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The Response of a Structural Target to an Explosive Charge Incorporating Foreign Objects: A Numerical Study

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Abstract

This dissertation reports on the results of a numerical investigation into the effect of incorporating foreign objects into explosive and its subsequent influence on the response of a target structure. The explosive, the container and the ball bearings were simplified representation of the key components of an improvised explosive device (IED). The numerical study was aimed at studying the ball bearing interaction with blast when incorporated into charge, and was based on previous experiments. In the experiments, 22g of plastic explosive charge (26mm in diameter with a length-to-diameter ratio of 1) was detonated inside a fully confined cylindrical mild steel container of 9.3mm wall thickness and 273mm outer diameter. Different experiments were carried out using charges with varying numbers of ball bearings arranged in different configurations. The ball bearings were either packed around the cylindrical charge in row(s), or were randomly embedded into the charge.

In the numerical simulations, i) a quarter symmetry model in the radial plane and ii) a half symmetry model in the axial plane were developed in ANSYS AUTODYN using Euler and Lagrangian meshes, based on the previous experiments. The cylindrical target and the ball bearings were modelled using Lagrangian elements, while the air and the PE4 plastic explosive were modelled using Eulerian elements. Ball bearings of fixed diameter 5mm, were placed at positions relative to the charge corresponding to the experimental conditions. The predicted crater depth created in the cylindrical target by ball bearing impact were compared to the experimental results. A comparative numerical study was then conducted to investigate how different factors influenced the ball bearing behaviour and the target response. The parameters tested included the total number and size of ball bearings incorporated in the explosive charge, the manner in which the ball bearings were distributed inside or outside the charge, and the length-to-diameter ratio of explosive used.

The numerical models provided insights into how the ball bearing interacted with the blast when incorporated into charge. 2D numerical simulation techniques were used to simulate the velocity distribution of a cased cylindrical explosive charge. The results of the numerical simulations were verified against previously reported equations for fragments and pre-formed fragments, which are based on experimental data which indicated a non-uniform velocity distribution along the cylinder axis. Overall, there was a good agreement between the 2D model and the experimental measurements, including the distribution of the lower velocity values near the cylinder edges. The ball bearing velocity - crater depth correlation was also compared to the

projectile velocity equations from literature. A good correlation was shown in all radial simulations. In the axial plane simulations, a good correlation was observed only when the projection angle of the ball bearing was nearly perpendicular to the charge surface. The effect of the ball bearing presence on the overall pressure observed in the confined space is also studied. The inclusion of ball bearings in the charge resulted in an overall decrease in peak pressure, and the percentage decrease was proportional to the total number of ball bearings. Charge covered in rows of ball bearings acted similar to encased charges, especially to charges with pre-fragmented casings. It was observed that an increase in length-to-diameter ratio of the charge led to an overall increase in blast magnitude.

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Glossary

ALE	Arbitrary Lagrangian Eulerian
ANFO	Ammonium Nitrate/Fuel Oil
AOAV	Action On Armed Violence
BISRU	Blast Impact and Survivability Research Unit
C4	Composition C4 Plastic Explosive
CJ	Chapman-Jouget
CNC	Cardboard-No-Cover
EBC	Equivalent Bare Charge
EGDN	Ethylene Glycol Denigrate
EOS	Equation of State
IED	Improvised Explosive Device
JWL	Jones-Wilkins-Lee
NASA	National Aeronautics and Space Administration
PE4	Plastic Explosive No. 4
SoD	Stand-off Distance
TATP	Triacetone Triperoxide
TNT	Trinitrotoluene Explosive
ZND	Zel'dovich-von Neumann-Döring

Nomenclature

α	Incidence angle of the blast detonation wave
$\bar{\eta}$	Effective pressure ratio
β	Oblique angle of projectile
η	Equilibrium pressure ratio to static burst pressure
γ	Adiabatic index of detonation product-fuel mix
γ_{infty}	Adiabatic index of pure air
ρ	Density
σ	Standard deviation from mean
σ_y	Yield stress
θ	Fragment projection angle, measured from the normal of surface
2E	Empirical Gurney velocity constant
BH	Brinell Hardness
C	Charge mass
C_s	Speed of sound
d	Charge inner diameter
D	Diameter of cylinder
E	Elastic modulus
h	Centre-to-centre distance (radial plane)
h_t	Wall thickness

$\mathbf{I}_{eff}$	Effective selective impulse
$\mathbf{K}_1$	Fish-Summer target constant
$\mathbf{K}_\infty$	Semi-infinite equation constant
$\mathbf{L}$	Length of cylinder
$\mathbf{L/D}$	Length-to-diameter ratio
$\mathbf{L/R}$	Length-to-radius ratio
$\mathbf{m}$	Projectile mass
$\mathbf{M}$	Explosive casing mass
$\mathbf{p}$	Crater depth
$\mathbf{P}_\infty$	Depth of penetration
$\mathbf{P}_\theta$	Atmospheric/ ambient Pressure
$\mathbf{P}_s$	Peak overpressure
$\mathbf{Q}$	Detonation energy released during explosion
$\mathbf{R}$	Radius of cylinder
$\mathbf{R}_b$	Distance from the centre of blast source
$\mathbf{t}$	Time
$\mathbf{T}$	Temperature
$\mathbf{t}_a$	Time of arrival
$\mathbf{t}_b$	Target thickness for ballistic limit
$\mathbf{T}_d$	Time delay between impulsive load and equilibrium pressure
$\mathbf{t}_{max}$	Time taken to reach peak deformation
$\mathbf{t}_p$	Time of start of negative phase
$\mathbf{t}_{sp}$	Target thickness for spallation limit
$\mathbf{t}_s$	Duration of peak overpressure
$\mathbf{t}_t$	Target thickness
$\mathbf{U}$	Detonation velocity
$\mathbf{v}$	Ballistic limit

$\mathbf{V}$	Volume of structure
$\mathbf{v}_E$	Charge volume
$\mathbf{V}_{Gurney}$	Gurney velocity
$\mathbf{V}_n$	Impact velocity normal to surface
$\mathbf{V}_o$	Fragment velocity
$\mathbf{W}_{TNT}$	Equivalent TNT mass of explosive
$\mathbf{x}$	Distance from detonation point along the cylinder axis
$\mathbf{y}$	Centre-to-centre distance (axial plane)
$\mathbf{Z}$	Scaled distance

Chapter 1

Introduction

1.1 Background and Motivation

Improvised Explosive Device (IED), as defined by the United States Department of Homeland Security [1], is a device placed or fabricated in an improvised manner, incorporating destructive, lethal, noxious, pyrotechnic or incendiary chemicals and designed to destroy, incapacitate, harass or distract. It may incorporate military stores, but is normally devised from non-military components [1].

The threat of IEDs has grown exponentially in the 21st century, as the methods and means of warfare have adapted to modern threats such as terrorism [2]. Action On Armed Violence (AOAV) recorded a 70% global rise in the number of civilians killed or maimed by IED attacks [2] between 2011 and 2013. The number of IED attacks occurring in populated areas such as a crowded street, a market or a residential neighbourhood, have increased over the years [3] as illustrated in Figure 1.1.



Figure 1.1: Infographic illustrating the percentage of IED attacks that occurred in populated areas between years 2011-2013 [3]

IEDs are, by definition, improvised. They can come in many forms, shapes and sizes ranging from pipe bombs to truck bombs [1]. Different IEDs contain a variety of projectiles such as nails, glass, bearings or even chemicals designed to increase the damage caused by the explosion. The versatile nature of IEDs makes it very difficult to conduct a generalised study on IED impact; this may explain why there is relatively little data available on the use and impact of IEDs. However, it is still crucial that such data is collected in order to quantify and address the harm caused by IEDs [4].

The available literature on IEDs is heavily biased towards their effects on human victims. However, research on the effect of IED explosion on the surrounding infrastructure is still necessary. Understanding the effect that IED explosions have on structures will allow for design and implementation of structures that can minimise civilian and military casualties and structural damage [5]. While the research into the development of protective measures has been predominantly experimental in the past, more numerical simulations are now being used, with the recent improvement in numerical code. The high cost of full scale experiments is a further motivation behind the use of small scale tests and numerical models.

1.2 Objective and Scope

A numerical study was carried out using ANSYS AUTODYN to investigate the effect of incorporating foreign objects, namely ball bearings, into explosive. The explosive, the container and the ball bearings were simplified representation of the key components of an improvised explosive device (IED). The numerical study was aimed at studying the ball bearing interaction with blast when incorporated into charge. A comparative numerical study was conducted to compare the results from different scenarios, using the same model formulation which is based on previous experiments designed and conducted by Kang [6]. The purpose of the comparative study was to gain insight into the effect of parameters such as the total number of bearings incorporated into the charge, the manner in which the bearings are incorporated into the charge, the size of the bearings, and the length-to-diameter (aspect) ratio of the charge.

The objectives of this study were to:

- Investigate how the destructive effect of IEDs may be quantified, based on different factors such as the total number and size of shrapnel (represented by the ball bearings) incorporated inside the IED, the manner in which the shrapnel is distributed inside the explosive charge, and the length-to-diameter ratio of explosive used.
- Formulate a numerical model based on previous experiments.
- Study the velocity and the trajectory of the bearings, as well as their interaction with the blast waves.
- Perform a parametric study to investigate the effect of varying the quantity and location of bearings in the explosive charge.

- Draw conclusions and make recommendations based on the outcome of the results.

The material models used in the numerical models were based on the experiments, which were restricted to a grade of locally-sourced mild steel (for the cylinders) and stainless steel (for the ball bearings). The geometry of the bearing made material testing impractical using the tools available. It was not necessary to consider the work-hardening effects of the mild steel. Therefore, the material data used for this report is limited to published material characterisation, and the strain-rate sensitivity and influence of thermal effects were not measured experimentally, but rather obtained from published literature.

ANSYS AUTODYN, a commercial explicit analysis tool, was used to simulate the blast tests. The fluid-structure interaction between the explosive, the bearings and target cylinder were modelled for each configuration.

1.3 Outline

A literature review is presented in Chapter 2, which covers the definition and reported usage of Improvised Explosive Devices (IEDs), detonation theories and blast loading, fragmentation of explosive charge casing, and projectiles. The previous experimental setup is discussed briefly in Chapter 3, with the experimental results reported in Chapter 4. The formulation of the numerical model is discussed in detail in Chapter 5. The numerical results are presented in Chapter 6, followed by the results of the parametric studies in Chapter 7. The numerical results are discussed further in Chapter 8. Chapter 9 and 10 present the conclusions and recommendations for future studies.

Chapter 2

Literature Review

The outcomes of the following literature review provide insight into decisions and assumptions made throughout this study.

2.1 Improvised Explosive Devices

2.1.1 Components of an Improvised Explosive Device (IED)

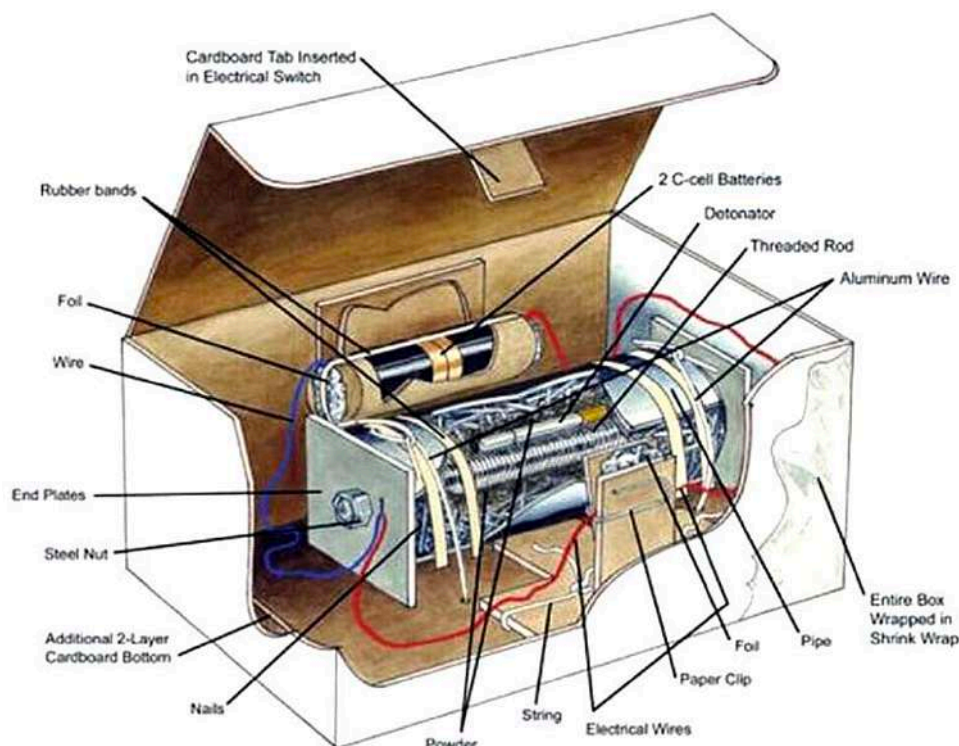


Figure 2.1: Components of a typical IED [7]

Figure 2.1 illustrates the typical key components found inside an IED which include: a trigger or a switch and its power source; a detonator; a main charge; and a container [3, 7]. The main charge is an explosive that provides the primary blast [3]. It can be constructed from conventional munitions such as artillery shells, which provide a ready-made fragmentation effect [7]. Otherwise the main charge is made using commercial explosives such as PE4, trinitrotoluene (TNT), Triacetone Triperoxide (TATP), urea nitrate (fertiliser), and fuel oil (ANFO) [1, 7]. Table 2.1 lists a summary of commonly used explosives in IEDs.

The container of an IED conceals its contents and provides fragmentation effect [7]. It may also be used to direct blast to a certain direction [3, 7]. The casings can come in a variety of sizes, shapes and materials, ranging from cigarette packets to large trucks or even an airplane [7]. Table 2.2 lists the range of common IED container sizes and their respective explosive capacities. Terrorist groups would design the explosive components to maximise the penetrative damage inflicted by the device. For example, shaped charges of different configurations ranging from high angle cones to dishes have been used to achieve specific penetrative effects [7]. These effects can be further developed by the use of a range of different metals which deform in different ways.

Most IEDs contain projectiles such as ball bearings, bolts, nuts, or nails, that act as shrapnel, which are used to enhance the fragmentation and the penetrative damage [3, 7]. Upon explosion,

Table 2.1: Examples of explosives commonly used in IEDs [7]

	COMMON USES	COMMON FORM	KNOWN IED USE
HIGH EXPLOSIVES			
Ammonium nitrate and fuel oil (ANFO)	Mining and blasting	Solid	Oklahoma City bombing
Triacetate Triperoxide (TATP)	No common uses; mixed from other materials	Crystalline solid	2005 bombings in London
Semtex, C-4	Primarily military	Plastic solid	Irish Republican Army bombings
Ethylene glycol denigrate (EGDN)	Component of low-freezing dynamite	Liquid	Millennium Bomber, intended for Los Angeles airport, 1999
Urea nitrate	Fertiliser	Crystalline solid	World Trade Centre 1993
LOW EXPLOSIVES			
Smokeless powder	Ammunition	Solid	Olympic Park bombings

Table 2.2: IED containers and their respective explosive capacity (converted to SI units) [1]

THREAT DESCRIPTION	EXPLOSIVE CAPACITY (kg)	BUILDING EVACUATION DISTANCE (m)	OUTDOOR EVACUATION DISTANCE (m)
Small package/letter	0.45	12.2	274.3
Pipe Bomb	2.3	21.3	365.8
FedEx Package	4.5	27.4	329.2
Vest/Container Bombs	9.1	33.5	518.2
Parcel Package	22.7	45.7	563.9
Compact Car	226.8	97.5	579.1
Full Size Car/Minivan	453.6	121.9	731.5
Van/SUV/Pickup Truck	1 814.4	195.1	1 158.2
Delivery Truck	4535.9	262.1	1 554.5

the shrapnel is propelled at very high velocities, causing significant damage to surrounding infrastructure. More densely and evenly packed projectiles are likely to cause more damage [7].

In addition to shrapnel, IEDs may contain chemical substances intended to maximise the damage. For example, the shaped pipe bomb that was discovered in the Spokane bombing attempt in 2011 contained shrapnel combined with an anticoagulant rat poison, apparently intended to prevent any resulting wounds from clotting [8].

The Boston Marathon bombing was an IED attack targeting a crowded area. Two homemade pressure cooker bombs were detonated near the finish line of the annual Boston Marathon in April 2013. [9]. Remains of a pressure cooker, as shown in Figure 2.2 a), were found at the scene. [10]. The device also contained fragments including nails and ball bearings [11]. Figure 2.2 b) shows the nails and ball bearings found near the site. Upon explosion, the pressure cooker vessel fragmented, effectively acting as shrapnel itself. The blasts of the bomb were powerful enough to blow out windows of adjacent buildings, and the lid of one of the pressure cooker bombs was found on a nearby rooftop.

Another recent example of an IED attack that took place in a crowded area is the Manchester Arena bombing incident from 2017, where an IED packed with nuts and bolts was detonated in the Manchester Arena foyer, killing 23 people; the victims of the bomb included those who

were up to 20 meters away from the detonation point [12]. It was reported that the shrapnel from the bomb dented brick walls and penetrated metal doors [12].

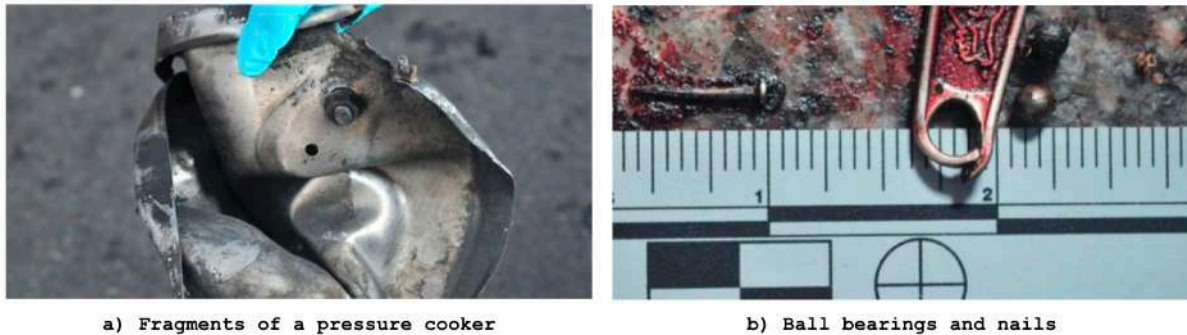


Figure 2.2: Shrapnel found at the scene of Boston Marathon Bombing [12]

2.1.2 How IEDs Affect the Surrounding Area

Like all explosive weapons, IEDs affect an area with blast, heat and fragmentation. If an IED is detonated in or near a structure, the explosion may shatter windows, destroy the building walls, shut down building systems such as power and ventilation, or result in the release of hazardous materials used within the building [1]. When an IED explodes, its casing as well as additional shrapnel included in the bomb are propelled away from the detonation point at high velocity. Such fragments, which originate from the explosive source, are called primary fragments [13]. Primary fragments, combined with the blast waves, may strike surrounding buildings, concrete, masonry or glass, and shatter even further, causing additional damage. This is known as secondary fragmentation [13].

The 1995 Oklahoma City bombing incident illustrates the damage caused by primary and secondary fragmentation. The Alfred P. Murrah building, the intended target of the bombing, was subjected to a pressure wave of up to 62,000 kPa [14]. To contextualise, it is estimated that most buildings cannot withstand blast pressures higher than 34 kPa [14]. The bomb destroyed or damaged 324 other buildings within a 16-block radius, shattering glass in 258 nearby buildings [15]. The shattered glass from the nearby buildings, combined with the projected shards from the bomb, effectively acted as small, sharp bullets. Annealed glass in older buildings had broken into sharp shards, proving even more hazardous when propelled at a high speed. The Australian Safety Solutions organisation reports that 5% of the deaths and 69% of the injuries outside the buildings for a radius of over 10 blocks from the blast centre were due to fragmentation damage [14].

The energy from a bomb explosion also creates a high temperature zone. High temperature may weaken building materials, leave burns on the human body or even cause secondary fires or explosions by reacting with fuel sources or flammable materials nearby [13]. For example in the Oklahoma City bombing, the blast of the bomb caused secondary fires, setting vehicles and buildings ablaze. 86 vehicles in the vicinity were destroyed [14].

2.2 Detonation Physics

2.2.1 Detonation Theories

The detonation of explosives is the result of the complex relationship between chemistry and hydrodynamics. The extremely high pressure and temperature conditions that occur behind the detonation front during the detonation process makes the experimental study of chemical kinetic processes governing explosives very difficult. The prediction of energy delivery and detonation velocity is accomplished by analytical detonation theories [16].

The Chapman–Jouguet (CJ) detonation theory assumes that the detonation products reach thermodynamic equilibrium instantaneously [16]. In CJ detonation, the Rayleigh line (which describes the conservation of momentum) intersects the shock Hugoniot (which describes the conservation of energy). The Rayleigh line gradient is proportional to the detonation velocity, and therefore the CJ state is the slowest propagating state that can intersect the shock Hugoniot such that energy is conserved [16]. Note that the Hugoniot is not a curve but a locus of end points. Figure 2.3 shows a graphical representation of the CJ state on a P (pressure) – ν (specific volume) plane.

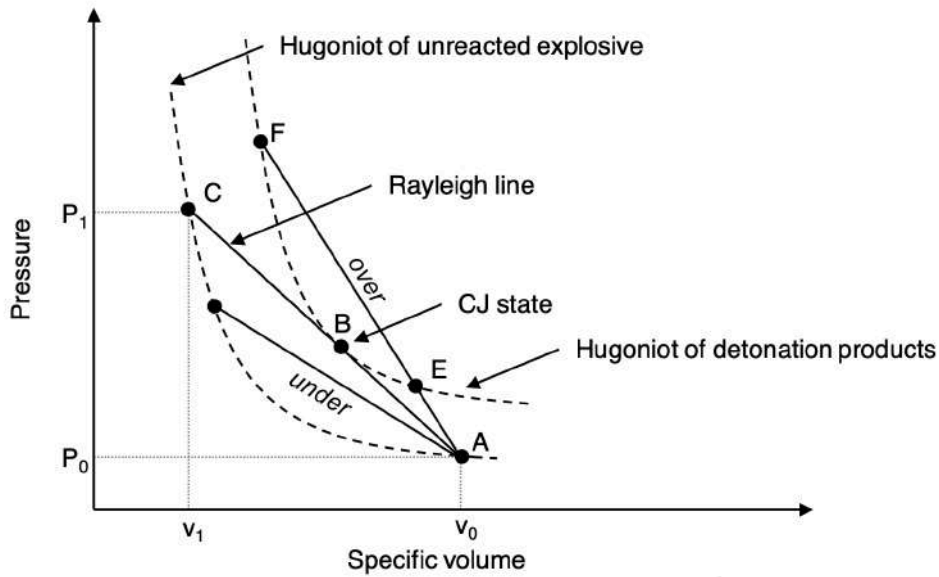


Figure 2.3: Graphical representation of the CJ state on a $P - \nu$ plane [16]

The Zel’dovich–von Neumann–Doering (ZND) detonation model is a one-dimensional model which assumes finite-rate chemical reactions and states that the detonation process follows the following steps [17]. First, the explosive is compressed to a high pressure (Von Neumann spike) by a infinitesimally thin shockwave. The spike is followed by the onset of exothermic chemical reaction zone. Once the CJ state is reached, the detonation product proceed to expand in the reverse direction.

The CJ and ZND detonation theories make a number of simplifying assumptions. It is assumed that the detonation wave is uniaxial; the wave front is discontinuous; the reaction is completed instantly; the reaction zone has zero thickness; the detonation products remain at the same state at all times, and the detonation velocity is constant. Given these assumptions, detonation may be viewed as a shock wave passing through an explosive material at a constant velocity. However, experimental results have shown that detonations are often characterised by unsteady, three-dimensional structures, which cannot be modelled accurately using a one-dimensional model [17]. Furthermore, the CJ and ZND detonation theories yield poor results when used with explosives with a reaction zone with a finite thickness and a relatively slow chemical reaction rate, as the assumption of instantaneous thermodynamic equilibrium is no longer valid [16].

Another theory that successfully solved many limitations of the CJ and ZND theories is the Wood-Kirkwood detonation theory, which essentially models ZND detonation waves in cylindrical geometry where flow is restricted to the central stream tube [18]. The Wood-Kirkwood detonation theory is a two-dimensional steady state kinetic detonation theory that assumes a cylindrical charge of infinite length. The theory considered the interaction of chemical kinetics with the detonation wave, and was obtained by solving the hydrodynamic Euler equations along the central streamline of the cylinder. The source term in the one-dimensional flow along the streamline was assumed to be the radial expansion. The shock front is assumed to have a radius of curvature, and a velocity vector at a point on the shock front has components in the r and z directions [18].

2.2.2 Explosions and Explosives

An explosion is the sudden, rapid and violent release of energy into the surrounding environment [19, 20]. Upon detonation, the localised accumulation of energy is abruptly dissipated into the environment in the forms of light, heat, sound and blast waves [21]. Explosives are materials that convert their potential energy by means of rapid chemical reaction and releases a significant amount of energy in the form of an explosion upon triggering. Explosives may be categorised by the different triggering mechanisms, which range from a simple percussion to the detonation of another explosive.

The type of explosive that is most commonly used in laboratory tests is plastic explosives (namely PE4 plastic explosive), which is very stable and requires the use of a detonator. The detonator makes use of a more sensitive explosive that is detonated by means of an electrical discharge, which then creates a detonation wavefront that triggers a chemical reaction within the explosive. This forms a zone of high pressure and temperature gases along the wavefront as it travels, forming a detonation shock wave in turn. The pressure values are characteristic of the type of explosive and is generally described using the CJ hypothesis.

Blast loading refers to the loading of a structure by a blast wave, a region of highly compressed air created by the high pressure and temperature gases from the explosion expanding and compressing the surrounding air mass, and is characterised by an instantaneous increase in

pressure from the ambient pressure (P_0) to the peak overpressure (P_s) [20]. The pressure history of a blast wave is extremely complex, and is usually simplified for practical purposes. A simplified free air pressure profile of a blast wave is illustrated in Figure 2.4.

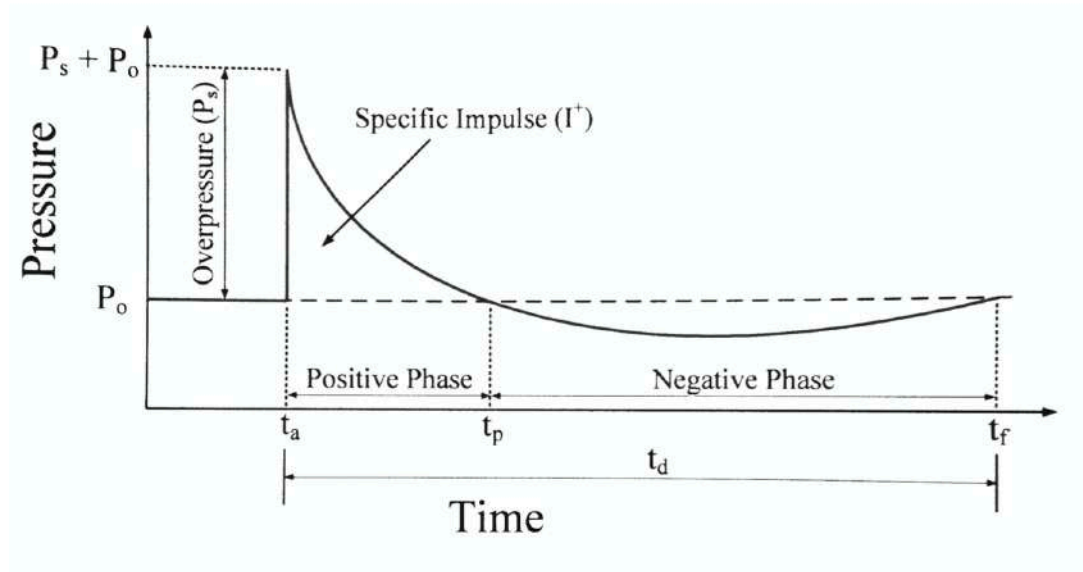


Figure 2.4: Free air pressure profile of a blast wave [21]

When the blast wave arrives (at t_a), the pressure jumps by the overpressure P_s in a almost discontinuous manner, to a peak incident pressure ($P_0 + P_s$) [19]. As the blast wave expands, the pressure decays in a quasi-exponential manner (positive phase) until it drops to a sub-atmospheric pressure at time (t_p) (start of negative phase); this is due to the contraction of the over-expanded gas and reversal of particle flow [21]. For most structures, the under pressure from the negative phase is often assumed to be negligible compared to the overpressure. The pressure eventually returns to atmospheric pressure at time t_f and equilibrium is reached. Another key characteristic of a blast is the impulse, which can be determined by integrating the overpressure with respect to time.

A blast load is categorised as impulsive, dynamic or quasi-static based on the duration of the applied load and the natural frequency of the vibration of the structure affected [22]. If the duration of the loading is substantially shorter than the natural frequency of the structure, the load is considered impulsive [22]. If the load is dynamic, the loading phase will last longer and the structure will have time to respond to the load [20, 22]. If the loading lasts significantly longer than the natural frequency, the load is considered quasi-static [22].

2.2.3 TNT Equivalence of Explosives

The plastic explosive PE4 is known to be near-identical to the more commonly used C4, since they differ only by the type of plasticiser used [23]. PE4 is known to be highly stable and easily formed, and is used by militaries worldwide. However, blast safety or design manuals like UFC

3-340-02 [24] uses the more ubiquitous Trinitrotoluene (TNT) as a benchmark explosive for their blast effect data. TNT has been used for military applications for more than a century, and has become a unit of equivalence for multiple explosive types. As such, TNT blast data may be used for other explosives by relating the explosive energy of a given explosive mass to that of an equivalent mass of TNT [25, 26].

