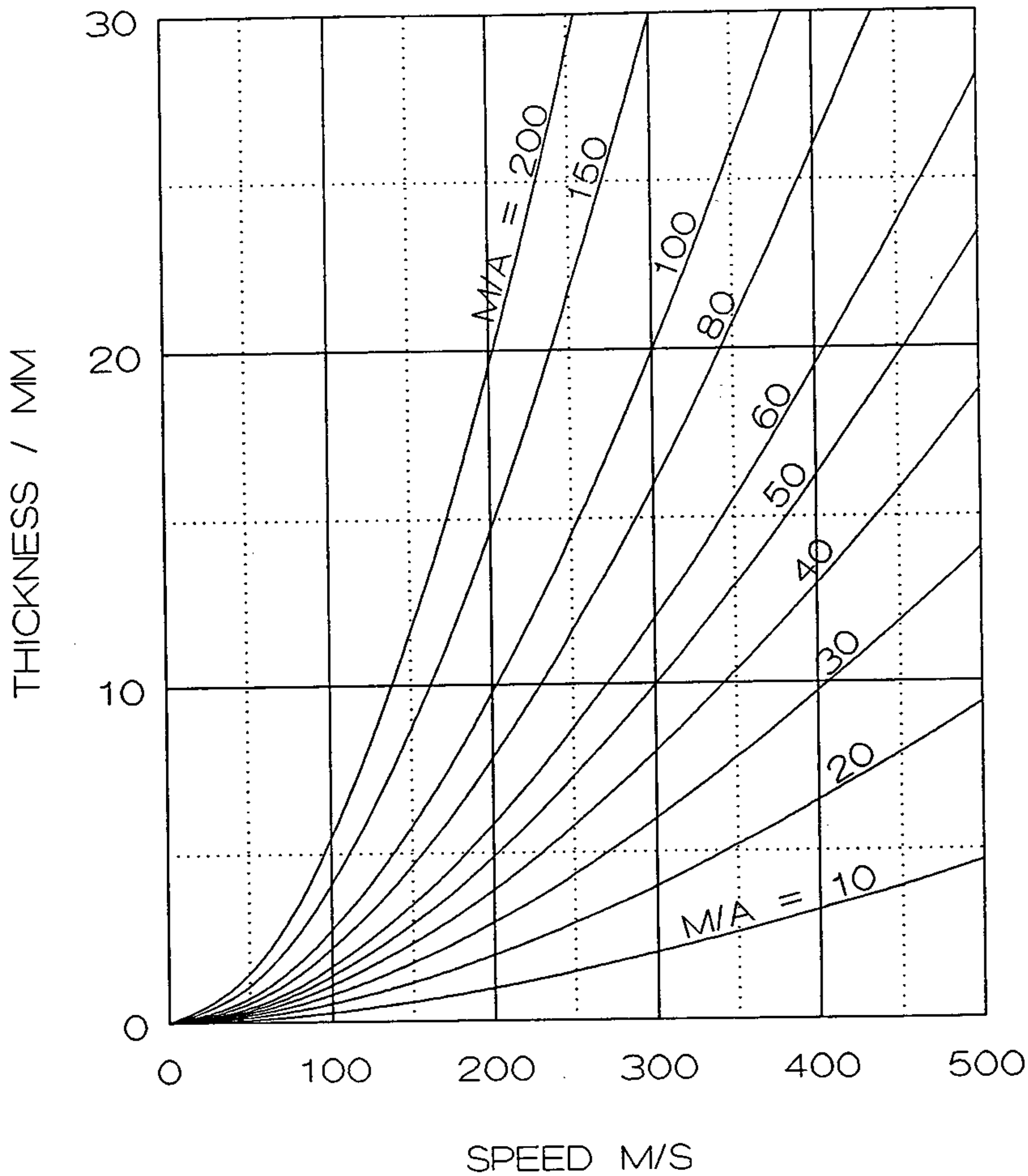

Figure 3.11. Perforation of polycarbonate — predictions of Recht's equation (3.3). (M/A = mass/fragment presented area, kg/m^2)

4 BLAST AND IMPULSE LOADING OF STRUCTURES

4.1 Introduction

In Section 3, it was suggested that 35% of the energy stored in the pressure vessel before failure is converted into the kinetic energy of missiles whilst the remaining 65% is converted into a blast (or shock) wave or into other modes of energy transmission, such as sound. (The distinction between a blast wave and a sound wave is that the former travels at supersonic speed.) Section 3 also examined the thickness required of walls and barricades to resist perforation by missiles. However, the impact of a missile with a wall also results in the transferrence of kinetic energy of the missile to the wall in the form of an impulse and this has a very similar effect to the impingement of a blast wave with the wall. In this Section, we consider blast and the effect that it and the impact of non–perforating missiles have on any structures placed in their path.

In comparison with the behaviour of missiles, our knowledge of blast waves is very limited. Most of the available information is for military explosives, conventional and nuclear, and for chemical explosions such as gas explosions. We must consider, therefore, how relevant this information is to the failure of pressure vessels. To keep matters simple, we assume that the fluids in the pressure vessel are non–combustible so the possibility of secondary chemical explosions is excluded. Most of the methods considered can, however, be extended to include them.

Experimental studies of exploding, frangible vessels filled with a pressurised gas have been undertaken by Boyer *et al* [4.12] and Esparza and Baker [4.10, 4.11]. Boyer *et al* measured the blast waves from shattering glass spheres pressurised with air, helium and sulphur hexafluoride. Esparza and Baker measured the blast waves from shattering glass spheres pressurised with air, argon and freon vapour. These experiments showed that the pressure–time history of a pressure vessel explosion is quite different from that for a condensed high explosive and that the formation of the blast wave is strongly dependent on

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the rate of release of energy. The most important difference between the vessel failure experiments and condensed high explosive results is that there is a negative phase impulse, following the initial positive impulse, that is almost as large as the positive impulse.

Consider an explosion taking place in the free air. The differences between the effects of condensed high explosives and pressure vessel failures are greatest at small distances away from the centre of the explosion. This is termed the near–field situation. At very great distances from the centre, the far–field situation, the behaviour becomes very similar. When pressure vessels are tested, however, the vessel is usually inside some form of containment structure. In this case the pressure–time history is dominated by the effects of interference between the blast waves reflected from the walls of the test cubicle and the distinctive character of the near–field blast wave is lost.

The effect of differences in the rate of release of the energy is more difficult to quantify. Condensed high explosives are set off by a detonator. There is therefore an initial high–speed release of energy over a very short period of time followed by a further energy release as the high explosive charge itself explodes over a slightly longer period. The rate of energy release from a pressure vessel depends on the nature of the fluid stored in the vessel and the mode of failure of the vessel. Two limiting cases are clear.

- The vessel contains a gas at high pressure and fails in a brittle manner. Brittle failure is a very rapid process which releases the gas over a very short period of time. This results in the formation of a blast wave in a similar way to a condensed high explosive.
- The vessel contains a liquid at high pressure and fails in a ductile fashion. Ductile failure is a relatively slow process and requires energy to be expended in opening a crack while the system energy is declining from the release of fluid. A point is reached where the stored energy is insufficient for the crack to grow further and the escape of fluid is choked within the area of the open crack. Typically, failure starts with the formation of a bulge on the side of the vessel which eventually splits longitudinally, see Figure 1.5, allowing the liquid to be released. This behaviour does not result in the formation of a blast wave. Bursting nylon or copper tubing are extreme examples of this case.

Most pressure test failures are likely to fall somewhere between these two extremes. Clearly, on economic grounds, it would be useful to be able to say that in a given situation no blast will occur and therefore there is no need for blast protection. In its narrowest sense, this is certainly true for vessels pressure tested with an involatile liquid such as water, since failure in these cases does not produce a supersonic shock, only a loud bang. However, the impact of missiles with protective walls has a blast–like character and the considerations of this chapter still apply (see, particularly, Section 4.8). In all other cases, blast must be expected, but its magnitude depends on the mode of failure of the vessel. The need for blast protection must therefore be assessed in the hazard assessment. When doing this, it should be borne in mind that a new vessel, manufactured from a nominally ductile material, may fail in a brittle fashion due to undetected flaws or porosity, hydrogen embrittlement of the weld heat–affected zone or precipitation of inter–metallic compounds. Vessels that are tested after having been in service may suffer from environmental stress cracking or fatigue and this cracking may be very difficult or impossible to detect using NDT (non–destructive testing). These factors often suggest the assumption of the worst possible case, namely, brittle failure.

In Sections 4.2 to 4.7, we review the physics of blast wave formation from condensed high explosives. This leads to a method for estimating the parameters of the blast wave from the energy released during an explosion whether of a condensed explosive or a bursting pressure vessel (see Section 2.7 for the energy equivalence between pressure system energy and TNT). In Sections 4.8 to 4.12, we discuss the response of structures to the dynamic loads imposed by blast waves and missiles. Although a full dynamic analysis is very complex, it turns out that most practical cases can be modelled in terms of two limiting forms of behaviour. For these, the structural response can be obtained from graphs.

4.2 Blast waves

When a vessel bursts, the enclosed fluid expands, pushing the surrounding air outwards and producing a pressure wave. If the energy release is sufficiently intense, the pressure wave will

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be a discontinuity in which the pressure changes from atmospheric pressure to a very high value over a very short distance. This constitutes a shock front which is referred to as the blast. After a very short distance, the speed of advance of the shock front outstrips the speed of advance of the fluid expanding from the vessel and the shock front is then propagated in the air. It is the discontinuity at the shock front that distinguishes shock waves from sound.

Since the density (or specific volume) and temperature are related to the pressure through an equation of state, discontinuities in density and temperature accompany the shock front. For normal shocks in air, all of these quantities are related through the Rankine–Hugoniot equations [4.1], but these are not reported here as no use is made of them in this report.

Most of our knowledge of blast waves has resulted from studies of condensed high explosives. Figure 4.1 shows the pressure–time history at a fixed location for a blast wave generated by a condensed high explosive source. It consists of an initial rapid rise in pressure, referred to as the positive phase or positive pulse, followed by a much weaker negative pressure phase. The negative phase can be thought of as being due to the particles (objects) over–shooting their equilibrium positions after passage of the positive pulse. As the positive phase passes a given point, the particle velocities and particle displacements are large. Yet, after passage of the negative pulse, the net displacement of particles is generally quite small. This is illustrated in Figure 4.2, where the position of the dog should be noted. The rapid changes in particle velocity imply rapid accelerations and, through Newton's second law, large forces acting on the particles in the path of the blast wave. It is these forces that give rise to the damaging effects of the blast.

4.3 Scaling equations

At the turn of the twentieth century, it was realised that the effects of an explosive blast depend upon the amount of explosive used, the type of explosive and the distance from the point of explosion (termed ground zero) to the position of measurement. Using data derived from accidental explosions, Hopkinson and Cranz [4.2] formulated the blast scaling law

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which is usually referred to as the Hopkinson–Cranz cube root law. According to this law, the safe distance for the location of inhabited buildings, public roads and railways from an explosion hazard is given by:

$$R = K\, W^{0\cdot333} \tag{4.1}$$

where R is the radius (m), W the mass of explosive (kg) and K a constant (m/kg$^{0\cdot333}$).

During the second world war, Kennedy [4.3] showed that this relation could be applied to many types of explosion by identifying W with the equivalent mass of TNT rather than the mass of the explosive in question. White [4.4] extended this idea further using the principle of similarity to compare two different explosions, suggesting that the effect of a blast in general could be represented as dimensionless groups and that two 'similar' explosions would have:

$$R/R_1 = t/t_1 = I/I_1 = (W/W_1)^{0\cdot333} \tag{4.2}$$

where t is the elapsed time and I is the impulse. Since there are far more data for TNT explosions that for any other explosive, correlation of blast effects are now universally expressed in terms of the group $Z = R/W^{0\cdot333}$ which is referred to as the scaled distance.

Brode [4.5], Henrych [4.6] and others have developed correlations from which the parameters of a blast wave can be estimated. For chemical explosions, blast wave parameters are as follows:

$$p_s = 14.072\, Z^{-1} + 5.540\, Z^{-2} - 0.357\, Z^{-3} + 0.00625\, Z^{-4} \qquad \text{for } 0.05 \le Z \le 0.3 \tag{4.3a}$$

$$p_s = 6.194\, Z^{-1} - 0.326\, Z^{-2} + 2.132\, Z^{-3} \qquad \text{for } 0.3 \le Z \le 1 \tag{4.3b}$$

$$p_s = 0.662\, Z^{-1} + 4.05\, Z^{-2} + 3.288\, Z^{-3} \qquad \text{for } 1 \le Z \le 10 \tag{4.3c}$$

$$\Delta p_{min} = -0.35\, Z^{-1} \qquad \text{for } 1.6 \le Z \tag{4.3d}$$

$$t_s = \left[\frac{980\, W^{0\cdot333}\, [1 + (Z/0.54)^{10}]}{[1 + (Z/0.02)^{3}]\, [1 + (Z/0.74)^{6}]\, [1 + (Z/6.9)^{2}]^{0\cdot5}}\right] \tag{4.3e}$$

$$t = 1.25\ W^{0.333} \tag{4.3f}$$

$$i_s = \int_{t_a}^{t_a+t_s} p(t)\ \mathrm{d}t \tag{4.3g}$$

$$\bar{i_s} = i_s\,[1 - 0.5\ Z^{-1}] \tag{4.3h}$$

where all symbols are defined on Figure 4.3, all pressures are expressed in Pa, all times in s and all impulses in Pa·s. For ease of use, $p(t)$ is often approximated as a triangular wave.

Other useful correlations include the distance r_d (m) to which debris may be propelled:

$$r_d = 45\ W^{0.333} \tag{4.4}$$

the distance r_s (m) within which sympathetic explosions may occur:

$$r_s = 0.5\ W^{0.333} \tag{4.5}$$

and the minimum safe distance r_m (m) for insignificant missile damage:

$$r_m = 120\ W^{0.333} \tag{4.6}$$

4.4 Reflection of blast waves

Blast waves may be reflected from a flat unyielding surface in three different ways, depending on the angle of incidence of the blast wave to the surface.

The simplest form of reflection is called a normal reflection since the propagation vector for the wave is normal to the surface. The main difference between the incident and reflected waves is that the reflected wave is propagated back along the path of the incident wave and therefore travels through a different medium to that of the incident wave. The properties of reflected shocks can be given in terms of a reflection coefficient, defined as the ratio Λ_n of the reflected over–pressure $p_r - p_x$ to that of the incident shock $p_i - p_x$, where p_x is the ambient air pressure before arrival of the shock. For shocks in air:

$$\Lambda_n = [8\ M_x^2 + 4]/[M_x^2 + 5] \tag{4.7}$$

where M_x is the Mach number of the incident shock. Two limiting values of the reflection coefficient are found:

$$\lim_{M_x \to 1} \{\Lambda_n\} = 2 \tag{4.8}$$

$$\lim_{M_x \to \infty} \{\Lambda_n\} = 8 \qquad (4.9)$$

Note that, since the reflection coefficients are greater than unity, the over–pressure in the reflected shock is greater than that in the incident wave. The particle velocity and temperature likewise increase but the Mach number for the reflected shock front is less than that for the incident shock.

Oblique reflection occurs when the incident shock is not normal to the reflecting surface. The reflection coefficients calculated for shocks in air at different angles of incidence are shown in Figure 4.4. It can be seen that the reflection coefficient initially decreases slightly with increasing angle of incidence but then suddenly jumps to a higher value before decreasing again.

The sudden jump in the reflection coefficient is the result of transition to the third type of reflection termed 'Mach stem reflection'. For shocks impinging at grazing incidence, the Mach number at which the transition occurs is $M_x = 1$ and the angle of incidence reaches a limiting value of 40^0 as $M_x \to \infty$. The reflection process is shown in Figure 4.5, where it is seen that a third shock front forms next to the reflecting surface and travels parallel to it. The incident and reflected shock fronts meet this new shock front, termed the Mach stem, at a triple point.

4.5 Yield magnification

So far, it has been assumed that the blast source is placed in an infinite homogeneous medium such that the blast wave travels out equally in all directions until it impinges on a reflecting surface. For an explosive source placed on the ground in an open field, the ground acts as a reflector and the energy released is constrained to propagate through the surface of a hemisphere rather than a sphere. Thus the effective explosion yield is twice that for a source in an infinite medium. This is referred to as the surface blast effect. Similar yield magnification effects can be obtained if the energy is directed into a cone of small solid

angle. This could occur for a vented cubicle, where the vent aperture defines the solid angle. In this case, the energy is confined to the cone in the proportion of the solid angle to 4π. Very large magnification effects can be produced in this way for the region immediately outside the cubicle and in line of the blast. However, diffraction results in the over–pressure further afield being essentially the same as if the cubicle had not been there at all (High [4.18]).

A second effect relates to the height above the ground at which the source explodes. This is depicted in Figure 4.6 and is referred to as the height of burst effect. If the source is at some height above ground zero, different points on a radius away from ground zero will receive a blast wave at different angles of incidence, with the angle of incidence increasing as the radius increases. At a certain radius, the reflections change from regular oblique reflections to Mach stem reflections, as shown in Figure 4.7. For a given amount of stored energy, there is a height for which the radius at which damage occurs is a maximum.

4.6 Geometric Effects

Blast sources from pressure vessel failure are unlikely to be spherical in symmetry due to:

- the position of the vessel within the test chamber;
- the possibility of a single crack opening due to ductile failure or loss, for example, of an end closure.

The only non spherical case that has been studied experimentally is for cylindrical high explosives. Wisotski and Snyer [4.13] and Reisler [4.14] have shown that the blast wave that results is quite complex due to the interference between waves propagated from the end and sides as shown in Figure 4.8. The blast typically exhibits multiple shocks and decays in quite a different way from a spherical wave as shown in Figure 4.9. Usually, these asymmetries only dominate in the near–field region and smooth out as the blast wave progresses. Baker [4.7] suggests that the distance at which spherical symmetry resumes can be determined through simulation but does not give details of how this can be done.

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4.7 Internal blasts

Internal blasts occur if the explosive source is confined within a chamber. This is the situation where a pressure vessel under test is placed in a test cubicle for total or partial containment.

Figure 4.10 shows an explosive source in a test chamber. The blast wave from the explosion will propagate outwards until it meets the chamber walls, which will then reflect it back into the chamber. Reflection may be normal, oblique or Mach stem, depending on the geometry, and the shock will reverberate around the chamber until the Mach number of the shock front becomes less than unity. An additional factor, when compared with an explosion in the open, is the release of gas within the cubicle. If the vent area of the cubicle is sufficiently small, this gas will build up an internal pressure. This pressure will decay relatively slowly, both through escape from the vent or gaps in the cubicle structure, if present, but also, in the case of a TNT explosion, through cooling of the hot gases formed during the explosion.

Figure 4.11 shows a test chamber in which charges of up to 5 kg of TNT were exploded [4.15] and Figure 4.12 shows a typical pressure–time history obtained with it for a cubicle with a small vent area [4.16]. Note that, although, initially, there is a lot of short–lived oscillation, there is no negative pressure pulse. This is at least in part due to the development of gas pressure within the chamber from the gases released by the TNT.

Baker [4.7] has shown that this complex behaviour can be approximated by a simple model. He assumed that the peak shock pressure is halved on each reflection, that the duration of each pressure pulse remains constant and that three pulses are sufficient to represent the reverberations, as shown in Figure 4.13. These three pulses are then combined into a single triangular pulse. The duration of this triangular pulse is typically of order 1 millisecond. Meanwhile, the gas pressure starts to build up, reaching a maximum in a time comparable with the shock pulse and is then followed by an exponential decay. If the chamber is vented, the pressure will eventually decay to atmospheric pressure. Combining the triangular shock

pulse and the gas pressure, one obtains the overall pressure/time curve shown in Figure 4.14. This illustrates the key features of the loads presented to the walls of the cubicle: the shock load is intense but of short duration, the gas load has a very much lower peak pressure but is much longer–lived. Computationally, the response of the structure to the combined shock and gas impulses may be obtained either by determining it for each pulse treated separately or by additively combining the two pulses into one pulse by assuming that the pressure peaks are coincident at time zero. The latter is, of course, more precise, but computationally more troublesome.

Test 'cubicles' can range from a single cantilevered wall at one end of the scale to a fully enclosed 4–wall plus roof cubicle at the other (all cubicles are assumed to have a floor). In all cases except the fully enclosed cubicle, the 'missing' walls provide sufficient venting to prevent any build up of gas following the explosion. Thus the slow release of gas referred to in the previous paragraph only takes place when a fully enclosed cubicle with a limited vent area is used. If the vent area is very small, this release can take several seconds.

The procedure we adopt to estimate the impulse load taken by a cubicle is to determine the peak pressure and impulse received by the walls of the cubicle during the initial shocks and to add on, in the case of a totally enclosed cubicle, with or without limited venting, the pressure load and impulse due to the released gas.

4.7.1 Internal shock loading of protective cubicles

Ayvazyan et al [4.17] have produced a very extensive series of charts for determining the shock loading on cubicles. Although the cubicles must be parallelopipeds, all length to width to height ratios are covered and the explosive source can be placed almost anywhere in the cubicle. The charts give the average loading on a given wall, thus assuming that the high loading in one part of the wall is quickly transferred to regions of lower stress. The authors maintain that this will happen with well constructed cubicles made from reinforced concrete or steel. These charts are clearly very valuable for the design of protective enclosures and it

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is unfortunate that their scientific basis is not given, although, since Baker is one of the co–authors of this document, we presume that it draws heavily on the ideas presented above.

Figures 4.15 and 4.16 have been derived from these charts for the case of a cubical chamber where the explosive source is placed at the centre of the cube (or at the centre of the notional cube in the case of a single wall). Figure 4.15 gives the average peak reflected pressure and Figure 4.16 the average reflected impulse on each wall. The letter codes refer to:

A	a single vertical cantilever wall
B	either wall of a 2–wall cubicle
	the side wall of a 3–wall cubicle
	the roof of a 2–wall cubicle with roof
C	the back wall of a 3–wall cubicle
	any wall of a 4–wall cubicle
	either wall of a 2–wall cubicle with roof
	the side wall or roof of a 3–wall cubicle with roof
D	the back wall of a 3–wall cubicle with roof
	any wall or roof of a 4–wall cubicle with roof.

The scaled distance used in both these figures is the ratio of the distance of the explosive source from the wall (one half of the internal length of the cube in our case) to the cube root of the energy released during the explosion. The same scaling, by the cube root of the energy release, is also used for the impulse. Although the curves are strictly applicable only to high explosives, we presume their validity for shocks produced by failing pressure vessels.

4.7.2 Gas impulse loading of protective cubicles

As indicated above, gas loading of a cubicle is only relevant when the cubicle totally or almost totally encloses the explosive source. An open wall is sufficient to reduce gas loading to essentially zero. However, walls of very light construction are often used to keep out the elements on nominally open walls, usually termed 'blow–out panels'. Ayvazyan [4.17]

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considers these blow–out panels to have negligible effect provided they will be displaced by a pressure which does not exceed 1 kPa (0.01 bar).

The computation of the gas load on a cubicle wall involves the determination of the initial peak gas pressure, at time zero, and its rate of decay. In the case of a TNT detonation, the gases formed by the detonation will be hot but not fully oxidised (TNT does not contain sufficient oxygen for this) so that combustion will follow, using oxygen from the air in the cubicle. Thus the appropriate energy change to use when calculating the final gaseous state is the energy of combustion, not the energy of detonation. However, for larger charges, the cubicle will contain insufficient oxygen for complete combustion, so a transition will take place to the energy of detonation being the appropriate energy term. Tests have shown that gas pressures calculated in this way for TNT explosions predict the initial pressures obtained by extrapolation of measured pressures to zero time as shown in Figure 4.12.

In the case of failure of a gas–filled pressure vessel, the temperature of the gas released will be very much lower than ambient. However, the gas released will mix with the ambient air in the cubicle and interchange heat with the walls, so we propose, as a slightly conservative solution, to assume that the gas released from the vessel instantaneously achieves ambient temperature. The calculation of the gaseous over–pressure at time zero is then quite straightforward.

The rate of decay of gas pressure depends on the vent area of the cubicle. If there is no vent, this initial over–pressure must be assumed to persist indefinitely. If a vent is present, an approximate equation describing the pressure–time history, Baker [4.7] is:

$$p(t) \quad = \quad p_1 \exp(-2.13\ \tau) \tag{4.10}$$

Here: p_1 is the peak gas pressure (bara);

p_0 is the ambient pressure (bara);

$\tau = \overline{A}\ t = \left[\dfrac{\alpha_e\, A_S}{V^{0.667}}\right] \left[\dfrac{t\ a_0}{V^{0.333}}\right] = \left[\dfrac{\alpha_e\ A_S\ \ t\ \ a_0}{V}\right];$

α_e is the ratio of the vent area to the wall area;

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A_S is the internal surface area (m^2);

V is the internal volume (m^3);

a_0 is the speed of sound at ambient conditions (approx. 340 m/s).

The duration of the impulse due to the gas release is the time taken for the pressure to fall from the initial pressure of p_1 to ambient pressure p_0 and is obtained from equation (4.10) as

$$t_{im} = c \ln (p_1/p_0) \tag{4.11}$$

and the impulse from

$$i_g = p_1\, c\, [1 - \exp(-t_{im}/c)] - p_0\, t_{im} \tag{4.12}$$

where

$$c = \left[\frac{V}{2.13 \quad \alpha_e \quad A_S \quad a_0}\right]$$

For convenience, equations 4.11 and 4.12 are plotted in Figures 4.17 and 4.18.

4.8 Missile impact loading

The issue of penetration and perforation of a protective barricade by a missile was dealt with in Section 3, however, the impact of a missile with a wall also imparts a transient impulse which the wall, and most importantly, its anchorages, must withstand, otherwise the wall may become a missile itself due to this cause alone.

The impact of a missile with a wall is usually highly localised, it is this which results in penetration or perforation, but the absorption of the energy/momentum of impact is spread over a much wider area of the wall. In order to maintain a consistency of approach to the effect of transient loading on protective walls, we will assume that this loading takes place uniformly over the whole wall. On this assumption, the impulse experienced by the wall is:

$$i_i = mv/A \tag{4.13}$$

where:

m = mass of missile (kg);

v = speed of missile (m/s);

A = area of the wall which is impacted by missile (m^2).

The impulse duration may be estimated by assuming that the missile is brought to rest linearly with time, thus the duration of the impulse is:

$$t_i = 2x/v \tag{4.14}$$

where:

x = the depth of penetration of the missile into the wall.

The peak 'pressure' is then, assuming a triangular impulse:

$$p_i = 2i_i/t_i \tag{4.15}$$

The expected depth of penetration of a missile, x, is not normally known, however, a sufficiently accurate estimate can be made by treating it as equal to the thickness of target required to resist perforation and this can be calculated by the methods of Section 3.

4.9 Example

A cubical test chamber of internal dimension 2 m made from reinforced concrete of thickness 300 mm with a vent in the roof of area 0.4 m^3 is used to test a pressure vessel of internal volume 0.02 m^3 containing gas at 1 kbar. The most likely failure mode of the vessel is judged to be the ejection of an end closure of mass 100 kg with an initial speed of 100 m/s which will penetrate the wall to a depth of 150 mm. It is assumed that 65% of the stored energy is available for the generation of a shock.

Calculation of shock:

From Figure 2.5, stored energy = 73 MJ/m^3, giving 1.46 MJ for this vessel;

Shock energy = $0.65 \times 1.46 = 0.95$ MJ;

Distance from vessel to wall = 1 m;

Scaled distance = $1/0.95^{0.333} = 1.02$ $m/MJ^{0.333}$;

From Figure 4.15, peak shock pressure = 0.7 MPa;

From Figure 4.16, scaled impulse = 430 $Pa{\cdot}s/MJ^{0.333}$;

Impulse due to shock = $430 \times 0.95^{0.333} = 422$ $Pa{\cdot}s$;

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Shock duration, for a triangular shaped pulse, $= 2 \times 422 / 0.7 \times 10^6$ s = 1.20 ms.

Calculation of gas pressure pulse:

Peak gas pressure $= 1000 \times 0.02 / 2^3 = 2.5$ barg (= 0.25 MPa gauge);

In equations 4.12 and 4.13, $c = 2^3 / (2.13 \times 0.4 \times 340) = 0.0276$ m²/s;

From equation 4.11, duration of pulse = 0.0276 ln (3.5 / 1.0) = 0.0346 s;

From equation 4.12, impulse

$$= 3.5\times10^5 \times 0.0276 \times [1 - \exp(-0.0346 / 0.0276)] - 10^5 \times 0.0346 = 3440 \text{ Pa}\cdot\text{s}.$$

Calculation of missile impulse:

From equation 4.13, impulse $= 100 \times 100 / 4 = 2500$ Pa·s;

From equation 4.14, duration $= 2 \times 0.15 / 100 = 0.003$ s = 3 ms;

From equation 4.15, peak pressure = 1.67 MPa.

We see that in this example, the peak–pressure load on the walls due to the impact of the missile is more than 6 times that of the gas but lasts for only one tenth of the time. The peak pressure due to the shock is intermediate in strength, but of shorter duration than both.

4.10 Structural response to blast

4.10.1 Introduction

The blast from an explosion damages a structure by causing it to deform. This deformation, however, is resisted by the inherent strength of the structure and its inertia. In general, a structure has many degrees of freedom, each with its own inherent strength and inertia, however, damage to the structure is caused almost entirely by the reponse of the mode with the lowest vibrational frequency. This greatly simplifies the analysis since only 'single degree of freedom' (SDOF) methods can be used. Additionally, since all parts of the structure start to move at the same instant, and therefore keep in phase if only a single vibrational frequency is involved, a single mass may be assumed.

4.10.2 Single degree of freedom systems

A typical dynamical system with a single degree of freedom is shown in Figure 4.19. The general equation describing the motion of the mass is:

$$m\,x'' + c\,x' + k\,x \quad = \quad F(\mathrm{t}) \tag{4.16}$$

where m is the mass of system, $x(t)$ its displacement as a function of time t, c the damping constant, k the spring constant and $F(t)$ the forcing function. The solution of equation (4.16) depends on the forcing function $F(t)$ and the initial conditions.

If $F(t) = 0$, damped simple harmonic motion takes place if the mass is set into oscillation by a displacement at $t = 0$:

$$x(t) \quad = \quad \exp(-\zeta\,\omega\,\tau)\left[\frac{x'_0 + \zeta\,\omega\,x_0}{\omega_\mathrm{d}}\sin(\omega_\mathrm{d}\,t) + x_0\cos(\omega_\mathrm{d}\,t)\right] \tag{4.17}$$

where $\zeta = c/[2\,m\,\omega]$, $\omega = \sqrt{k/m}$, and x'_0 is the velocity and x_0 the displacement at $t = 0$. The solution for different values of ζ is shown is Figure 4.20. Protective structures generally have values of ζ in the range 0.1 to 0.4 and are usually termed 'underdamped'.

Solutions can be obtained for more general forcing functions $F(t)$ by superposition. Applying this procedure to a blast load modelled as a triangular pulse, see Figure 4.21, the results can be expressed, for the case where the damping, $c = 0$, in the form:

$$DLF \quad = \quad x(t)\,\frac{k}{F} \quad = \quad \left[1 - \frac{t}{t_\mathrm{d}} - \cos(\omega\,t) + \frac{1}{\omega\,t_\mathrm{d}}\sin(\omega_\mathrm{d}\,t)\right] \tag{4.18}$$

where F is the force on the structure at $t = 0$ and DLF is the dynamical load factor and is the relative increase in displacement produced by the blast wave over and above that which would be produced by an equivalent static load. A similar, but more complex, result may be obtained when damping is non–zero. Although no real structure will have truly zero damping, we will base our analysis on the zero–damping case for reasons of simplicity and conservatism as far as the specification for protective structures is concerned.

4.10.3 Resonance

Since the forcing function can be represented as a Fourier series of harmonic contributions, the behaviour of the single degree of freedom system excited by a single component $F(t) = F\sin(\omega\, t)$ is of concern. Using this function, the equation of motion can be solved to yield, again for the undamped case:

$$DLF \;=\; \left[\left[1-\left(\frac{\omega_f}{\omega}\right)^2\right]^2+\left[2\,\zeta\,\frac{\omega_f}{\omega}\right]^2\right]^{-0\cdot 5} \tag{4.19}$$

This is plotted in Figure 4.22 for varying $\frac{\omega_f}{\omega}$ from which it can be seen that, for a particular frequency ω_f, the magnitude of DLF becomes large: this is the condition of resonance. For systems in three dimensions with distributed mass, there will be many frequencies at which resonance occurs. These are the normal modes or eigenvalues of the generalised equations of motion. However, in our simplification of the behaviour of the structure to a single frequency system, resonance will only occur at a single frequency. If the forcing function is a triangular pulse rather than a periodic one, build–up of the amplitude of vibration will not take place to the same extent. Nevertheless, where the duration of the triangular pulse is comparable with the period of vibration of the structure, some resonance–like enhancement of the displacement will take place.

4.10.4 Limiting responses

Analysis of a lumped mass single degree of freedom system shows that there is a strong relationship between the natural frequency of a structure (and hence its natural response period) and the positive phase duration of the blast wave.

Two limiting situations are possible. If the positive phase is long compared with the natural period, the structure is said to be loaded quasi–statically. In this case, the maximum displacement $x_{\max}$ is solely a function of the peak blast load F and the stiffness of the structure and does not involve the positive phase duration or the mass of the structure.

If the positive phase is short compared to the natural period, the blast load will be over before the structure has moved at all. This is referred to as impulsive loading.

Intermediate between these two limits is a region where $t_d \simeq t_m$, where t_m is the time at which maximum deflection occurs. In contrast to the limiting behaviour, it is usually necessary to solve the equations of motion numerically to obtain the response in this region.

All three forms of behaviour can be shown graphically as a plot of the dynamic load factor DLF against the product $\omega\, t_d$. The three regions of quasi–static, impulse and dynamic response are identified as regions I, II and III, respectively, in Figure 4.23. Consideration of the energy shows that, for quasi–static loading, $DLF = 2$ whereas, for impulse loading, the limiting tangent to the curve has unit slope so $DLF = \omega\, t_d$.

This diagram illustrates the importance of not basing damage on peak shock pressure alone; if loading is impulsive rather than quasi–static, severe over–estimation of damage could result.

4.10.5 Pressure–impulse diagrams

The information shown in Figure 4.23 can be plotted in a more useful way by using dimensionless groups as in Figure 4.24. The curve now separates regions in which damage will occur $x > x_{\max}$ from regions where no damage is observed $x < x_{\max}$. Graphs of this kind are referred to as pressure–impulse diagrams and the curve is called an iso–damage curve.

Pressure–impulse diagrams can be constructed on theoretical grounds, experimentally and from observation of the outcome of real explosions. Iso–damage curves have been constructed from a study of damage caused to housing by bombs dropped during the second world war. Baker [4.7] has presented a diagram of this kind, expressed in terms of impulse and peak pressure rather than deflections (see Figure 4.25) in which five categories of damage are identified:

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• category A: almost complete destruction;

• category B: such severe damage as to necessitate demolition: 50–75% of external brickwork destroyed or unsafe;

• category C_b: damage rendering housing temporarily uninhabitable, partial collapse of roof and one or two external walls, load–bearing partitions severely damaged and requiring replacement;

• category C_a: relatively minor structural damage yet sufficient to make house temporarily uninhabitable: partitions and joinery wrenched from fixings;

• category D: damage calling for urgent repair but not so as to make house uninhabitable: damage to ceilings and tiling: more than 10% of glazing broken.

These curves can be used to assess the potential for causing damage during pressure testing in an open, unprotected, area.

4.11 Pressure–impulse diagrams for protective structures

Protective structures for use in pressure testing will, in most cases, be fabricated from flat, rectangular plates or slabs of materials such as mild steel or reinforced concrete. These plates may surround the item being tested on anything from 1 to 6 sides. A full dynamic analysis of the response of such a structure to blast is a job for the specialist. However, it is arguable that such a detailed analysis is inappropriate since we normally have only an approximate knowledge of the details of the blast wave and an 'engineering solution' is perhaps quite acceptable. Following Baker [4.7, 4.17], we pursue this path to devise an approximate design procedure for protective enclosures.

In order to make the procedure more tractable, we make a number of assumptions:

• the protective cubicle is cubic in shape but may have one or more sides missing,

• the item being tested is placed at the centre of the cubicle,

• the edges of the cube are sufficiently strong so that failure is of the walls of the cubicle and not the edges.

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The first two assumptions enable us to use the information given in Section 4.7 to determine the loading on the walls of the cubicle. The third greatly simplifies the dynamic analysis and, in practice, does not impose severe constructional penalties.

When a flat plate, clamped or supported on all four edges, is subjected to an increasing uniform load, elastic failure first takes place at the centre of the plate where the stress is largest. The plate does not, however, totally fail at this stage, but it does start to plastically deform and, as the load is increased further, the plastic region expands until it reaches the edges and the whole plate is now completely plastic. Failure of the plate follows shortly afterwards. This elastic–plastic behaviour greatly complicates the dynamic analysis of the plate, so the engineering solution advocated by Baker is to ignore the plastic behaviour of the plate and to treat it as elastic within the body of the plate and to use the elastic failure criterion only when the plastic–elastic boundary reaches the edges of the plate. The results of a dynamic analysis of a plate which behaves in this way is summarised in the pressure impulse diagram, Figure 4.26, which shows the iso–damage line. In this diagram:

X = length of each side of plate (m);

h = thickness of plate (m)

p = peak impulse pressure (Pa);

i = impulse (Pa·s);

d = density of plate (kg/m^3);

E = Young's Modulus of elasticity of plate (Pa);

σ_y = dynamic yield strength of plate (Pa);

f = a numerical factor which equals 0.48 for a square plate.

The difference between the solutions for simply supported and clamped edges is small and is ignored. It will be noted that the axes are expressed in terms of readily available quantities, rather than the deflections of Figure 4.24, thus making the application easier. To the extent that this solution is really an asymptotic one, i.e. it determines the limiting behaviour in the two directions, the curved portion in the middle, where resonance becomes important, is not well defined. However, this does not seriously detract from the results provided that one

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remembers that it is an engineering solution rather than an accurate one.

Values of E and σ_Y are readily available for metals under normal tensile test conditions, but when loading is very fast, yielding takes place at rather higher stresses than normal. Figure 4.27 shows that for structural steel, it can be as much as 1.58 times higher when loading takes place in less than 5 ms. Thus in the example given in Section 4.9, the appropriate values to use for σ_Y will be 1.58 times the normal yield stress for the shock (duration = 0.8 ms) and 1.30 for the gas pulse (duration = 34.6 ms).

The determination of values of E and σ_Y for reinforced concrete is not quite so straightforward. Although E is usually determined in tension for metals, it is usually determined in compression for concrete. However, E in tension is usually taken to have the same value as E in compression, so the compression value should be used here. The contribution of concrete to the tensile strength of reinforced concrete is usually assumed to be negligible. The tensile strength is therefore solely due to the steel reinforcing. However, not all of the reinforcing contributes to the tensile strength in a particular direction. Reinforced concrete walls, as used in blast protection, usually have a layer of steel reinforcing close to each face with interlacing tying them together. Each of these two layers of reinforcing will contain steel bars running orthogonal to one another, generally parallel to the edges of the wall. When the wall is loaded on one side by a blast, the flexure of the wall will tend to place the side of the wall nearest the blast into compression and the one furthest away into tension. The interlacing is there to prevent these stresses forcing apart the two layers of reinforcing and resulting in severe weakening of the wall. Failure of the wall is due to tensile yield and the layer of reinforcing on the loaded face contributes nothing significant to this since it is in compression. Thus only the layer of reinforcing in the unloaded face contributes to the tensile yield strength, σ_Y, and of the steel bars in the two orthogonal directions in this layer, on average, only those in one direction makes a significant contribution (imagine the wall placed in tension parallel to one edge; the bars orthogonal to that direction will have a negligible effect on the failure stress).