Weckert and Anderson [27] conducted blast tests to compare TNT and PE4 by the degree of plate deformation and peak overpressure achieved, and reported that $1kg$ of PE4 was equivalent to $1.37kg$ of TNT. However, the equivalent mass differed when different criteria of comparison were used. For example, $1kg$ of PE4 was equivalent to $1.09 - 1.21kg$ of TNT to achieve the same plate acceleration. Therefore some authorities such as the US National Counterterrorism Centre [28] distinguish between pressure and impulse equivalency values, which are stated to be 1.3 and 1.5 respectively. Typically, the PE4 to TNT equivalency is assumed to be 1.34 [29].

2.2.4 Blast Scaling

Another important factor in all blast effects is the distance between the point of detonation and the point of interest, referred to as the stand-off distance (SoD). Hopkinson and Cranz [19] developed a method called the Hopkinson scaling law, which states that when different masses of the same explosive are detonated under the same conditions at identical scaled distances, similar blast waves are produced. Therefore, any distance from the centre of blast source (R_b) can be written as a scaled distance using Equation 2.1.

$$Z = \frac{R_b}{\sqrt[3]{W_{TNT}}} \quad (2.1)$$

Z represents the scaled distance, and W_{TNT} represents the equivalent TNT mass of the given explosive. As the mass of the explosive in the equation represents the blast energy, the term can also be replaced by another suitable measure. The scaled distance parameter (Z) correlates a particular explosion to a standard charge of the same material, and other blast-related parameters may be written as a function thereof.

Kinney and Graham [19] presented Equations 2.2 and 2.3 that express the scaled overpressure and its duration as functions of W , Z and the atmospheric pressure P_0 .

$$\frac{P_{max}}{P_0} = \frac{808(1 + (\frac{Z}{4.5})^2)}{\sqrt{1 + (\frac{Z}{0.048})^2} \sqrt{1 + (\frac{Z}{0.32})^2} \sqrt{1 + (\frac{Z}{1.35})^2}} \quad (2.2)$$

$$\frac{t_s}{\sqrt[3]{W_{TNT}}} = \frac{980(1 + (\frac{Z}{0.54})^{10})}{(1 + (\frac{Z}{0.02})^3)(1 + (\frac{Z}{0.74})^6) \sqrt{1 + (\frac{Z}{6.9})^2}} \quad (2.3)$$

2.2.5 Blast Loading

A blast load may be classified as 1) unconfined or 2) confined based on the degree of confinement of the explosion. [24] An unconfined explosion occurs in free air or in air at some distance from the ground or surrounding structures, and the blast wave is generally free to propagate away from the detonation point. Unconfined loading, where the projectiles can scatter freely into the surroundings [21], will result in fewer shrapnels interacting with the target structure.

A confined blast is an explosion that occurs within or in close proximity to a structure that limits the propagation of the blast wave [30]. A confined blast results in more damage than the equivalent mass unconfined blast [16]. The damage of the confined blast depends on various geometrical parameters of the container and the explosive charge, such as their dimensions, shape, the charge location, the venting characteristics and explosive characteristics [31].

A confined blast load is usually comprised of a short-duration shock load and a long- duration gas pressure load [16]. The incident blast wave is reflected off the walls of the structure, causing the initial high-magnitude short-duration shock load. The remainder of the shock loads are caused by the repeated reflection of the reflected waves within the structure. The extent of re-reflection depends on the degree of confinement.

The degree of confinement also determines the long-term secondary load exerted on the walls of the structure. This load is caused by the accumulation of detonation products and gases within the structure [31]. After the short-duration detonation process, a longer-lasting combustion process may occur, called afterburning. Confinement conditions promote afterburning, since the pressure and temperature decay is slower due to the lack of venting and the confinement promotes the mixing of detonation products and the oxygen, which increases the possibility of afterburning [32]. Afterburning raises the temperature within the structure, release more energy into the system and generates a gas pressure which exerts a long-duration load on the walls [31]. When compared to shock loading, gas pressure loading is smaller in magnitude but has a significantly longer duration.

Confined explosions may be further categorised according to the blast load interaction with surrounding structures. Fully-vented explosions occur within or adjacent to a structure with at least one surface that is open to the atmosphere. The initial shockwave is amplified owing to the interaction with the structure. The amplified shockwave and the detonation products are then vented to the atmosphere and propagates away from the structure.

Partially-confined explosions occur within a structure where a limited portion of a surface is open to the atmosphere [16]. The amplified shockwave and the detonation product are vented to the atmosphere, but after a finite period. The presence of high pressure and temperature gas products confined within the structure results in the accumulation of quasi-static pressure, which is also referred to as “gas pressure”. The duration of the quasi-static pressure is longer than that of the shock pressure.

Fully confined explosion refers to explosions that occur within a totally or near-totally confined

structure that has no significant openings. The shockwave is amplified within the structure and is unable to be vented to the atmosphere. The duration of the quasi-static pressures is significantly longer than any of the other confinement configurations. Fully confined explosions are less investigated than open or partially confined explosions, due to their complex internal blast loading condition [21].

2.2.5.1 Fully Confined Explosion

In fully confined explosions, repetitive shock loads are generated as the initial blast wave is repeatedly reflected off the interior walls. Figure 2.5 show typical examples of pressure-time history of an explosion in a fully confined structure, where the repetitive reflection of the waves is clearly evident [33]. For confined explosions, where the angles of wave incidence with the surfaces of the structure vary significantly, “Mach reflections” are typically observed [25]. Mach reflections refers to the “skimming” of the wave off the reflected surface, where the reflected wave catches up to the incident wave and merges to form another wavefront called the “Mach stem”, illustrated in Figure 2.6 [25].

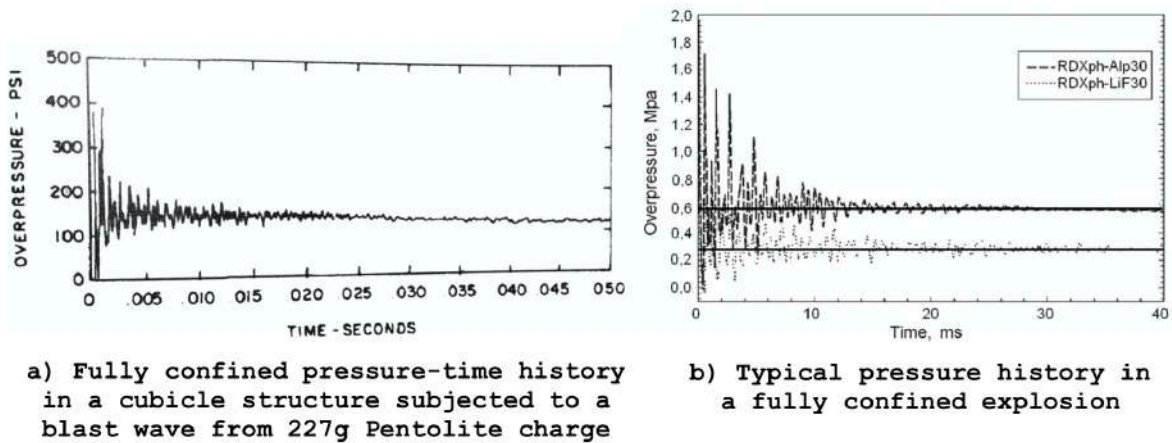


Figure 2.5: Pressure time history graphs under confined loading condition [21]

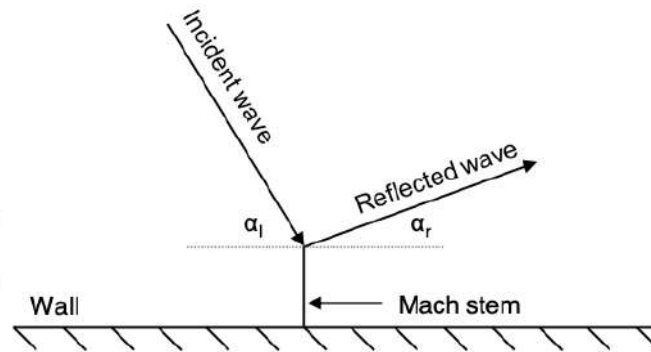


Figure 2.6: Mach reflection of a plane shock wave [25]

No venting occurs in a fully confined structure; therefore the detonation product and the hot gas exert a constant static load on the walls of the structures [31]. Gregory [33] investigated the pressure distribution on the walls of a fully-confined cylinder vessel, illustrated in Figure 2.7. It was reported that the distribution of pressure generally increased towards the corners of the container. The findings were qualitative. No quantitative measure to predict the distribution with respect to space or time was given.

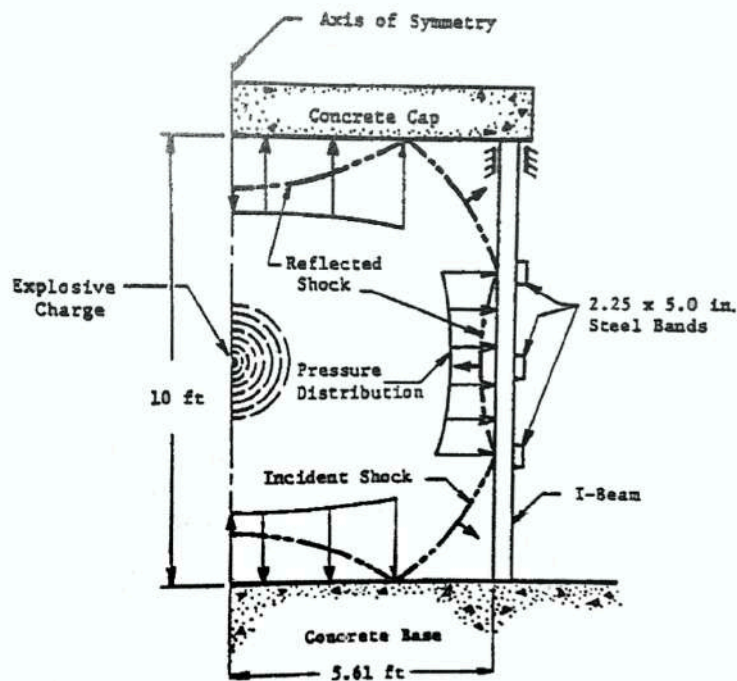


Figure 2.7: Shock wave reflections on the interior walls of a cylindrical confinement vessel [33]

Karpp et al. [34] performed simulation of the detonation of high explosive inside spherical containment vessels of 352mm in diameter using hydrocode to investigate the pressure loading on the walls. A spherical charge of 25.4mm diameter was located centrally inside the structure. The explosive products were modelled using the JWL EOS, and the air was approximated using the linear polynomial EOS.

Figure 2.8 shows the results of the simulation, which shows the formation of two consecutive shocks post-detonation. Prior to interacting with the shell wall, two shocks, namely the main and secondary shocks were observed approximately $40\mu s$ after detonation. The secondary shock traveled in the opposite direction relative to the moving explosive product and formed a high-density region between the main shock and the air-explosive interface. The high-density region is indicated in Figure 2.8 b).

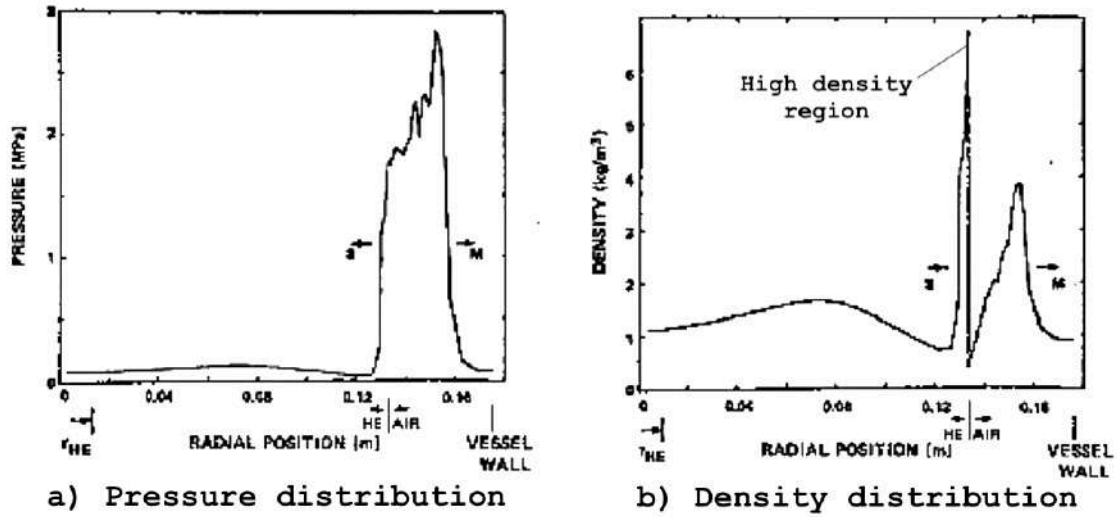


Figure 2.8: Pressure and density distributions 40 μ s after detonation [34]

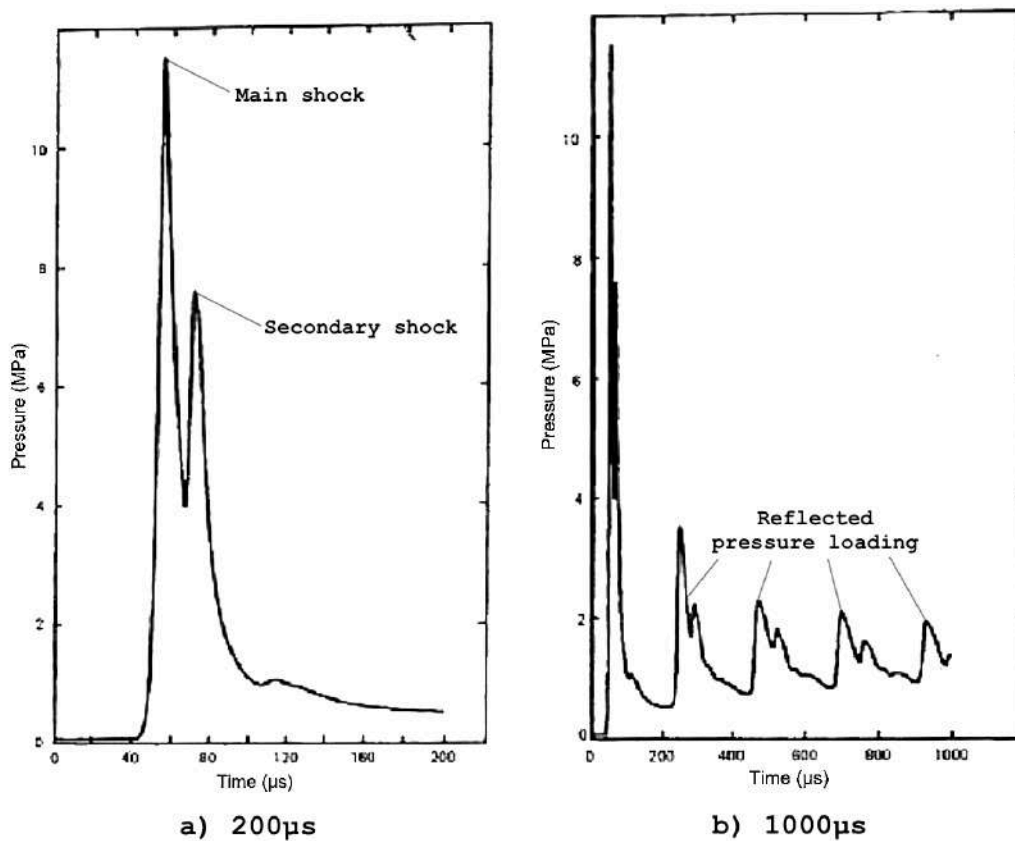


Figure 2.9: Pressure pulse on shell wall for different times following detonation [34]

After the main shock is reflected off the container wall, it interacts with the air-explosive interface. Since the interface is highly dense, a significant portion of the shock wave is once again reflected back to the wall, resulting in another loading pulse. The two loading pulses are

shown in Figure 2.9 which shows the wall pressure for the duration of $200\mu s$ after detonation. The rest of the main shockwave was shown to be reflected from the centre of the containment vessel, and impinge on the wall again at approximately $240\mu s$. This reflected pressure loading occurs repeatedly, as shown in Figure 2.9. The “double-peaking” phenomenon is observed for each load. It was also observed [34] that the size of the explosive charge resulted in a different pressure pulse shape, and at charge size of 14% of container radius, the second “peak” from air-explosive interface reflection was larger than the initial peak. The simulation was shown to overestimate the experimental values, especially that of the initial shock front.

Feldgun et. al [35] studied the residual blast pressure (gas pressure, quasi-static pressure), with the aim of understanding the characteristics of an interior explosion within a room with limited venting. Different models for the prediction of gas pressure were presented, including a thermodynamic model relating the gas pressure to the total released energy and the heat capacity ratio. This model was shown to be in best agreement with the experimental data and was also successfully applied to simulation of confined blast response of flexible structures.

In another study, Feldgun et al. [36] conducted a numerical investigation on the late-time pressure relief behaviour from partially confined explosions. AUTODYN was used, and the ideal gas model was used for the air and the JWL EOS was used for the TNT model. In the experiment used for comparison, the corners of the structure were truncated; therefore the numerical model investigated both the truncated and non-truncated models. It was shown that both in 2D and 3D, the pressure-time history was affected qualitatively and quantitatively by the truncation, especially at gauges closer to the corners. The comparison between the pressure-time history of truncated and non-truncated 2D models is shown in Figure 2.10.

The numerical models were validated using experimental data as well as the UFC 3-340-02 manual [24]. The simulated pressure-time history showed a good correlation with the experimental data especially in the initial peaks, but underestimated the pressure responses significantly in the intermediate and late-time range. Feldgun et al. [36] stated that the underestimation may have been due to the mesh size sensitivity of the model, or because the model did not take the afterburning effect into account. Afterburning refers to the phenomenon where a longer-duration combustion process releases further energy into the system following an explosion [36].

Feldgun et al. [36] also proposed an equation for the calculation of residual quasi-static pressure following a fully-confined explosion.

$$P = \left[\frac{P_\theta}{\gamma_\infty - 1} (V - v_E) + Q\rho_E v_E \right] \frac{\gamma - 1}{V} \quad (2.4)$$

where P_θ is the atmospheric pressure, γ and γ_∞ are the adiabatic indices of detonation product-fuel mixture and of pure air respectively, V is the volume of the structure, and v_E and ρ_E are the charge volume and density respectively. Q represents the detonation energy released during explosion.

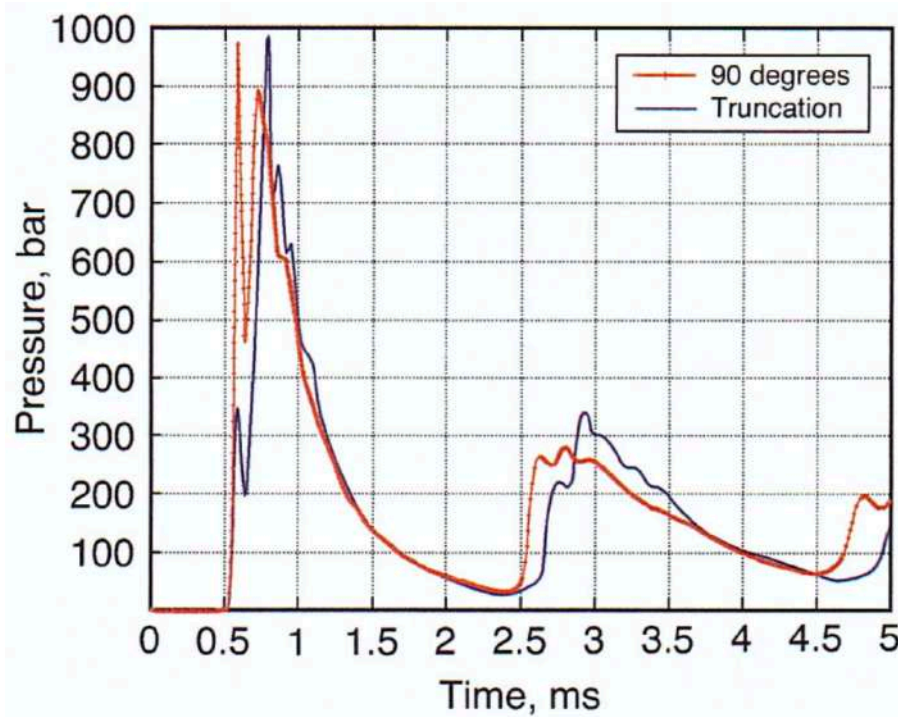


Figure 2.10: Typical pressure-time history from 2D simulations with and without truncations by Feldgun et al [36]

Edri et al. [32] studied the influence of afterburning on the simulation of pressure-time history, and proposed that the after burning effect may be captured by altering the value of Q depending on the ratio of structure volume (V) and charge mass. Furthermore, Edri et al. [32] updated the numerical model by Feldgun et al. [36] by specifying additional energy into the explosive over a given time interval to capture the additional energy introduced by afterburning. When afterburning was not taken into account, the pressure peaks occurred later in the simulation, as depicted in Figure 2.11, showed poor correlation with the experimental data. However, with the model modified to account for afterburning, the correlation between numerical and experimental data was significantly improved as shown in Figure 2.12, in terms of the arrival time as well as the magnitude of the pressure. It was thus shown that significant error may be introduced in numerical models when afterburning is not accounted for.

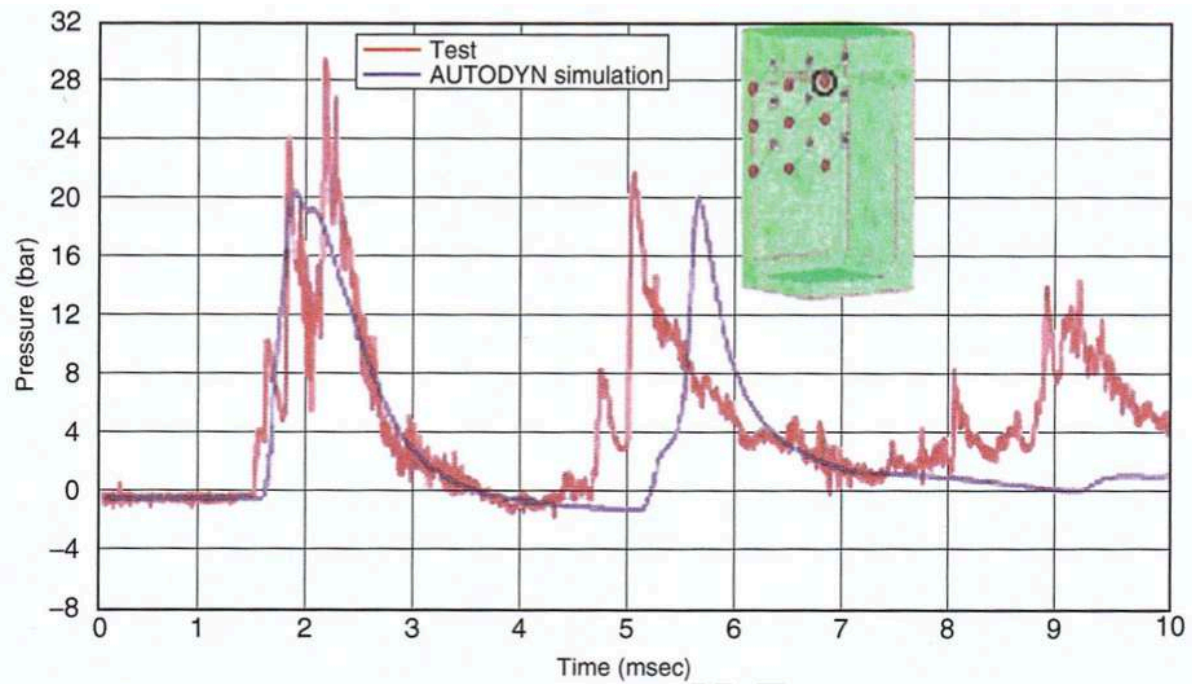


Figure 2.11: Experimental simulated pressure-time history comparisons for simulations ignoring afterburning by Edri et al. [32]

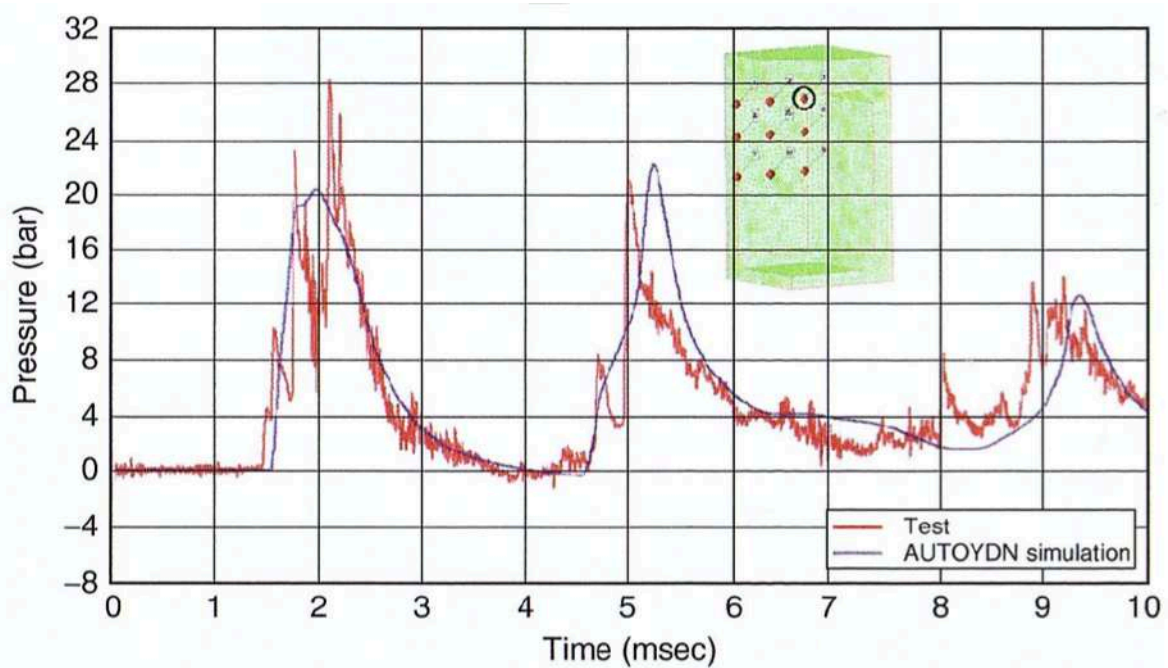


Figure 2.12: Experimental simulated pressure-time history comparisons for simulations considering afterburning by Edri et al. [32]

Pickerd et al. [37] analysed the structural response and failure of welded partially and fully confined containers subjected to internal blast loading. The findings indicated that an accurate prediction on failure may require information on both the peak pressure and the impulse of the blast loading, not an estimation based on quasi-static loading.

2.2.5.2 Confined Blast Loading Simulation

Modern standards in blast-resistant design assumes spherical charge shape and does not take into consideration the effect of charge shape. Confinement is an aggravation factor of explosion effects, and must be taken into consideration when designing or analysing infrastructure under blast loading. Hu et al. [38] developed a numerical model in AUTODYN to predict blast loading inside unvented structure under varied charge shape and charge orientation, as well as confinement chamber geometry and volume. The model was validated with available external blast testing data, as well as confined box blast experiment data.

The study involved parametric studies investigating the variation in reflected overpressure within the confinement box, by changing charge orientation, charge weight, and charge shape. The parametric study was conducted using small 3D boxes of various dimensions with a charge placed in the centre. Gauges were placed to measure reflected overpressure histories, and are labelled with numbers in Figure 2.13. The cylindrical charges were point-detonated from the centre of the charge. Contrary to the assumption of uniformly distributed overpressure and one peak shock reflection from UFC-3-340-02 [24], large variation in peak reflected overpressure and multiple shock reflections were observed along the chamber walls. Figure 2.13 shows the peak overpressure distribution along centre lines of the walls of the box when a cylindrical charge of 100g ($L/D = 2$) of vertical orientation along z-axis was placed at the centre. The pressure distribution showed significant variation of peak pressure between the centre and the corner in any given plane, the overpressure being up to 3 times higher in the corners.

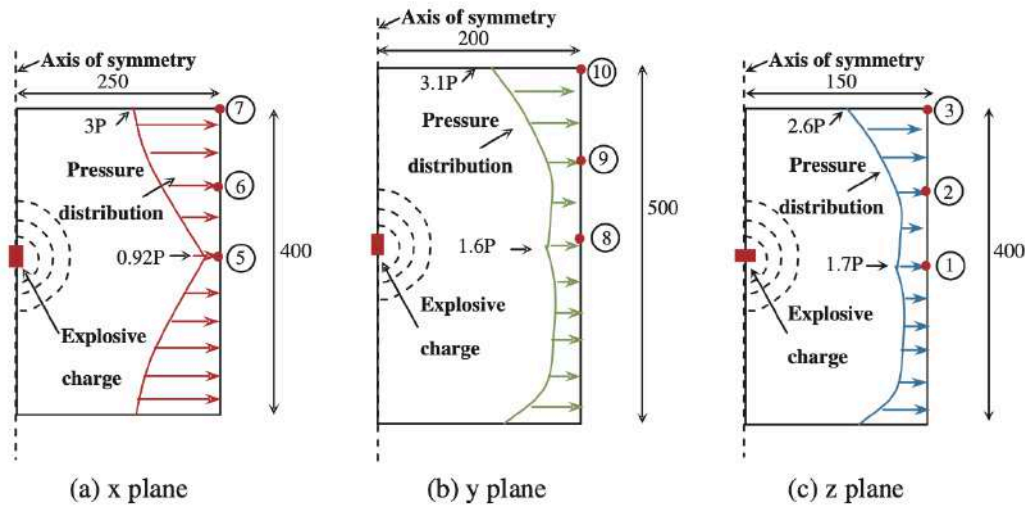


Figure 2.13: Peak overpressure distributions along centre lines of the 3 planes of the small box with a 100g cylindrical charge ($L/D = 2$) z-axis orientation [38]

Five different charge weights and six different length-to-diameter ratios (L/D) were tested to investigate the effect of variation in charge length-to-diameter (L/D) ratio. The results showed that peak reflected overpressure in the radial direction increased significantly with an increase in L/D ratios until $L/D = 4$, but decreased with further increase in L/D value (shown in

Figure 2.14). The trends in the axial direction showed that the smaller values of L/D generally resulted in higher overpressure, as shown in Figure 2.15. The results indicated that a great portion of energy was directed axially for the charge with low L/D value, while more energy was directed radially for higher L/D values.