Example.

A reinforced concrete wall has 1% steel, by volume, running parallel to each edge on each of the two faces of the wall. I.e., the total volume of steel in the wall is 4%, ignoring interlacing. Then, for the calculation of the behaviour of the wall under dynamic loading,

$$\sigma_y(\text{reinforced concrete}) = 0.01 \times \sigma_y(\text{steel}).$$

4.11.1 Validation

High [4.18] conducted a series of tests on the strength of cubicles intended to be used in the containment of hazardous high pressure experiments. The tests involved the detonation of a small TNT charge inside 1/6th scale models of the planned cubicles. The models were heavily instrumented with pressure transducers to monitor the blast and successively larger quantities of explosive were used until the cubicles were destroyed. Tests were conducted on a brass cubicle, which was not taken to destruction, and reinforced concrete ones, which were.

The brass cubicle was 24"×24"×30" of wall thickness 0.25" with a vent area of 24"×12". The charge of 0.25 oz of RDX/TNT was not placed centrally, but about 6" from the 24"×24" wall. For the purposes of comparing with predictions of the method described in Section 4.10, the cubicle was treated as a cube of the same volume as the actual one, and all pressures and impulses were averaged over the whole cell to estimate their values for a central charge.

Using the values given by Ayvazyan [4.17], the shock wave energy is estimated as 35×10^3 J and the thermal energy release as 64×10^3 J. The cubicle has a volume of 0.28 m^3, and equivalent length of side of 0.65 m, a vent area of 0.27 m^2 and initially contains 11.3 moles of air. After detonation of the charge and combustion of remaining combustible products from the detonation, the thermal energy released would heat up the air and hence increase its pressure from 10^5 Pa to 1.9×10^5 Pa thus generating an overpressure of 0.9×10^5 Pa. Using

equations (4 10 to 4 12),

the duration of the gas pulse = 1.96 ms;

the impulse due to the gas pulse = 79 Pa·s.

Assuming a charge at the centre of the cube,

the scaled distance of the charge from the wall is = 0.99 $m/MJ^{0 \cdot 333}$

and from Figures 4.15 and 4.16,

peak shock pressure = 0.75×10^5 Pa;

shock impulse = $400 \times 0.035^{0 \cdot 333}$ = 131 Pa·s;

shock duration (triangular pulse) = 0.35 ms.

The average experimentally recorded peak pressure was 11×10^5 Pa and the impulse was 175 Pa·s, each with an estimated accuracy of 20%. These are to be compared with a predicted peak pressure of 7.5×10^5 Pa and a total impulse of 210 Pa·s. This agreement is considered good in view of the large number of approximations made and is more than adequate when assessing protective structures.

For brass,

σ_y(static) = 259×10^6 Pa;

$E = 101 \times 10^9$ Pa;

$d = 8430$ kg/m^3;

assume relation between dynamic and static σ_y is as for steel.

For shock loading of cubicle,

normalised impulse ($= iE^{0 \cdot 5}/f\sigma_y h d^{0 \cdot 5}$) = 0.35;

normalised peak pressure ($= pX^2/f\sigma_y h^2$) = 40.

For gas pulse,

normalised impulse = 0.23;

normalised peak pressure = 4.8.

Both of these points are within the survival region of Figure 4.26 and this is consistent with the experimental observation of no significant damage.

Two reinforced concrete cubicles were tested, one more extensively than the other. We concentrate on the former. This cubicle was 30"×30"×17" with 2" thick walls and a vent measuring 17"×15". Details of the reinforced concrete used are a little sketchy: the concrete was thought to have a compressive strength of 3000 to 3500 psi, the reinforcing steel an ultimate tensile strength of 20–30 tons/sq in and reinforcement in the range 0.61 to 0.85 % in each direction and each face.

These figures have been re–interpreted as follows:

cubicle volume = 0.25 m^3;

side of cube = 0.63 m;

vent area = 0.16 m^2;

wall thickness = 0.0508 m;

d for concrete = 2400 kg/m^3;

E for concrete = 24×10^9 Pa;

σ_y(static) for steel reinforcement = 2500×10^5 Pa;

reinforcement (for tension members) = 0.73 % by volume.

Tests conducted with a 1/4 oz charge would lead to the following loadings, calculated as before:

peak gas overpressure = 1.01×10^5 Pa;

duration of gas pulse = 3.21 ms;

impulse due to gas pulse = 143 Pa·s;

peak shock pressure = 8×10^5 Pa;

impulse due to shock = 131 Pa·s;

duration of shock = 0.33 ms.

For reinforced concrete,

$$\sigma_y(\text{static}) = 250\times10^6 \times 0.0073 = 1.8\times10^6 \text{ Pa};$$

$$\sigma_y(\text{dynamic}) = 1.8\times10^6 \times 1.38 = 2.5\times10^6 \text{ Pa}.$$

For shock wave,

normalised impulse = 1.10;

normalised peak pressure = 2.67.

For gas pulse,

normalised impulse = 1.20;

normalised peak pressure = 0.34.

Comparing these values with the iso–damage curve in Figure 4.26, we see that the shock wave point lies essentially on the boundary between survival and deformation whereas the gas pulse one lies well on the survival side. The experimental result was reported as 'no damage', in agreement with the predictions.

Increasing the charge to 0.53 oz increased these results to:

For shock wave,

normalised impulse = 1.9;

normalised peak pressure = 5.3.

For gas pulse,

normalised impulse = 3.8;

normalised peak pressure = 0.72.

This time the shock wave is clearly in the 'deformed' region with the gas pulse remaining in the 'survival' region. Experimentally, this charge resulted in 'slight cracking' in various parts of the structure.

Increasing the charge still further, to 1.03 oz, resulted in substantial damage to the cubicle,

corresponding to predictions that both the shock wave and the gas pulse would be in the 'deformed' region.

These results show that the method given in Section 4.11 has predicted the onset of failure in the reinforced concrete cubicle with remarkable precision. High's experimental results also indicate that on a 'once in a lifetime' basis, a reinforced concrete cubicle should be able to withstand an energy release of about 4 times that which it can sustain without showing obvious signs of damage and which may, therefore, withstand the loading repeatedly.

4.12 Human response to blast loading

The same principles discussed above for damage to buildings can be applied to human injury. Three types of injury are recognised.

- Primary injury is due directly to blast wave over–pressure and duration. Over–pressures are induced in the body following arrival of the blast and the level of injury sustained depends on a person's size, gender and possibly age. The severity of injury is greatest where density differences between adjacent body tissues are greatest. This includes the lungs, ears, larynx, trachea and the abdominal cavity.
- Secondary injury is due to the impact of missiles.
- Tertiary injury is due to displacement of the entire body which will be subject to strong accelerations and impact loadings.

Data are often presented in the form of survival curves (see Figures 4.28–4.29) or as pressure–impulse diagrams for specific types of injury (see Figures 4.30–4.36). As might be expected, different curves must be used depending on whether the person is standing up or lying down.

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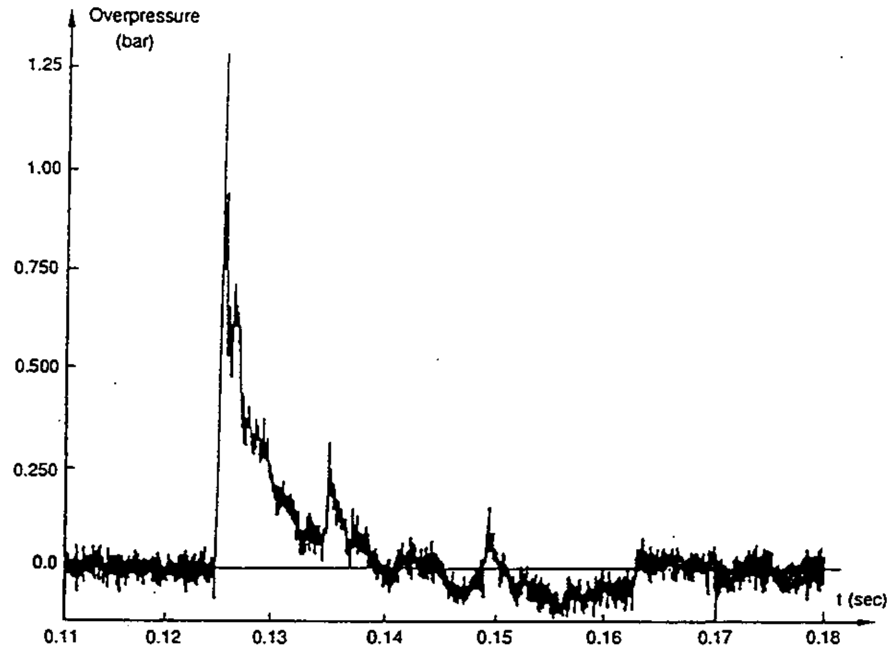

Figure 4.1. Overpressure recorded during an explosion.

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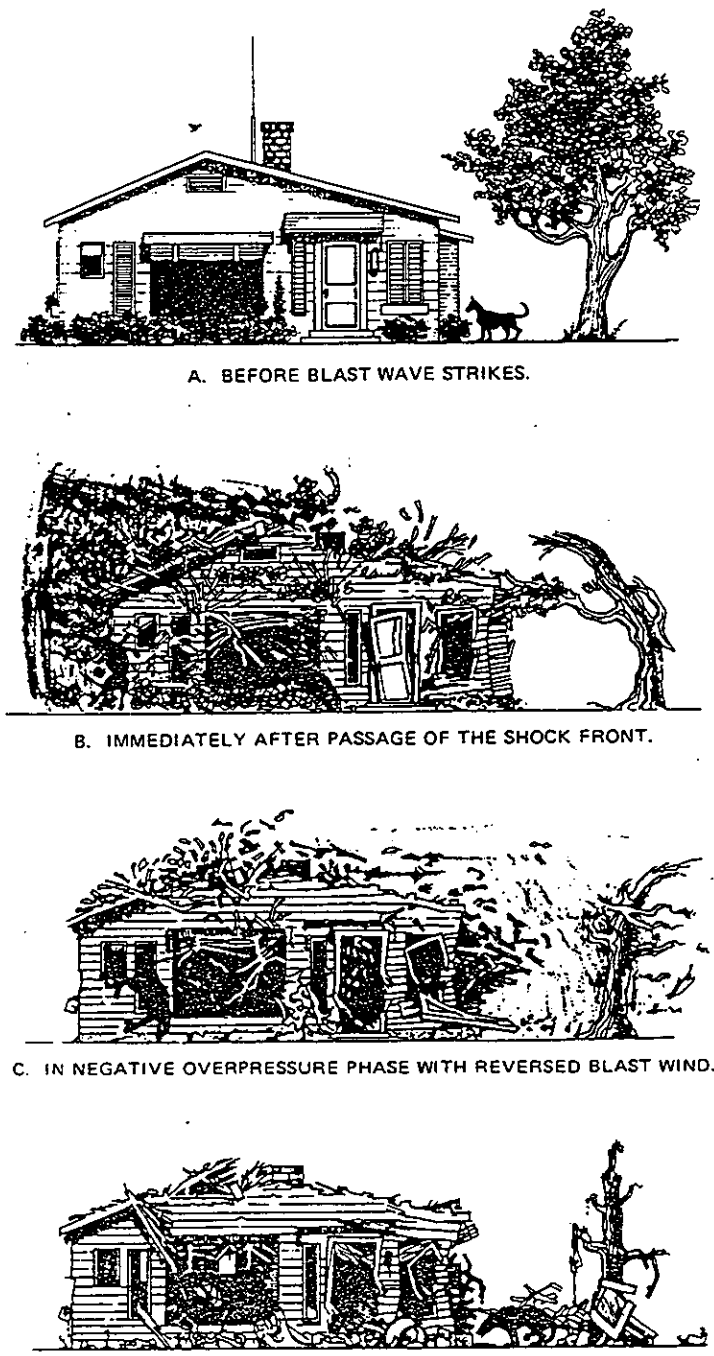

A. BEFORE BLAST WAVE STRIKES.

B. IMMEDIATELY AFTER PASSAGE OF THE SHOCK FRONT.

C. IN NEGATIVE OVERPRESSURE PHASE WITH REVERSED BLAST WIND.

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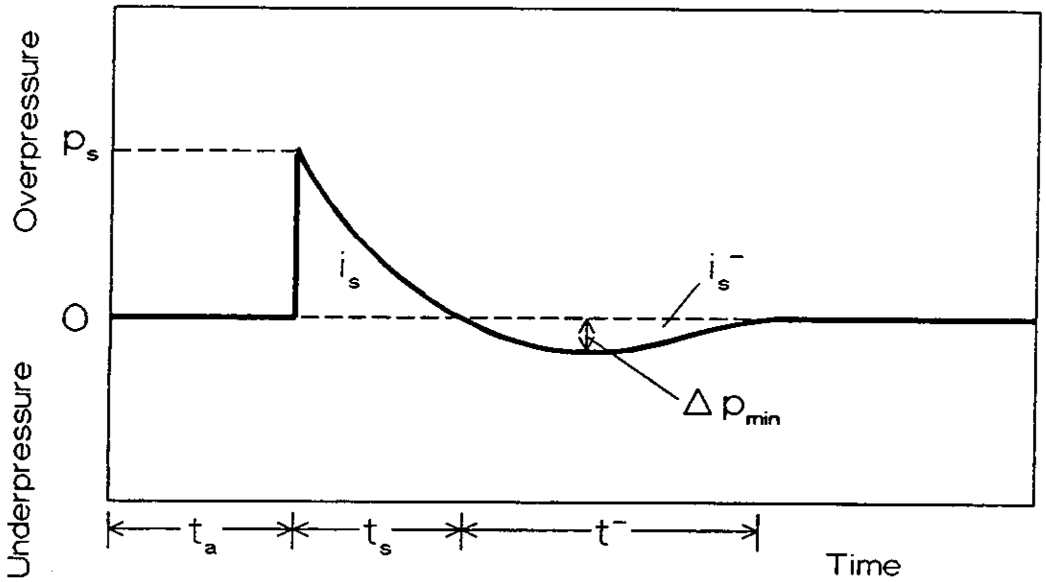

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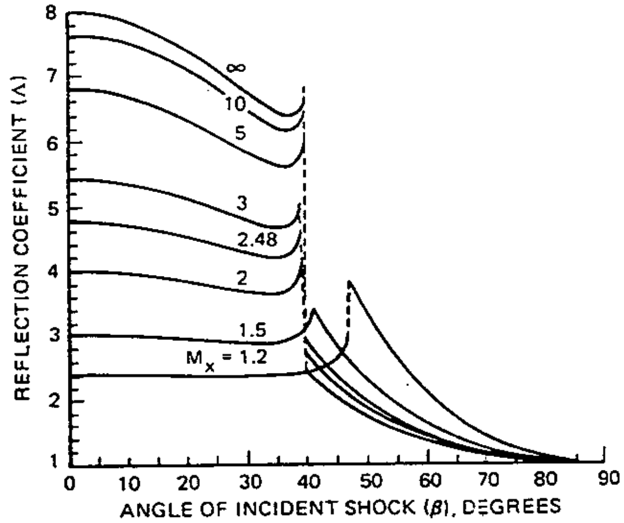

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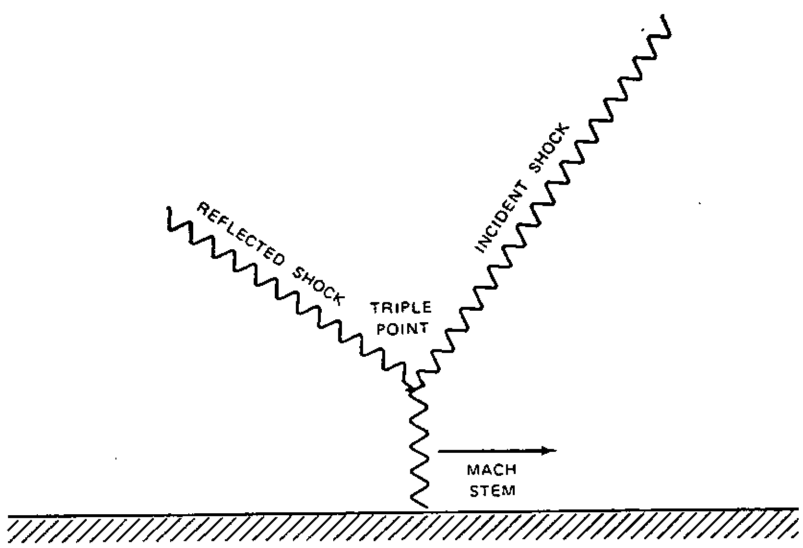

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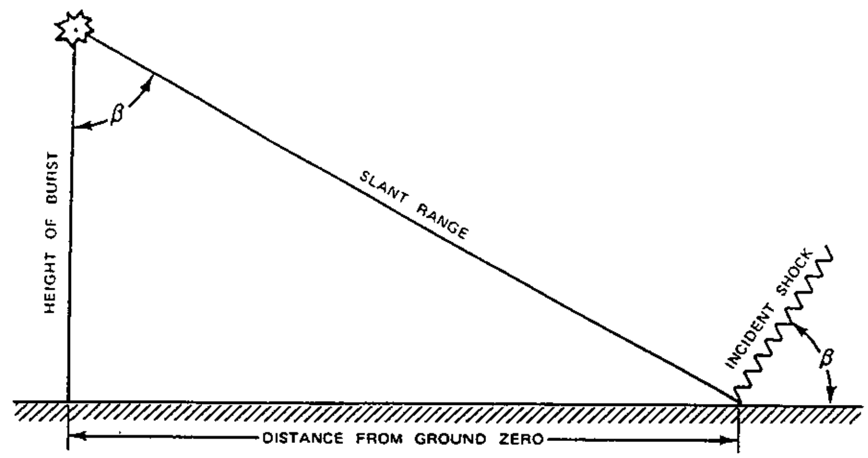

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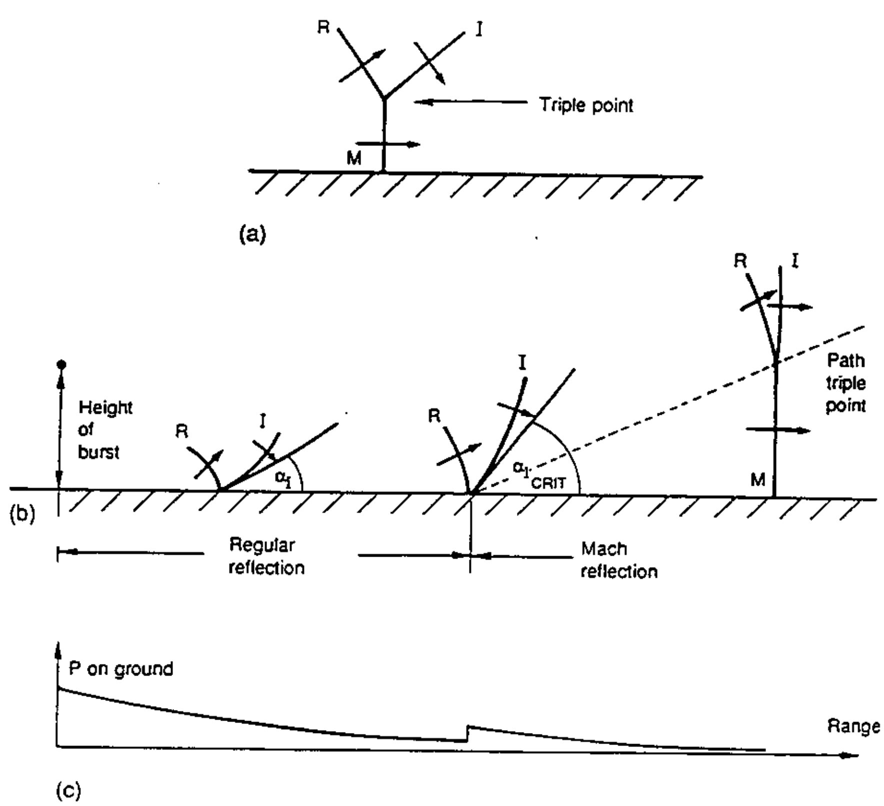