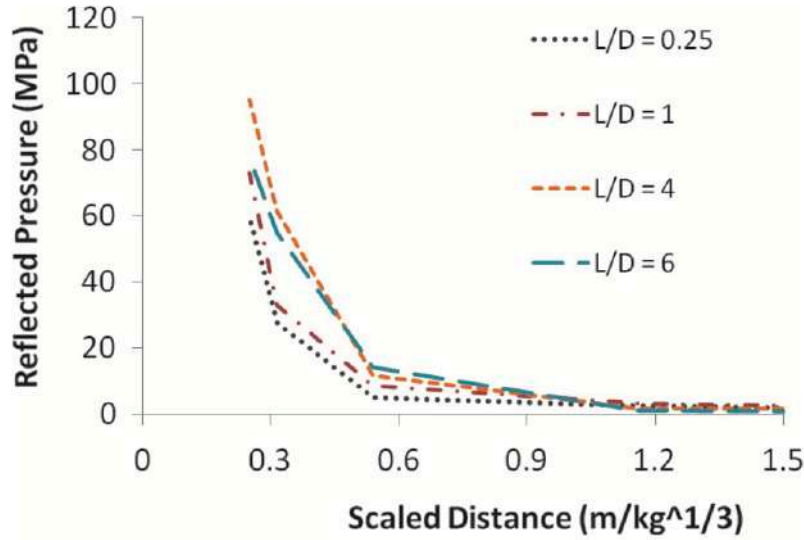


Figure 2.14: Peak reflected overpressure as a function of L/D in $0.125m^3$ cube in the radial direction [38]

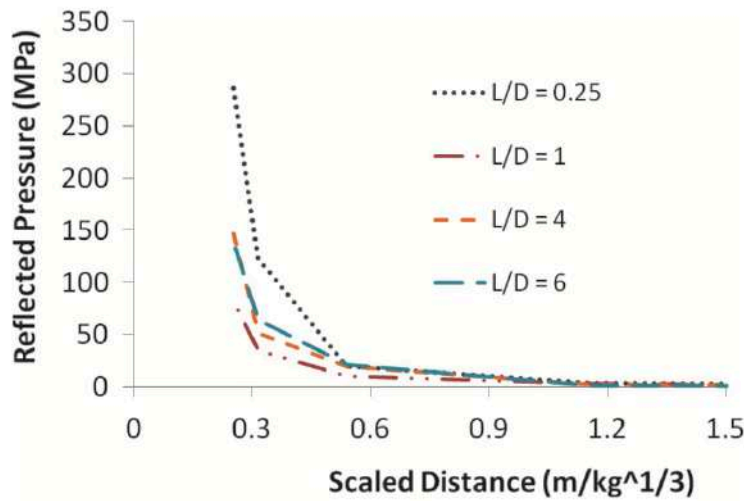


Figure 2.15: Peak reflected overpressure as a function of L/D in $0.125m^3$ cube in the axial direction [38]

The effect of chamber geometry on overpressure was also investigated. The chamber volume was kept constant at $0.125m^3$. Three different chamber geometries were tested, namely a cuboid, a cylinder and a cube. A spherical charge was placed at the centre of the chambers in each test. It was observed that the cylindrical chamber resulted in the highest reflected overpressure in z direction among all three chamber geometries tested.

Salvado [39] performed ALE simulations using AUTODYN and LS-DYNA hydrocode softwares to conduct parametric studies on confined chambers, where various cuboid chamber “slenderness ratios” [39] were investigated. The results showed that a shorter “stand-off” distance between the chamber wall and the charge resulted in an increase in the observed peak pressure. It was shown that the subsequent reflected pressure intensity after the initial impact was generally lower than the initial blast peak pressure.

At the vertices, strong reflections were observed due to the pressure waves arriving simultaneously from three orthogonal directions. Even for non-cubic boxes, the wave interference was evident, as shown in Figure 2.16, where the corners of the box nearest the explosive were shown to experience the largest pressures. Salvado [39] stated that the angle between the stand-off direction and the main planes of the internal faces of the box were correlated with the peak pressures obtained at the corners, and thus contributed to the pressure enhancement effect observed in certain cases which could not be explained by the stand-off or z scaled distance alone.

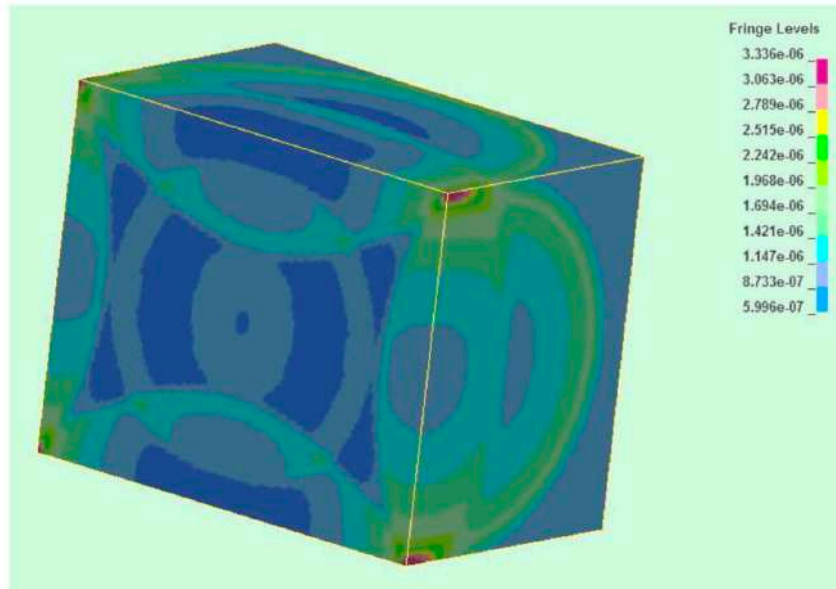


Figure 2.16: Reflected wave pressure peak at corner (tracer point 3) for box 2 with explosive located half way between the centre of the box and the face normal to the z axis [39]

2.2.5.3 Cylindrical Confinement Vessel

A right-circular cylindrical vessel is one of the most commonly used type of confinement vessel due to ease of manufacture, the convenience of their symmetrical shape and their relative efficiency in absorbing blast loads [40]. Duffey and Mitchell [40] investigated the response of mild steel air-filled cylindrical containers with open ends subjected to internal blast loading. The geometry of the cylinder implemented in this study is shown in Figure 2.17.

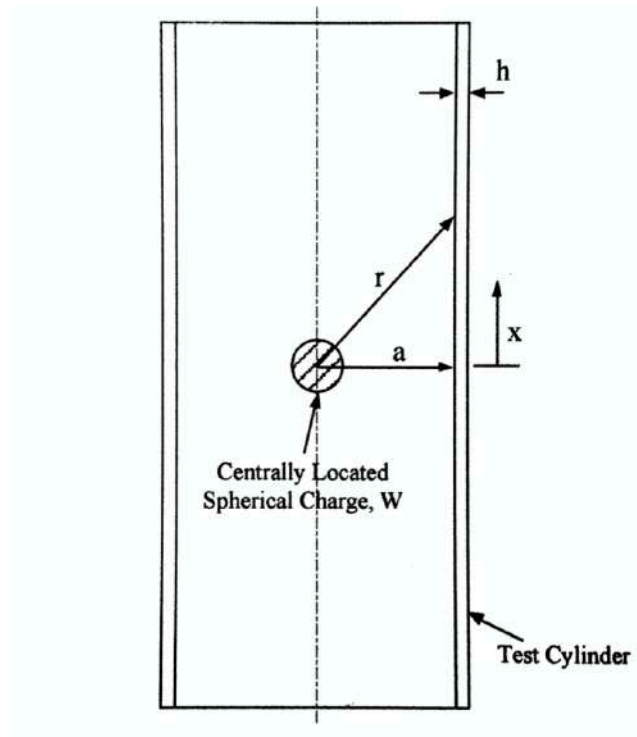


Figure 2.17: Schematic of open cylindrical vessel geometry [40]

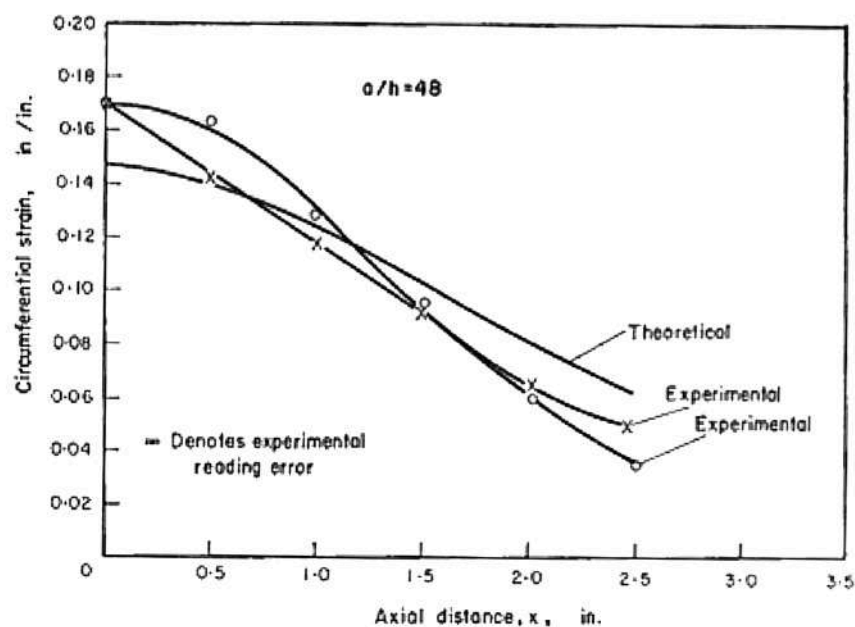


Figure 2.18: Circumferential strain versus axial distance results for mild steel open cylinders subjected to internal blast load (80g Pentolite) [40]

The focus of the investigation was on the radial strain as a function of axial coordinates and charge mass. Duffey and Mitchell [40] proposed Equation 2.5 to determine the radial strain as

a function of the radial length in an open-ended cylinder subjected to purely impulsive load.

$$\varepsilon(x) = \frac{I_{eff}^2(x)}{2\rho h_t^2 \sigma_y [1 + (\frac{I_{eff}(x)}{\rho h_t R D})^{\frac{1}{p}}]} \quad (2.5)$$

where I_{eff} is the effective selective impulse, ρ is the density of the cylinder material, h_t is the wall thickness, σ_y is the yield stress, R is the cylinder radius and D, p are strain rate sensitivity constraints.

The theoretical strain results obtained from Equation 2.5 were in reasonable agreement with the experimental results, as shown in Figure 2.18. It must be noted that Equation 2.5 is valid for bare charges in air-filled cylinders only. The effect of shrapnel was also not considered, and it was stated that the presence of shrapnel may cause failure at a lower charge levels than predicted by the proposed equation.

Benham and Duffey [41] studied the wall response of a closed steel cylinder due to detonation of a centrally located bare charge detonated in a setup shown in Figure 2.19. The amount of radial deflection, and hence the circumferential strain, was shown to be proportional to the effective ratio of the long term pressure and the static burst pressure of the cylinder.

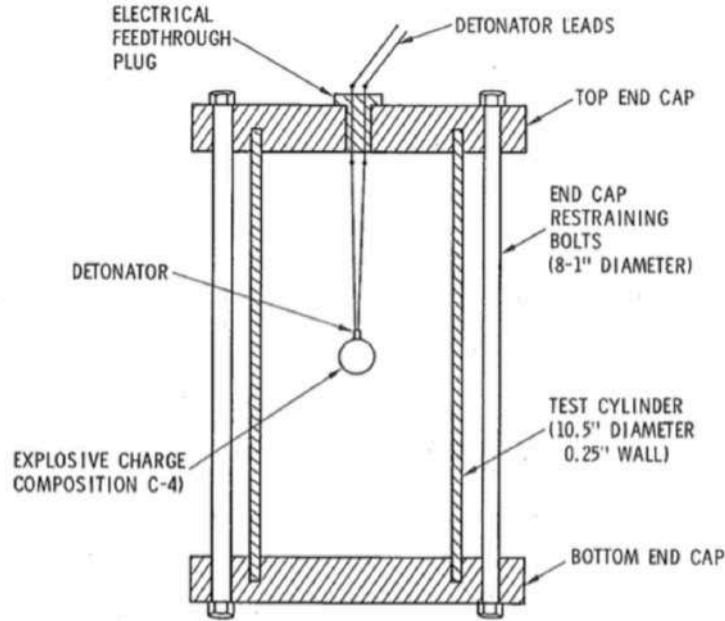


Figure 2.19: Schematic of closed cylindrical vessel experimental setup [41]

Benham and Duffey [41] also presented Equation 2.6 to determine the type of loading which governed radial deformation.

$$\bar{\eta} = \eta \left(\frac{t_{max} - T_d}{t_{max}} \right) \quad (2.6)$$

where $\bar{\eta} = 0$ for $T > t_{max}$. $\bar{\eta}$ is the effective pressure ratio, η is the ratio of the equilibrium

pressure to the static burst pressure, t_{max} is the time taken to peak deflection and T_d is the delay between the impulsive load and the equilibrium pressure.

It was shown theoretically and experimentally that for cylinder of $R/h = 21/22$ inches, the load would be purely impulsive for $0 < \bar{\eta} < 0.3$ and impulsive with long-term pressure for $0.3 < \bar{\eta} < 0.8$. Unpredictably large deformation occurred for $0.8 < \bar{\eta} < 1$ and bursting occurred for $\bar{\eta} > 1$. The equations derived were valid only for cylinders where the length was greater than twice the radius ($L > 2R$). The axial stress and axial bending moment were both neglected, and the deformed profile of the cylinder was also not put into consideration [41].

Langdon et al. [42] reported on the response of partially confined right-circular stainless steel cylinders ($L/R = 4$) to internal air-blast loading. The loading conditions were varied by changing the axial location and the mass of bare PE4 charge. When the charge was located at the axial centre, the permanent diametric deflection showed a linear increase with increasing charge mass, as illustrated by Figure 2.20. The scatter in the data shown in Figure 2.20 was attributed to the slight asymmetry of loading and consequent asymmetry in the structural response [42]. In general, the cylinder response was symmetrical, as illustrated in Figure 2.21. Longitudinal tearing occurred at charge mass of 60g, and complete failure (petalling) occurred at charge mass of 75g [42].

Further numerical analysis showed that the interaction of the pressure waves reflected off the cylinder walls and the closed end caused the pressure to be driven out of the open end. The quasi-static pressure accumulation, therefore, had little effect on the deformation of the cylinder [42]. However, at full confinement where the pressure waves cannot leave the vessel, the quasi-static pressure is likely to become non-negligible.

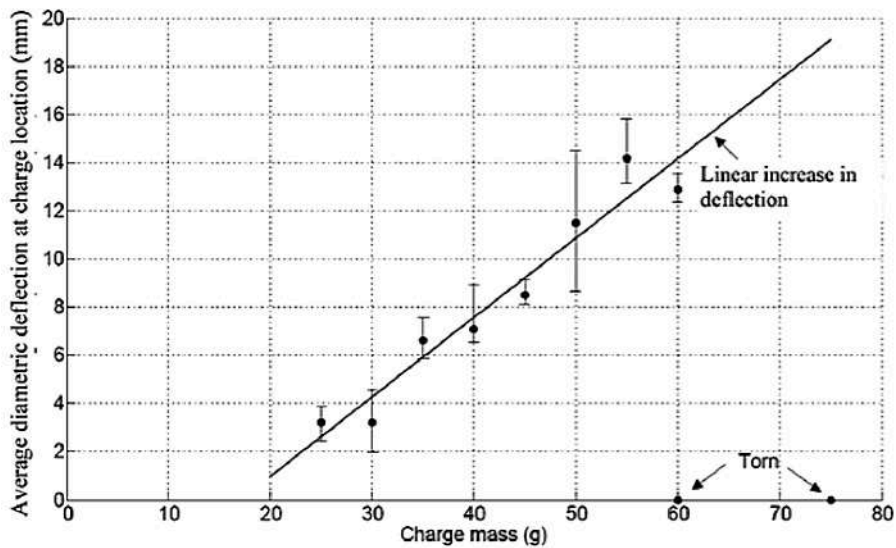


Figure 2.20: Graph of average diametric deflection at the charge mass location versus PE4 charge mass for $L = 150mm$ (error bars indicate maximum and minimum measurements [42])

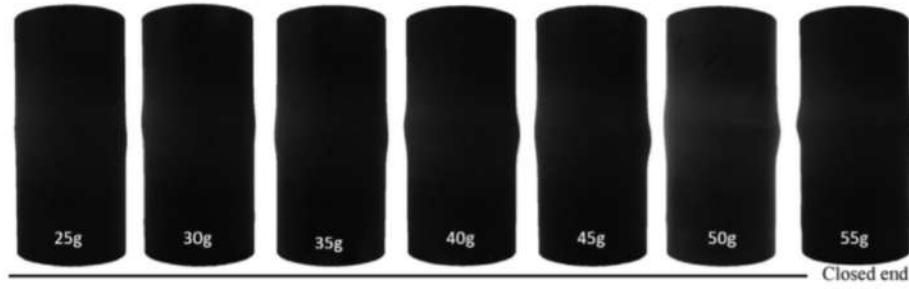


Figure 2.21: Deformed profiles of the blast loaded cylinders (the clamped and closed end at the bottom) [42]

Rushton et al. [43] carried out theoretical, numerical and experimental work on internal blast loading of steel pipes. Open-ended cylindrical mild pipe specimens of length 800mm , diameter 324mm and wall thickness of 9.5mm were used. Cylindrically shaped PE4 charges were placed at the centroid of the pipe and detonated from both ends to achieve symmetrical loading. Numerical models were developed in AUTODYN to investigate the effect of charge shape on the structural response. The results showed that as the length-to-radius ratio (L/R) of the cylindrical charge increased, the peak pressure and the impulse increased significantly. When the charge shape was changed to a sphere instead of a cylinder while the mass was kept constant, the observed pipe deformation was less severe. Rushton et al. [43] concluded that the shock transmission through the explosive had a significant influence on the impulsive loading.

2.3 Fragmentation

Little to no research are publicly available on the topic of IED shrapnel damage on structures. However, shrapnel from IED explosions are similar to blast-induced fragments, and literature related to fragmentation damage is readily available.

2.3.1 Fragment Velocity Equations

Gurney [44] proposed a formula to estimate the initial fragment velocities of spherical, cylindrical and infinite plane metal casing enclosing high explosives as a function of the ratio M/C , where M is the mass of the casing and C is the mass of the charge. A number of simplifying assumptions were made, such as uniform gas density, lack of material strength and the late release of detonation products. The potential energy of the explosive prior to detonation was also assumed to be equal to the kinetic energies of the detonation products and the metal post-detonation. The formula for the fragment velocity of a cylindrical warhead (on-axis initiation) is shown in Equation 2.7 [44].

$$V_0 = \sqrt{2E} \sqrt{\frac{C/M}{1 + 0.5C/M}} \quad (2.7)$$

where V_0 is the fragment velocity (m/s), $2E$ is an empirical Gurney velocity constant of the explosive charge, and C/M is the ratio of charge-to-metal casing masses.

Taylor [45] also developed a relationship for predicting the fragment projection angle (relative to the normal of metal surface) θ , given V_0 and the velocity at which the detonation wave sweeps along the explosive-metal interface, U , as shown in Equation 2.8.

$$\theta = \sin^{-1}\left(\frac{V_0}{2U}\right) \quad (2.8)$$

Gurney [44] neglected the effect of propagation of detonation wave in the explosive, and assumed the simultaneous rupture of the metal casing. Therefore the fragment velocity distribution was assumed to be uniform. However, this assumption did not take into account the “end effect” of the rarefaction waves from the casing ends. Studies on fragmentation velocities of encased charges have shown that the velocity of fragments are lower at the ends of the cylindrical charge due to what is called the "end effect". Charron [46] described that end effects resulted from the rarefaction waves generated in the system. When detonation occurred, a shock wave passed through the explosive, transforming the chemical energy of the explosive to kinetic energy and forming a reaction zone of finite thickness behind the shock wave. The explosive process continued through the reaction zone and beyond into the burnt gases so that rarefaction waves were created. [46] The detonation process is illustrated in Figure 2.22.

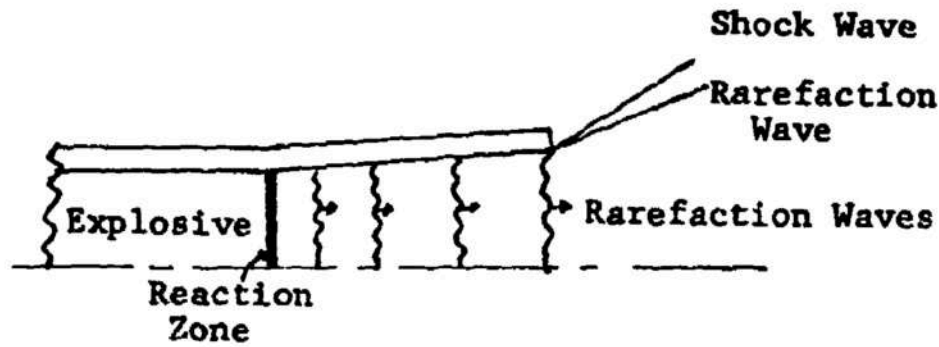


Figure 2.22: Progression of detonation in a cylindrical encased charge [46]

Because of the high detonation velocities involved in the process, the explosive matter was completely detonated before the casing broke apart, and the rarefaction waves formed at the initiated end of the explosive travelled in a direction opposite that of the detonation shock. The central section of the detonation products was still confined and the pressure loss was less in the central section than at the initiated end. When the detonation shock reached the opposite end, the gases were no longer confined and the rarefaction waves caused the loss of pressure. In cases where L/D was small, the rarefaction waves reached farther inside the casing towards the middle and the pressure decreased over a wider axial distance. Because the pressure was decreased, the velocity of the fragments was decreased.

König [47] used a semi-empirical approach to study the effect of rarefaction waves on the axial distribution of fragment velocity and its direction. A correction term was added to the Taylor angle, based on the concept that the fragment velocities near the end are reduced due to rarefaction waves. The results of the study indicated that the axial pressure drop across the encased cylinder was the dominating driving factor behind the divergence of the fragments in solid casings. For pre-fragmented casings, however, the accelerative drag due to the axially outward component of the release-wave gas flow over the fragments was the dominating factor, and the axial pressure drop was a minor factor in comparison.

Anderson [48] carried out numerical studies with cylindrical casings of different fill and length-to-diameter (L/D) ratios. A time-dependent, two-dimensional, finite-difference code (HEMP) was used to model the fragmentation, and also implemented a gas leakage model to simulate the explosive gas leakage around the fragments after the casing had shattered. The findings showed that when the flow could not be considered as one-dimensional (for example, when $L/D < 2$), the fragment velocity could not be predicted accurately with the Gurney [44] and Taylor [45] formulae. It was also shown that an enhanced containment of the explosive gas resulted in higher final fragment speeds.

Kong [49] conducted numerical research on cylindrical casings, both with and without end caps, using AUTODYN hydrocode to model the detonation process and the fragment distribution. It

was shown that the wave propagation and reflection was directly related to the casing expansion and rupture. Kong [49] observed that the fragments from the cylinder end opposite the initiation end (i.e. far end) showed higher velocity in its normal direction than the near end, and also expanded more rapidly, mainly due to the additive effect of the initial ‘free’ detonation waves and the reflected detonation waves.

Since the Gurney formula does not take into account the end effect or the incident angle of the detonation wave, the formula yields different results to the experimental data. Many researchers have developed correction formulae based on the Gurney formula, taking these factors into account. In order to correctly predict the fragment velocity at cylinder ends, Zulkonski [50] used the least squares method and exponential curve fitting to obtain reduction factors for the existing Gurney formula and presented the following equation (Equation 2.9).

$$V_0 = (1 - e^{-2.361\frac{x}{d}})(1 - 0.288e^{-0.4603\frac{(L-x)}{d}})V_{Gurney} \quad (2.9)$$

where x is the distance from the detonation point along the cylinder axis, L is the cylinder length, d is the charge inner diameter and V_{Gurney} is the velocity predicted by Gurney (Equation 2.7).

Charron [46] proposed another correction formula for the Gurney velocity based on “geometric equivalence”, which is the filled ratio of the reduced amount of explosive near the cylinder edges. The modified equation is shown in Equation 2.10.

$$V_0 = \sqrt{2E} \sqrt{\frac{F(x)C/M}{1 + 0.5F(x)C/M}} \quad (2.10)$$

where $F(x) = 1 - (1 - \min(\frac{x}{2R}, 1.0 \cdot \frac{L-x}{R}))^2$ and R is the cylinder radius.

Both Equations 2.9 and 2.10 share the problem of yielding zero velocity at the detonation end, which does not agree with the experimental results. Huang et al. [51] noted the zero-velocity issue and proposed a new modified semi-empirical formula based on the experiments by Feng [52] which used flash radiograph technique to measure the fragment velocity distribution after detonating cylindrical casings from one end. Huang et al. [51] focused on the influence of the cylinder end rarefaction wave on the initial velocity of the fragments. The correction formulae for the detonation end and for the non-detonation end was based on Zulkonski’s [50] modified equation. The correction coefficients were determined using curve-fitting from the experimental data.

$$V_0 = (1 - 0.361e^{-1.111\frac{x}{d}})(1 - 0.192e^{-3.03\frac{(L-x)}{d}})\sqrt{2E} \sqrt{\frac{C/M}{1 + 0.5C/M}} \quad (2.11)$$

Since the experimental data reflects the effect of rarefaction waves, the resulting equation also

considers the influence of rarefaction end effect. The proposed formula, Equation 2.11, was shown to give better fragment velocity prediction than previous modified Gurney equations, when compared against existing experimental data [51].

Another factor that influences the fragment velocity is the incident angle of the detonation wave. Li et al. [53] conducted a numerical study on the effect of rarefaction waves, wave incidence angle and eccentric initiation in a cylindrical casing filled with charge. Figure 2.23 from Li et. al [53] shows that the variation in initiation point causes a variation in the incident angle (α) of detonation wave at an arbitrary point (labelled as P in Figure 2.23) on the casing.

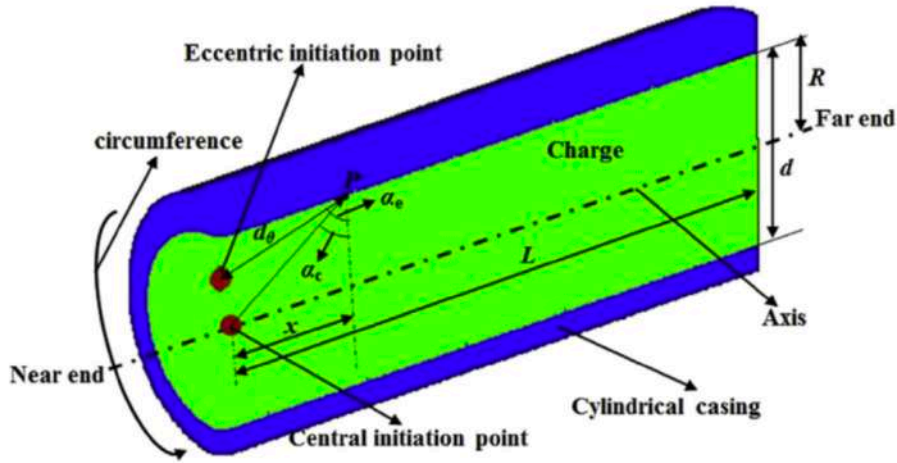


Figure 2.23: Cutaway schematic diagram of a cylindrical casing with charge [53]

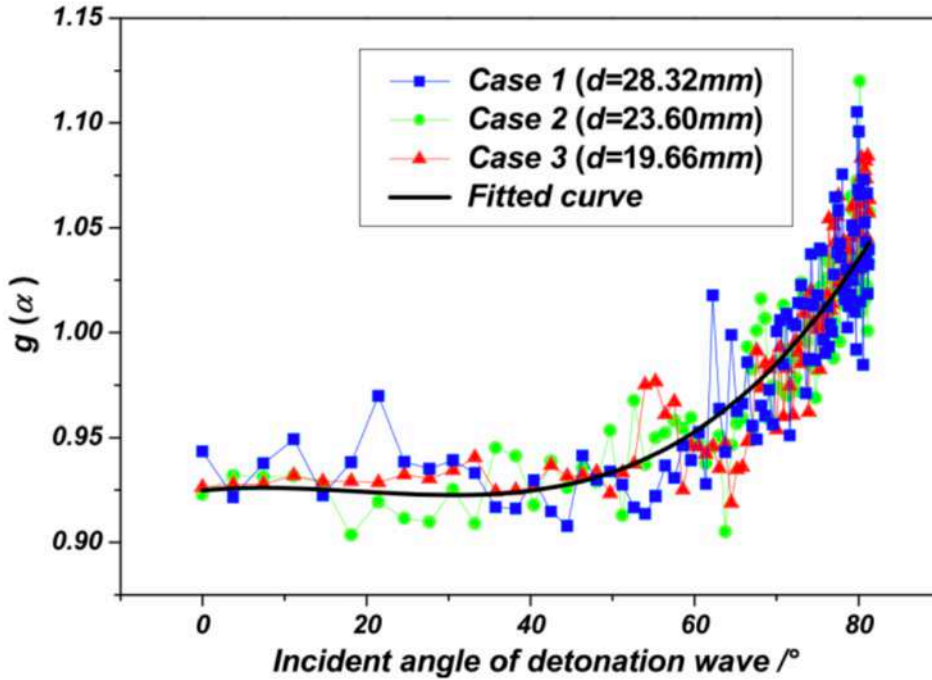


Figure 2.24: Curve of $g(\alpha)$ versus incident angle of detonation wave [53]

The study assumed that Equation 2.11 is composed of the Gurney formula and a correction term, $f(x/d)$, which is a function of the ratio between x (the axial distance between the near end and P) and diameter d . In other words, $f(x/d) = (1 - 0.361e^{-1.111\frac{x}{d}})(1 - 0.192e^{-3.03\frac{(L-x)}{d}})$. $f(x/d)$ was further decomposed into two terms: 1) the rarefaction wave effect term $h(x/d)$, and 2) an incident angle term, $g(\alpha)$, a function of the incident angle α . Numerical modelling was used to derive the detonation wave incident angle influence formula $g(\alpha)$, and showed that the formula is only dependent on the angle of incidence (α), and that it follows a trend shown in Figure 2.24. It was evident that the angle of incident term $g(\alpha)$ increased with the value of α . Therefore, for a fixed rarefaction influence factor, an increase in angle would result in an increased velocity.