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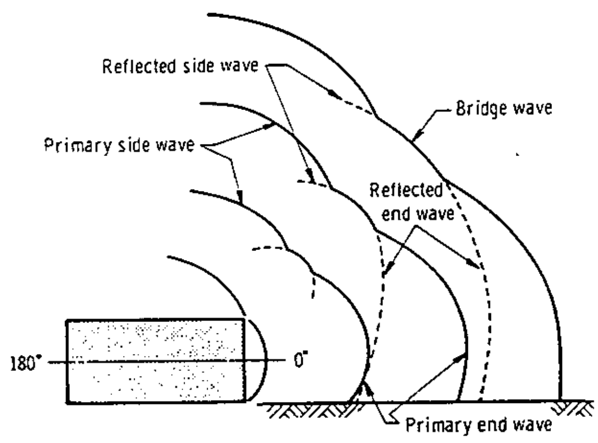

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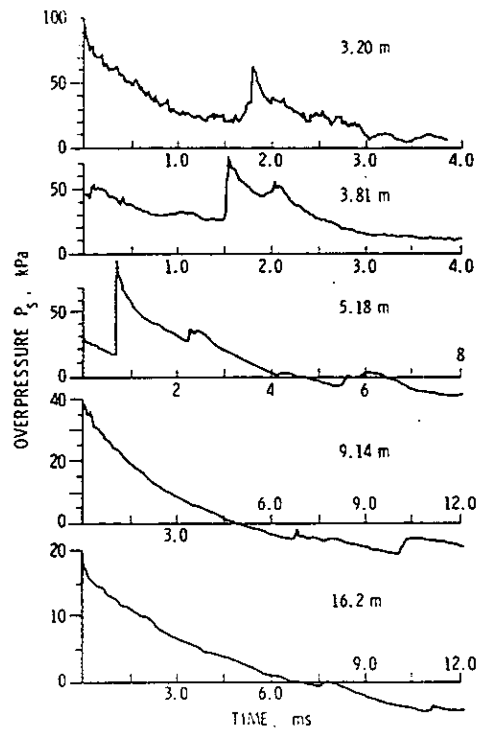

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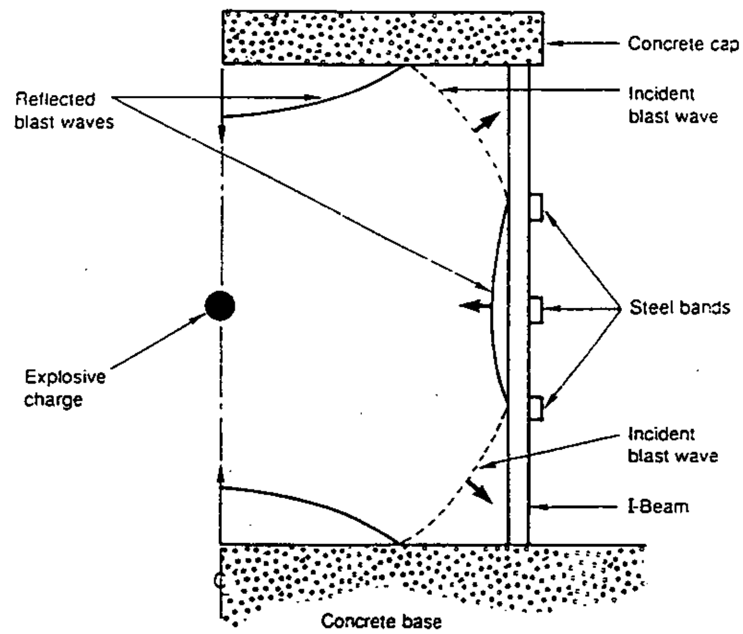

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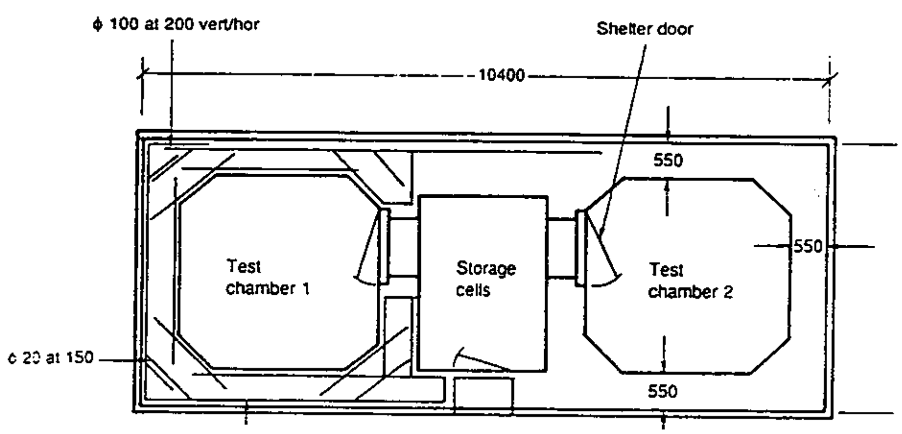

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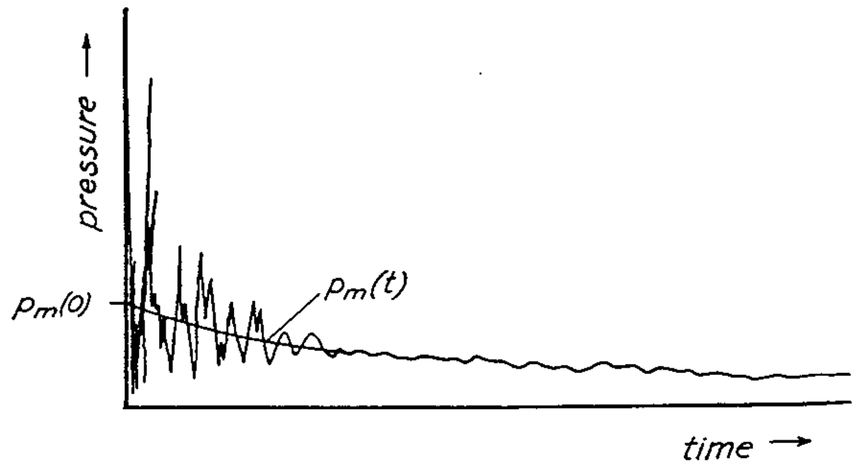

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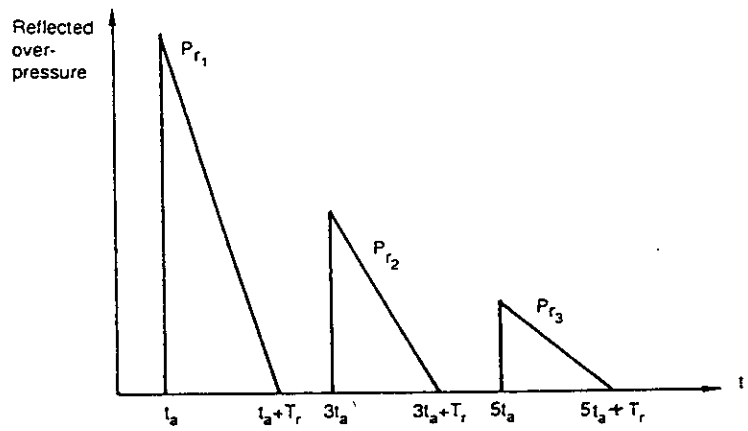

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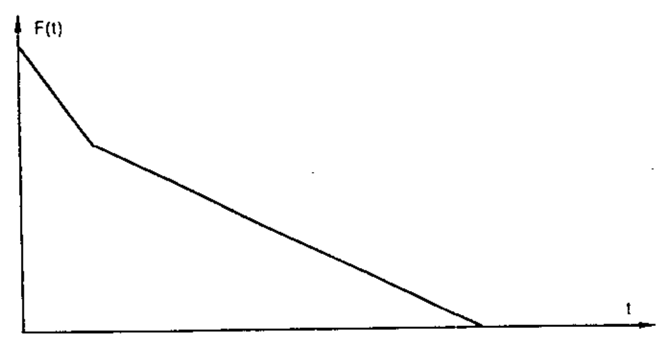

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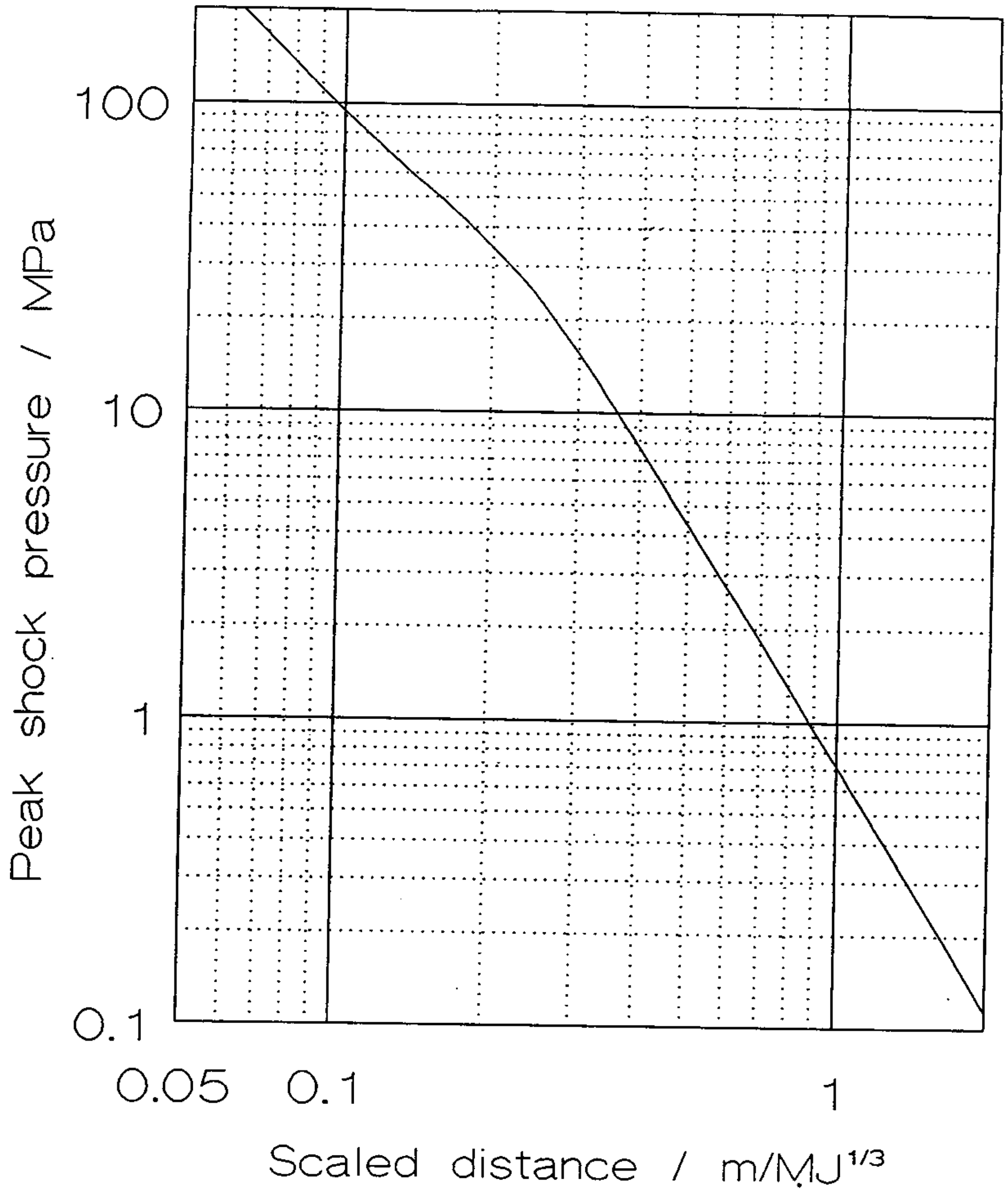

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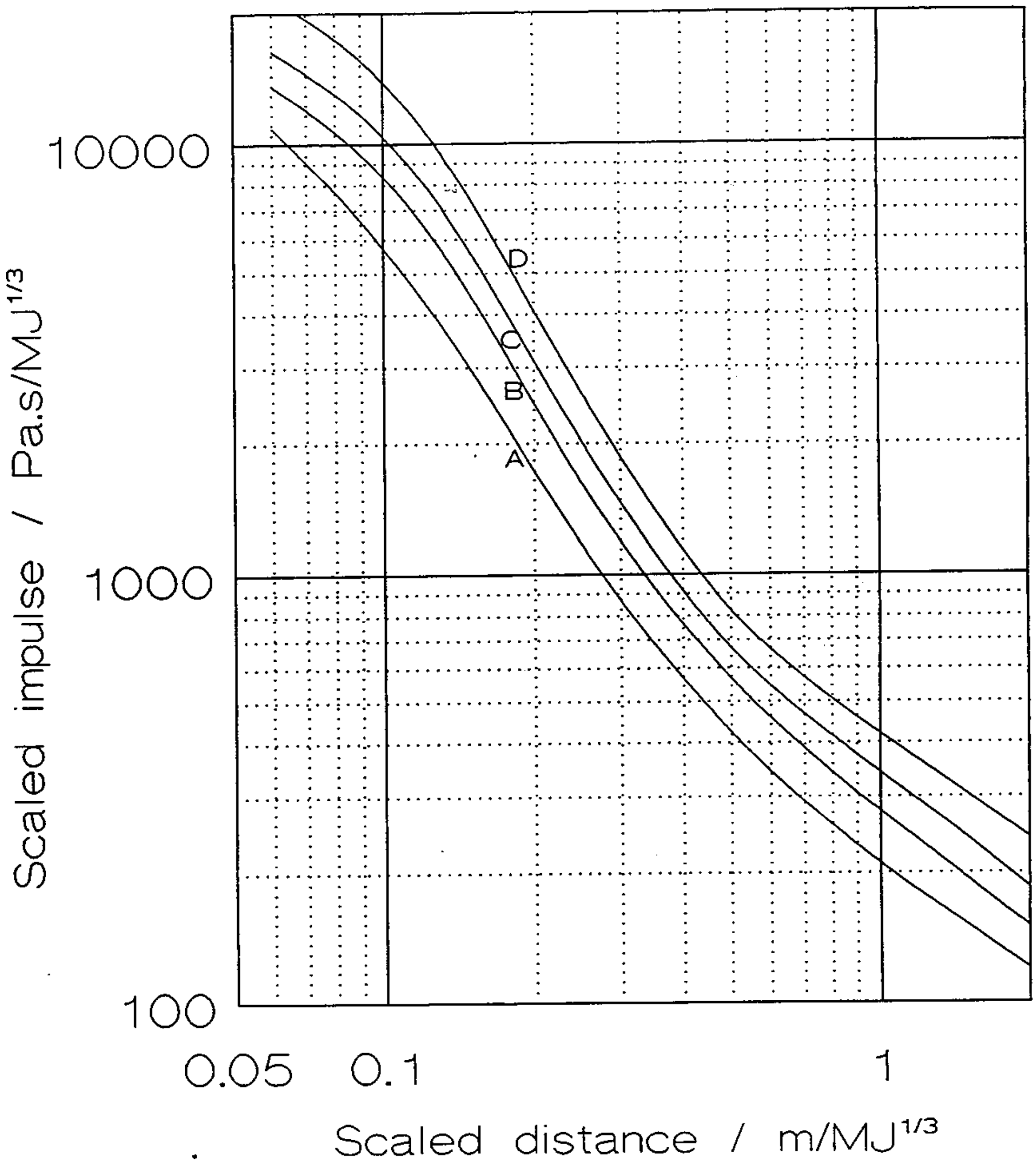

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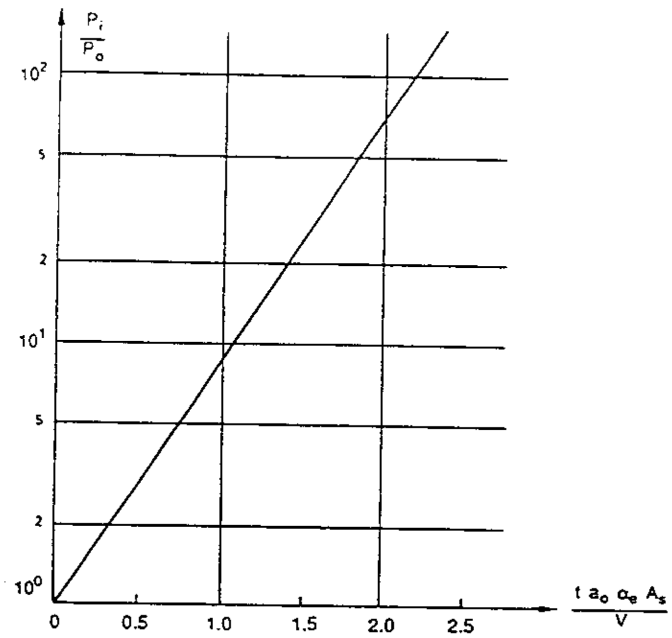

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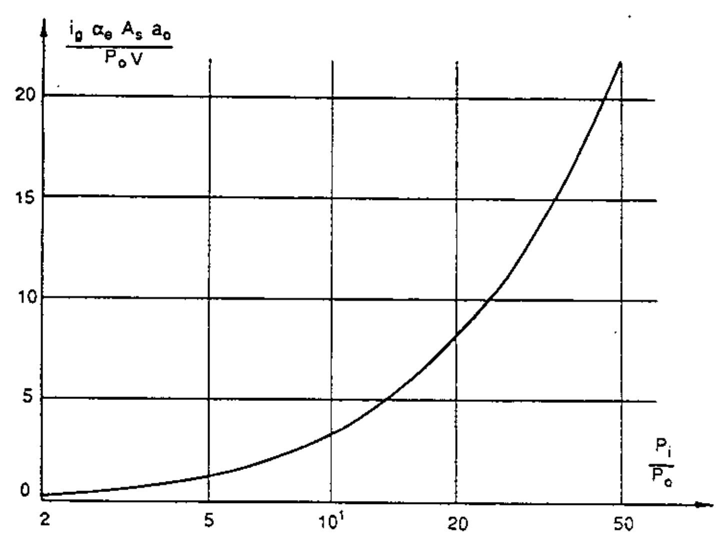

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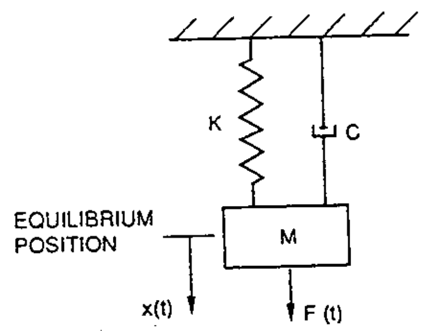

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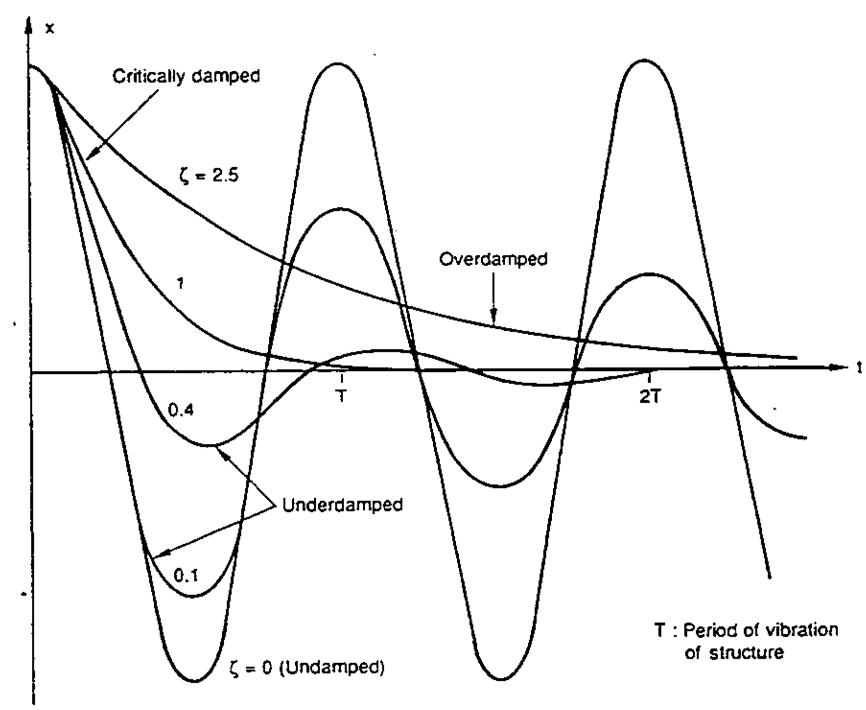

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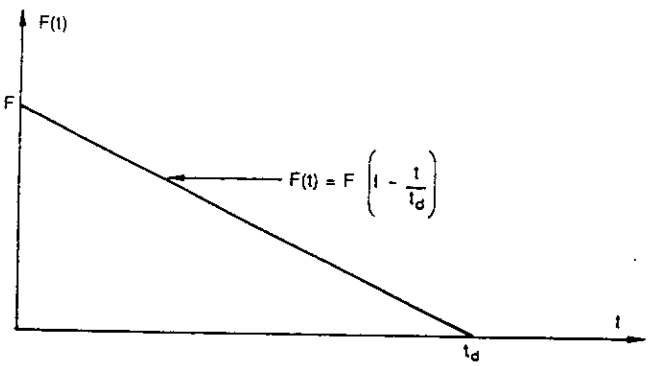

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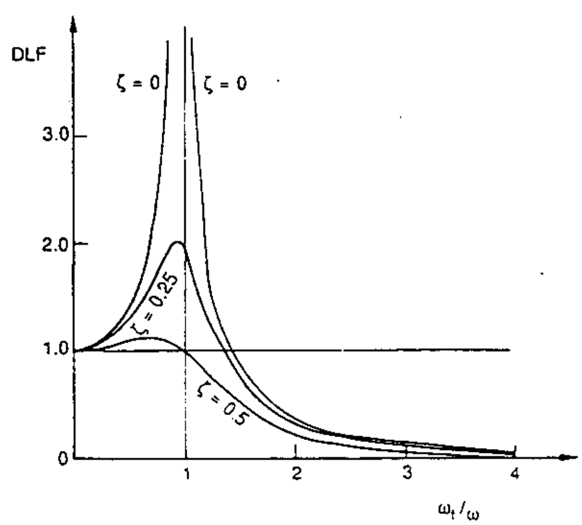

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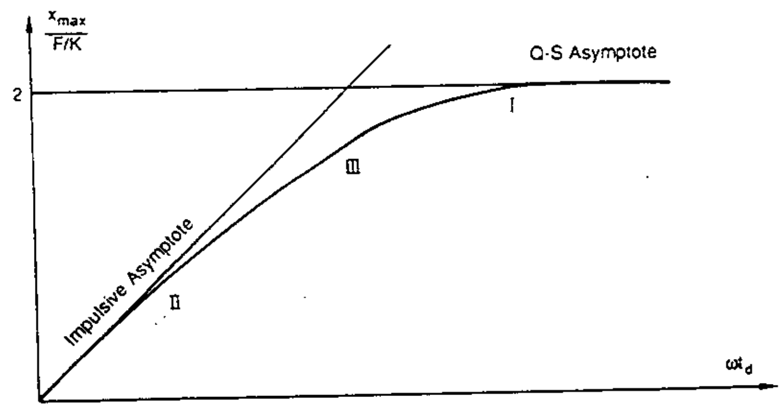

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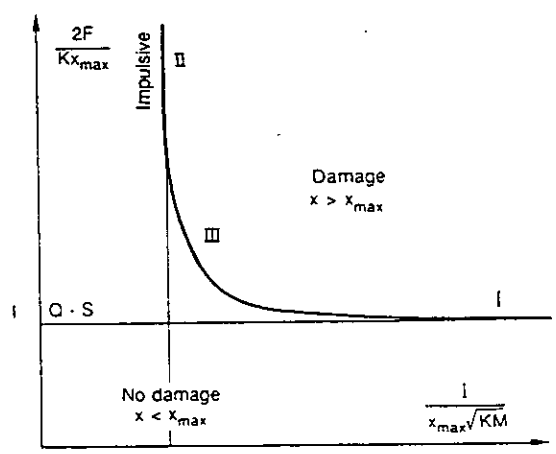

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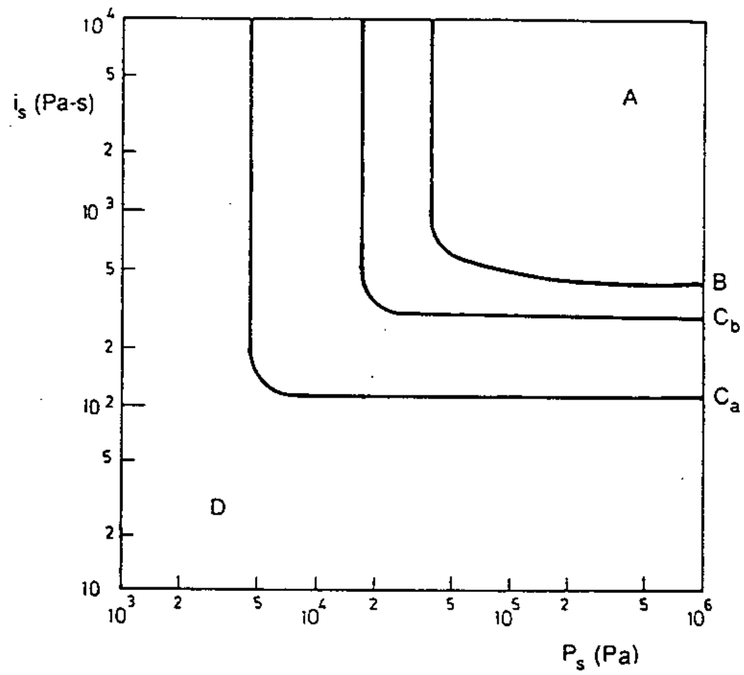

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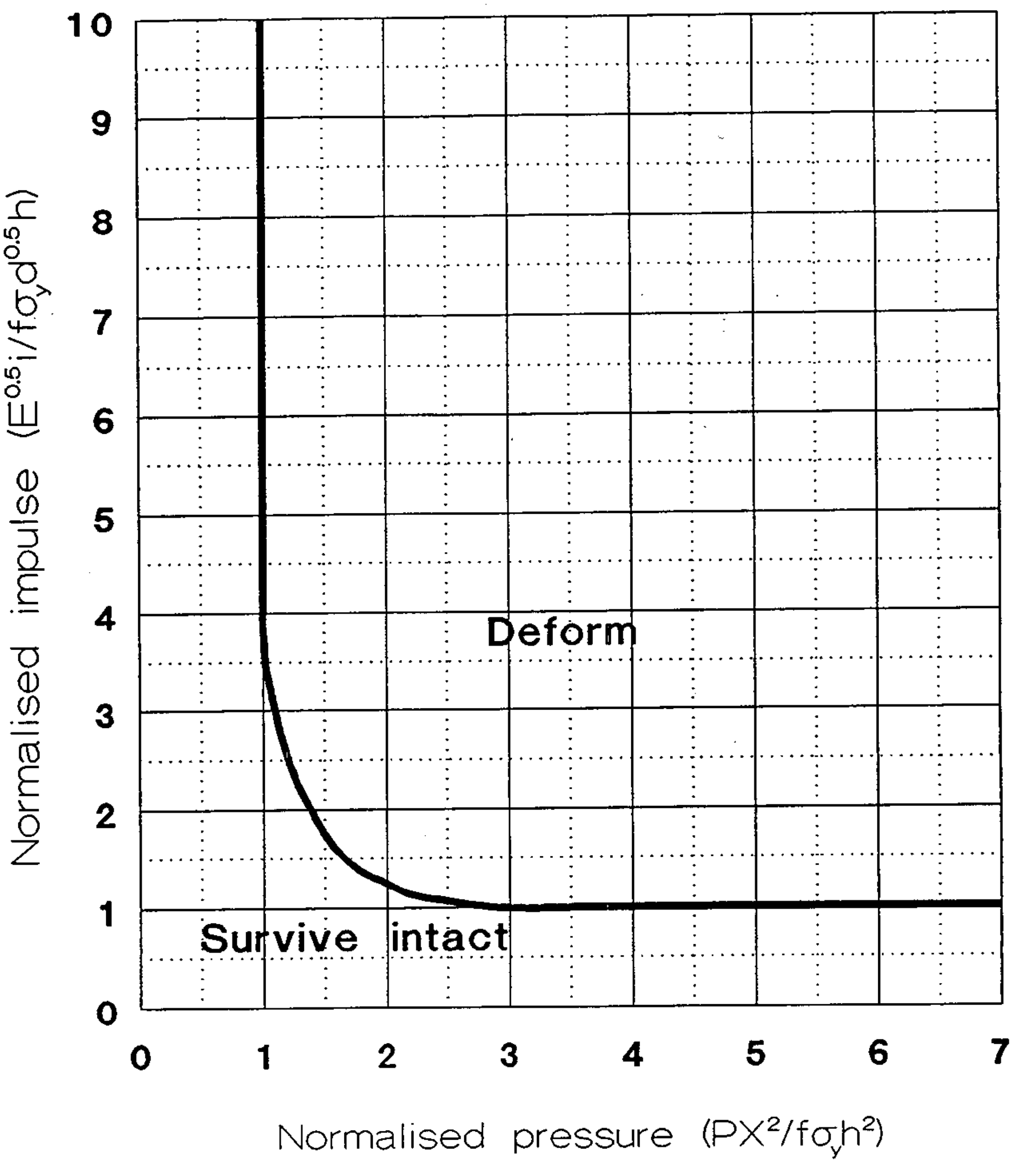

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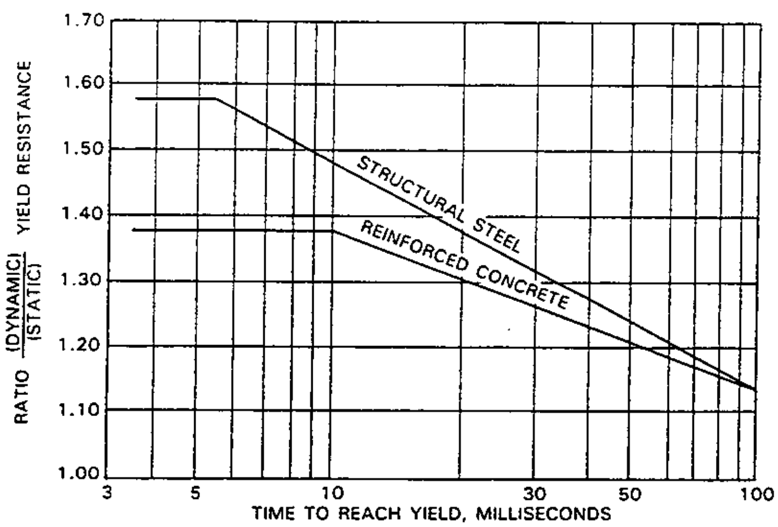

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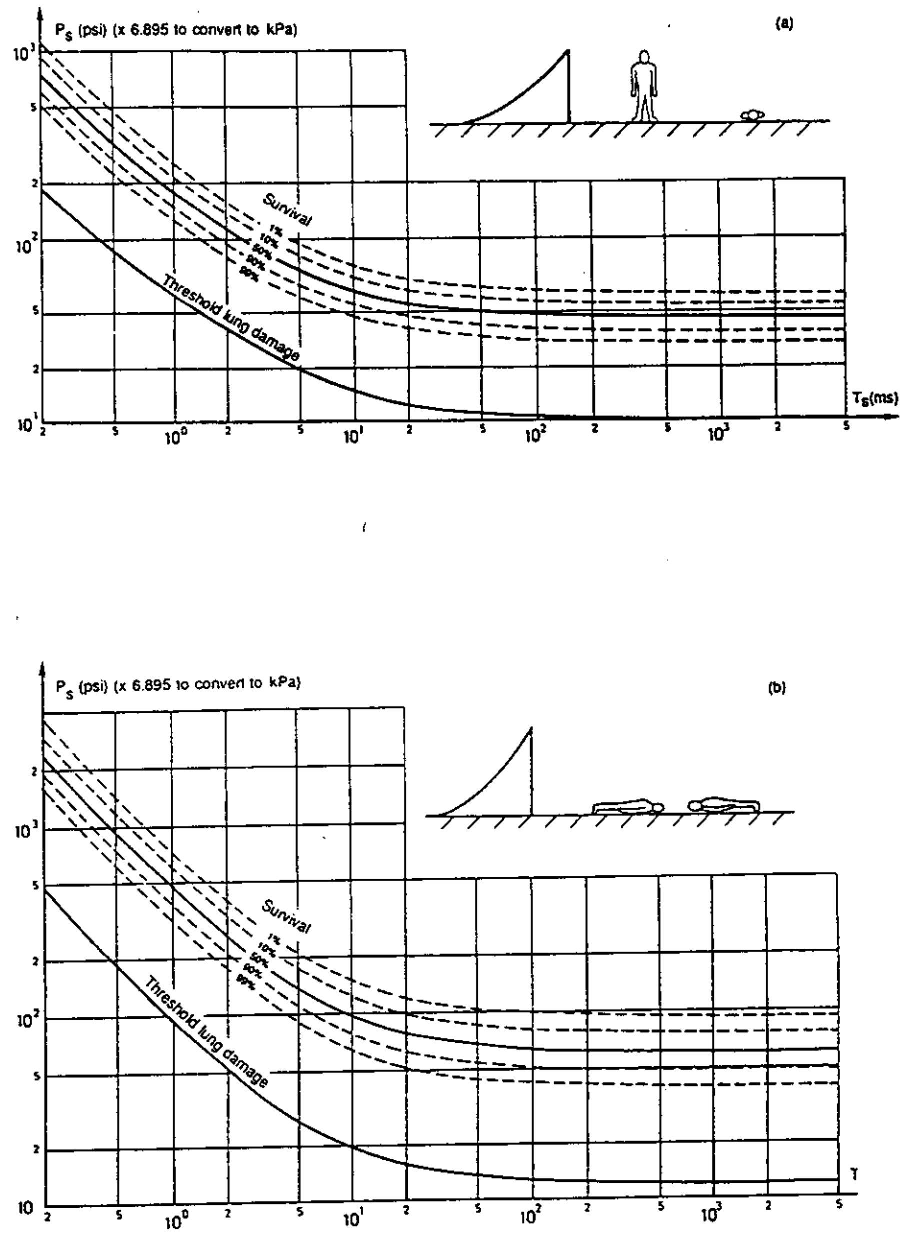

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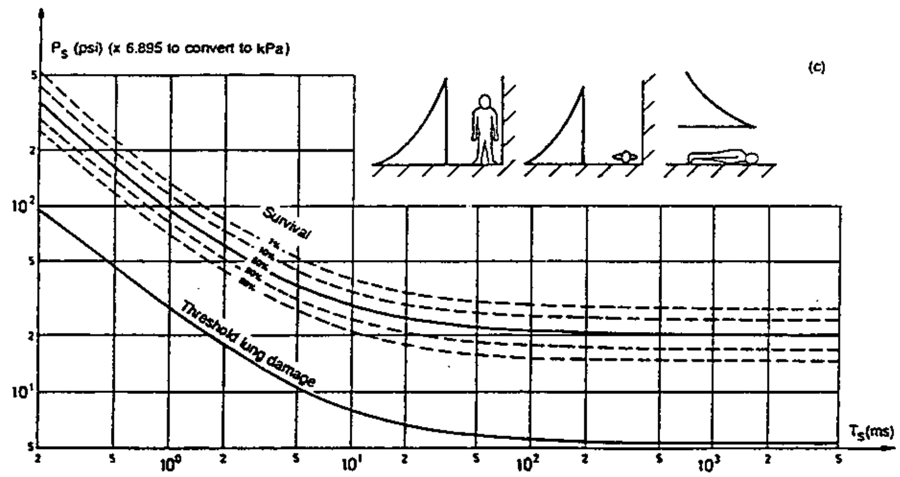

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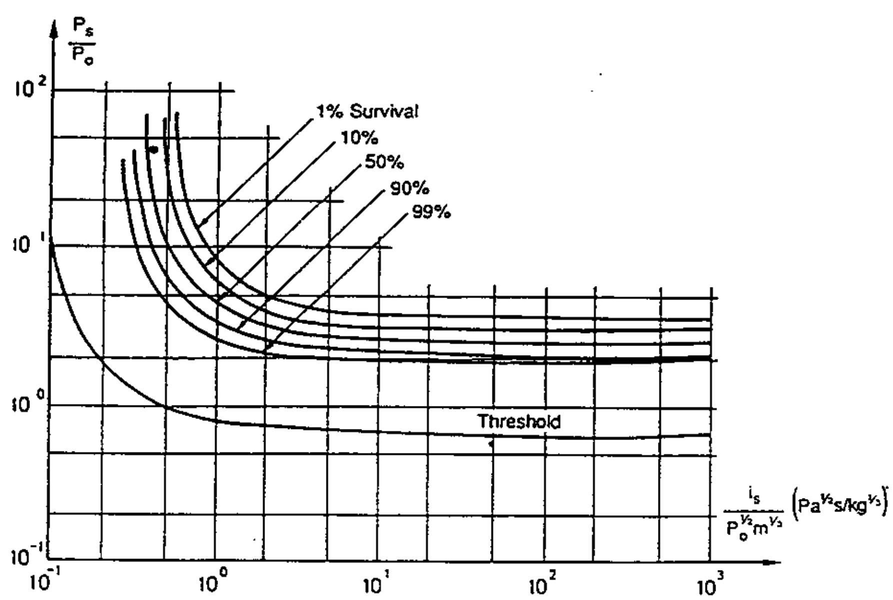

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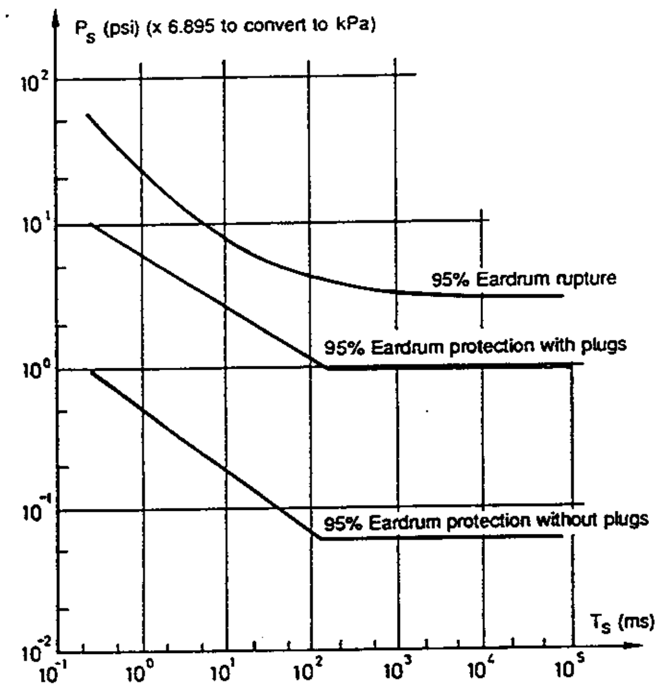