2.3.2 Effect of Pre-formed Fragments with Gaps

Taylor [45] devised a method to determine how the efficiency of a bomb as a projector of fragments may be quantified in terms of the amount of detonation energy transferred as the kinetic energy of the casing. The limiting condition for fragment velocity was considered to be the point at which complete casing rupture had occurred. Therefore, the further the case expanded without rupture, the amount of energy transferred to the casing was increased, leading to higher fragment velocity and greater ‘efficiency’. The efficiency was greater if the elastic-plastic expansion of the casing lasted longer.

When discrete fragments were placed on a cylindrical body, air gaps occurred between the fragments in the circumferential direction. Therefore, there was no containment effect and the detonation products would immediately pass through between the the component members of the case. The energy transferred to the fragments would therefore be substantially reduced.

Waggener [54] studied fragment velocities using solid cylinder and pre-formed fragments, where steel cubes (i.e. pre-formed fragments) were placed directly on Comp B explosive charge. It was concluded that warheads utilising pre-formed fragments produced fragment velocities approximately 10% lower than their solid case equivalents. Pearson [55] conducted similar studies involving pre-formed fragments and low strength cylindrical check seals (inserted between the explosive and metal components for containment effects), the percentage reduction in pre-formed fragment velocity reached up to 20% when compared to fragments from solid cylinders.

The velocity loss in pre-formed fragments have also been related to the value of the Gurney constant $\sqrt{2E}$ used in the Gurney equation [44]. Henry [56] showed that the effective value of the Gurney constant was reduced by 10 to 15% when the casing being tested was pre-fragmented, to account for the early gas leakage. Kennedy [57] similarly noted that the reduction in Gurney constant value for pre-formed fragments was up to 20%. In another work, Kennedy and Chou [58] studied the conditions where there were gaps in explosive metal systems. It was concluded that if there were gaps between the explosive and the metal parts, the velocity of the metal decreased in a manner that could be predicted by considering the explosive to be operating at a density equivalent to the mass of the explosive divided by the total volume of the

explosive and the gaps. The discrete fragment casing around cylindrical charge was analogous to composite of metal and air. The effective value of the Gurney constant could be calculated from this relationship, and the reduced value of the Gurney constant consequently led to a reduction in the Gurney velocity value. Furthermore, Charron [46] suggested the use of existing Gurney velocity formulas with only part of the charge mass, to obtain lower calculated velocities for pre-formed fragments.

Figure 2.25 shows the results of the discrete-fragments, $C/M = 0.937$, $L/D = 2.0$, end initiation test from one of the numerical studies conducted by Anderson in [48]. The fragment velocity vs. relative axial position plot shows that when the casing was pre-fragmented, the fragments velocity was much lower than the theoretical Gurney velocity value. The value had also decreased significantly at either ends. The simulated projection angle vs. relative axial position plot from Figure 2.25 showed that while the projection angle towards the middle of the cylinder corresponded to the Taylor angle, the projection towards the cylinder ends deviated significantly from the Taylor angle value. Anderson [48] stated that the fragment separation and gas leakage occurred at smaller expansion ratios for discrete fragment cylinders, and the effects of explosive gas venting were thus more pronounced.

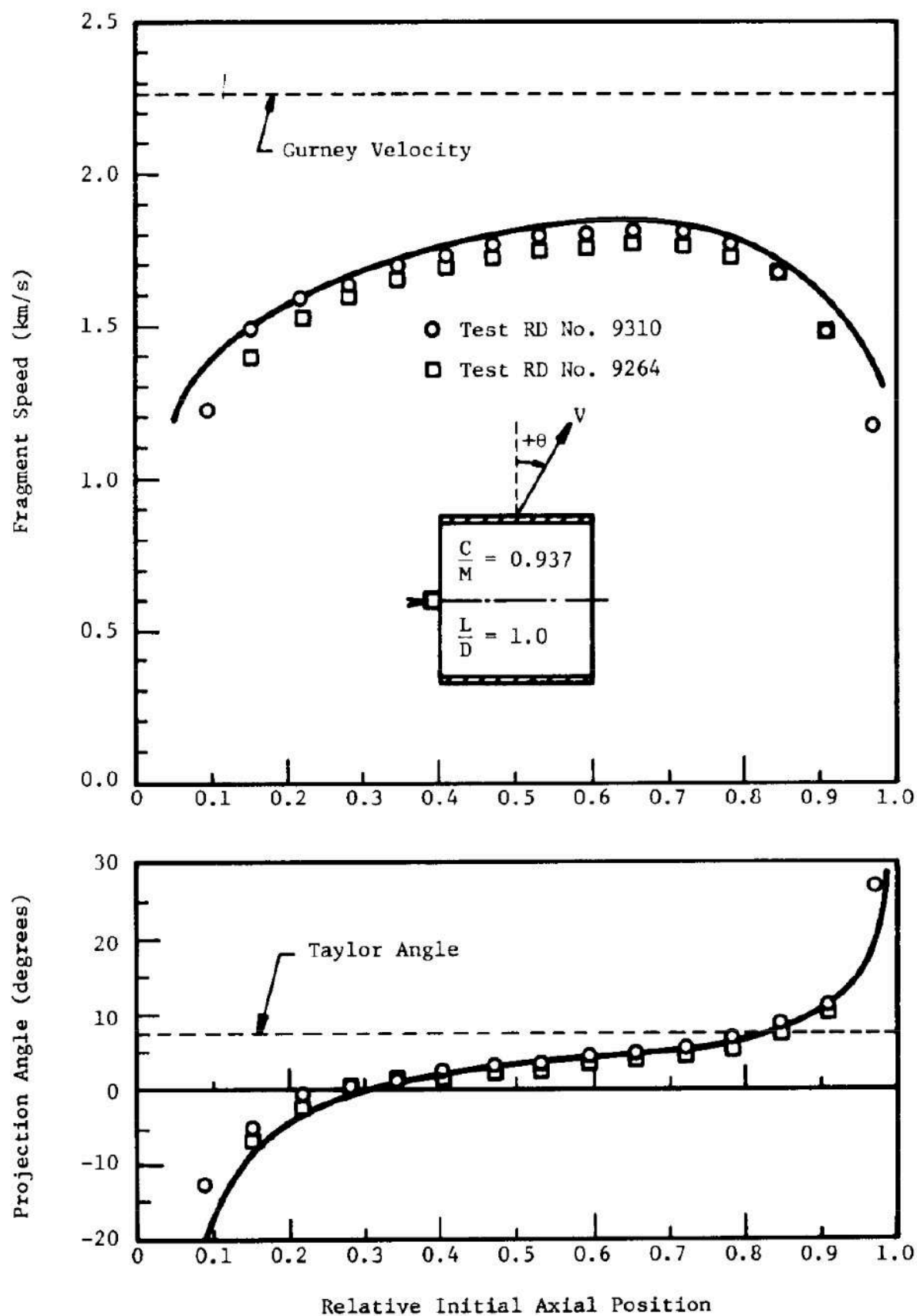


Figure 2.25: Fragment velocity vs relative initial axial position, discrete fragments, $C/M = 0.937$, $L/D = 2.0$, end initiation [48]

2.3.3 Charge Encasement

Often times during explosions structural damage are caused not only by the blast waves, but also by the fragments formed when the casing ruptures. Explosives combined with casing typically cause more damage [59]. Casing fragments travel at very high speeds and may cause significant damage, depending on its size. Damage caused by munitions or warheads are typically a combination of both blast and fragmentation. The casing may be constructed to achieve a specific effect, whether it be naturally fragmenting or pre-formed.

During blast, the fragmentation process absorbs a significant amount of the energy released by the explosive. Therefore, blast generated by a cased charge is typically weaker than a bare charge of the same charge mass, regardless of the amount of damage caused. The fracture process also determines the amount of reduction in blast magnitude. If the casing fractures prematurely or the fragments are pre-formed, the gases escape through the gaps and the energy transferred to the fragments is reduced, and the blast is in turn not reduced as significantly [59].

The equivalent bare charge (EBC) is defined as “the amount of bare explosive that would produce the same blast as a cased weapon” [60]. EBC equations such as the Fisher equation (Equation 2.12) and the modified Fisher equation (Equation 2.13) are used to estimate the blast generated by cased explosives.

$$EBC_{Fisher} = C(0.2 + \frac{0.8}{1 + \frac{M}{C}}) \quad (2.12)$$

$$EBC_{ModifiedFisher} = C(0.6 + \frac{0.4}{1 + \frac{M}{C}}) \quad (2.13)$$

where EBC is the Equivalent Bare Charge (kg), C is the mass of explosive (kg) and M is the mass of the casing (kg). It is recommended to use the Fisher equation for thick cases or large fragments, and the modified Fisher equation for pre-formed fragments and other casings that provide less confinement.

Dunnett et al. [61] reviewed a number of EBC prediction methods and summarised the equations on a plot of EBC/C vs. M/C . It was shown that while all the equations exhibited the same trend, implying that increasing the casing mass results in a decrease in the bare charge mass required to produce the same magnitude of blast, no other distinct correlations could be found.

Crowley [62] derived a new empirical formula for calculating the EBC (Equation 2.14) through comparison with experimental data, after expressing doubt in the validity of the existing EBC equations. It was also shown that the EBC was dependent on the casing material properties,

such as the casing yield stress.

$$EBC = C((1 - f) + \frac{f}{1 + 2\frac{M}{C}}) \quad (2.14)$$

$$f = 1 - 4.63(\frac{\sigma_y}{P_i})^{\frac{2}{3}} \quad (2.15)$$

$$f = \frac{E_G}{E_0} \quad (2.16)$$

where σ_y is the yield stress of the casing material, P_i is the pressure on the internal wall of the casing at fracture, E_G is the energy converted to kinetic energy of the casing material and explosive gases following detonation, and E_0 is the detonation energy of the explosive.

Hutchinson [59] applied the principle of momentum partition between the charge casing and the explosive gases to derive a new set of equations for cylindrical and spherical charges (Equations 2.17 and 2.18 respectively). It was assumed that the casing material reached a stable velocity prior to fracture, and the factor f from Equations 2.15 and 2.16 must be appended to Equations 2.17 and 2.18 as a multiplier in the instance of premature casing fracture.

$$EBC_{cylinder} = C(\frac{0.5}{0.5 + \frac{M}{C}})^{0.5} \quad (2.17)$$

$$EBC_{sphere} = C(\frac{0.6}{0.6 + \frac{M}{C}})^{0.5} \quad (2.18)$$

2.3.4 Experimental Studies of the Effect of Fragmentation on Structural Response

Chung Kim Yuen et al. [63] investigated the effect of charge encasement on the response of quadrangular steel plates blast loaded in air, using four encasement types namely cardboard, cardboard-no-cover (CNC), plastic and metal. The PE4 charge was cylindrical in shape, with a diameter of 45mm , and a height of 6.5mm . The charge mass (16.5g) and geometry were kept constant throughout the investigation. Domex 550 MC plates of dimensions $400\text{mm} \times 400\text{mm} \times 2\text{mm}$ with $300\text{mm} \times 300\text{mm}$ exposed area were used as target plate.

Shrapnel damage was observed on the target plate surface when the plates were exposed to the charge encased in metal. The hot casing fragment colliding with the plate resulted in localised damage in a circular pattern around the burn radius, shown in Figure 2.26, formed by the encased blast. The metal casing caused significantly more damage than the other types of casing even though the metal encased charge had a larger SoD from the plate. No shrapnel damage was observed in other types of encasement.

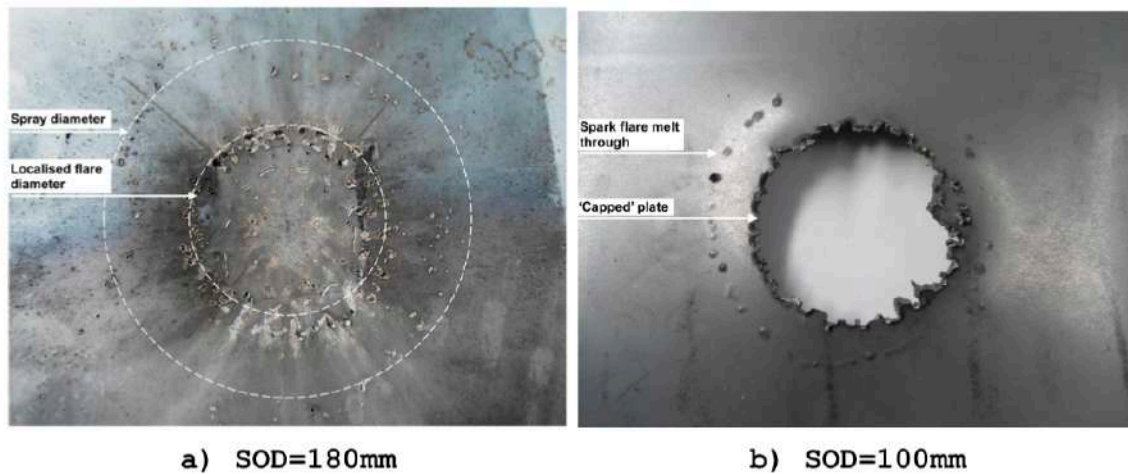


Figure 2.26: Typical response of plates subjected to charges with metal casings detonated in air [63]

Kong et al [64] conducted experiments with an encased charge on a multi-layered protective structure to study combined blast and fragmentation loading in the setup shown in Figure 2.27. The “cabins” within the layered structure were either filled with air or water. A cylindrical mild steel metal casing was filled with TNT to model a warhead, and the encased charge detonated in the middle compartment of the outermost layer of the structure. Target damage analysis following detonation showed that the shrapnel traveling radially from the cylindrical casing had formed a narrow band of perforation damage adjacent to the charge location, as shown in Figure 2.28. The reinforced plates of the structure were weakened by the fragmentation damage, and was more prone to damage by subsequent blast loading. Therefore the region perforated by a larger number of fragments were shown to exhibit greater blast damage as well.

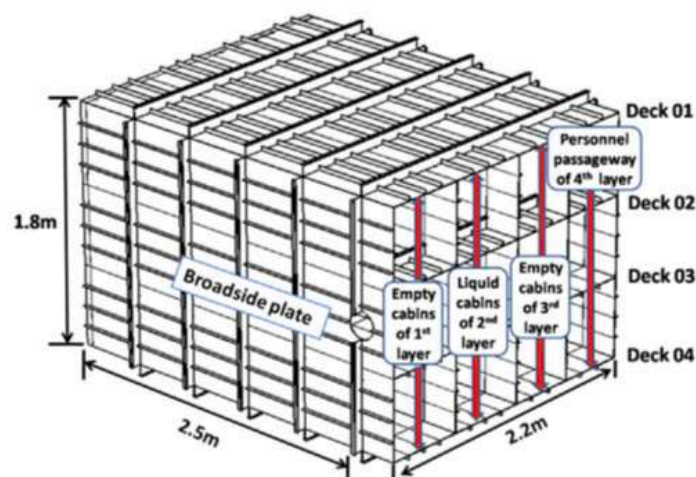


Figure 2.27: Multilayered structure used in Kong et al.'s experiments [64]



Figure 2.28: Perforation damage observed adjacent to charge location, as observed by Kong et al. [64]

2.3.5 Fragment Velocity Analysis from Experiments

Ranwaha and Chung Kim Yuen [65] conducted an experimental study to characterise blast-induced fragments and to investigate the protective performance of different cellular materials. A front plate was subjected to a localised blast load to generate a cap fragment. The fragments were directed to impact a rear plate, which was placed behind an energy-absorber made of cellular material [65]. The size of the fragments were varied by using different charge size and mass. Figure 2.29 shows the fragments collected post-impact with the rear plate. The speed, the dimensions and kinetic energy of the released fragments were recorded. The damage caused

by the fragment was quantified by the means of the maximum deflection of the rear plate [65].

The measured fragment speeds were shown to be unpredictable for a given charge diameter [65]. At a charge mass of $7g$, the variation in fragment speed was $\pm 96m/s$, with a mean speed of $371m/s$; and the variation increased to $\pm 215m/s$ at $11g$ charge mass, with a mean speed of $630m/s$. Figure 2.30 shows the speed of the released fragments at increasing charge masses at charge diameter of $36mm$. Even though there was considerable experimental variation, an increasing trend in the measured fragment speed with increasing charge mass was clearly evident. The charge mass required for the tearing of the rear plate was not always at the higher end of the charge mass range, probably owing to the inconsistency in the material property of mild steel and the unpredictable, spinning motion of the fragments [65].



Figure 2.29: Various states of fragments after impact with the rear plate [65]

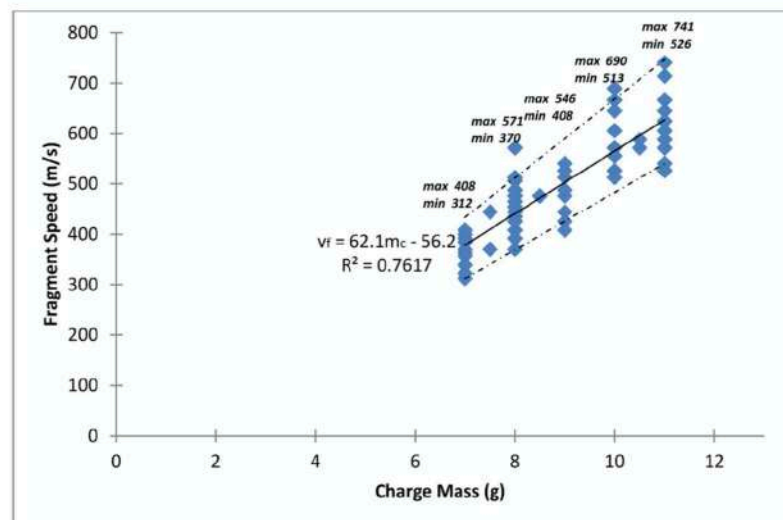


Figure 2.30: Speed of fragment plotted against charge mass for charge diameter of $36mm$ [65]

Teeling-Smith and Nurick [66] studied the tearing of thin circular metal plates subjected to impulsive loads. Steel plates of $100mm$ diameter and $1.6mm$ thickness were clamped between

two support flanges and a sheet explosive was placed on a polystyrene pad in two concentric circular rings and detonated centrally. This arrangement was assumed to produce uniform blast pressure loading on the plate. The velocity of the ejected “plug” (disk fragment) was recorded. The explosive mass and the corresponding velocities of the disc are presented in Table 2.3, as an approximate indication of the range of fragment velocity at a given explosive mass. The variation in the data showed that fragmentation caused by explosive charge detonation was unpredictable in nature.

Table 2.3: Mass of sheet explosive used and the corresponding recorded velocity [66]

Explosive mass (g)	Velocity of disc (Minimum-Maximum) (m/s)
14	242.4 - 271.7
15	250.0 - 292.7
16	210.5 - 300.0
17	266.7 - 324.7
18	285.7
20	315.8 - 362.3
21	375.0
22	393.4
23	375.0
26	423.7

2.3.6 Numerical Modelling of Fragment Velocity Distribution

Grisaro and Dancygier. [67] investigated the velocity distribution of fragments caused by explosion of a cased charge using numerical simulation techniques to simulate the velocity distribution of a cased cylindrical explosive charge. The study used three different numerical approaches in AUTODYN, namely 2D simulations, 3D simulations and 3D simulations using the SPH method. Additional simulation of a pre-formed fragmentation was also investigated. The experimental data from Huang et al. [51] was used as benchmark data for the validation of the numerical model.

Casing rupture into large number of fragments could not be simulated in 2D simulations, as each differential area represented an axisymmetric ring [67]. However, since the distributions of the mass and size of the fragments were excluded from the scope of the study, 2D simulations could be used. Figure 2.31 shows the 2D axisymmetric numerical model, consisting of an air mesh modelled using the ideal gas EOS, and a Lagrangian mesh for the steel casing material modelled using the “Steel 1006” model from the AUTODYN library. The velocity and location of the casing at any given time after detonation were monitored with the numerical gauges.

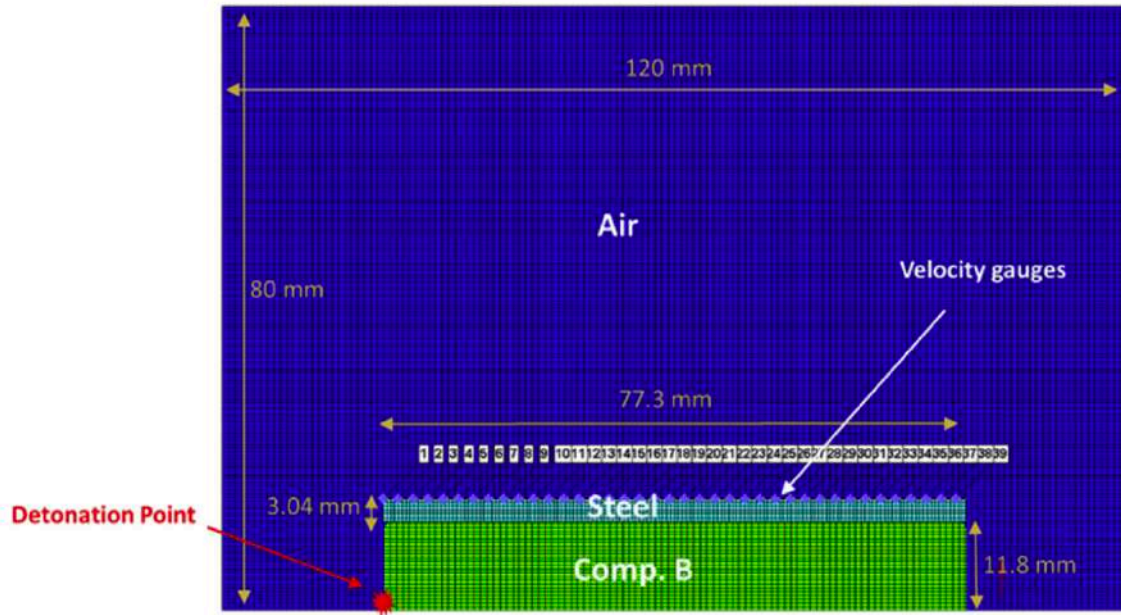


Figure 2.31: 2D axisymmetric numerical model and gauge locations [67]

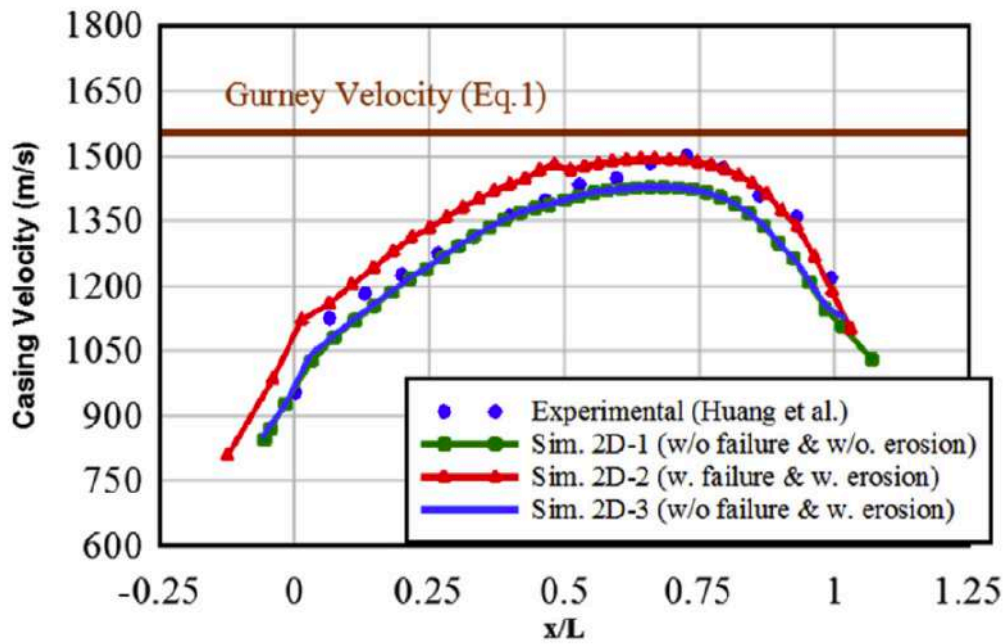


Figure 2.32: Comparison of 2D simulations with experimental data [67]

The results indicated a non-uniform velocity distribution along the cylinder axis. Figure 2.32 shows the comparison between the 2D simulation results (at $t = 0.0329ms$) and the experimental data from Huang et al. [51]. A “good” [67] agreement was shown between the results. The predicted Gurney velocity, also plotted on the same set of axes shown in Figure 2.32, was higher than the experimental and simulation results especially near the cylinder ends. This

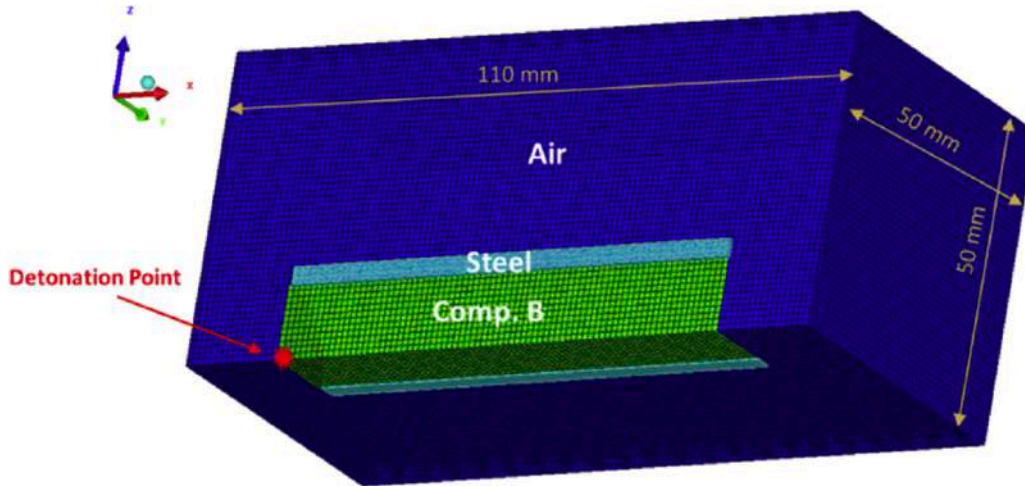


Figure 2.33: 3D axisymmetric numerical model [67]

was expected as the Gurney formula is intended for long cylinders, which makes it unsuitable for the cylinder used in the simulation ($L/D = 2.6$). Grisaro and Dancygier [67] noted that the simulated velocity dropped to lower values outside the original length of the cylinder, a phenomenon which was not addressed by any of the pre-existing formulae. It was also noted that the velocity along the cylinder axis was time-dependent; at a certain point in time, the casing reached its maximum value and continued to expand with constant velocity.