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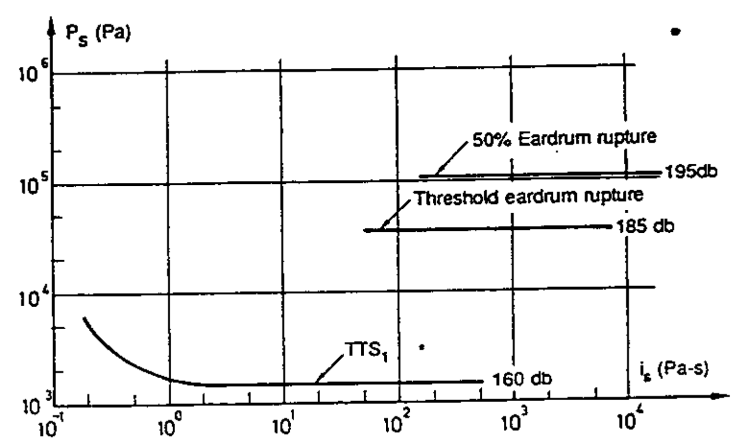

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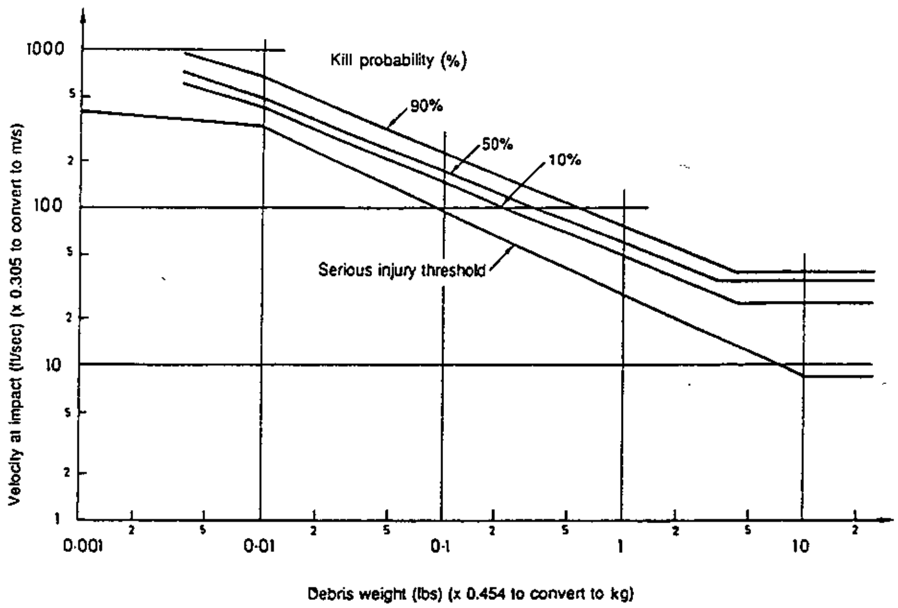

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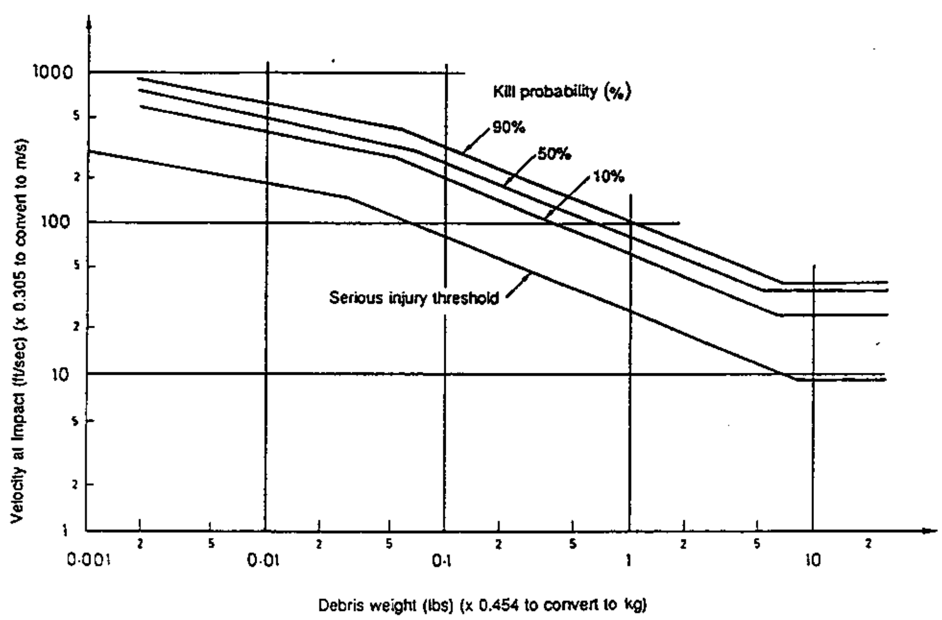

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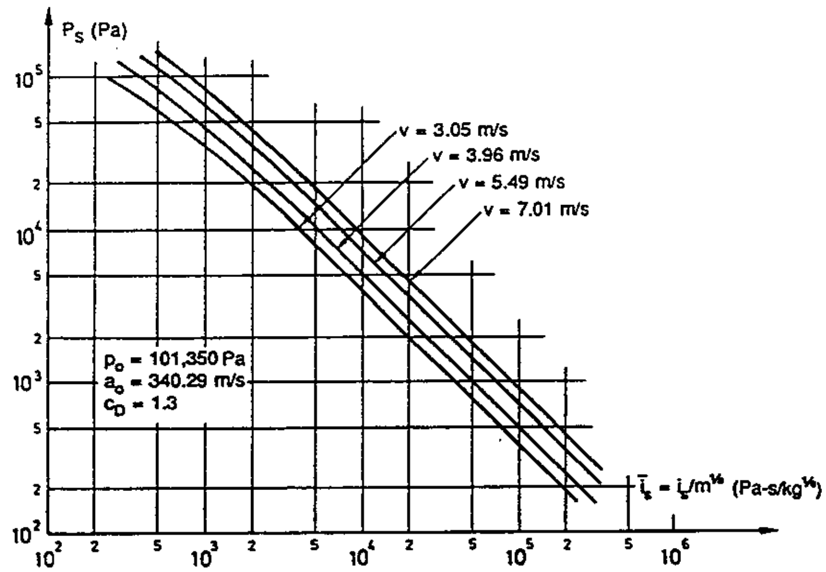

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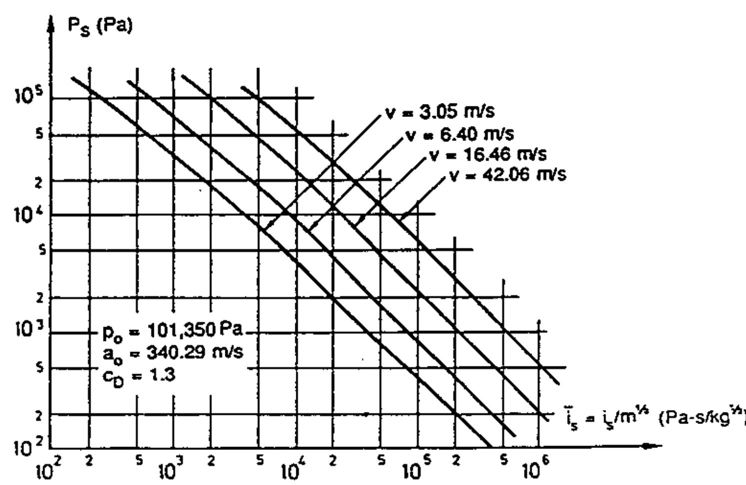

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5 PRESSURE TEST EQUIPMENT

5.1 Hazard assessment

All pressure testing must be subjected to a thorough hazard assessment whether the testing is one–off or an all purpose facility is being designed and whether the pressure testing equipment is portable or fixed. Many factors go into such an assessment, as will be discussed below, but in general, the greater one's ignorance about the item being tested, the more protection that will be needed. It is difficult to think of any pressure test that will need no protection at all, although in a few cases it may well be quite minimal.

The size range of items to be tested may vary from small laboratory items of a few cm^3 in volume to larger industrial items of several tens of m^3. Pressures may vary from a few bar to several kilobar. The energy content of the items being tested may vary, therefore, over a range of as much as 10^6. It may be tempting to consider the lowest energy content cases as inherently low risk and requiring little or no protection. This supposition is false. The low energy case can still eject a small projectile with high velocity which can cause a severe injury even if the blast is negligible. Thus, in general, protection required during pressure testing may be as little as a sheet of polycarbonate or as large as a heavily reinforced concrete bunker.

The largest single issue in conducting a hazard assessment of a pressure test is the energy stored in the pressurised system and the severity of any failure which results, that is blast and fragments, whether of the item being tested or of the equipment being used to carry out the test. The magnitude of the hazard can be reduced by a number of measures, the most important of which is to test with water or oil whenever possible, rather than using air or other gases, due to the inherently lower energy content of the former, but nevertheless, this does not reduce the hazard to one of negligible proportions. Another general policy is to keep the internal volume of the item being tested as small as possible. In tests of small items, it is often possible to do this by filling up the inside of a vessel with solid material, although it is

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appreciated that in most industrial situations this is probably impractical.

However, consideration can be given to conducting the more dangerous tests (e.g. the proof test) on sections of the system and only putting them together for the final leak or function test. Hazard assessments are required, of course, for each separate section of the test and for that of the complete assembly undergoing its leak or function test.

In most cases, a choice will need to be made between a fixed pressure testing facility and a temporary/mobile one. Except for on–site testing, where a temporary facility may be the only feasible choice, the decision on which way to go will usually be determined by a combination of safety and economic factors, so there will often be a need to pursue the safety assessment along both routes.

Most fixed pressure testing facilities are designed to handle the testing of a range of items to a range of pressures with a range of pressurising fluids. In order to design these facilities it is necessary to conduct at least a partial hazard assessment on each item for which testing is anticipated to determine the parameters which influence the design of the fixed facility. In the case of a temporary/mobile facility, the assessment will usually be of a one–off test.

Normally, the two most important factors in the hazard assessment are the magnitude of any blast formed by a failure of the item being tested and the penetrating power of any missiles formed during that failure. Both of these factors depend on a knowledge of the stored energy in the system being tested. The quantitative determination of these issues for a given failure mode has been discussed in Sections 2 to 4.

In conducting the hazard assessment on each item or system being tested it is necessary to consider every potential mode of failure: the possibility of brittle failure, of ductile failure, of loss of a major section and of loss of a minor component. Although the first two of these possibilities involves total destruction of the item, the latter two do not and, for example, there will usually be several ways in which a major section can be lost and even more for

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minor components such as a temporary plugs. Each and every possibility must be examined separately. In some cases, it might be possible to come to the conclusion that a given failure mode is so unlikely that it can be ignored; for example, if one is sure that all of the materials used in the manufacture of the item are highly ductile at the test temperature, that the material properties have not been changed deleteriously by welding or heat treatment and that the item is of sufficiently simple design for the non–destructive testing (NDT) carried out on it to demonstrate, with total confidence, that the item is free from defects, then the possibility of brittle failure is remote and this mode of failure can be ignored.

Where missiles are concerned, the hazard assessment must take into account not just the speed of the missile and the thickness of material required to stop it but also the direction of flight of the missile. This latter will usually be identifiable for small components such as plugs and end caps, but may be more difficult for larger vessel sections; for a brittle failure, missiles in all directions must be assumed.

In designing the test facility, whether fixed, temporary or mobile, the protection must be sufficient to withstand the worst cases, both of blast and missiles. This does not mean that all–round protection to the same level is necessarily required, only that the level of protection in a given direction must be sufficient to withstand any assault coming from that particular direction, but all directions which receive harmful blast or missiles must normally be protected. One exception is a the test in conducted on an open test site where personnel and public are excluded in non–protected directions, as, for example, when the test is conducted behind a blast wall or barrier. A variant on this is the placement of personnel and their equipment inside a protective bunker with the blast free to go in all directions around them. It should be remembered, however, that the exclusion zone when a large item is being tested may well be measured in kilometres. The minimum objective in all cases, whatever the means of protection used, must be to prevent injury to the operator, any personnel in the neighbourhood (defined as widely as is necessary) and the general public. In most cases there will also be a need to reduce consequential economic loss following a failure during test.

Finally, the hazard assessment must also include the pressurising equipment used in conducting the test and the way in which it is used. The equipment side of this assessment comprises the pressurising equipment itself and any ancillary equipment used in mounting the items under test. The assessment should throw up any need for handling equipment (such as cranes), special assembly tools (such as hydraulic bolt tensioners and torque spanners) and any need for interlocks (for example between the application of pressure and the door of the enclosure where there is the possibility of unauthorised access).

5.2 Protective enclosures for fixed installations

It is convenient to divide protective enclosures into small ones, which would normally only contain the item being tested, the pressurising equipment being outside the enclosure, and larger ones, which may well be walk–in, which could contain the pressurising equipment (except for the control valves and pressure gauges, which would always be outside) as well as the item being tested. If the pressurising equipment is outside the enclosure, the hazard assessment must point to negligible risk in so doing; if it is inside, it becomes part of the item being tested and common hazard considerations apply, which may well be less onerous than a requirement for negligible risk posed by exposed equipment. However, if the pressurising equipment is inside the enclosure, it may be damaged by fragments from a item which fails during test and, apart from the consequential economic loss, the possibility of secondary hazards arises and these must be taken into account in the hazard assessment.

The objective of the enclosure is the containment of the effects of vessel or component failure during a pressure test. This does not mean any sort of failure, only those which are likely to occur with the items actually being tested. Thus if the hazard assessement indicates that brittle failure is so remote that it can be discounted, then there is no point in making provision for this type of failure in the enclosure. Another point to be considered is how often one expects the worst possible failure to happen. If it is infrequent, is it really necessary to build an enclosure that is capable of any number of such failures rather than one which will withstand but a single failure? Roughly speaking, an enclosure designed for

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repeated explosions (blast) of a given size will withstand one of four times that size in a once in a lifetime explosion (see Section 4.11.1). Of course, once the latter happens, the enclosure has to be demolished and a new one built in its place.

Sections 3 and 4 above have dealt with the two factors which a protective enclosure must withstand, namely, blast and missiles and with the response of various structures to them. Most of the response information relates to reinforced concrete and steel, with some on brickwork and soil (this indicates the military origin of much of this information) and it is logical to prefer the use of these materials when providing protection during pressure testing. This is not to say that other material could not be used successfully, merely that we have few ways of quantifying their capacity in this role.

In designing an enclosure which will withstand specified levels of blast and missiles, it is important to take into account the essential difference between these two types of assault. Blast, to a first approximation, produces a uniform pressure loading over the whole of the inner surface of the enclosure. Missiles produce heavy local damage over a radius which is only two or three times the radius of the missile itself in addition to a more general impulsive loading of the structure. Thus, in designing a structure to withstand blast, the two most important issues are: is each panel sufficiently rigid to withstand the uniform loading without excessive deflection and are the corners of the structure sufficiently strong to hold the panels in place? The main issue with regard to missiles is whether the panel thickness is sufficiently large to withstand perforation. A second issue is whether the panels are adequately anchored to withstand the impulse imparted by the missile.

Generally, blast advances on a spherical front, weakening as it does so (the inverse square law). Thus, the larger the enclosure, the smaller the pressure to which each wall of the enclosure is subjected. This fact also points to the desirability of placing the item being tested in the middle of the enclosure rather than near one side, otherwise, additional strength will be needed on that wall. If the test enclosure is totally enclosed so that none of the blast escapes, the effect of the blast on the surroundings can be ignored. However, the

situation with a vented cubicle, or in the extreme case, a safety wall, is very different and the effect of the escaping blast must be considered.

Other than in a brittle failure, missiles created during a pressure test will tend to travel in preferred directions and the hazard assessment should have identified the size, speed and direction of each potential missile. Advantage can be taken of this by aligning the item being tested such that the most damaging missiles impact on the strongest walls of the enclosure. This means that not all walls need to be of the same strength, provided, of course, that it does not compromise the enclosure's ability to withstand blast.

If the vessel failure results in a supersonic shock, this will be the first to reach the walls of the cubicle. It will be followed by missile impacts, the gas overpressure impulse and reflected shocks in an order which is difficult to predict. In principle, the determination of the structural response of the cubicle should be on the basis of this composite assault. In practice, this is simply not feasible. In Section 4.7 we saw how to combine the initial supersonic shock with its reflections, thus reducing the number of impulses to be considered to three – supersonic shock, gas and missile. We suggest that enclosure design should be such that it can withstand each of these impulses singly and that engineering judgment is used to decide whether extra strengthening is required to withstand the combined assault.

Protective enclosures can take many possible forms. They could be small reinforced cupboards or merely protective screens at the low energy release end of the scale to large walk–in concrete bunkers at the other. It is often possible for the user to design his own protection at the lower energy levels since he can use overdesign to cover his limited knowledge of the subject without incurring excessive cost. At the other end of the energy scale, the design will usually require the services of a professional civil engineer. This will be particularly so where reinforced concrete bunkers are concerned since detailed design of these involves more than just the wall thickness and the amount of steel placed in tension (see Section 4.11) but also how the reinforcing on the front and back faces of each wall are

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interlaced and how the corners of the structure are reinforced. Examples of what is involved may be found in references [5.1, 5.3–5.5].

The determination of the type of protective enclosure tends to be one of trial and error since it involves economics as well as engineering. However, the general procedure should be as follows:

• Determine the stored energy in the system, using the methods given in Section 2.

• Determine the shock wave energy and the mass, kinetic energy and direction of flight of all missiles, again using the methods of Section 2.

• Choose a construction material (steel, reinforced concrete, etc) and determine the thickness required to prevent missile perforation for each missile (Section 3).

• If the shock wave energy is non–zero, choose a cubicle size and determine the expected impulsive loading on the walls of the cubicle using the methods of Section 4.7.

• Determine the missile impulsive loading on the walls (Section 4.8).

• Choose a cubicle wall thickness, perhaps that required to resist perforation as a first try, and determine whether it is capable of withstanding the shock and impulse loading to which it will be subjected (Section 4.11).

• In the light of the results just obtained, revise the design as appropriate and re–assess.

It is at this last step that non–engineering considerations come into play, such as:

• whether or not the cubicle should be vented and if so by how much,

• the damage to the surrounding area if a vented cubicle were used, including the effects of both blast and missiles,

• whether protection on 'once in a lifetime' basis is required or whether the protective structure should withstand repeated vessel failures,

• allowance for uncertainties in the assessments of hazards and in the design procedures given in Sections 2–4,

• cost.

General guidance on these issues is difficult to give, but we suggest that where only small items are being tested, preference should be given to designs involving total enclosure, since

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cost is not likely to be large in any case, but where very large items are being tested, total enclosure may well be infeasible.

Experience leads to the following suggestions:

- For energy releases of up to 0.1 MJ, closed cubicles made of 6 mm mild steel plates attached to an angle iron frame are often very convenient. Higher energy loadings may be possible in some cases, particularly if destruction of the cubicle is acceptable. The volume of such a cubicle would typically be within the range 1–10 m^3.
- The scale up of such a cubicle to larger capacities and larger energies is certainly feasible but it would probably necessitate the use of thicker steel plate and RSJs to provide the structural strength rather than angle.
- For energy releases in the 10 MJ range, a reinforced concrete cubicle is likely to be less expensive than an all steel one.
- A pit has particular attraction as the basis for a closed cubicle since one can then make use of the soil around the walls of the pit to support the wall and to absorb missiles, which can now be allowed to penetrate the wall. In principle, therefore, the walls can be significantly weaker than would be necessary in a free standing cubicle. Of course, the weaker the walls, the more extensive the damage they would experience and the greater the cost of repair. If the pit is to be used as a closed cubicle, it will require a lid which is removable, which is strong enough to withstand blast and missiles and which is adequately bolted down so that it does not itself become a missile.
- Cubicles which are not completely closed have a number of attractions. The walls of a vented cubicle will normally experience a lower blast pressure than a closed one so that the walls do not need to be as strong. Another possibility is the use of a labyrinth entrance rather than a door (a labyrinth is required to prevent egress of missiles; a missile's energy is usually considered spent after two collisions with walls). The obvious problem is the possibility of blast and missiles emerging from the opening(s). It is difficult to calculate what these might be in any given set of circumstances but if this can be done, then it is certainly a viable option.

Particular issues which should be considered are as follows.

- Large steel RSJs attached securely to the wall and strategically placed in the line of flight of the most damaging missiles can absorb a lot of kinetic energy. In particular if the line of flight is vertically upwards, the beam of an overhead travelling crane can be an excellent absorber of energy, although the cost of repairs may be large.
- Any door in the wall of the enclosure must be suffiently strong not to be blown outwards. Hinges are unlikely to be able to provide this strength. Two solutions are eminently practical:

    ▸ place the door on the inside wall of the enclosure, make the door larger than the opening and strengthen the edges of the opening so that they are able to take the full force of the blast,

or: ▸ provide pins or bolts all around the periphery of the door to lock it in place during the test.

The former solution requires the door to be inward opening or sliding, the latter is more appropriate for an outward opening door.

- If the enclosure is made of steel plate with angle iron corners, put the plate on the inside of the angle. The bolts holding the plate to the angle are then never placed in tension and, therefore, will not fail. If the plate is placed on the outside, one must ensure that the bolts are sufficiently numerous and strong never to fail in tension when subjected to blast from the inside, otherwise the snapped bolts become particularly dangerous secondary projectiles outside of the enclosure.
- Windows in the wall of the enclosure should normally be avoided and closed circuit television used instead. However, if a window is installed it should be of a suitable missile resisting material, polycarbonate or armoured glass, and all viewing of operations within the enclosure should be through a mirror which is outside of the enclosure and which enables the operator to be out of line–of–fire of any missiles coming through the window.

5.2.1 Examples of protective enclosures

A number of protective enclosures have been described in the literature. The following lists a

number of these, but the list is not meant to be exhaustive.

Baker *et al* [5.9] have considered the design of structures subject to blast loading which are suitable for the construction of plant rooms and similar buildings for chemical plant.

Bergman [5.1] has presented a design for a structure capable of containing the detonation of 5 kg of TNT as used in the Swedish explosive manufacturing industry. This is shown in Figure 4.11. The main wall thickness is 550 mm and is heavily reinforced with steel bars in the inner and outer skin to resist bending. It is thought that this type of structure probably represents the upper limit on the size of explosion that can economically be fully contained without venting.

High [5.2] has described the type of vented cubicle that was developed by ICI to contain high pressure polyethylene plant. Tests on a model of this cubicle were analysed in Section 4.11.1.

A very interesting design of vented cubicle was developed by Monsanto [5.7] with the aid of the US Armour Research Foundation. Its roof is shaped as half of a cylinder, as shown in Figure 5.1, with the objective of deflecting the blast wave outward though a vent. Immediately outside an inner shield wall is a sand pit in which the blast and missiles are deflected. The sand is able to absorb most missiles and will help to attenuate the blast wave.

The test pit shown in Figure 5.2 is from Manning [5.8] and is used by a UK steel manufacturer for testing pressure vessels.

Bergman [5.1] shows, see Figure 5.3, an example of a structure which is designed to collapse safely by opening up on hinges. This is intended for explosives manufacture where missiles will not be a major concern. Presumably this is contained within an earth bank which will deflect the shock wave upwards and would contain missiles. It represents an intermediate stage between a cubicle and placing the vessel in an open field.

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Detailed information on the design of structures to protect against the effects of external nuclear explosions are given by the American Society of Civil Engineers [5.6]. This provides a wide range of design data and procedures but note that loading is external and not internal as would be the case for most pressure test cubicles. The design philosophy is also based on the assumption that only a single nuclear event need be withstood.

5.3 Protective equipment for testing on site

Testing on site presents particular problems in the provision of protection and should normally be limited to leak or function testing of items which have undergone proof testing elsewhere. Massive steel or concrete structures are usually infeasible and one is often reduced to basing protection on naturally occurring features. These tests are nearly always one–off and a full hazard assessment of the test is essential. The quantification of stored energy and the magnitude of any blast which follows failure and the determination of probable mass, speed and direction of any missiles which are formed do not differ from what is required for a fixed installation. The difference lies in the way these hazards are handled in order to make the test low–risk.

Factors which should be considered when conducting tests under these circumstances include the following.

- It is generally very much easier to provide protection against missiles than against blast, so avoid the use of gas as a pressurising medium.
- Perform the test out of hours when very few people are about. The implication here is that failure during the test may result in damage to the equipment within the factory or premises and perhaps substantial economic loss, but risk of death and injury is kept to a low value.
- Use existing walls to provide some protection. This should only be done after very careful consideration of the consequences of a failure. Brick walls are not usually very strong and blast or impact of a large fragment with them may result in destruction of the wall and perhaps even the premises. They could, however, provide adequate protection against the effects of small missiles.

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- Consider installing shields in critical directions to stop missiles. The calculation of the thickness of such a shield does not differ from that for a fixed facility, but the way in which the shield is supported must be considered very carefully. A large missile travelling at speed has a lot of kinetic energy and on impact with a shield that kinetic energy will be shared between the missile and the shield. If the shield is free to move, it will become a secondary missile with potential to cause damage or injury. The way in which it moves may also be important; for example, does it become a moving vertical wall as if on castors or does it topple over and crush someone?
- A large diameter steel pipe can often provide an excellent impromptu enclosure for testing long items of substantially smaller diameter. However, consideration must be given to what might come out at the ends of the pipe and the damage which it can do.
- Strong mats (whether of the household variety or specially made ballistic mats) can provide protection against small missiles of not too high a speed but very little quantitative information is available as to their effectiveness.

5.4 Pressurising equipment

Pressurising equipment will normally include a pump or compressor or other source of pressurised fluid or gas, valves to control the flow and gauges to monitor the pressure both within the pressurising system and within the item being tested. Connection between these items may be with metal piping or flexible hose. There is no single configuration of pressurising equipment which will apply to all cases, but the following is a list a considerations which represent good practice.

- Use only components which are rated for the full source pressure and conduct a pressure test on the system before using it to test other equipment.
- Ensure that adequate overpressure devices are available, both on the pumps and on the delivery system and that they have adequate relief flow lines to a safe venting area. This is vitally important when pressure testing is done at several levels of pressure with the possibility of pipes and gauges experiencing pressures in excess of their working pressure.
- Use control valves or needle valves in the flow line to the item being tested so that rate of

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pressure increase during the test can be kept low. Items being tested should never be subjected to the sudden pressure shocks which can result from the use of ball and similar wide opening valves.

• Use metal piping as far as possible. This piping must be adequately clamped, to walls, floors and so on, so that, in the event of pipe–failure, say if the pipe breaks away from a fitting, there is no possibility of 'pipe–whip'. Pipe–whip can cause very severe injuries and must be avoided at all cost. People are often unaware of how large are the forces exerted on the end of a pipe when it is emitting a jet of fluid at high pressure.

• Use flexible hoses as little as possible and, where they are used, make sure they are not damaged, for example by people walking over them. Inspect the hoses at frequent intervals for signs of damage and discard if any are found. Pipe–whip with hoses is much more severe than for metal pipes, so limit the dangers by keeping the hoses as short as possible.

• Avoid the use of several different manufacturers' high pressure components, particularly where the components are not compatible but are sufficiently alike to cause confusion.

• Where testing at a range of pressures is conducted, consider marking every single component in the pressurising system with a code indicating the maximum working pressure. This includes every single fitting and every individual piece of pipe or hose. At the very least, do this for all free or interchangeable items which are used to make the final connections to the item being tested. If there is a limitation as to the type of pressurising fluid which can be used on a component, this usually applies only to hoses, ensure that the component is so marked.

• Where the range of items and the test pressures is extensive, it is usual to carry a stock of components such as blind flanges and plugs with which to close ports and openings not involved in the pressure test. The connection of high pressure fluid may also be through a flange or plug with small bore connector. Care must be taken that only items of adequate pressure rating are used and that they are not mis–used. After each pressure test, the temporary fitting must be examined visually for damage and if found, the fitting must be discarded. In some circumstances, it will be desirable to set up an inspection room with the necessary tools to check dimensions to a higher degree of accuracy and to maintain a log of usage and inspection of each fitting. The need for this should be thrown up by the hazard

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assessment.

• Ensure that the pressurising system has an adequate number of pressure gauges. A counsel of perfection is to have one gauge in every section that can be isolated, so that one can have a complete picture of the state of the system at all times.

• The item being tested must have its own pressure gauge so that one can check that the pressurised fluid has indeed reached the item during the test and also so that one can be sure that the item has indeed been depressurised when one comes to the end of the test. This requirement can be quite exacting when the test pressure is high since a gauge of adequate range to cover the test pressure will probably have low sensitivity as the pressure reaches atmospheric. If the item being tested has a valve of any description between sections, and this includes a non–return or check valve, a gauge should be attached to each section whether the valve is closed or open. The presence of a gauge in each section may be necessary in order to conduct the pressure test properly, as for example when one of the objectives of the test is to detect leaks across a closed valve, but the presence of a gauge in each section provides an important safety function in that one can check whether each section of the item is depressurised before it is dismantled after the test. Situations in which this multiple gauge requirement is relaxed must be positively identified as such during the hazard assessment and not be merely a default procedure.

• Provide adequate clamping to secure the item being tested. An excellent way of doing this is to bolt it directly onto a fixed test–bed using the same number and type of bolts as will be used to connect the item to the rest of its pressure system when it is in use. Sometimes, in an attempt to speed up routine testing operations, a special quick–release clamping arrangement is used instead of bolts. Where this is done, a hazard assessment must be done to ensure its adequacy. Where free pipework is used to connect the source of pressurised fluid to the item being tested, the item itself should be firmly clamped to the bench, test–bed or floor. The general objective in all of these means of clamping is to restrain all major fragments in the event of failure. It must be remembered that the force involved can be very large, for example, failure of the pipe to flange weld on a 6 inch pipe when tested at 100 bar will result in a thrust of order 20 te. The hazard assessment of the clamping arrangement must also take into account the possibility of the clamps themselves becoming missiles, for

example if a G–clamp of inadequate strength is used it can open up under the stresses imposed upon it and fly off at considerable speed.

• Provide adequate instrumentation and, where the test is carried out behind a barricade, consider the provision of visual means of following the test; this often gives advance warning when something is going wrong. Armoured windows may be satisfactory for tests on small items but the operator must view the proceedings through a mirror so that in the event of a projectile passing through the window, he is not in line of fire. Better, in general, is the use of closed circuit television for this purpose.

5.5 References

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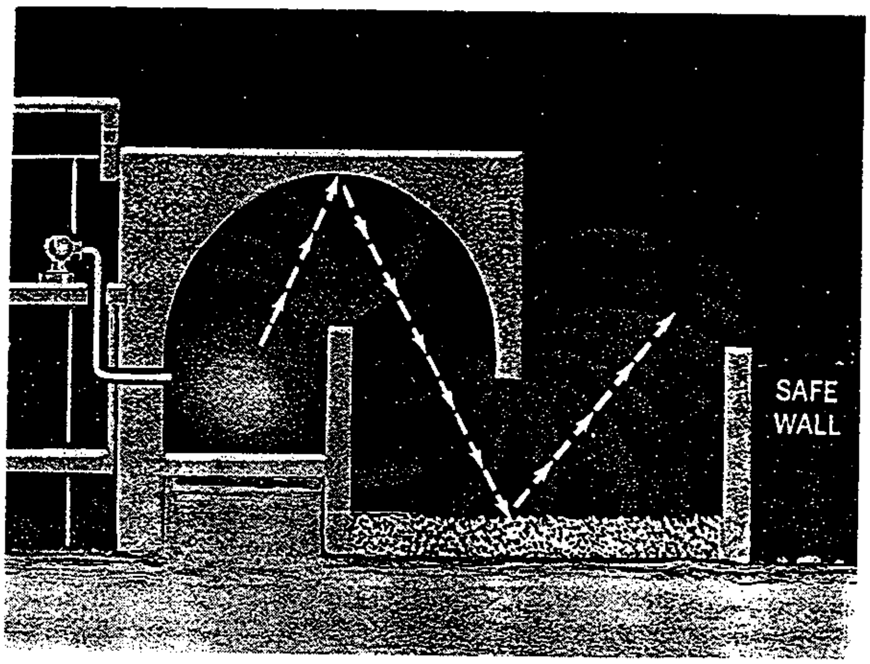

Figure 5.1. Vented cubicle with sand pit.
(After Browne [5.7])

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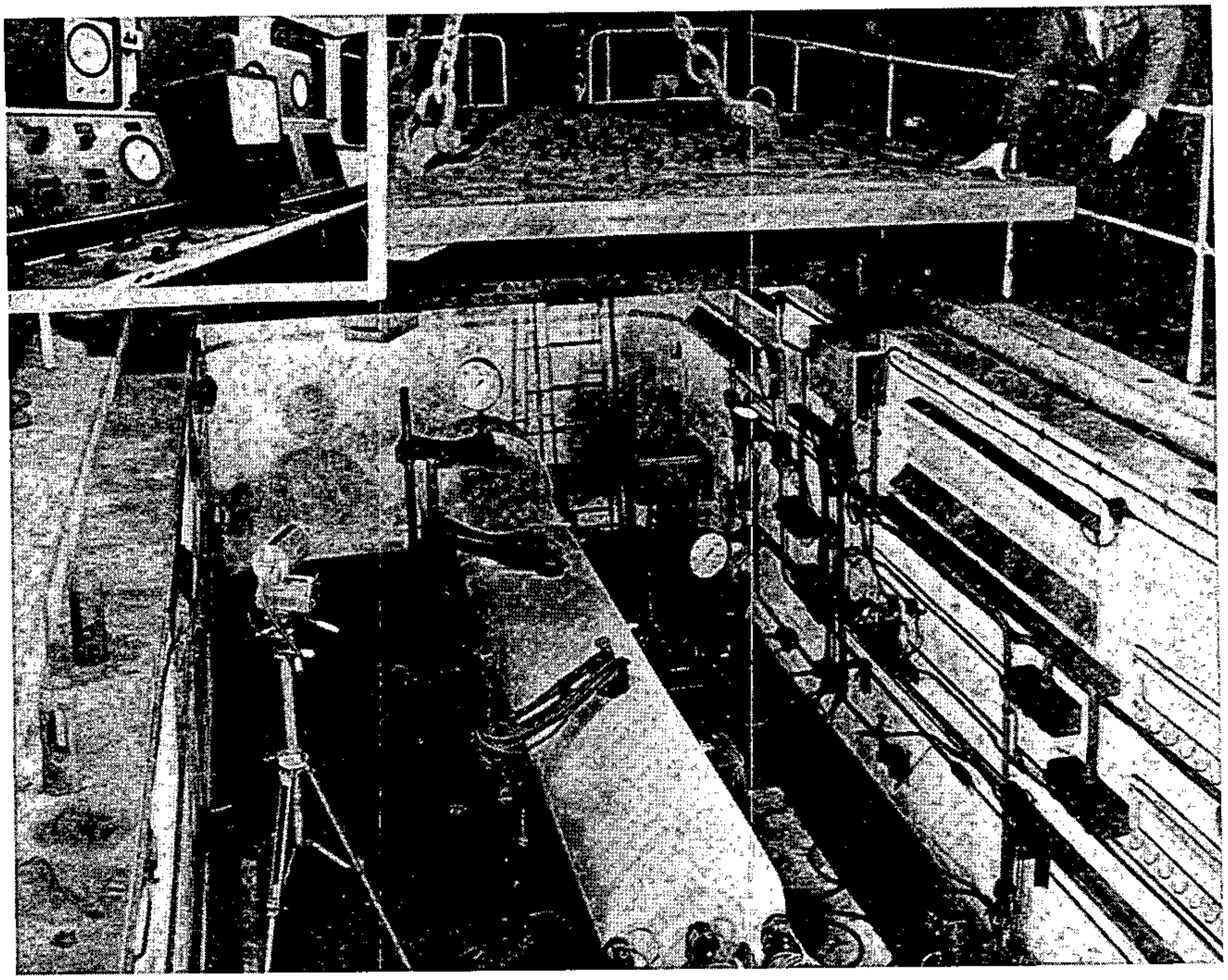

Figure 5.2. Safety pit for testing high pressure vessels.
(After Manning [5.8])

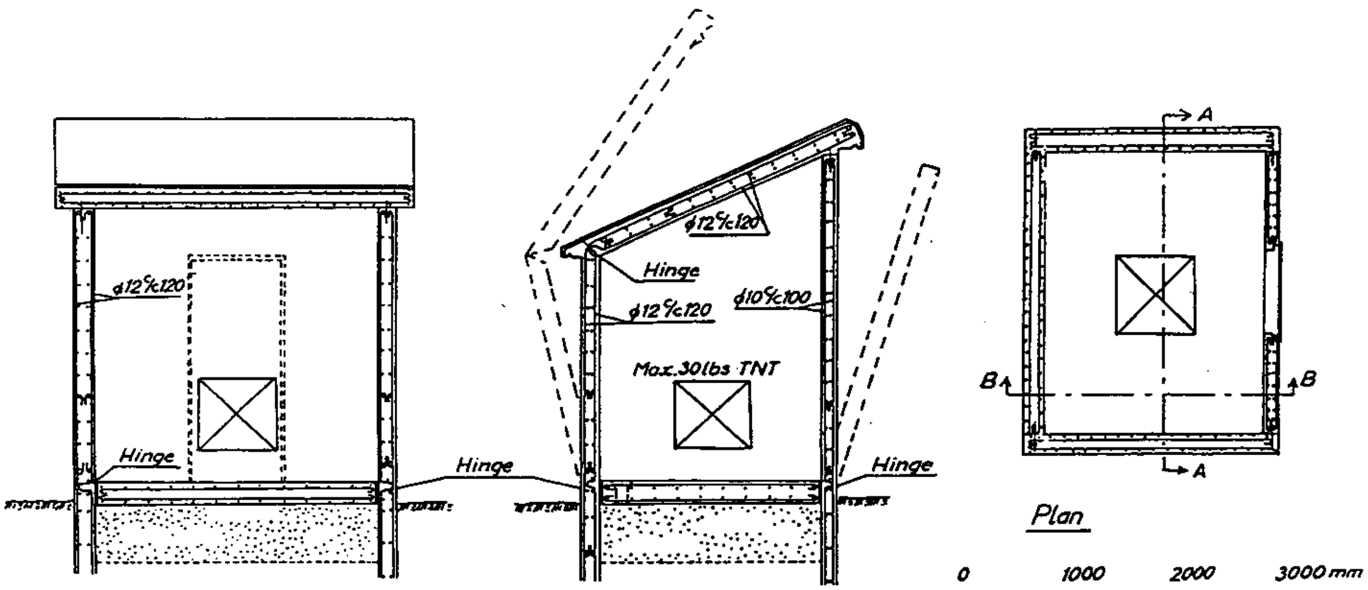

Figure 5.3. Reinforced concrete structure designed for 14 kg of TNT.
(After Bergman [5.1])

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6 TEST OPERATIONS

6.1 Design of test programme

The first step in a pressure test is the production of a written test programme for approval by management. This must be completed and approved before any physical testing takes place.

The written test programme may apply to an individual vessel or component or to a group of similar vessels or components. By similar we mean that the vessels will have been manufactured to the same drawings, be of the same nominal dimensions and have been manufactured in the same way. For example a manufacturer should only need to design a test for a given product.

The Pressure Systems Regulations require that pressure vessels and systems should be designed in such a way that they can be inspected. The responsibility here lies with the vessel/system designer. It is also the responsibility of this equipment designer to define the test objectives, the method of test and to communicate these to the test designer. It may happen on occasions that equipment designed in a particular way will prove very difficult to test safely. Problems of this kind are best resolved through dialogue between the equipment designer and the test designer before the equipment is manufactured.