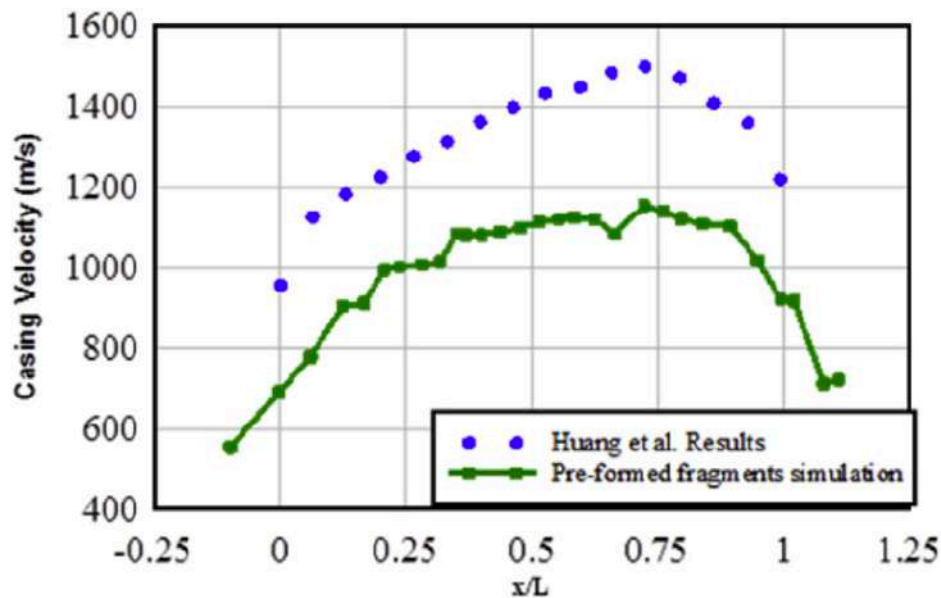


Figure 2.34: Velocity distribution at $t = 0.0329ms$ using pre-formed fragments [67]

Grisaro and Dancygier. [67] also included the 3D simulation of pre-formed fragments. The setup shown in Figure 2.33 was used. The casing was divided into 252 pre-formed, equally

sized fragments with no spaces between them. Despite the similarity in the general trend of the different sets of data, the velocity values obtained from the simulation was significantly lower than the experimental data from Huang et al. [51], as illustrated in Figure 2.34. This was due to the early separation of fragments and the faster dissipation of the explosive energy.

Grisaro and Dancygier. [67] presented a normalised form of the formula by Huang et al. [51] (Equation 2.19).

$$\frac{V_0}{V_{Gurney}} = (1 - 0.361e^{-1.111\frac{x}{d}})(1 - 0.192e^{-3.03\frac{(L-x)}{d}}) = \phi(\frac{x}{L}, \frac{L}{D}) \quad (2.19)$$

The normalised function $\phi(\frac{x}{L}, \frac{L}{D})$ represents a “shape distribution function” [67] of a given cylinder as a function of the L/D ratio. The plot of the normalised formula is shown in Figure 2.35 for a range of L/D ratios. It was shown that for lower L/D values, the velocities observed were much lower than the Gurney velocity prediction. As the L/D ratio increased, however, the velocity along the cylinder length (except in the vicinity of the ends) was shown to approach the Gurney velocity.

Figure 2.36 shows the normalised shape functions, which was obtained by normalising the velocities of each simulation with respect to the corresponding maximum value. It was shown that the normalised results showed good agreement, coinciding into a single “shape function”, including the pre-fragmented simulation despite its lower velocity values. As the ratio L/D was fixed in all simulations shown in Figure 2.34, this showed that the normalised velocity V/V_{max} may be represented as a function of the normalised position x/L .

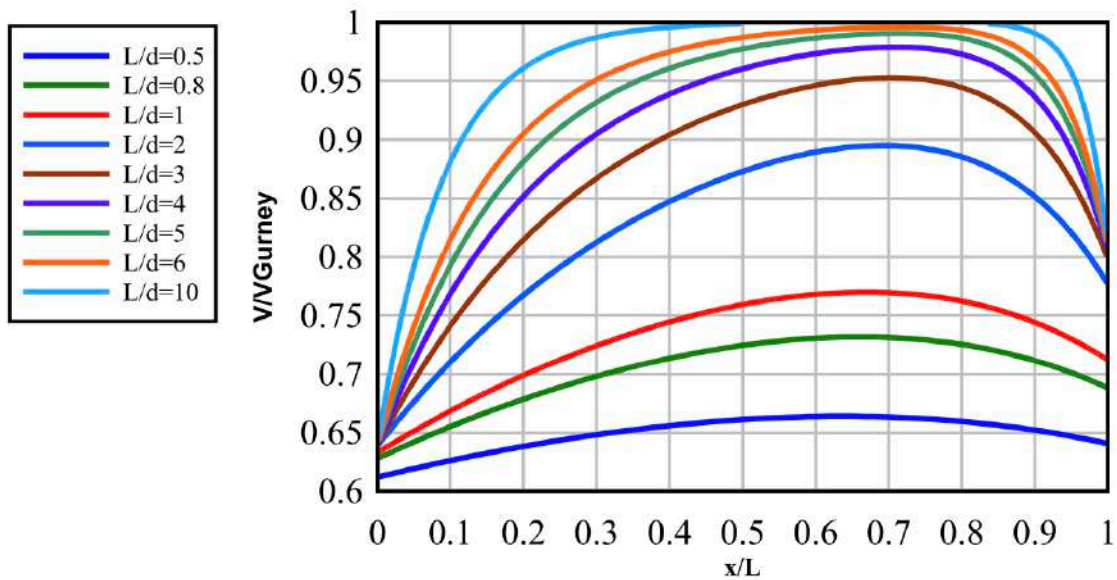


Figure 2.35: Normalised velocity distributions for various L/D ratios [67]

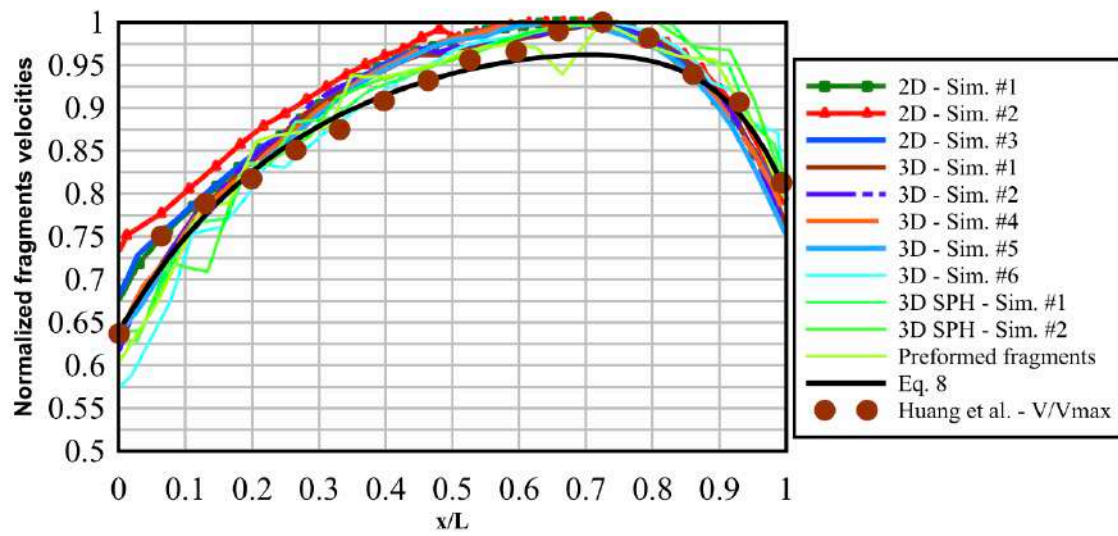


Figure 2.36: Normalised fragments velocities from various numerical simulations [67]

2.4 Projectiles

Shrapnels, especially ball bearings, are analogous to ballistic projectiles travelling at moderate to high speed. It may therefore be useful for the purposes of this project to report on how projectiles interact with their targets, and how the damage inflicted on the target may be quantified using theoretical models.

2.4.1 Projectile Impact

Projectile characteristics may be divided into methods of launching and intended effects. Spherical projectiles exhibit an infinite number of symmetry axes, and are used widely in ballistic laboratory experiments.

Another important structural property of a particular projectile/target combination is the ballistic limit v which is generally defined as an impact velocity where 50% of a series of identical projectiles breach the target [68]. However, there is lack of consensus in what constitutes “breaching” of a target. Some researchers have defined breaching as the point at which the striker projectile nose is visible from the distal side, or that the striker projectile completely passes through the obstacle [68].

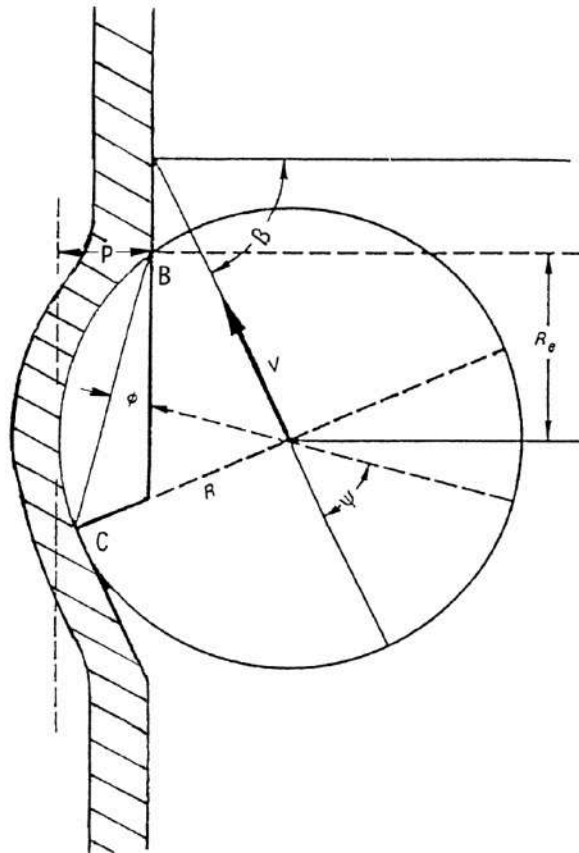


Figure 2.37: Diagram of the effect of forces acting on a rigid sphere striking a deformable plate at oblique angle β [69]

Ideal collision refers to the normal impact of projectile in pure translational motion against a stationary object. Ideal impact situations almost exclusively occur in laboratory conditions only, and in reality target surfaces do not present themselves at normal incidence to the projectile trajectory. Even if the target is stationary, the projectile may be fired from a mobile source that effectively produces obliquity. Many projectiles may experience spin during launching or striker tumbling due to glancing contact with the target.

The angle between the target normal and the trajectory of the projectile is called the oblique angle, and the angle between the target normal and the projectile axis is called the impact angle. In the case of a spherical projectile, the oblique angle and the impact angle are used interchangeably [69]. Angle β , shown in Figure 2.37, represents the oblique angle and the impact angle.

Oblique impacts on targets at angle generate asymmetric loading, which rotates the striker projectile. This may produce additional stresses and cause projectile fracture, which can increase the ballistic limit by 30%. [68]

Finnigan et al. [70] conducted experiments involving the impact of mild steel spheres of 6.35mm diameter projected at obliquity angles between 0° and 75° at plates of 2024-T3 aluminium, stainless steel 304, and mild steel at various speeds ranging from 0.2km/s to 1.2km/s. Figure 2.38 shows the plot of ballistic limits and target breakup onset of the metal plates as a function of obliquity angle. As the obliquity increased, the ballistic limit increased while the target breakup limit decreased. In previous studies by Backman and Finnigan [71], it was reported that high oblique impact of spheres on thin aluminium strips produced cracking at speeds below the ballistic limit, which was comparable to the trend reported by Finnigan [70].

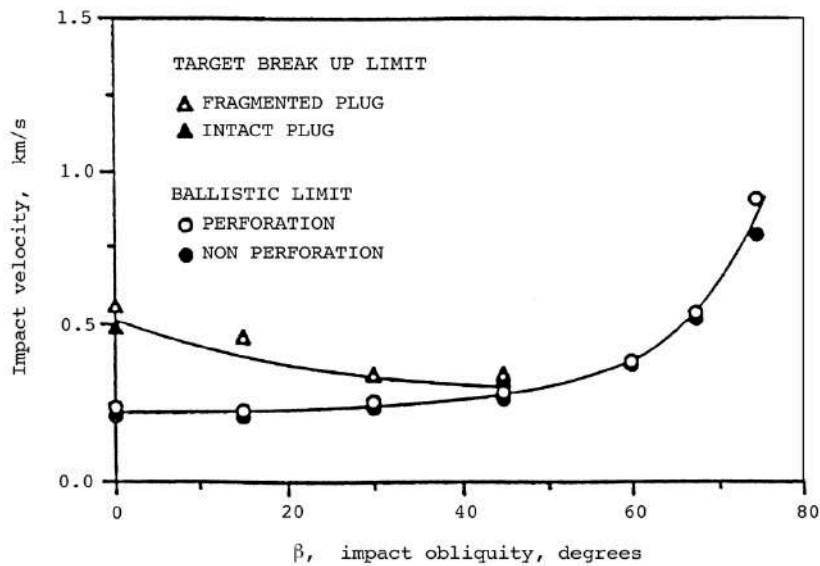


Figure 2.38: Ballistic limit and target breakup limit for 6.35mm diameter mild-steel spheres striking 1.59mm thick 2024-T3 aluminium [70]

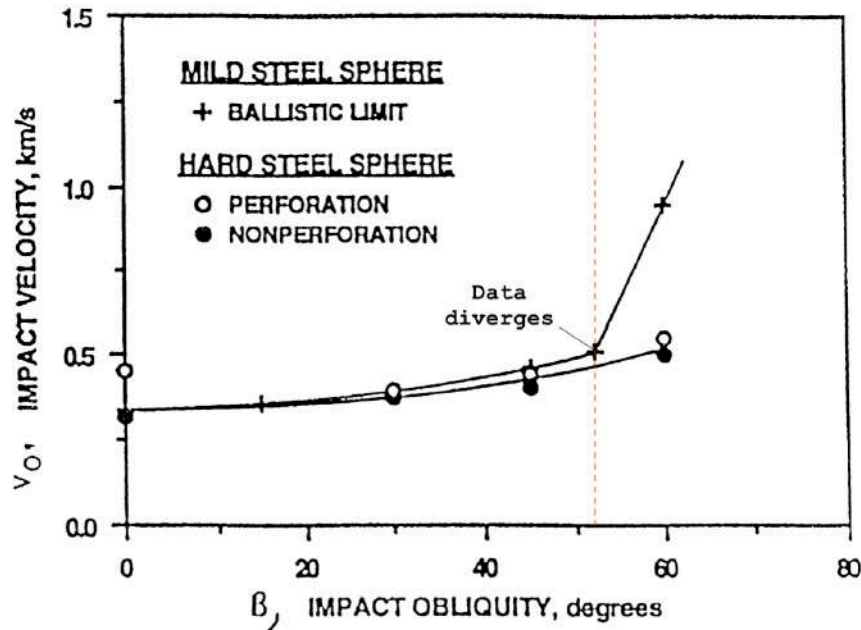


Figure 2.39: Comparison of ballistic limit values as a function of obliquity for 1.50mm thick mild steel targets impacted by 6.35mm diameter hard- and mild-steel spheres [70]

The result of hard and mild steel sphere impact on mild steel target is shown in Figure 2.39 . The material behaviour for the different spheres was approximately the same for an obliquity angle of up to around 50°. Beyond an oblique angle of 50°, the data diverged. The sphere experienced minimal shape changes upon contact with the aluminium, but exhibited noticeable deformation when striking the two steel plates, as shown in Figure 2.40.

Kivity [72] compared the numerical predictions of both the DYTRAN and the AUTODYN codes using experimental results involving the impact of Armco 80 iron and hard steel (ball bearings) against hard-rolled homogeneous plate . The projectile was 15mm in diameter, and had an impact velocity of 1.5km/s. The obliquity angle was kept constant at 70°. The target dimension was 200 × 100mm × 6mm thickness. The experimental conditions represented the impact of 1) an elasto-plastic sphere against a rigid target, and 2) a brittle striker against an elasto-plastic deformable target. The AUTODYN results are presented in Figure 2.41 at three points in time after initial contact. The impact of "soft" sphere against rigid target resulted in a significant deformation of the projectile, where the sphere deformed plastically as it "slid" against the target. A very shallow crater was formed. The hard sphere impact against deformable target generated substantial projectile fragmentation and target crater formation.

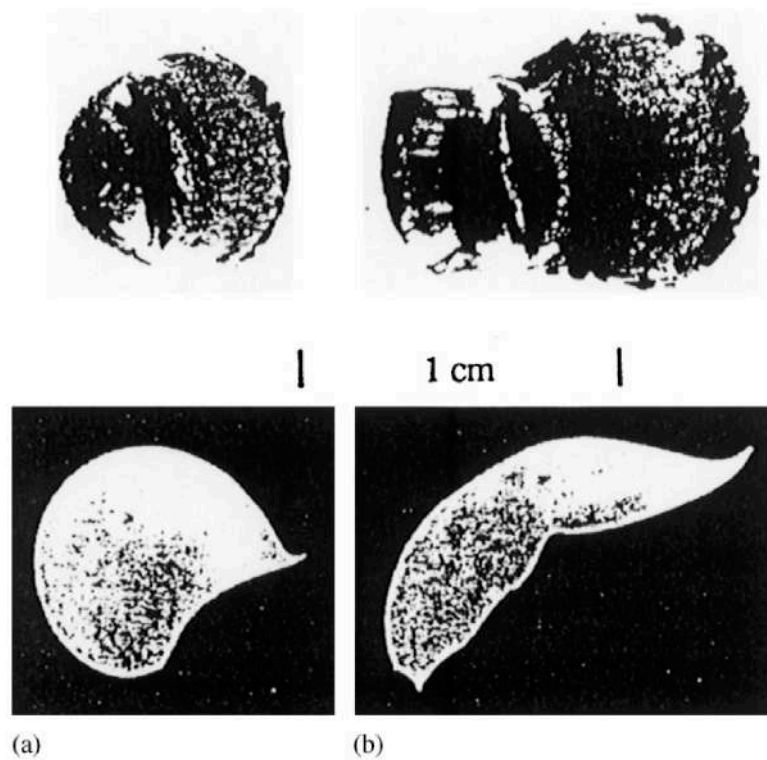


Figure 2.40: Photographs of projectile deformation patterns parallel (top) and perpendicular (bottom) to the impact surface for 60° impacts of 6.35mm diameter mild steel spheres against 1.5mm thick mild-steel targets at a) 0.70km/s and b) 0.91km/s . Direction of travel = left to right [70]

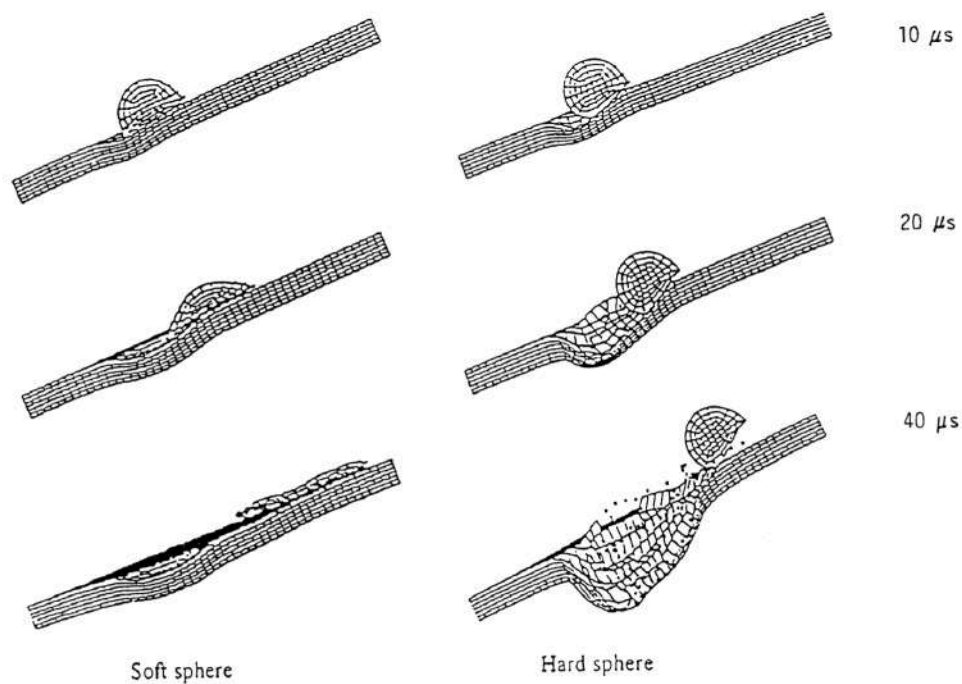


Figure 2.41: Impact of 15mm diameter Armco 80 iron and hard-steel spheres against hard-rolled homogeneous armour at 1.5km/s and an oblique angle 70° AUTODYN calculations at 10 , 20 and $40\mu\text{s}$ after impact [72]

2.4.2 Damage Modes and Target Characteristics

The Meteoroid damage assessment report by NASA [73] summarised the perforation and spallation of thin plates. When a thin, ductile metal plate was perforated by a solid, compact projectile at low impact velocities, a plug of approximately the same diameter as the projectile was punched out. However, when the thickness of the plate was large relative to the crater dimensions and the impact velocity was high, the exit surface was not affected at all.

At reduced thickness, a bulge appeared on the opposite face when the plate was impacted, and fragments of the disturbed material were formed. These fragments are referred to as detached spall, and the condition as spall threshold [73]. When the spallation cracked through the thickness of the target, the hole had an approximate shape of an hourglass. The hole was increasingly cylindrical as the thickness decreased. Figure 2.42 illustrates the depth of penetration and spall for 2024-T3 Aluminium target of varying thickness.

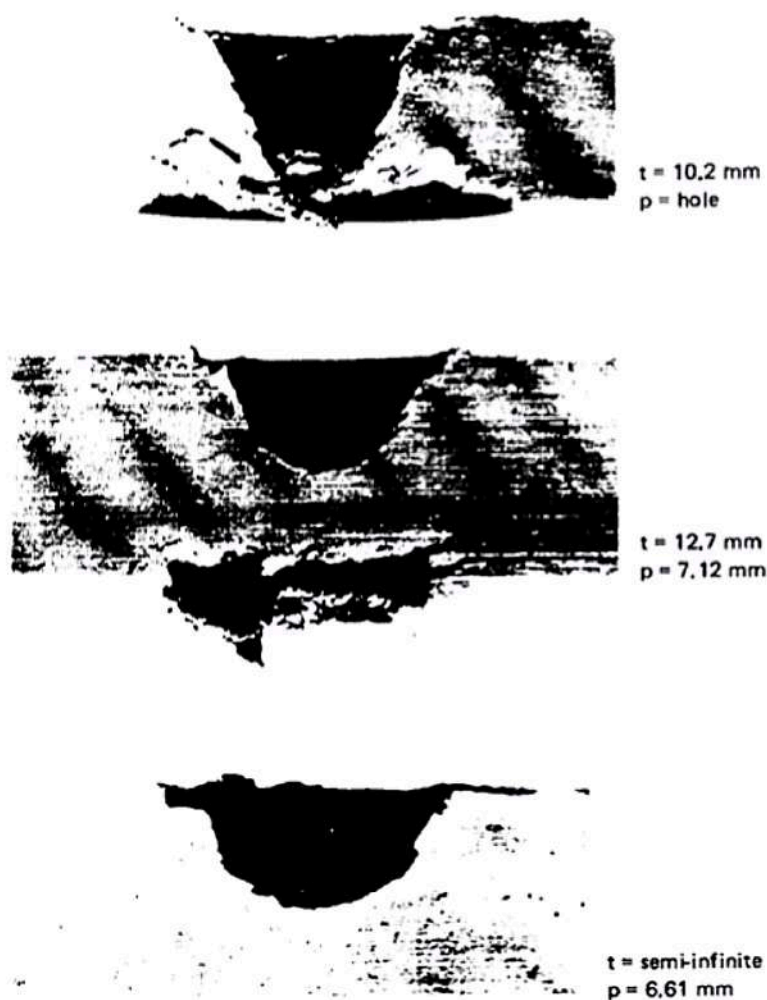


Figure 2.42: Effect of thickness on depth of penetration and spall of 2024-T3 aluminium targets caused by 3.2mm diameter aluminum projectiles at 7.4km/s projectile velocity [73]

Semi-infinite bodies have lateral and thickness dimensions that are large with respect to the dimensions of the impact crater [73]. Due to its simplicity, this type of structure is often used in theoretical and experimental investigations. In a semi-infinite body, the significant damage is limited to the depth of crater and solid compact projectiles in the low-velocity regime form a narrow cylindrical crater by mechanical deformation.

It was stated in the Meteoroid damage assessment report that if the depth of penetration was limited to 25 percent or less than that of the plate thickness, the structure was considered semi-infinite [73]. Fish and Summers [74] also reported that it was customary to predict that the target would be perforated by a projectile if its thickness was 1.5 times the crater depth that the projectile would make in a semi-infinite body.

2.4.3 Target Penetration Depth Design Equations

Design equations are useful for estimating the penetration thickness of thin plates, as well as semi-infinite bodies. The thin-plate equations may be used to determine the spallation and ballistic limits, i.e. the minimum threshold point of spallation and penetration respectively. However, it is worth noting that prevention of spallation and/or deformation will require thickness much larger than the threshold dimension.

Of the 5 equations presented by Hayashida and Robinson [75], the Fish-Summers equation and the JSC (Modified Cour-Palais) equation were shown to be the most accurate, correctly predicting the penetration result for five out of six tests.

Fish and Summers [74] summarised the results obtained from a parametric study of thin-sheet penetration conducted at the Ames Research Centre at NASA. In these tests, 2017-T4 aluminium alloy balls of 1.59mm diameter were used as standardised projectiles. The aluminium balls were sabot-launched from 22-caliber light gas guns. The time and space coordinates of the projectiles were recorded by shadowgraph stations. The velocity in the tests ranged from 0.5 to 8.5km/s. The Fish-Summers Equation, was derived from the results [74] is given by Equation 2.20 [75].

$$t_t = K_1 m^{0.352} V_n^{0.875} \rho^{\frac{1}{6}} \quad (2.20)$$

where t_t is the target thickness (cm), K_1 is a constant for the target (0.57 for aluminium alloys; values for other metals may be found in [75], m is the projectile mass (g), ρ is the projectile density (g/cm³) and V_n is the impact velocity (normal component of the projectile relative velocity (km/s)).

In Equation 2.20, target material properties are considered by the constant K_1 , and a 0.70 factor is used to determine the plate thickness to prevent penetration from spalling.

The JSC (Modified Cour-Palais) Equation, modified from the equation for the Apollo project

by Burton Cour-Palais at JSC, is given by Equation 2.21 [75].

$$p = 5.24d^{\frac{19}{18}}BH^{-0.25}\left(\frac{\rho_p}{\rho_t}\right)^{0.5}\left(\frac{V_n}{C_s}\right)^{\frac{2}{3}} \quad (2.21)$$

where p is the crater depth on target (cm), d is the projectile diameter, ρ_p is the projectile density (g/cm^3), ρ_t is the target density (g/cm^3), BH is the Brinell hardness of the target, V_n is the impact velocity (normal component of the projectile relative velocity (km/s)) and C_s is the speed of sound, $\sqrt{\frac{E}{\rho_t}}$. t_b , the target thickness for ballistic limit, is equal to $1.8p$; similarly, t_{sp} , the target thickness for spallation limit, is equal to $2.2p$.

The semi-infinite body equation is used when designing a structure where the depth of penetration is limited to a small fraction (less than 25%) of the plate thickness. The empirical equation for predicting depth of penetration in a semi-infinite body as recommended by [73], is given by Equation 2.22.

$$P_\infty = K_\infty m^{0.352} \rho_m^{\frac{1}{6}} V_n^{\frac{2}{3}} \quad (2.22)$$

where P_∞ is the depth of penetration (cm), K_∞ is a constant for the target (0.42 for aluminium alloys; values for other metals may be found in [75]), m is the projectile mass (g), ρ_m is the projectile density (g/cm^3) and V_n is the impact velocity (normal component of the projectile relative velocity(km/s)).

2.4.4 Crater Geometry

Nishida et al. [76] examined the effects of projectile diameter and impact velocity on length distributions of fragments. Spherical projectiles made of aluminum alloy 2017-T4 with diameters of $1.6mm$, $3.2mm$ and $7mm$ were impacted upon thick targets made of aluminium alloy 6061-T6 at impact velocities ranging from 2 to $7km/s$. Fractured projectiles collected from test chamber after impact experiments were referred to as fragments.

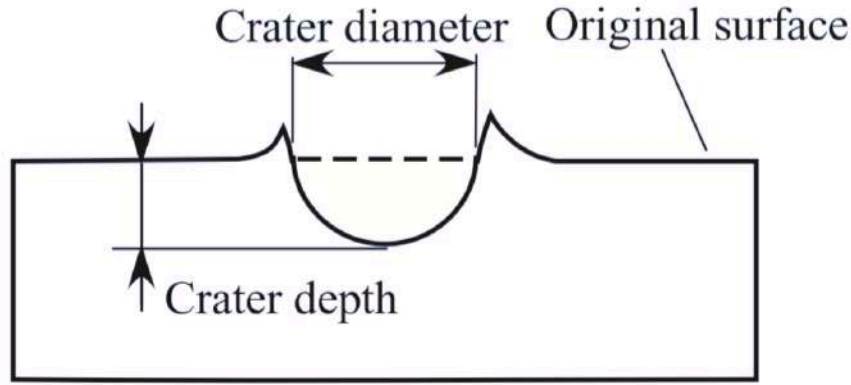


Figure 2.43: Definition of crater diameter and depth [77]

It was observed that when both sides of the JSC equation (Equation 2.21) were raised to the third power, the cube of crater depth was proportional to the square of the impact velocity. The craters were almost hemispherical, and crater volume were directly proportional to the kinetic energy of projectiles when the spherical projectiles struck the aluminium targets at velocities over 2km/s [77]. Figure 2.43 shows a schematic of the crater shape with the crater diameter and depth. Nishida et al. [76] speculated that due to the hemispherical shape of the crater, the cube of crater depth was roughly equivalent to the crater volume, and the crater volume was closely related to the total amount of fragment mass. Therefore, it was possible that the square of the impact velocity was related to the total amount of fragment mass [76]. However, the total amount of fragment mass or fragment volume could not be determined solely from fragment size and mass distribution, because the fragment size and mass were dependent on the fracture mechanism of the materials.

2.5 Summary

2.5.1 Improvised Explosive Devices

The key components of IEDs include a switch and its power source, a detonator, a main charge, and a container. Most IEDs contain projectiles such as ball bearings or bolts and nuts that act as shrapnel, used to enhance the fragmentation effect and damage [3, 7]. IEDs also affect an area with blast, heat and fragmentation. Upon explosion, the casing as well as the projectiles are propelled away from the detonation point at high velocities, causing significant damage to targets [13].

2.5.2 Detonation Physics

Blast loading is characterised by an instantaneous increase in pressure from the ambient pressure to peak pressure, followed by an exponential decay in magnitude towards atmospheric pressure [20]. Pressure reflections occur when the blast wave impinges on a surface. The magnitude of the reflected pressure wave is greater than the incident wave. The reflected pressure is increased as the number of reflective surfaces increase, such as at the internal corners or vertices of structures [25]. Blast loading is broadly categorised as unconfined and confined [24]. An unconfined blast is characterised by a single interaction with a blast wave, whereas a confined blast generates multiple high magnitude shock loads. The resultant damage to the structure increases as the degree of confinement increases [24]. The explosive charge geometry and dimension, as well as the container geometry have an effect on the peak overpressure [35, 36, 38, 39].

2.5.3 Fragmentation

Shrapnel from IED explosions are analogous to blast-induced fragments, as it also travels at high speeds propelled by blast, and cause additional damage to surrounding structures. Gurney [44] proposed a formula to estimate the initial fragment velocities of encased charges as a function of the charge and casing mass, based on a number of simplifying assumptions. The formula neglects both the rarefaction wave effect and the effect of the wave incidence angle, and must be used with caution as it may produce inaccurate results. Experimental results have shown that in cylindrical charges, the fragment velocity decreases significantly towards the charge ends ("end effect") [46–48]. More recent, modified versions of the Gurney formula are able to predict the rarefaction "end effects" relatively accurately [46, 50, 51].

Fragments absorb significant amount of energy during blast, and the blast generated by an encased charge is typically weaker than that of a bare charge of the same mass [59]. When the casing is pre-fragmented, the blast load is not reduced as significantly, as the detonation energy escapes through the air gaps instead of being transferred to the fragment. However, for this reason, the fragment velocity is decreased when compared to a solid casing [45].

2.5.4 Projectiles

Ball bearings found in IEDs are analogous to spherical ballistic projectiles travelling at moderate to high speed. The ballistic limit, defined as the velocity at which the projectile breaches at least 50% of the target thickness, increases as the target impact angle increases [68]. Craters formed by spherical projectiles are typically hemispherical, and numerical simulations have shown that hard projectile impact against deformable target generates substantial projectile fragmentation and target crater formation [72]. Design equations such as the Fish-Summers [74], JSC [75] and semi-infinite equations [75] are useful for estimating the penetration thickness of plates given a projectile velocity.

2.5.5 Motivations to the Current Study

Based on the findings of the literature review, it has been shown that it is impossible to replicate the exact effects of an IED both experimentally and numerically as there are no two IEDs that are similar. IEDs in itself are unpredictable as they can be found in unlimited varieties not only in shape and sizes but also using different types of explosives. Furthermore, foreign objects incorporated into the IEDs also come in different materials, shapes, sizes and quantities and are often located at random places within the charge. Consequently, the resulting fragmentation from IEDs can occur in any shapes and sizes travelling at any unpredicted velocities. Numerical modelling is, therefore, the most effective way to gain better understanding in behaviours of IEDs and its subsequent effects on the surrounding. To model this complex phenomenon, numerous studies have simplified the IEDs for basic understanding. Key components of an IED are not included in any current numerical models. The resulting damage from the detonation of an IED are also not isolated and identified in terms the effects of blast loads or high temperatures or fragmentation. Foreign objects are simplified as ball bearings and the target as a flat plate. These numerical models are validated against simple experiments.

Chapter 3

Experimental Design

The experimental design and setup by Kang [6], which the numerical study is based on, are discussed in this chapter.

3.1 Test Specimens

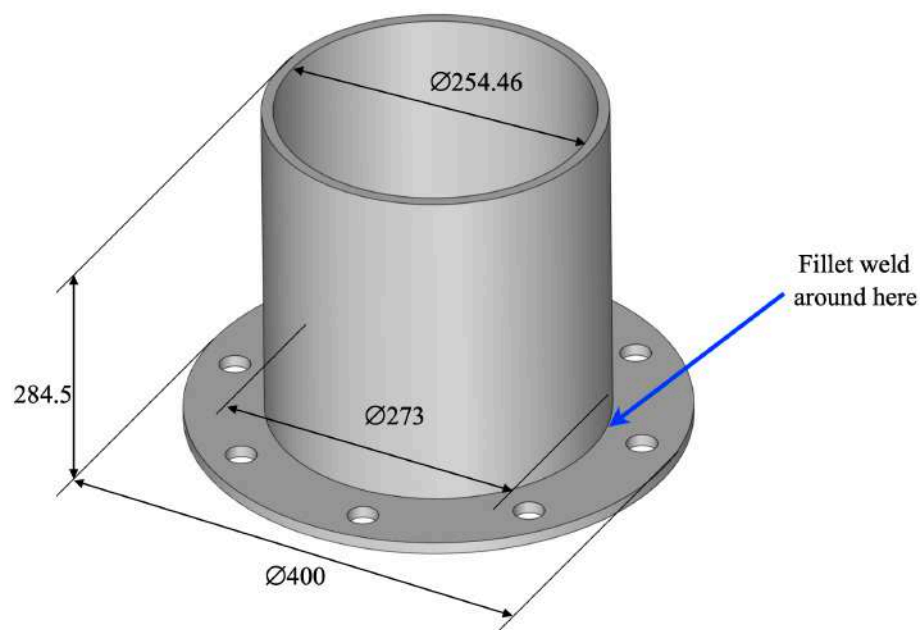


Figure 3.1: Schematic of dimensions of the container and the location of fillet welds

Confined mild steel cylindrical containers were selected for the experiments not only because of the ease of manufacture but also for safety reasons to capture the uncontrolled scattering of shrapnel. The explosion was therefore fully confined, and the cylinder walls were thick enough

to prevent bulging of the cylinder upon blast loading. The wall thickness of the cylinder was estimated based on existing literature. The estimated penetration depth was calculated using the Fish-Summers equation (Equation 2.20), the JSC equation (Equation 2.21) and the semi-infinite equation (Equation 2.22). The cylindrical container had an external diameter of 273mm and a wall thickness of 9.3mm . The cylinder inner diameter used was 254.46mm (equivalent to 10 inch), and was chosen from a preset value provided by the supplier. The flange had outer diameter of 400mm , and inner diameter of 273mm . The body of the container had a length of 284.5mm length. The flange was fitted over one end of the cut pipe and welded onto it with 6 short fillet welds around the circumference. Figure 3.1 shows the schematic of dimensions of the container, as well as the circumference around which the fillets were made. The aspect ratio (L/D) of the confined volume inside the cylinder was equal to 1. Detailed engineering drawings of the parts used in the experiments have also been included in Appendix A for reference.

3.2 The Explosive Device

The base charge consisted of 22g of plastic explosive (PE4) shaped in a cylinder that had an aspect ratio (L/D) of 1. The diameter and length of the charge were 26mm, approximately 10% of the inner diameter of the cylinder (254.46mm).

Spherical steel ball bearings, 5mm in diameter, were added to the explosive to create the “IED” that would produce approximately hemispherical indentations on the target. The AISI 420 stainless steel ball bearings were chosen, as they are readily available off the shelf. The ball bearings were chosen so that they were large enough to generate enough momentum to leave appreciable indentations on the cylinder wall, and to not shatter nor melt upon the detonation of the explosive.

Different experiments were carried out using charges with varying numbers of ball bearings arranged in different configurations to provide a range of response. The ball bearings were either packed around the cylindrical charge in row(s) secured by scotch tape, as shown in Figure 3.2 a), or were randomly embedded into the charge, as shown in Figure 3.2 b). In total, 20 ball bearings could fit in a single row around the charge circumference. Figure 3.3 shows a schematic of the different number of ball bearings placed around the charge.

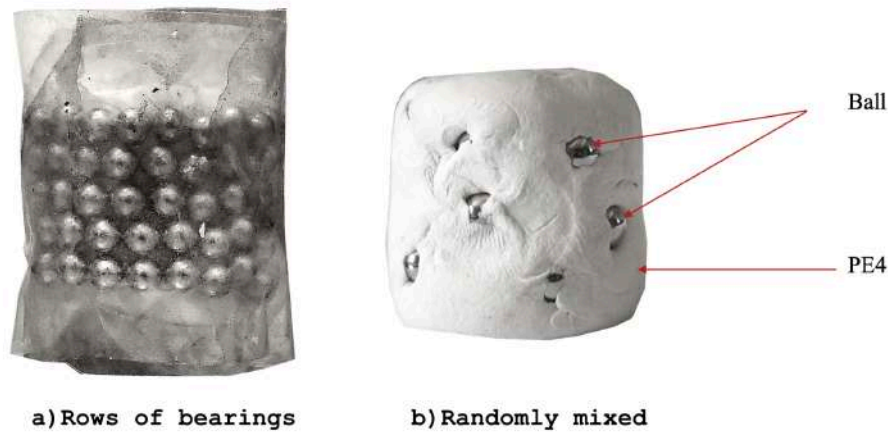


Figure 3.2: Arrangement of ball bearings around PE4 charge

A baseline test was conducted with a bare charge (with no ball bearings) to evaluate the response of the confined target containers to the explosive load only. In total, 7 tests, as summarised in Table 3.1, were conducted. In all cases, the charge was mounted onto a column of polystyrene and placed and detonated in the centre of the cylindrical target in a fully confined environment.

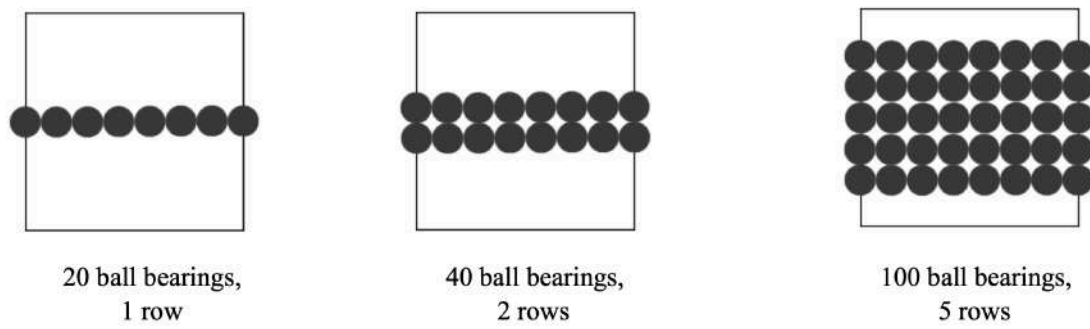


Figure 3.3: Schematic of the different number of ball bearings placed around the charge

Table 3.1: Summary of experiment setup data

Test Number	Charge condition	No. of ball bearings	Mass ratio (PE4:Steel)
1	Bare charge	0	1:0
2	1 row	20	2:1
3	2 rows	40	1:1
4	Random	40	1:1
5	Random	40	1:1
6	5 rows	100	2:5
7	Random	20	2:1

3.3 Test Setup

A backing plate of thickness 20mm was fitted onto the mounting plate of the blast pendulum with four spacers and secured with four M12 bolts. No impulse measurements were taken. The container was loaded onto the pendulum for the sake of mounting. A cylindrical boss of diameter 254.46mm and thickness 50mm was screwed onto the backing plate, leaving an internal circular boss of 30mm for the container to fit onto. The experimental setup is shown in Figure 3.4.

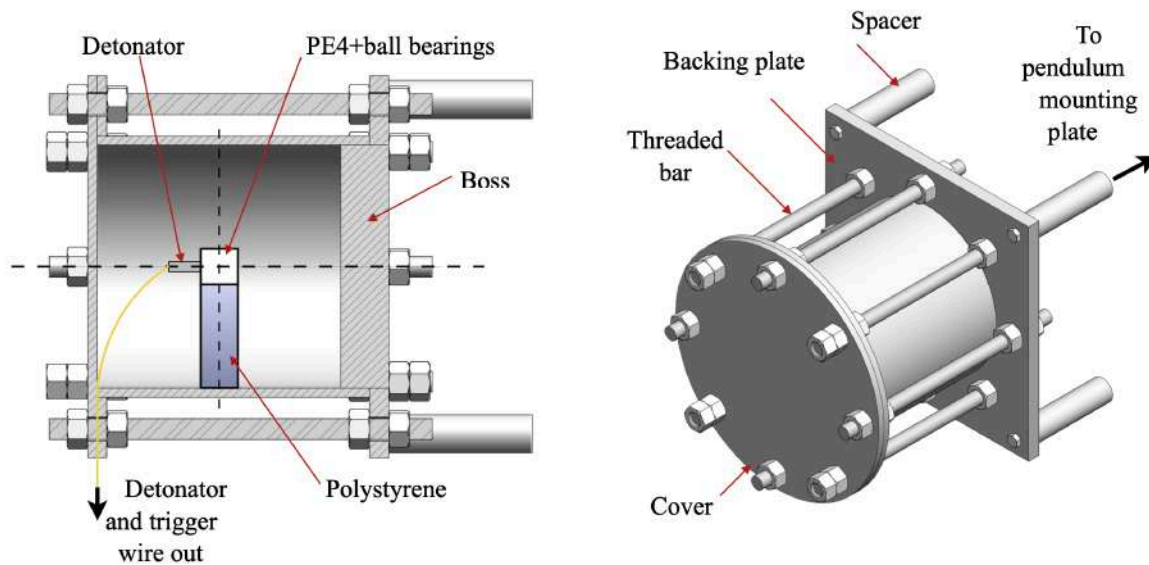


Figure 3.4: Schematic of the charge placement within the confined cylindrical container and the fully confined cylindrical container setup



Figure 3.5: Photograph of the placement of the polystyrene stand holding the cylindrical charge

The flange-less end of the cylindrical container was then fitted onto the boss. The region fitted over the boss was 30mm long, leaving a 254.5mm deformable section (effective L/D ratio of 1). M24 threaded bars of length 400mm were fitted through the holes in the flange of the cylinder and through the holes in the backing plate to secure the cylinder in place. When the container was correctly positioned and locked into place, the explosive charge prepared with ball bearings was positioned on top of a piece of polystyrene located in the centre of the container. Double sided tape secured the polystyrene column to the inside of the cylinder. A photograph of the polystyrene column holding the charge is shown in Figure 3.5. Once the charge was secured in place, the cover plate was placed over the flange end of the cylindrical container and secured with M24 bolts. No seal was applied between the flange and the cover plate. The detonator trigger cable was fitted between the cover and the flange. Photographs of the completed setup are shown in Figure 3.6.



Figure 3.6: Photographs of the completed experimental setup

Chapter 4

Experimental Results

4.1 Response of the Cylinder to Blast

No significant damage was observed in the fully confined cylindrical container when exposed to the “bare” charge. The wall of the cylinder showed no visible permanent deformation.

After the blast, the ball bearings were collected inside the cylinder. A majority of the ball bearings had undergone significant irreversible deformation, from either the blast load itself or the impact from hitting the cylinder wall or both. Some of the ball bearings were observed to have simply cracked on the surface and not entirely lost their spherical shapes whilst some had been compressed into disk-shapes and had petalled, or even completely shattered.

The ball bearings had also blackened, losing their metallic lustre. The degree of damage was more significant when a larger number of ball bearings were involved the test. More significant damage was also observed under random charge conditions than in row charge conditions. Figure 4.1 shows the samples of ball bearings pre-blast and after the blast.

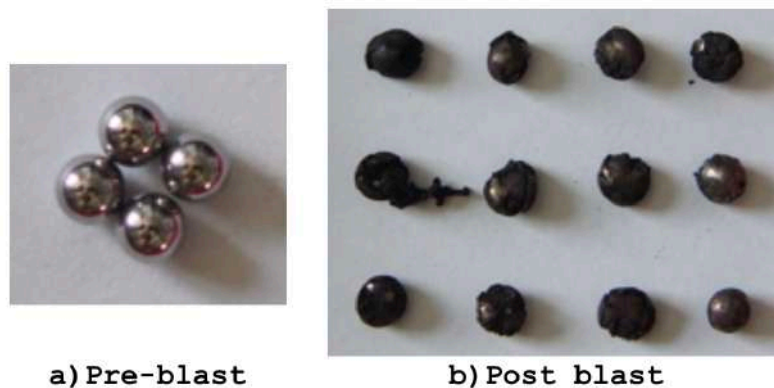


Figure 4.1: Samples of the ball bearings



Figure 4.2: Craters on the inner wall of the cylindrical container from test #4 (Random test)

For the “mixed” charge, visible damage in the form of hemispherical dents, shown in Figure 4.2, was observed on the cylinder wall caused by the ball bearings. No visible permanent deformation of the cylindrical container was observed. The distribution patterns and depth of penetration of the different craters formed on the cylinder wall was mapped out and analysed for each test.

The cylinder specimen from each test was cut in half after it was dismounted from the pendulum. The photos of the halved cylinder specimens have been included in Appendix B for reference. The flattened 2D profile of the distribution pattern of the craters was obtained by manually marking the location of each crater onto paper tape stuck on the inner wall of the cylindrical container post the blast test. The distance between adjacent craters, as well as the height of each crater from the flange-less end of the container were measured and recorded. The data was then plotted as a true 2D profile of the crater distribution pattern, as shown in Figures 4.3 to 4.8 for tests number 2 to 7 as summarised in Table 3.1.

4.2 Analysis of Experimental Data

4.2.1 Crater Formation on Cylinder Wall

For analysis, the cylinder profiles were divided into 3 equal sections of 84.7mm each (lower, middle and upper), the upper end being the flanged end, as labelled in Figure 4.3. Figures 4.3, 4.4 and 4.5 show the distribution of the craters obtained from test 2 (1 row of ball bearings), test 3 (2 rows of ball bearings) and test 6 (5 rows of ball bearings) plotted in graphical form respectively. The vertical dotted line on the graph indicates the line along which the cylinder was cut.

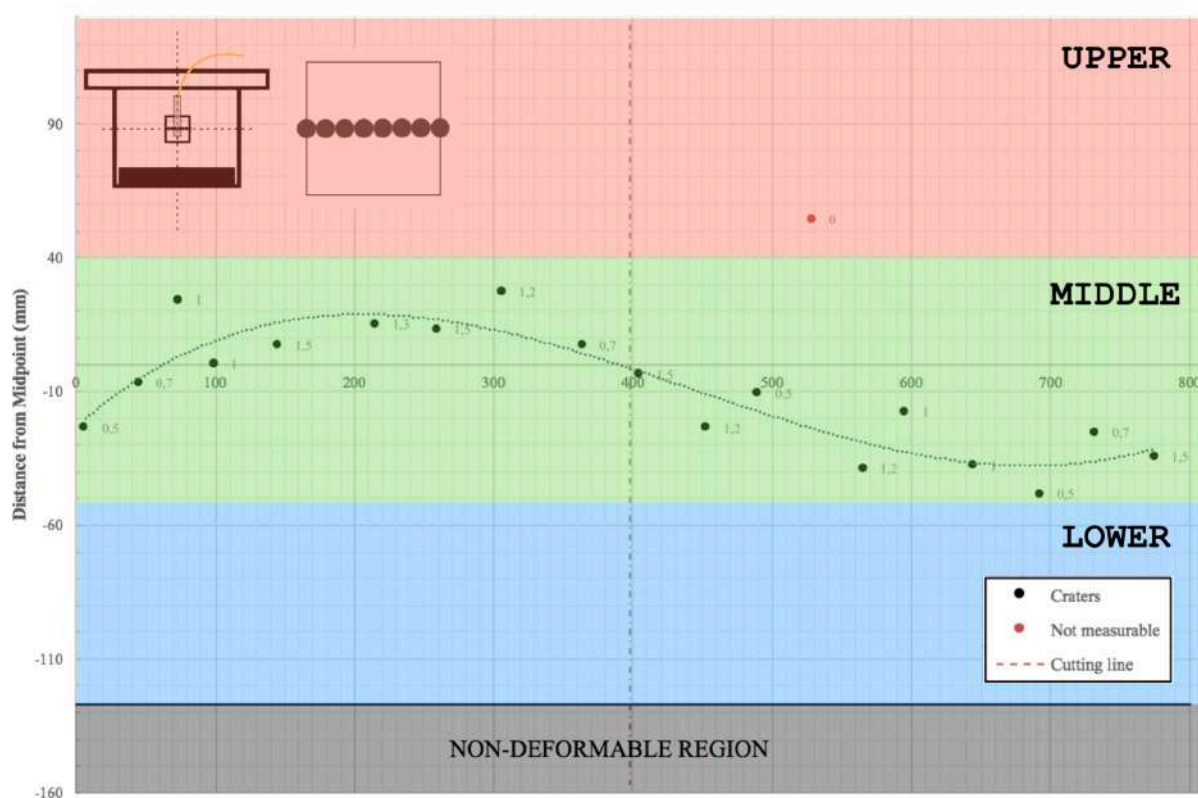


Figure 4.3: Distribution of craters from 1-row test

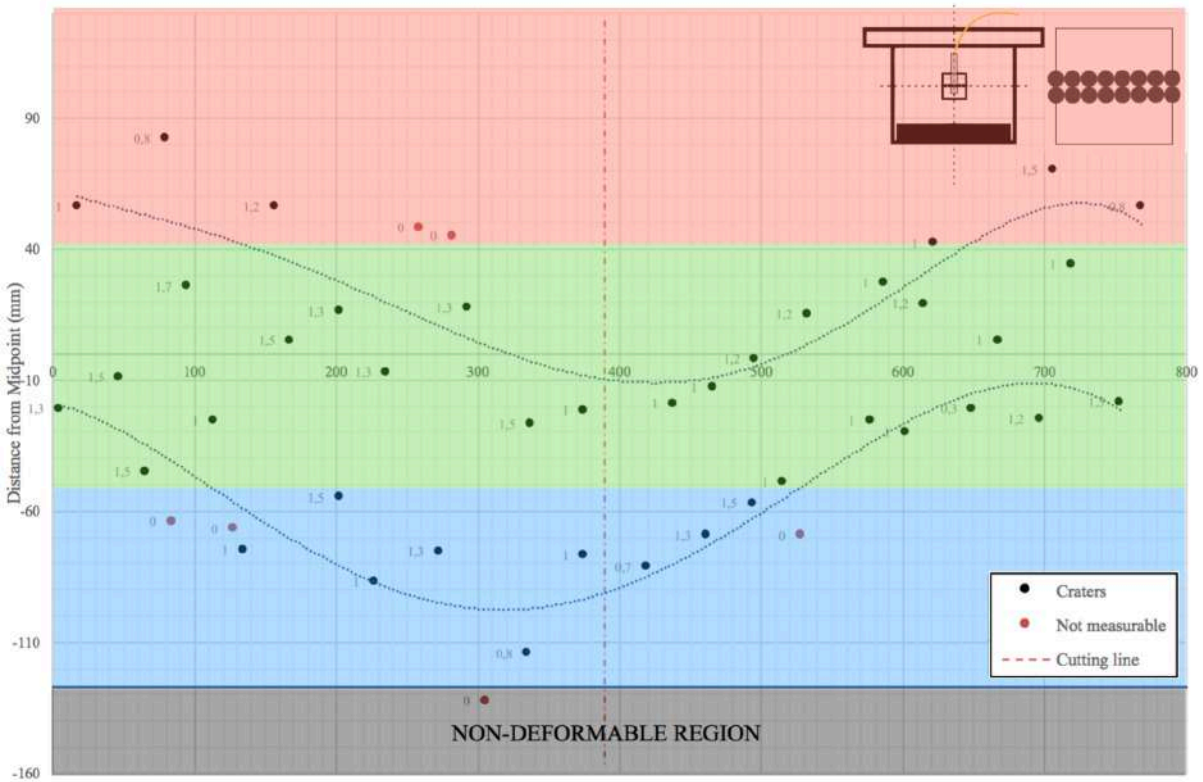


Figure 4.4: Distribution of craters from 2-rows test

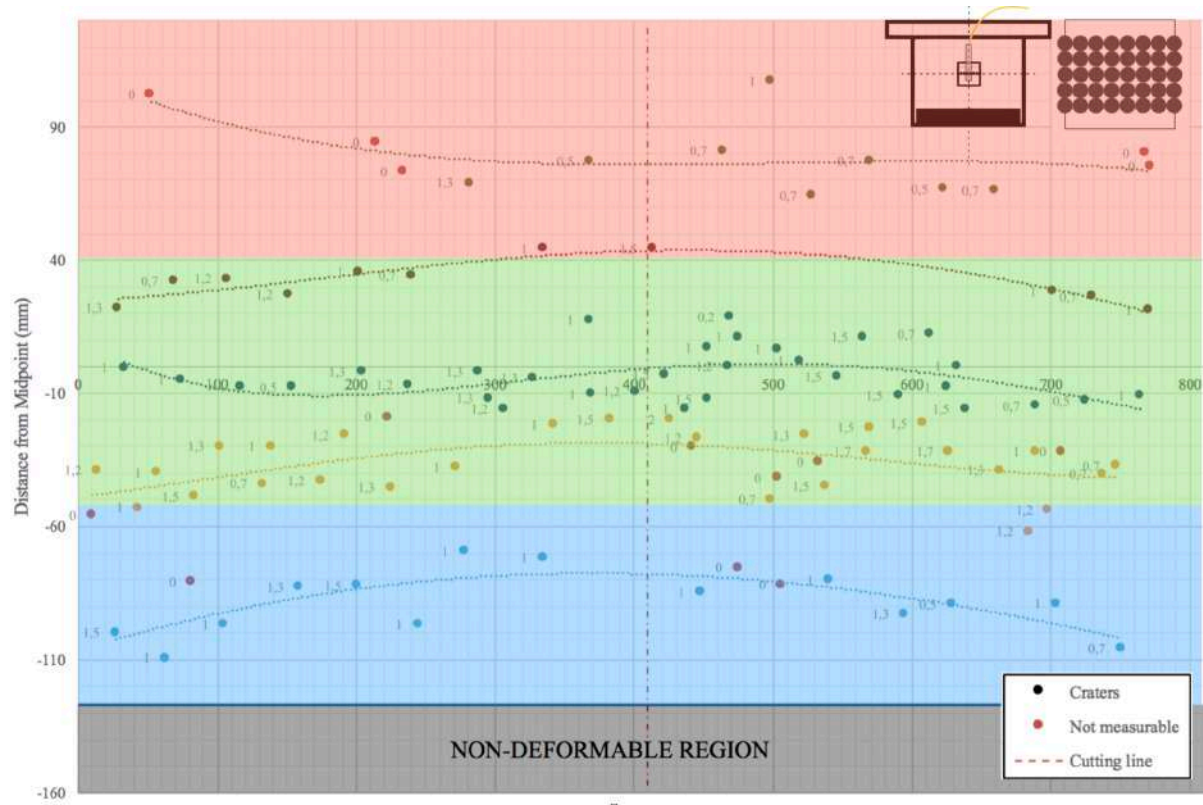


Figure 4.5: Distribution of craters from 5-rows test

The results showed distinct lines of craters to indicate the number of rows of ball bearings attached to the explosive charge. Ideally, if the charge and the rows of ball bearings were to be perfectly aligned with the half-length of the container, the craters would have formed a straight line on the wall (which would form a circle if joined at the two ends). Similarly, if the charge was perfectly shaped, placed centrally inside the cylinder and detonation occurred in the centre of the charge the depth of all the craters would ideally have been uniform since the ball bearings would have been travelling the same distance at the same velocities before they hit the cylindrical target wall. However, this is difficult to achieve. The curved shape of the lines of best fit through the crater data points implied that the charge and/or the ball bearings were slanted and positioned off-centre. The angle at which the detonator was inserted may also have affected the pattern. The observation, especially from Figures 4.4 and 4.5 still confirmed that packing the ball bearing in rows would create a row-like pattern on the target structure itself.

In contrast, the distribution patterns of the craters obtained from random-charge condition tests 4, 5, and 7 shown in Figures 4.6, 4.7 and 4.8 were more chaotic and random as expected. The results showed the unpredictability and danger of blast loading. When the explosive was packed randomly, the formation of craters on the surface of the cylindrical container was equally random. No distinguishable pattern could be extracted.