6.2 Test requirements

In Section 1, the various types of pressure test were defined. These are: research, proof, leak and function. However, it is frequently necessary to make a variety of ancillary measurements during a test. Most common are measurements of strain while the vessel is under pressure. These are used to verify that the actual strain is commensurate with the state of stress assumed in the design calculations and to identify areas of local plastic yielding.

Some design codes, such as those for gas bottles, require a measurement of volumetric strain by a fluid displacement method. A more general example is where large vessels have been fabricated from rolled and welded steel plate. In this case it is often difficult to maintain close dimensional tolerances and the final vessel may be slightly at variance with the nominal dimensions assumed in the design calculations. In this case it is common practice to examine regions of complex strain by strain gauging and other methods. Such tests are employed to prove that the vessel will not fail by mechanisms such as strain–ratchetting and may involve the vessel being taken through several pressurisation cycles.

6.3 Hazard assessment

The purpose of the hazard assessment during the test design is to determine if the hazards associated with the test can be contained within the test facilities that are available. We assume that a hazard assessment has already been carried out in the design of the test facilities and that the containment structures have been designed to protect against some specified (quantified) hazard.

The main steps in the hazard assessment are:

- estimation of the energy stored during the test;
- estimation of the worst case missile penetration in the structure;
- identification of any specific hazards over and above those of blast and missile formation;
- estimation of the likely modes of failure.

The stored energy, missile penetration and blast should be estimated using the methods outlined in Sections 2 to 4.

The possible modes of failure must be determined by consultation between the tester and the vessel designer. In situations where the pressure tests are conducted by a different company from that which designed and manufactured the vessel, the manufacturer must release sufficient information for this assessment to be made.

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Special attention should be given to the testing of vessels and components which have been refurbished after a period of service. Such vessels may have been:

- subjected to cyclic loading;
- subjected to corrosive environments;
- repaired by welding.

and may, therefore, be in a more hazardous state than would a new vessel.

If the vessel or component has been subject to cyclic loading consideration must be given to its remaining life. To do this, details of the metallurgy and service history are required. If the vessel has been subject to a corrosive environment, NDT inspection should be carried out in order to identify the presence of sub–critical cracks which may have been induced by environmental stress cracking.

If a vessel has been repaired by welding it should be considered to be remanufactured. It is essential that this should have involved:

- a thorough design evaluation in accordance with the appropriate design codes;
- written welding procedures should have been produced and that the welding process should have been qualified according to the appropriate codes of practice;
- similar metallurgical and quality control standards should be in force as were employed in the original manufacture.

6.4 The working fluid

The working fluid needs to be selected on the basis of:

- energy stored by compression;
- toxicity;
- flammability;
- compatibility with the components under test.

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Under most circumstances liquids are to be preferred to gases due to their lower compressibility and thus lower stored energy. Water is preferred whenever possible. If flammable fluids such as petroleum fractions must be used consideration should be given to the possibility of fire and explosion resulting from fine sprays of liquid escaping from leaking fittings or hairline fractures. Mercury is sometimes used with laboratory equipment and has the lowest compressibility of all the commonly available liquids. It is highly toxic and is incompatible with aluminium or copper containing alloys such as bronze, beryllium copper, brass, monel and duralumin. With all of these alloys rapid failure may result from liquid metal embrittlement. Soluble oil is some times used instead of water for high pressure tests since pure water can damage pumps due to its poor lubricity.

For most tests potable water is satisfactory. The use of sea–water and ground–water is less satisfactory due to the risk of chloride stress–corrosion cracking. Vessels made of materials particularly susceptible to stress–corrosion cracking should be tested using de–ionised water.

For some equipment, such as instrumentation, liquids may have a deleterious effect on the item under test and a gas has to be used. Similarly, tests performed at pressures above about 5 kilobar are often performed using gases since most liquids will have solidified at this pressure. Nitrogen is generally used since it is non–toxic, non–flammable and relatively cheap. Helium is used on some occasions where very small leaks have to be found.

Wherever possible, the volume of the working fluid should be minimised by filling the vessel with core bars. Care needs to be taken in doing this that the core bars cannot move and block off one or more of the vessel ports trapping high pressure fluid in the vessel.

Always check that item being tested, plus its supporting structure, can withstand the additional weight of the pressurising fluid and any solid filler material used. This is usually only a problem for large vessels operating at relatively low pressures when tested with water.

6.5 Test set–up design

Drawings and parts lists should be prepared by the test designer to show how the vessel under test should be mounted, how it is to be connected to the pressurising system, what special tools, blanking pieces and fixtures are required and where all valves and pressure gauges are to be fitted. At a minimum, there must be a pressure gauge on the item being tested and this gauge must be readable from the operator's position outside the test enclosure/facility. It is good practice to install a pressure gauge in each major isolatable section of the system for monitoring purposes. These details should be thought through in advance with reference to the hazard assessment and should not be left to chance that the fitter will sort it out.

Vessels which are to be tested with water, or any other liquid, should be mounted in such a way that all of the air is displaced from the vessel during filling. Air pockets trapped in dead legs and pockets can add appreciably to the energy stored in the system during the test and may invalidate the assumptions made in the hazard assessment.

6.6 Preparation for the test

6.6.1 Inspection of vessels and components

The written test programme should have identified the nature and sequence of inspections required prior to the pressure test. However, we list here a number of simple checks which should be carried out to filter out obviously 'rogue' items where something seriously has gone wrong with manufacture.

When a vessel arrives for test the test supervisor should check that the vessel presented corresponds to that shown on the drawings and documentation. Where certificates of conformity are available for the materials used these should be checked. If the item contains welds, they should be checked for obvious defects.

Threads should be checked to be certain that they run freely without binding and the clearance of bolt holes in flanges should be checked to make sure that the bolts do not rub on the side of holes.

Dimensional checks are nearly always called for. The objective is to check that the vessel has been manufactured within the tolerances shown on the drawings and as a means of detecting plastic deformation by making measurements before and after testing. It is good practice to mark–up the drawing with the actual values of key dimensions so that they can be compared after the test. Special attention should be given to mating components, seals and threads.

6.6.2 Assembly

The pressure vessel should be assembled in accordance with the drawings. Flanges, manholes and other openings should be blanked off using the correct blanking pieces called for in the test–programme parts list. Care should be taken to ensure that seals and other small items are assembled in the correct order and correct orientation.

All screw threads should be treated with an anti–scuffing thread lubricant unless this is specifically prohibited on the drawings. All bolts should be tightened in sequence to the correct torque as detailed on the drawings and test programme. Note should be taken of the fact that the torque required for the test will probably be higher than that required for normal operations. Care must be exercised to identify the correct torque if both are specified on the drawings. Only new bolts should be used and they should be inspected for obvious damage prior to fitting. The bolts should be in accordance with the specification given on the parts list. In general they will be high tensile bolts with a rolled thread. They should not be substituted with common mild steel bolts drawn from general stock.

Bolts should normally be tightened using a torque wrench or a hydraulic bolt tensioner. Before tightening the bolts the test certificates for the torque wrenches should be checked to ensure that they have been calibrated recently. The use of flogging spanners for tightening

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critical bolts should be discouraged. Small diameter pipework should be tightened carefully to avoid damaging the fittings through overtightening. Particular care should be paid to reducers where the larger end of the fitting will require a higher tightening torque than the smaller.

Many pressure vessel components are large and heavy but need very delicate handling. Appropriate lifting equipment and safe systems of work should be employed to prevent damage to the components and injury to the technicians. Good workshop practice should be followed at all times.

Bleed valves should be fitted to the vessel as shown in the written test programme. When assembling valves and pipework to connect up the system great care should be taken to ensure that the correct components are used and that they have a pressure rating that is appropriate for the test. If possible, use only fittings which have their pressure rating stamped on them. Particular attention should be given to 'bite' type pressure fittings (such as Swagelock, Gyrolock, Parker and so on) to make certain that the ferrules are made by the same manufacturer as the fitting body and that they are assembled according to the manufacturers instructions. Ferrules supplied by different manufacturers are superficially very similar but may fail in service if parts from different manufacturers are mixed.

The piping should be sized so that the vessel and components can be connected with the minimum number of fittings. If reducers have to be used a reducer should be chosen which will perform the reduction in a single step. The use of strings of reducers in series should be discouraged. The use of taper fittings is also to be discouraged.

Wherever possible pipes should be clamped down to prevent whipping if a joint should fail. The use of hoses should be kept to an absolute minimum and they should be mounted so that there are no sharp bends or twists in the hose.

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6.6.3 Conformity with test programme

After assembly the system should be checked by a supervisor or engineer to confirm that it is correct and in accordance with the written test programme. If necessary the test programme may require certain key parts of the assembly to be witnessed and checked by a supervisor during assembly. The supervisor should sign the test certificate to record that everything is in order. This step should not be overlooked. It is very easy for even the most skilled and experienced fitters and engineers to make elementary mistakes if they are distracted. Two pairs of eyes are better than one.

6.7 Conduct of test

Detailed step–by–step instructions for conducting the test should be available in the written test procedure. The person conducting the test should read through these prior to the start of the test to make sure that they are understood. Any questions or ambiguities should be referred back to the test designer for clarification before starting the test.

It is common practice for the test to be witnessed by the representative of a client or an insurance company. It is the responsibility of the test supervisor to ensure that such persons are:

- equipped with suitable personal protection (hard hats, ear plugs, overalls and so on);
- standing in a place of safety thoughout the test;
- able to witness the measurements (pressures, times, strains and so on) that are specified in the test.

It is important to note that such people are there to witness what takes place during the test, not to take part in its conduct. They are passive observers. Test operators and supervisors must not deviate from the written test programme at the request of the witness unless this is approved in writing by the test designer. Any approved modifications to the test must be appended to the test documentation.

For complex tests of large vessels, it is normal for technicians from a number of companies to be present to perform specialist measurements such as strain–gauging. The same general rules apply as for the witness. There should be a test supervisor present in such cases who has the authority to ensure that all the regulations are complied with.

Most test programmes will call for the pressure to be increased in steps of about 10% of the safe working pressure. After the first pressurisation step, to 10% of the safe working pressure, it is permissible to enter the test area to make a visual inspection for major leaks, provided that the possibility of brittle failure can be discounted and that the test being carried out is not a research test. If a leak is found, the item must be depressurised before any attempt at correcting the leak is made. This is to reduce the possibility of over–stressed components when the final test pressure is reached. Under no circumstances should bolts on flanges or closures be tightened while the vessel is under pressure.

Once all major leaks have been cured at this pressure, the test area must be fully secured: all protective doors and hatches closed and interlocked as necessary. No admission is acceptable during the next stages of the test.

Pressurisation can now continue, and after each subsequent pressurisation step, the vessel should be left to stand for a while to check that it is not leaking. This can now only be inferred from instrument readings, usually of the vessel pressure. It is often difficult to tell if leakage is present, particularly for small leaks. On first pressurisation some volume of the working fluid will be required to push seals into their proper places. This often takes place quite slowly and can be misinterpreted as leakage. In addition, the act of compressing a gas or a liquid produces a rise in the temperature in the fluid. When the vessel is left to stand between pressure increments, the fluid cools to the temperature of the vessel and the pressure will fall and this too is often misinterpreted as indicating leakage. If leaks are suspected, the best policy is to depressure the vessel and then inspect for leaks. A piece of paper tissue wiped around a joint is very effective in detecting small leaks when pressure testing with liquids. When the cause of the leak has been identified and rectified the vessel

should be tested again under the same conditions under which it failed. Once the test stage has been passed successfully, the next stage can be undertaken.

The exact nature of the pressurisation cycle required for the test should have been specified by the test designers, but it will normally involve of order three cycles of pressurisation and depressurisation to ambient pressure with dwell times of from several minutes at the maximum pressure in the case of a simple proof test, to hours or even days in a difficult leak test.

If close visual inspection under pressure is required, it may only be carried out after the pressurised cycles in the previous paragraph have been carried out. Depressurisation following the last pressurisation may be halted at the safe working pressure (this partial depressurisation must represent at least 10% of the safe working pressure) and the enclosure entered for visual inspection of the item being tested.

One serious hazard to which the operator is exposed when he enters the enclosure is the presence of fine fluid jets emerging from cracks in the item being tested. These may be too small to pick up as loss of pressure on the pressure gauge, too small to see, but nevertheless they can puncture the skin and inject pressurising fluid into the bloodstream. Options are as follows.

- Keep the body well away from the pressurised system and use implements to test for leaks, such as a brush with soap and water for a gas system, a paper tissue for a liquid system.
- Under no circumstances wipe the hand over the vessel or pipework to test for leakage.
- Wear suitable protective clothing.
- Eye protection is indicated whatever method is used.

6.8 Depressurisation

Failure to depressurise the system before starting to dismantle the vessel has resulted in numerous fatalities and must be treated as a hazardous operation. The main problem is in

ensuring that all parts of the system have been depressurised and that no pockets of trapped pressurised fluid are present in the system. It is for this reason that it is advocated that pressure gauges are installed in all major isolatable sections of the system.

Depressurisation should be carefully thought through at the test design stage and bleed valves should have been included to ensure that all parts of the system can be depressurised. Sometimes it is necessary to modify the vessel design to ensure that this can be done. An example of this is in the removal of pistons with multiple seals from cylinders. Some internal parts are best omitted from vessels if they are likely to move and block ports. The same applies to core plugs added to reduce system volume. These should be designed so that they cannot block holes.

The possibility that the working fluid may solidify during a pressure test should be considered. Some pure substances such as water will solidify at sufficiently high pressure at room temperature whereas the permanent gases such as nitrogen and helium will not. When the pressure is released, any such solids formed quickly melt, presenting no problem. Complex fluids such as oils, on the other hand, may partially solidify by precipitation of wax. Once the pressure is released the wax does not readily dissolve in the oil and may move into small passages such as tubing and valves, causing blockages. Problems of this kind are more likely to be encountered in function testing, where oil may be used, rather than in proof testing, where water is more likely to be the testing medium.

When the pressure is released from a compressed gas, the gas will cool quite significantly, both upstream and downstream of the release valve. If there is any water present in the system, it may be possible to form gas hydrates. These are solid materials which are only stable under pressure but may remain solid up to temperatures of 50 C. If they form, they can produce blockages which trap compressed gas within the system.

Dealing with hydrate blockages can be very difficult and dangerous. The best policy is usually to equilibrate the pressure on both sides of the blockage and then allow it to warm

up slowly. It may be necessary to heat the outside of the vessel in order to do this. The most dangerous time is the point at which the hydrate slug just starts to melt. Since heat transfer will be through the walls of the vessel, the first place that melting will take place is next to the vessel wall. The result is that the hydrate slug will lose adhesion with the vessel wall and, if there is a pressure differential across the slug, it may be accelerated as a projectile which will impact on the inside of the first bend that it meets. Vessels and pipework have been fractured in this way.

The low temperatures formed in the gas phase will cool the metal parts of the system, possibly below their ductile–brittle transition temperatures if these are not far below ambient. Generally, the lowest metal temperatures will be obtained within and downstream of the release valve, so one must be absolutely sure that any metal here can withstand the necessary temperature duty. A brittle failure at this point may cause serious injury, particularly if the valve is outside the protected area. Austenitic stainless steel does not go brittle at any sub–ambient temperature and is the preferred material for release valves.

Seriously low metal temperatures in vessels upstream of the release valve are less frequently observed due to the large thermal mass of the vessel, except when there is some condensation of the pressurising gas due to the generation of low temperatures, or the metal transition temperature is relatively close to ambient. The same is not, however, true of the pipework leading from the vessel to the release valve since the thermal mass of the pipe is small and heat transfer to the cold gas very good. By way of example, tests conducted on a 100 litre vessel containing nitrogen at 100 bar produced upstream temperatures some 80 K below ambient during depressurisation.

6.9 Dismantling

Dismantling should be carried out as carefully as assembly. When removing bolts from flanges, any unusual tightness should be noted since this may indicate that plastic deformation of the threads has taken place. It is a good idea to keep all components, such as

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bolts, in order so that they can be referred to the appropriate holes for future inspection. Threads are often examined with go–not go gauges to check that they are still within tolerance. As a matter of routine, the bolts used during the pressure test should be scrapped when all inspections have been completed. They should never be retained for future use.

Seals should be examined for evidence of extrusion of the elastomer, cuts and abrasions and any other damage. The orientation of parts should be marked so that any damage to the seal can be associated with its point on the housing. Always use soft tools, plastic or brass, for removing seals so as not to damage the housing. On new vessels it is very common to find seals failing due to swarf or sharp edges in the housing. This is particularly the case for housings on the bore of small holes which are difficult to fabricate and into which it is difficult to insert the seal.

Any unusual observations should be reported on the test certificate. The test designer and vessel designer should consider all of the data collected during the test before the vessel is passed.

7 PERSONNEL TRAINING

7.1 Introduction

The skills required for pressure testing can be divided into two:

- those required of the person(s) devising the test;
- those of the person conducting the test.

The first calls for skills comparable with those of a professional engineer and the second those of a skilled technician. We do not, however, believe that a particular formal specification should be specified for either of these jobs, but that one should identify the necessary skills each person should have and that it be a requirement that the people concerned should have them. We would note in passing that, as far as formal professional engineering qualifications are concerned, we are aware of no engineering course which, of it own, would provide all that is required for the person devising the pressure test: additional knowledge and experience, appropriate to the range of items being tested, will always be required.

The role of the professional person is to liaise with vessel designer, if appropriate, assess the items to be pressure tested as regards their suitability for test and their modes of failure, determine the level of protection needed to conduct the test safely and to decide how the test should be carried out. In taking this last decision, he may well have to take into account the skill and expertise of the person actually conducting the test; a test conducted by a relatively unskilled technician may well need to be conducted in a different way from that performed by someone with substantially greater skill. The role of the professional person is, therefore, primarily one of conducting the hazard assessment referred to in earlier sections of this report. In some circumstances, it may also be his responsibility to ensure that all recommendations of this assessment are carried out but, in others, this may be a separate management function.

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7.2 Designing the test

In conducting a hazard assessment, the professional person will need to consider:

- the metallurgy of the item being tested;
- the modes of failure which an item with this metallurgy might be liable to;
- the energy release and speed of missiles which can result from such a failure;
- the adequacy, or otherwise, of existing protective structures to resist attack by these effects, or alternatively, be able to specify what is needed to provide such protection;
- the conduct of the actual test on each different item so that the operator can conduct the test without significant risk to himself or others: this will usually involve drawing up a list of operating instructions.

It is not suggested that the person conducting this hazard assessment should be intimately familiar with every aspect of the items on this list. For example, he need not be a metallurgist able to conduct his own metallurgical examination, but he must be able to understand the implications of the results provided by the metallurgist. Thus, if the metallurgist states that the steel used has a Charpy impact value of, say 30 J, he must be able to assess the likelihood of a brittle failure (27 J is usually taken as the boundary between brittle and ductile behaviour, so 30 J, being barely above this boundary, indicates that brittle failure cannot be ruled out). The conduct of this hazard assessment should be within the capability of any professional person with an appropriate science or engineering background provided that he has broadened his knowledge to include the issues listed above.

7.3 Conducting the test

It is anticipated that the person actually conducting a pressure test will normally be someone with technician rather than professional training. He should not, therefore, be asked conduct a hazard assessment, whether of the whole test system or of the individual test he is about to carry out. This must be done by the professional person specified above. However, the technician must not be just a pair of hands following instructions. He needs to be able to

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interpret observations made during the test, particularly when these indicate that there is some departure from the expected behaviour, since these departures may be giving warning of a more serious failure. It must be remembered that failure can happen quite quickly with only a short period of warning, so the operator must be able to determine his own actions without recourse to higher authority in these cases.

This technician training can, of course, be conducted largely on the job, but we can see considerable advantage in some more formal training, particularly as regards an understanding of what is going on during a pressure test, how things can go wrong and what to look out for. It could also, in part at least, better prepare him to deal with the situation in which he is asked to test an item which is not safe to be tested, whether through an error or omission by his superiors.

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CRR 168

£44.00 net

ISBN 0-7176-1542-1

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