Figure 4.6: Distribution of craters from random-charge test (40 ball bearings, Test #4)

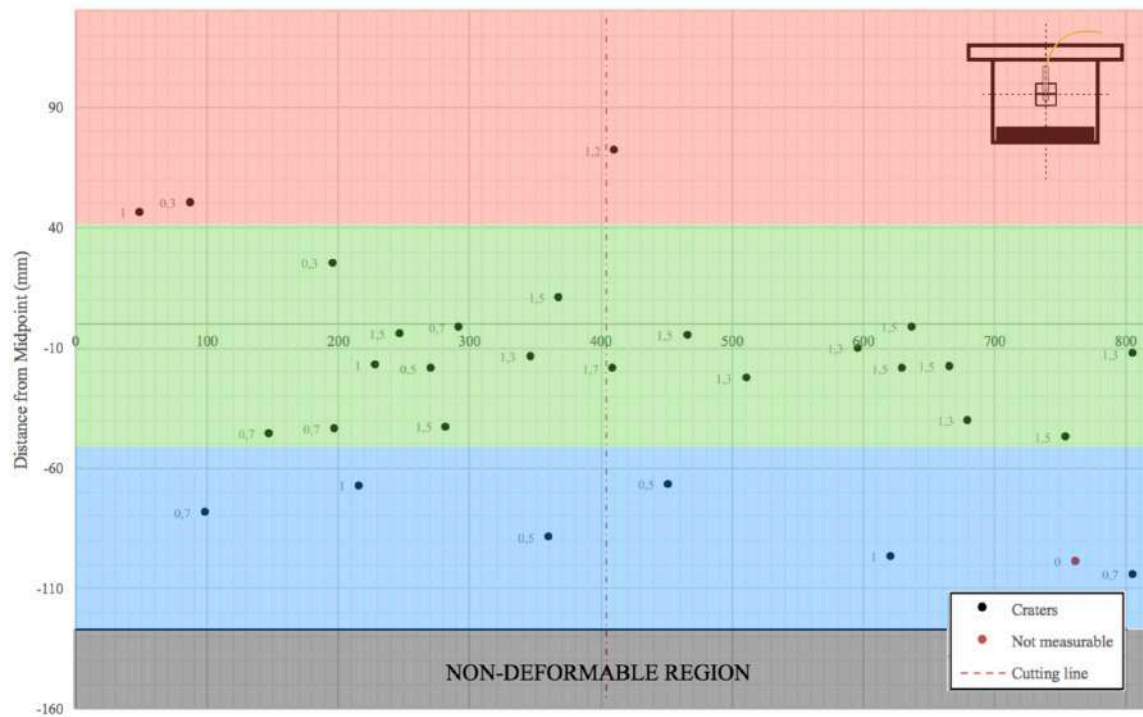


Figure 4.7: Distribution of craters from random-charge test (40 ball bearings, Test #5)

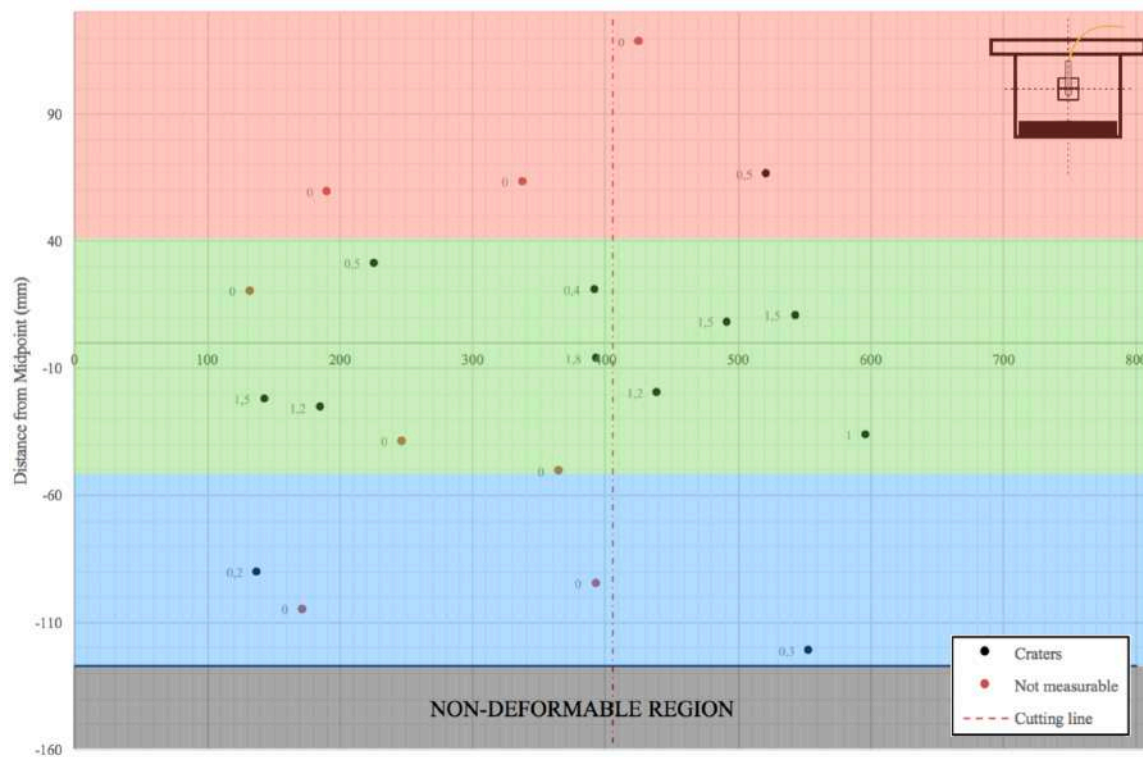


Figure 4.8: Distribution of craters from random-charge test (20 ball bearings, Test #7)

Test 4 and test 5 were conducted with the same number of ball bearings randomly incorporated in the charge. The distinct differences between the crater distributions from the two tests, shown in Figures 4.6 and 4.7, implied that there could be no predictable outcome when the ball bearing distribution was random as expected.

4.2.2 Crater Depths

The depths of all the craters were measured using Vernier callipers. The results showed variation in the depth of the crater even if the ball bearings were located within the same circumferential row. The mean depth and the % frequency of depth measurements within 1, 2 and 3 standard deviations from the mean (denoted as $\% \pm 1\sigma$, $\% \pm 2\sigma$ and $\% \pm 3\sigma$ respectively) were calculated and listed in Table 4.1. The mean depth of craters ranged from $1.0mm$ to $1.2mm$. No distinct correlation between the charge condition and the mean depth could be found.

For tests 2, 3 and 6 (row conditions), the % frequency of depth within one standard deviation from the mean ranged between 60% and 65%. For tests 4, 5 and 7 (random conditions), the % frequency varied more, ranging between 52% and 73%. The distribution for the row conditions group was more concentrated around the mean than it was for the random conditions group. However, it cannot be said for certain whether this was due to the packing condition or by coincidence.

Table 4.1: Mean depth and standard deviation data

Charge condition	Mean depth (mm)	$\% \pm 1\sigma$	$\% \pm 2\sigma$	$\% \pm 3\sigma$
1 row	1.03	61.1	100	100
2 rows	1.15	65.0	97.5	100
Random	1.21	72.7	97.0	100
Random	1.07	51.7	100	100
5 rows	1.09	60.2	97.9	100
Random	0.97	66.7	100	100

The correlation between crater depths and charge conditions was shown more clearly when each row of ball bearings was studied separately. The mean crater depth data per row in the non-random charge tests is summarised in Table 4.2. (The rows were numbered from the detonation end i.e. the flanged end). In the 2-row test, the mean depths of both rows of ball bearings were nearly identical. However, in the 5-row test, the depth increased further away from the detonation end, and peaked at row 4. The mean crater depths were the lowest for the rows closest to the cylinder ends of the explosive charge. The mean depths of the rows around the mid-section of the cylinder increased as the number of ball bearing rows increased.

Table 4.2: Mean depth per row from row-condition tests

	Row Number	Mean Depth (mm)
Test 2	1	1.03
Test 3	1	1.14
	2	1.15
Test 6	1	0.76
	2	1.03
	3	1.08
	4	1.22
	5	1.06

4.2.3 Crater Distribution Density

As most of the ball bearings were located in the mid-section of the cylindrical charge fewer craters were expected further away from the mid-plane of the cylinder. The number of craters in each section was recorded and was compared to the total number of craters.

Table 4.3 summarises the percentage density of craters in the upper, middle and lower sections of the cylindrical target wall for each test. The majority of the craters (55% - 79%) were clustered in the middle section of the cylinder in agreement with the results obtained by Grisaro and Dancygier [67] who had a ratio ranging between 65%-76%. The lower section had more craters than the upper section, because the detonator was inserted from the flanged end of the cylinder (i.e. the upper section). The blast waves propagated away from the detonator, and therefore more ball bearings were propelled towards the lower end of the cylinder. Test 4 (random condition) was the only exception with more ball bearings in the upper section than in the lower, likely due to the uneven distribution of ball bearings within the explosive charge. In the case of the randomly arranged ball bearings the distribution of the craters was not analysed further, because of the random nature.

Table 4.3: Percentage total number of craters in each section

% Total Number of Craters in Given Section						
Section	Test 2	Test 3	Test 4	Test 5	Test 6	Test 7
Lower	0	37.5	21.21	37.93	29.03	0
Middle	100	60	78.79	55.17	69.89	100
Upper	0	17.5	33.33	10.34	16.13	0

4.2.4 Effect of Packing Condition on Ball Bearing Deformation

The ball bearings were significantly more damage when packed randomly compared to the row packing on the outer wall of the charge. This was likely because in both random charge conditions the ball bearings were packed within the explosive and was closer to the detonation point. The degree of damage was also observed to be more significant with increasing number of ball bearings. With higher number of ball bearings, the charge became more confined exhibiting higher loads as observed for encased explosive [8].

4.2.5 Ball Bearing Velocity

Prior to testing, the Fish-Summers equation (Equation 2.20) and the JSC equation (Equation 2.21) were used to predict the depth of the craters using the given charge and projectile mass, at various estimated velocity values, since the exact velocity was unknown. The depth estimates are shown in Table 4.4. Based on the mean depth of the craters, $1.1mm$, obtained from experiment the impact velocities of the projectiles were estimated to range between $234m/s$ and $279m/s$ using the same equations.

It must be noted that the accuracy of these equations used in the design and the analysis cannot be guaranteed, since the equations were not formulated for the conditions of this experiment. More in depth analysis needs to be carried out to predict the depth of penetration and/or velocity of the projectile more accurately. However, as a first estimate, Equations 2.20 and 2.21 have proven to be useful.

Table 4.4: Estimated crater depth

Assumed velocity (m/s)	Estimated depth (mm)	
	Fish-Summers	JSC
250	1.05	1.01
350	1.41	1.22
450	1.76	1.49

4.3 Summary

All observable damage inflicted on the cylinder wall was caused by the ball bearings hitting the container walls. Hemispherical craters were formed on the walls. Distinct row-like patterns were observed when the ball bearings were packed in rows around the cylindrical charge. Random packing of ball bearings produced a distinctively different result to when they were packed in a row, and showed no predictable pattern.

The mean depth of craters ranged from $1.0mm$ to $1.2mm$. In the 2-row test, the mean depths of both rows were nearly identical whereas in the 5-row test, the depth increased further away from the detonation end, and peaked at row 4. The mean crater depths were the lowest for the rows closest to the cylinder ends of the explosive charge. The mean depths of the rows around the mid-section of the cylinder increased as the rows of ball bearings increased. The majority of the craters (55% - 79%) were clustered in the mid-section of the cylinder, and generally more craters were found in the cylinder lower section than in the upper section. The mean depth obtained from experiment was around half of the depth assumed during the design stage. The impact velocity, calculated using the experimental mean depth, was approximated to range between $234m/s$ and $279m/s$.

Chapter 5

Formulation of the Numerical Model

This chapter describes the formulation of the numerical model as developed in AUTODYN. The different material models are first selected before the geometry and the model is constructed. The boundary conditions and initial conditions are thereafter implemented.

5.1 Model Framework

In order to model explosions which involve large deformations, the interaction between fluids and solids need to be modelled. It is therefore crucial to make use of the correct solver for each of the fluid and solid entities in the simulated problem.

The numerical models are constructed in ANSYS AUTODYN R19.0, a hydrocode designed for non-linear dynamic analysis. AUTODYN is capable of solving dynamic, highly non-linear problems using Lagrange, Euler, Arbitrary Lagrange Euler (ALE) and mesh-free solvers [78].

5.1.1 Lagrangian and Eulerian Solvers

In a Lagrangian mesh, the mesh nodes are seeded on the geometric entity to be modelled. These nodes are fixed in position on the object but not relative to one another, and the mesh is free to deform with the material. Lagrangian elements distort with the body, and the mesh geometry changes as the body deforms. The Lagrangian approach works well for "smooth" non-massive deformations such as structural responses of solid materials, and hence this solver is typically used to model solid continuum structures. [79] However, if large distortions are encountered, computational inaccuracies are introduced by large mesh distortion.

For materials that deform rapidly, such as gases, a Lagrangian mesh would lose its initial node density with increasing element size, resulting in large inaccuracies and unrealistic model objects. The Eulerian approach is preferred for modelling of gases and liquids. Eulerian meshes work on a fixed structured mesh constructed from a fixed grid of nodes. The mesh is then filled

with the material being modelled, which can move between the cells formed by the nodes. The state of each cell is determined by the state of the material it contains. This approach does not suffer from mesh distortion issues associated with Lagrangian meshes, and is ideal for the modelling of hydrodynamic events where large deformations are to be expected [78]. A typical Eulerian mesh time step is considerably more computationally expensive than a Lagrangian time step. This is because an Eulerian time step requires an additional material transport step where materials are transported across cell boundaries. The volume fraction of each material in a given cell is also tracked, with a maximum of 5 different materials in a single cell [78].

Lagrangian solvers can be used with Eulerian solvers or Arbitrary Lagrangian-Eulerian (ALE) solvers to analyse fluid-structure interaction. The Eulerian and Lagrangian meshes can be coupled such that the blast that propagates through the explosive and air is loaded onto the Lagrangian target.

5.1.2 Erosion in Lagrangian Meshes

In some models, materials have to be defined on Lagrangian subgrids even though it is clear that these materials will be subjected to very large distortions. For example, a ballistic projectile re-entering the atmosphere will suffer gross distortion of the outer layers because of re-entry heating; an accurate calculation of this scenario should allow for the ablation of the outer layers [78]. Target and projectile material in hypervelocity impact problems is another example that is subject to gross distortion. In both examples, calculations on an Euler subgrid will suffer unless an accurate interface tracking routine is used, and therefore a Lagrange subgrid definition needs to be used.

During the subsequent calculation some of these Lagrangian cells can become grossly distorted and seriously impair the progress of the calculation. As a means to counter this issue, Lagrangian meshes allow for element deletion where the largely distorted element is removed from the analysis. This discard procedure is known as erosion. It is however important to note that this is not true modelling of a physical phenomena in general. It is rather a numerical palliative introduced to overcome the problems associated with the mesh distortion [78].

5.1.3 Framework Implementation in Numerical Model

In the context of this project, the interaction between the target cylinder and the bearings is analogous to a ballistic projectile impacting a target, which means that these materials may be subject to gross distortion. An accurate interface tracking routine is still required, however, and Lagrangian mesh structures are better suited for these parts. Large deformations are expected to occur especially in the target cylinder mesh. Therefore erosion technique will be used in these parts to overcome the mesh distortion issue.

An Eulerian solver is the most suitable for the explosive and the air within the confined

container, as large deformations are to be expected due to the blast loading. Interaction of the explosive products, air, ball bearing and the target cylinder was accommodated by selecting the 'Automatic' Langrange-Eulerian coupling option in AUTODYN.

5.2 Material Models

5.2.1 Air

5.2.1.1 Ideal Gas Equation of State

Air was modelled as an ideal gas using the parameters contained within AUTODYN's material model library. This model requires user input for the internal energy of the air, which determines the initial pressure of air at the beginning of the simulation. Atmospheric pressure ($101.3kPa$) corresponds to an internal energy of $2.068 \times 10^5 kJ.kg^{-1}$. The ambient temperature of the air is $15^\circ C$ ($288.2K$). Air, being a gas, does not require any strength model as it does not offer any resistance to shear distortion and cannot support negative pressures. Other parameters used for the ideal gas EOS for air are listed in table 5.1.

Table 5.1: AUTODYN material properties for the ideal gas EOS of air [80]

$\rho_a(kgm^{-3})$	$T(K)$	$C_p(kJ.kg^{-1}K^{-1})$	$C_v(kJ.kg^{-1}K^{-1})$
1.225	288.2	1.005	0.718

5.2.2 Explosive: PE4/C4

The plastic explosive PE4 is not present within the AUTODYN material library. However, PE4 and C4 have the same explosive characteristics and only differ in terms of the plasticisers used. Balden and Nurick [81] has shown that PE4 may be substituted with C4 material models in AUTODYN. Rigby et al. [82] also performed numerical simulations in LS-DYNA and used C4 material model as identical to PE4. The material model for C4 available in the AUTODYN library was therefore used.

5.2.2.1 Jones-Wilkins-Lee Equation of State

The Jones-Wilkins-Lee (JWL) EOS is used to describe the detonation of the plastic explosive, and is given by:

$$p = A(1 - \frac{\omega}{R_1 V})e^{-R_1 V} + B(1 - \frac{\omega}{R_2 V})e^{-R_2 V} + \frac{\omega e}{V} \quad (5.1)$$

where p is the pressure, V is the volume ratio and e is the energy per unit volume. A , B , R_1 , R_2 and ω are constant parameters to be calibrated. The default parameters provided in the material library [80] were used without any modifications. These values are summarised in table 5.2.

Table 5.2: AUTODYN material properties for C4 [80], used to model PE4

$\rho_e(kgm^{-3})$	A (GPa)	B (GPa)	R_1	R_2	ω	$CJV_{detonation}$ (ms^{-1})	$C - J \frac{Energy}{Volume}$ (kJm^{-3})	$C - JP$ (GPa)
1601	609.8	12.95	4.5	1.4	0.25	8193	9.0×10^6	28

To ensure accuracy, AUTODYN automatically converts the explosive products to an ideal gas after they have approached ideal gas conditions [80]. The ideal gas equations better models gases at low (near-ambient) pressures than the JWL EOS, and is also computationally less expensive.

5.2.3 Mild Steel Cylinder

Mild steel is currently not one of the standard materials found in the AUTODYN materials library. The material characteristics, grade or chemical composition of the mild steel used in the experiments were not available from the manufacturer. A mild steel sample cut out from one of the cylinders used in experiment was tested on a Zwick/Roell ZHV hardness test machine. The hardness test was repeated five times to ensure accuracy and repeatability. The average of the five test results was used as the final hardness value. A Vickers hardness of 147 VPN was obtained from the hardness tests.

Iqbal et al. [83] has conducted a comprehensive material characterisation on a mild steel samples of 12mm thickness and 142 VPN. The material behaviour of mild steel was studied under varying state of stress, strain rate and temperature, and the data was used for the calibration of the Johnson Cook constitutive model. The calibrated model was also validated by simulating the ballistic performance of 12mm thick mild steel targets against 7.62 AP projectiles.

The AUTODYN material model for Steel 1006 was modified using the material parameters obtained for mild steel by Iqbal et al. [83] to numerically model the target cylinder. Steel 1006 model was selected because it has similar yield strength to mild steel, and the Shock EOS for Steel 1006 obtained from the AUTODYN library was used. The parameters are listed in table 5.3.

Table 5.3: Steel S-7 shock EOS parameters, used for mild steel

Gruneisen Coefficient	$C_1(m/s)$	S_1	$T_{ref}(K)$	Specific Heat $Jkg^{-1}K^{-1}$
2.17	4.569×10^3	1.49	300	452

5.2.3.1 Johnson Cook Strength Model

The Johnson-Cook model parameters from Iqbal et al. [83] were used to numerically model the mild steel cylinder because the mild steel sample from experiments was similar in hardness

and thickness to the mild steel sample used in the study by Iqbal et al. [83]. The Johnson Cook model includes the hardening effects at high strain-rate loading and the softening effects of adiabatic temperature increases due to plastic work in the material. The Johnson-Cook strength equation is given by:

$$\sigma_e = [A + B(\epsilon_e^p)^n][1 + C \ln(\frac{\dot{\epsilon}_e^p}{\dot{\epsilon}_0})][1 - \hat{T}^m] \quad (5.2)$$

$$\hat{T}^m = \frac{T - T_{room}}{T_{melt} - T_{room}} \quad (5.3)$$

where σ_e is the yield stress, ϵ_e^p is effective plastic strain, $\dot{\epsilon}_0$ is the strain rate for the quasi-static test, T_H is the homologous temperature, and A , B , C , n and m are material constants. The Johnson-Cook parameters summarised in table 5.4 have been determined from material characterisation and validated by Iqbal et al. [83].

Table 5.4: Johnson Cook strength model parameters for mild steel [83]

ρ_e (kgm^{-3})	G (GPa)	A (MPa)	B (MPa)	n	C	m	T_{melt} (K)	$\dot{\epsilon}_0$ (s^{-1})
7.85	78.57	304.33	422.01	0.345	0.0156	0.87	1800	0.0001

5.2.3.2 Johnson Cook Failure Model

Iqbal et al. [83] derived the Johnson Cook failure model parameters from material characterisation. These parameters were also successfully validated with a numerical ballistic evaluation on 12mm mild steel targets in [83]. Therefore, since the failure model has been validated under similar context as the current study (the interaction between the target cylinder and the bearings is analogous to a ballistic projectile impacting a target), the Johnson Cook failure model from [83] was set as the material failure criteria for the target cylinder. The Johnson-Cook failure model equation is given by:

$$\epsilon_{ef}^p = [D_1 + D_2 e^{D_3 \eta}][1 + D_4 \ln(\frac{\epsilon_e^p}{\epsilon_0})][1 + D_5 \hat{T}] \quad (5.4)$$

$$\eta = \frac{\sigma_m}{\sigma_e} \quad (5.5)$$

where ϵ_e^p is effective plastic strain at failure, and D_1 , D_2, D_3, D_4 and D_5 are material constants. The failure model parameters are summarised in table 5.5.

Table 5.5: Johnson Cook failure model parameters for mild steel [83]

D_1	D_2	D_3	D_4	D_5	T_{melt} (K)	$\dot{\epsilon}_0$ (s ⁻¹)
0.1152	1.0116	-1.7684	-0.05279	0.5262	1800	0.0001

5.2.3.3 Effect of Erosion Criteria

In the context of this project, the interaction between the target cylinder and the bearings is analogous to a ballistic projectile impacting a target, which means that these materials may be subject to gross distortion. Large deformations were expected to occur especially in the target cylinder mesh. Therefore erosion technique must be used in these parts to overcome the mesh distortion issue.

Erosion is characterised by total material failure. From a numerical viewpoint, total failure occurs when the equivalent strain exceeds a user prescribed erosion strain, $\tilde{\epsilon}_e$. Erosion strains of 1.0, 1.5 and 2.0 have been tested in literature for various steel. Stecher and Johnson [84] found that, in general, the simulation results were relatively insensitive to the value chosen. AUTODYN also has an option to prevent the erosion of degenerate elements. However, the AUTODYN manual [78] warns that using this option can lead to unphysical results, especially when large gap size is used. Instantaneous geometric strain was set as the erosion criteria, with the geometric strain limit of 2.0. This erosion model has been used successfully by Hernandez [85] in modelling thick metal targets impacted by stainless steel spherical projectiles.

5.2.4 Ball Bearing

An accurate capture of the deformation mechanism of the ball bearing prior and post impact was excluded from the scope of this project. However, to investigate the effect of the pressure propagation and loading on the trajectory of the ball bearing, the bearing was represented in a Lagrangian subgrid. The ball bearing material used in experiment was AISI 420 stainless steel, which is not included in the AUTODYN material library. Therefore, the Steel S-7 model from the library was used to simulate the ball bearing, because of their similarity in yield strength and density. AUTODYN uses the shock equation of state for the steel S-7 model to calculate the hydrostatic pressure component of the stress. The shock EOS parameters used for the bearing are shown in table 5.6.

Table 5.6: Shock EOS parameters for steel S-7 [80]

Gruneisen Coefficient	$C_1(m/s)$	S_1	$T_{ref}(K)$	Specific Heat $Jkg^{-1}K^{-1}$
2.17	4.569×10^3	1.49	300	477

5.2.4.1 Johnson Cook Strength Model

Steel S-7 model from the AUTODYN library uses the Johnson Cook strength model. All parameters shown in table 5.7 were used unaltered from the default value set in AUTODYN.

Table 5.7: Johnson Cook strength model parameters for steel S-7 [80]

ρ_e (kgm^{-3})	G (GPa)	A (GPa)	B (MPa)	n	C	m	T_{melt} (K)	$\dot{\epsilon}_0$ (s^{-1})
7.75	81.8	1.539	477	0.18	0.012	1.00	1763	1.00

5.2.4.2 Failure and Erosion

Failure and erosion criteria were not set for the ball bearing, as the ball bearing was not expected to fail. It must also be noted that accurate modelling of the bearing material behaviour was excluded from the scope of this project.

5.3 2D Numerical Model Description

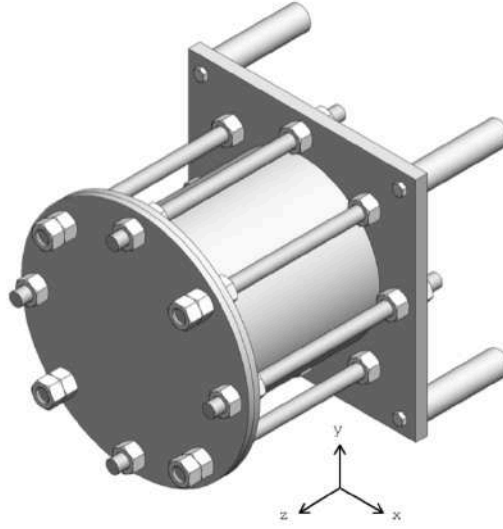


Figure 5.1: 3D view of the cylindrical container experimental setup

An axisymmetric model was used in an attempt to capture the behaviour of the ball bearings and the target cylinder following the detonation of the cylindrical explosive. The use of axisymmetric detonation models can mitigate drawbacks related to computational cost while still retaining a high level of accuracy.

The cylindrical geometry of the experimental setup, shown in Figure 5.1, means that there is symmetry about the central axis of the cylinder and the cylindrical charge sitting in the middle. It was noted, however, that each ball bearing needed to be represented as a sphere, and they did not share the same central axis as the cylinder. A 3D model would better represent the experiments, but would take significantly longer to run. 2D simulations are relatively simple and require less computational resources than 3D simulations. The lack of parallel processing for all models discourages the use of extensively complex models due to the considerable computational expense. Therefore, the scope of the numerical study was limited to capturing the ball bearing and blast interaction, rather than to accurately replicate the target cylinder behaviour after impact.

The x – y plane from the 3D setup, shown in Figure 5.2 on the right, produces the 2D radial cross section. Due to the two symmetry planes present in the radial cross section, only one quarter of the geometry needs to be modelled. Figure 5.2 shows a schematic of the 2D axisymmetric model used to simulate the experiments with rows of ball bearings around the charge. Included in the simulations were target cylinder, ball bearings, PE4 charge, and region of air in the blast vicinity. The pressure gauge points are also shown and labelled. For the bare charge scenario, the bearings were omitted.

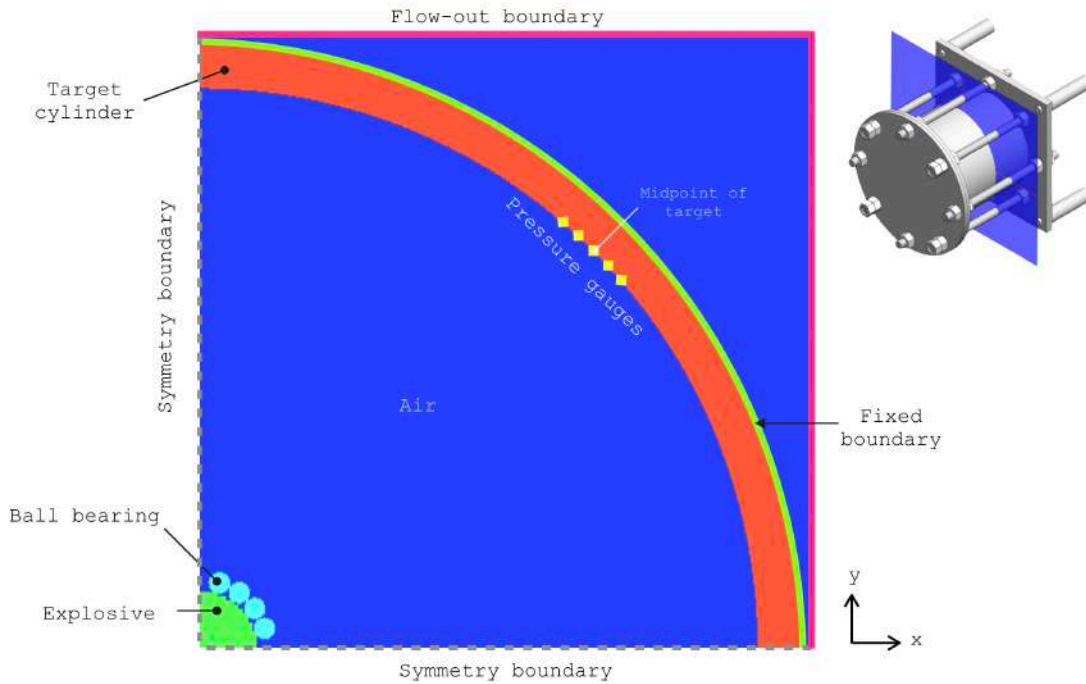


Figure 5.2: Simplified schematic of the 2D radial plane blast model used in the final model formulation

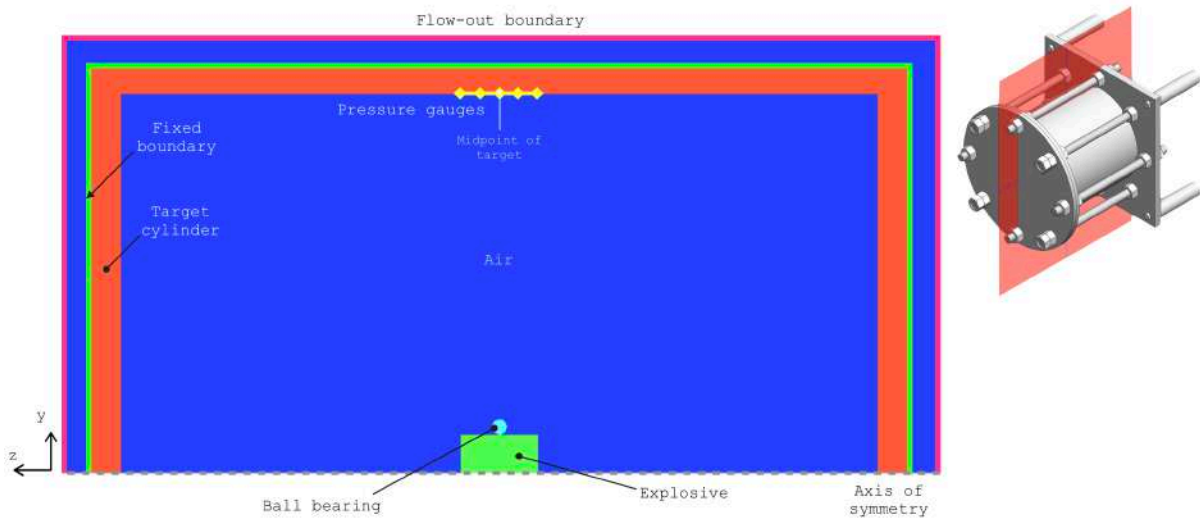


Figure 5.3: Simplified schematic of the 2D axial plane blast model used in the final model formulation

Similarly, the $y - z$ plane from the 3D setup, shown in Figure 5.3, produces the 2D axial cross section. Due to symmetry conditions, only half of the axial cross section had to be modelled. Figure 5.3 shows a schematic of the 2D axisymmetric model in the axial plane, for the single-row

ball bearings scenario, also showing the gauge points placed for blast pressure measurements. In the axial plane, it was noted that due to the position of the ball bearing relative to the axis of symmetry, axisymmetric geometry could not be fully exploited. For the ball bearing to behave as a separate, circular object and not as a "ring" in the simulation, planar geometry had to be utilised in AUTODYN. Under these conditions, it was not possible to model the curved geometry of the target nor the spherical shape of the ball bearings accurately. However, it is still possible to model the bearing and blast interaction, and to study the bearing velocity and spatial distribution in this plane. The aim of the numerical models was to capture the ball bearing motion and its subsequent interaction with the explosive gas product, rather than to accurately replicate the behaviour of the target cylinder, which did not deform significantly in the experiments. The target geometry was captured more accurately in the radial plane; therefore the target cylinder interaction with other parts was studied more in depth in the radial plane.

5.4 Boundary and Initial Conditions

The cylindrical target in the experiments was fully confined, and a cover was present on the cylinders. However, no seal was present and gas product could escape nevertheless. The numerical model was simplified and the gas leakage was not modelled. It was assumed that no explosive products or the projectiles would leave the modelled domain encased by the target cylinder. Symmetry conditions were required for the cylindrical container and the Eulerian mesh containing the air and the explosive. Confined or internal explosions using PE4 typically include afterburning [32], but the numerical model was simplified and the effect of the afterburning was not included.

Using reflective boundaries to model rigid faces is a common practice to decrease the number of nodes and reduce the computation time. The two parallel annular bases of the cylindrical container, used to clamp and fully confine the container, showed no sign of deformation under bare charge conditions. However, in some of the experiments, the ball bearings were observed to have hit the cover plate and the clamping boss. Therefore, even though the annular bases of the cylinder could be assumed as rigid, they cannot be simply modelled using reflective boundaries since the interaction between the ball bearing projectile and the target cylinder was included in the scope of this investigation.

The walls of the target cylinder were modelled using a Lagrangian model. The nodes on the perimeter of the target model that coincided with the symmetry plane were constrained to allow movement only within the plane of symmetry.

5.4.1 Geometric Limits and Boundaries

In the experiments, the cylindrical container was effectively clamped in an encastre boundary condition by the boss in the backing plate, the cover, the threaded bars, and bolts. Furthermore, no bulging or deformation of the cylinder was observed in the experimental tests with the exception of craters formed by the ball bearings.

When the numerical models were run without constraining the outer face of the Lagrangian target in a preliminary run, the target displayed severe deformation post-blast. This may be attributed to the fact that 2D geometry of the numerical models could not accurately represent the 3D cylindrical geometry of the target container as the geometry of the target cylinder in the experiments has a clamping effect on the outer walls. Consequently, the outer wall of the cylinder was fixed to simulate the insignificant deformation of the cylinder. The nodes along the outer perimeter of the Lagrangian model were completely constrained in both x and y directions.

5.5 Mesh Independence Study

A series of mesh independence studies was performed to determine an acceptable mesh size combination for the air mesh, the ball bearings and the target cylinder. The final crater depths on the target cylinder were used as the measure for acceptable mesh sizes, as well as the CPU time required to complete the simulation. Nevertheless, a representative explosive model was necessary to produce an accurate target response, as the interaction between the explosive products, the ball bearing and the target cylinder determines the blast response of the structure. Deformation of the bearing and the damage in the cylinder wall was therefore used as the measure for selecting an appropriate mesh resolution. The radial plane was used for the mesh sensitivity study.

5.5.1 Preliminary Blast Pressure Test

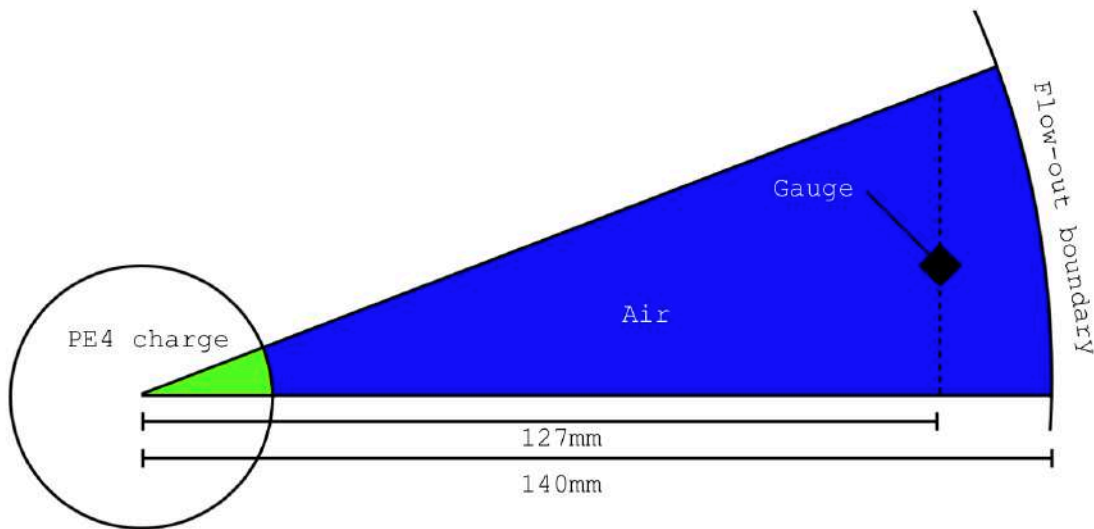


Figure 5.4: Free-air axisymmetric explosion model (Not drawn to scale)

Before the mesh independence study was performed, a preliminary test was completed to assess the accuracy of the blast pressure produced in the Eulerian mesh. The capability of the air mesh to accurately capture the blast pressure is useful as an indication of the accuracy of the simulated deformation. A suitable mesh size must be used for any given blast test setup. Should the mesh be too coarse, the pressure-time profile would be stretched out without indication of a distinct blast front arrival time [86]. Another indication of an overly coarse mesh is the rate of the initial pressure rise being smaller than 90 degrees. Conversely, finer mesh shortens the blast wavefront arrival time. Once the mesh size is sufficiently small and further decrease in mesh size does not influence the arrival time, the mesh size is said to have converged.

The size of elements in a numerical model is dependent on the specific scenario. The first step to modelling the impact of an explosion on a structure is to create a 2D FE model of free-air explosion. Draganić and Varevac [86] used a model composed of air, outflow boundary and circular TNT charge, the radius of which is determined by the specific reference density and given weight. The radius of the air-filled wedge was arbitrarily defined within reasonable magnitude to shorten calculation time. The use of axial symmetry ensures that the computation time is minimised. The results showed that the pressure decreases with increasing distance between the detonation point and the target structure, and the pressure values were less sensitive to air mesh size. For smaller scaled distances, however, the mesh sensitivity and the scatter in results were greater.

The blast was modelled using the free-air axisymmetric explosion model described by Draganić and Varevac [86]. This model is composed of circular charge, surrounding air, and outflow boundary. Using axial symmetry, only one sector was modelled as shown in Figure 5.4. The model was set up using a one-dimensional wedge in AUTODYN 2D axisymmetric solver. The angle of the wedge was pre-defined by AUTODYN. The dimension of the wedge was user-defined and depended upon charge weight and location of pressure measurement. The radius of the charge was defined as:

$$V_{charge} = \frac{mass}{density} = \frac{4}{3}\pi(R^3 - r^3)$$

where R is the outer radius of wedge and r is the inner radius of wedge. The inner radius of the wedge was set as $0.001mm$ such that the charge-filled section of the wedge would effectively represent a filled sphere.

The outer radius R was set to $140mm$. Air meshes with element sizes ranging from $0.02mm$ to $14mm$ (i.e. 10 elements to 7000 elements respectively) were tested for a charge mass of $22g$. A "Flow-out" boundary condition was defined at the end of wedge to allow pressure wave to leave the domain without reflecting any pressure back into the air domain. The detonation point was set at the centre of explosive to initiate the explosion at time zero. The peak pressure was measured using a gauge point located at a stand-off distance of $127mm$ (i.e. the radius of the cylinder from the experiments), and was compared to the empirical solution obtained by Kinney and Graham's overpressure equation, Equation 2.2 [19]. The gauge was located off the boundary surfaces to mitigate any errors that accumulate along the domain boundaries.

Figure 5.5 shows the plot of percentage error in recorded peak pressure against the number of elements across the wedge radius. Each data point is labelled with the corresponding number of elements, and the solid blue line indicates the 20% error threshold. Figure 5.6 shows both the percentage error from Kinney and Graham's empirical equation [19] and the runtime of each simulation plotted against the number of elements.

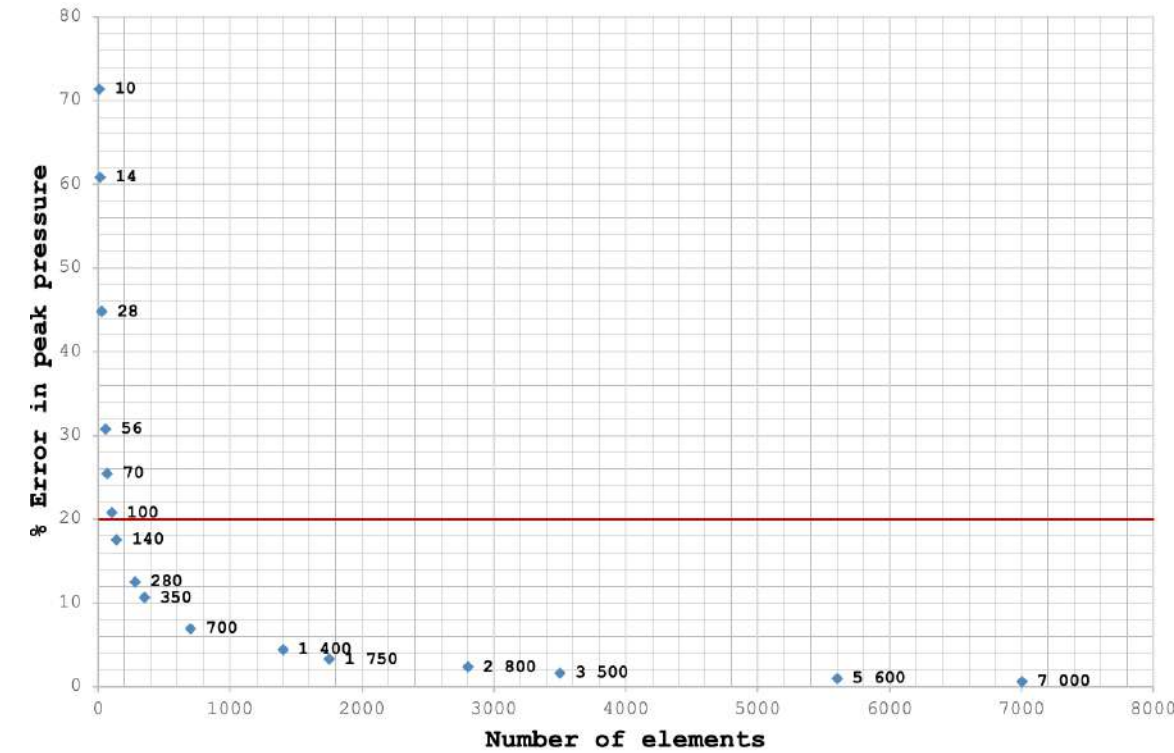


Figure 5.5: Graph showing the % error in recorded peak pressure versus the number of elements across the wedge radius (labels show the number of elements)

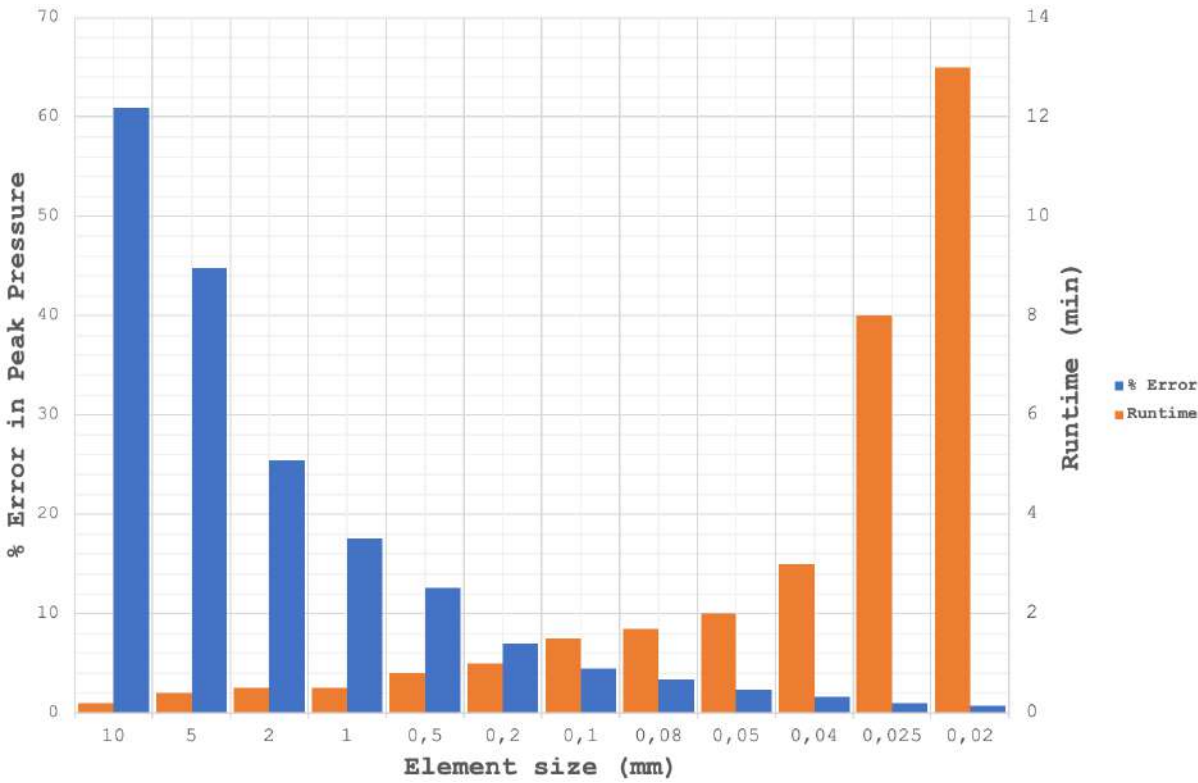


Figure 5.6: Graph showing the % error in recorded peak pressure and runtime for each element size

The simulated peak pressures were observed to approach the empirical solution as the air mesh became more refined. The percentage error in simulated peak pressure was less than 20% for mesh size $1mm$ and less (i.e. 140 elements or more across wedge radius). The accuracy of the predicted peak pressure improved as the mesh size was refined, with the $0.02mm$ air mesh capturing the most representative peak pressure with an error of less than 1%. However, the CPU time required also increased exponentially with refined meshes less than $1mm$ in size. Since a simple, 1-D wedge was used in this numerical model, it was expected that the models developed to simulate the experiments would require significantly more CPU time to complete.

The results of the blast pressure test was used as a preliminary criterion in choosing the minimum air mesh size. It must be noted that the full numerical model involves a larger domain to be modelled, and includes additional components such as the target cylinder and the ball bearing(s), thus requiring a significantly longer computational time. Therefore, only mesh size of $0.1mm$ or larger were considered for the air mesh of the full numerical model.

5.5.2 2D Radial Plane Model

The 2D radial plane model setup used for the mesh convergence study is shown in Figure 5.7. A $140mm \times 140mm$ air domain was used and a quarter-"ring" of $127mm$ inner radius and $10mm$ thickness, representing the target cylinder in the radial plane, was placed within the air domain. The nodes along the x and y axis were assigned corresponding symmetry boundary conditions, while the nodes along the outer edge of the ring were assigned fixed boundary conditions to represent an ideal clamped condition. A charge size of $13mm$ radius, corresponding to a mass of 22g, was used. A Lagrangian ball bearing model of $5mm$ diameter was placed outside the charge radius and the charge was detonated from the centre (i.e. coordinate $(0,0)$) at time zero.

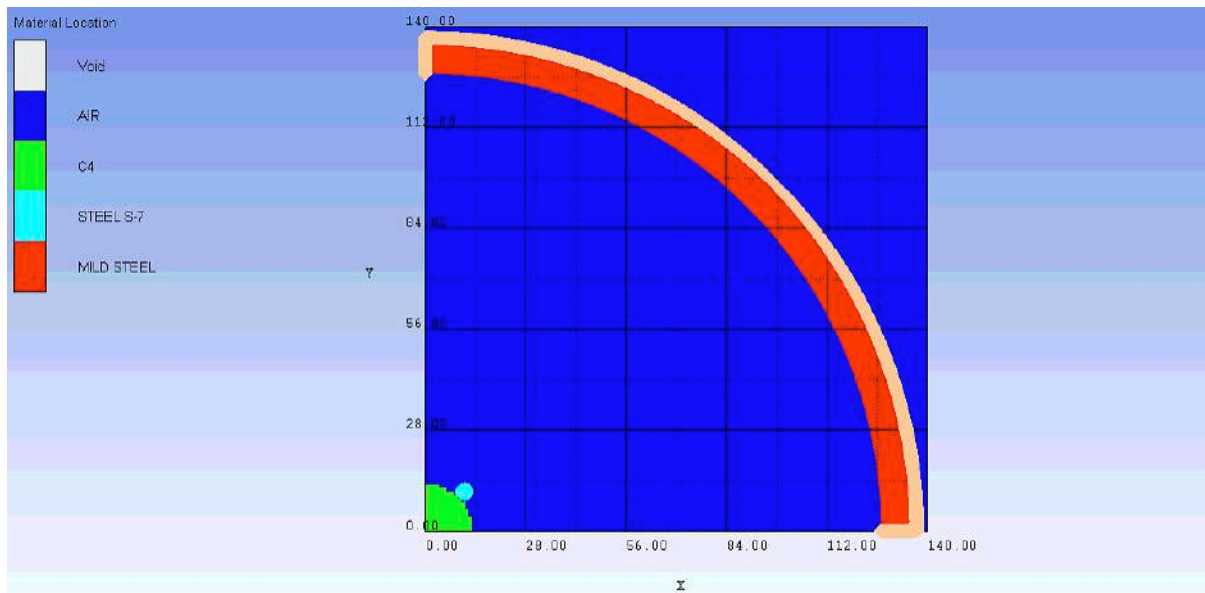


Figure 5.7: 2D Numerical blast model for mesh independence study in the radial plane

5.5.3 Air Mesh

Previous explosive modelling work by Balden [87], Balden and Nurick [81] and subsequent work by Chung Kim Yuen et al. [88] have shown that to ensure proper detonation of the explosive so that the CJ pressure is attained, at least 10 to 20 elements should be used through the smallest dimension of the explosive. The explosive was modelled using an Eulerian mesh, and it was considered preferable for the mesh size through the explosive to be a factor of its dimension. The minimum dimension of the explosive was 13mm . Assuming the minimum number of elements through the dimension to be 10 elements, the largest operable mesh dimension was 1.3mm . However, the radius of the cylindrical container used in the experiment did not have 1.3mm factors. Since square elements (in 2D) are favoured in numerical work, and as the vast majority of the cells were outside of the explosive, the second dimension (the radial length) of the cells was set as equal to the height of an air cell. Therefore a square size of 1mm was preliminarily chosen for both the air and explosive mesh.

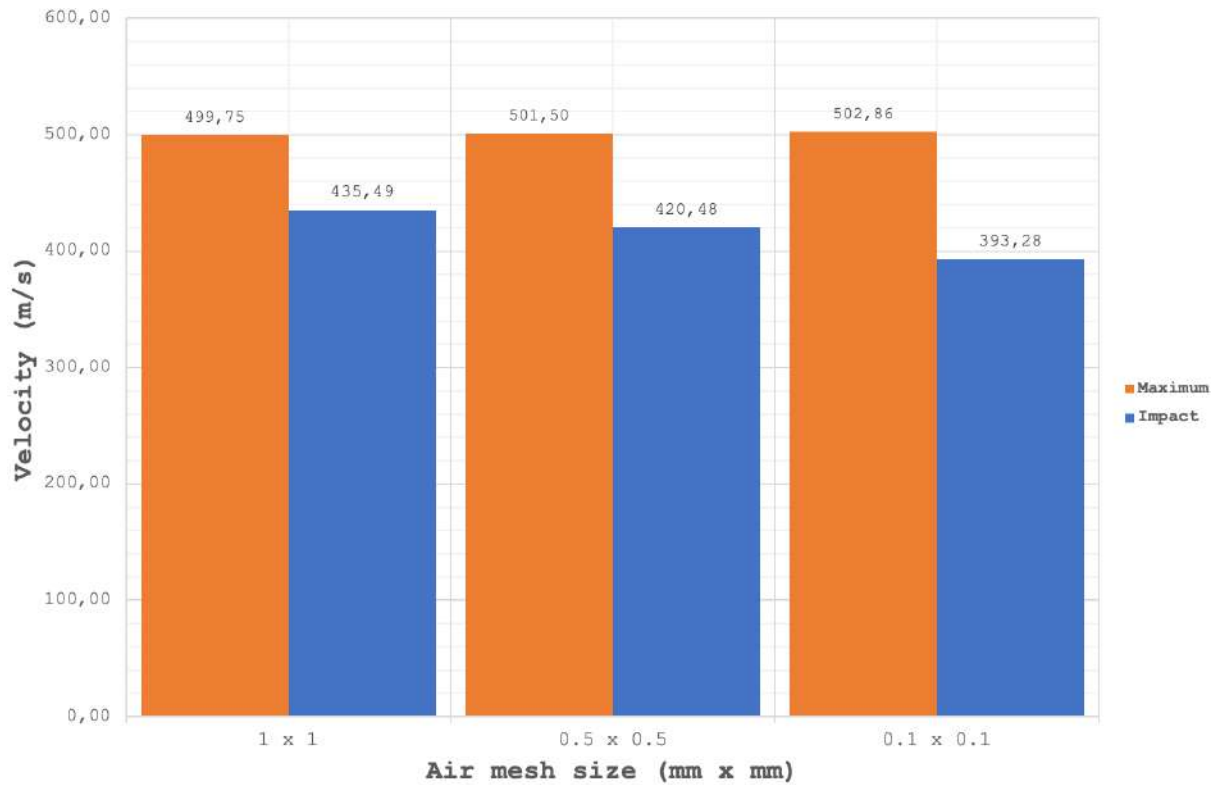


Figure 5.8: The maximum and impact velocities of the ball bearing for three different mesh sizes in the radial plane

A mesh convergence test was run to check for the accuracy of the numerical solution. Three different Eulerian meshes were used in the convergence tests, namely $0.1\text{mm} \times 0.1\text{mm}$, $0.5\text{mm} \times 0.5\text{mm}$ and $1\text{mm} \times 1\text{mm}$. The unit of comparison chosen was maximum and impact (resultant) velocity of the ball bearing post-detonation, as these are the factors that determine the crater depth in the target cylinder. Preliminary simulation runs also showed that with the boundary conditions applied, the nodes of the target cylinder that were in direct contact with the Euler

mesh were not displaced more than 0.001mm due to the blast wave interactions. Therefore the interaction between the air mesh and the target cylinder was negligible compared to the interaction between the air mesh and the bearing. The target cylinder mesh size was not of importance in the mesh convergence tests, since the simulations were terminated before the bearing impacted the target cylinder wall.

Figure 5.8 shows the maximum and impact velocity results from the radial plane simulations. Figure 5.9 shows the corresponding runtime for each test. The simulation results showed that the maximum velocity value of the ball bearing was relatively insensitive to the mesh size, but the impact velocity value of the ball bearing decreased for more refined mesh elements. In other words, the percentage drop between the maximum and the impact velocity increased for more refined air and explosive mesh. The percentage difference between the impact velocity values corresponding to $1\text{mm} \times 1\text{mm}$ mesh and $0.1\text{mm} \times 0.1\text{mm}$ mesh was 9.7%.

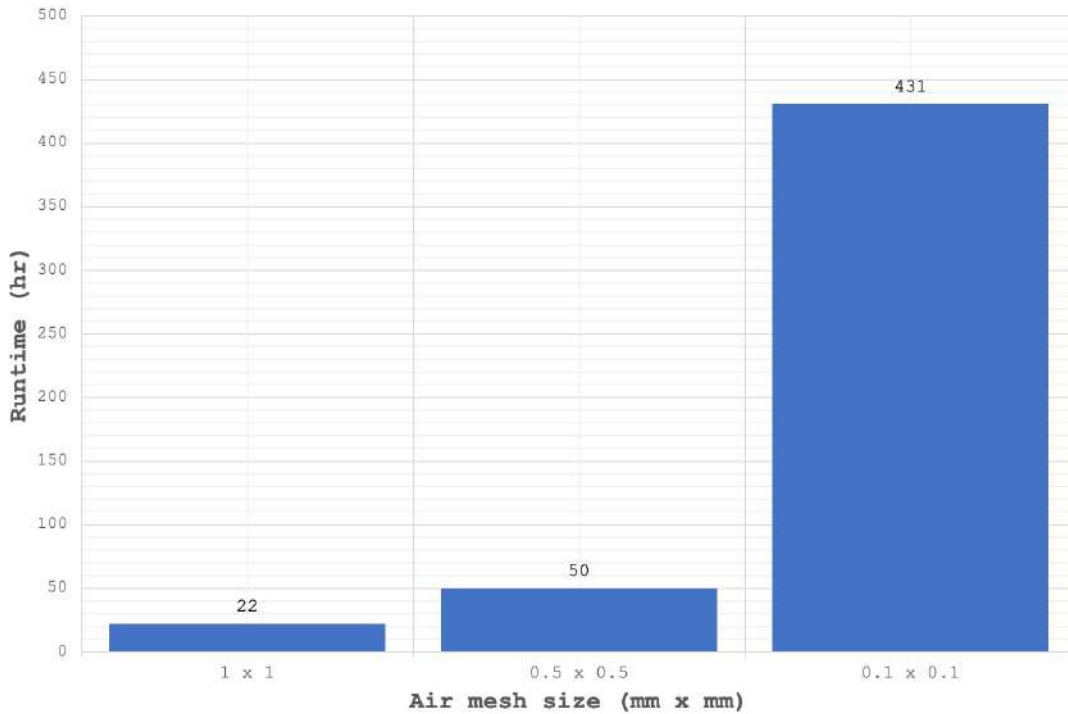


Figure 5.9: The computation runtime taken for the ball bearing to impact the target cylinder in each model in the radial plane

The $0.1\text{mm} \times 0.1\text{mm}$ mesh was considered too computationally expensive and discarded as a possible mesh size. The impact velocity from the $0.5\text{mm} \times 0.5\text{mm}$ mesh was closer in value to the impact velocity result corresponding to the finest (0.1mm) mesh, and using the $1\text{mm} \times 1\text{mm}$ mesh resolution resulted in an impact velocity that was 6.5% lower than that of the 0.5mm square mesh simulation. However, the $0.5\text{mm} \times 0.5\text{mm}$ mesh was also approximately 2.3 times more computationally expensive than the 1mm square mesh. Given that the runtime must be manageable in order to complete an extensive modelling range, the percentage difference between the velocity values was considered an acceptable compromise. The large variation observed in experimental data also reinforced the use of a relatively coarse mesh. A square

mesh of $1\text{mm} \times 1\text{mm}$ was therefore selected for the explosive and air parts.

5.5.4 Part Meshes

The target cylinder and the ball bearings were both modelled using Lagrangian elements. A mesh independence study was performed to determine the appropriate mesh size for the ball bearing and the target cylinder. The crater depths observed in the target cylinder in the radial plane was used for comparison, since the depth values are known from experiment. Multiple craters were observed in the target cylinder from the experiments. However, for the sake of computation time practicality, only one ball bearing was modelled in the mesh independence study and the average crater depth value from the single-row experimental test was used as benchmark.

Detonation mesh of $1\text{mm} \times 1\text{mm}$ square Eulerian elements was used in the blast model setup shown in Figure 5.7. A charge of 13mm radius, corresponding to a mass of 22g , was used. A Lagrangian ball bearing model of 5mm diameter was placed outside the charge radius. The charge was detonated in a space confined by a Lagrangian target of 127mm inner radius with 10mm wall thickness.

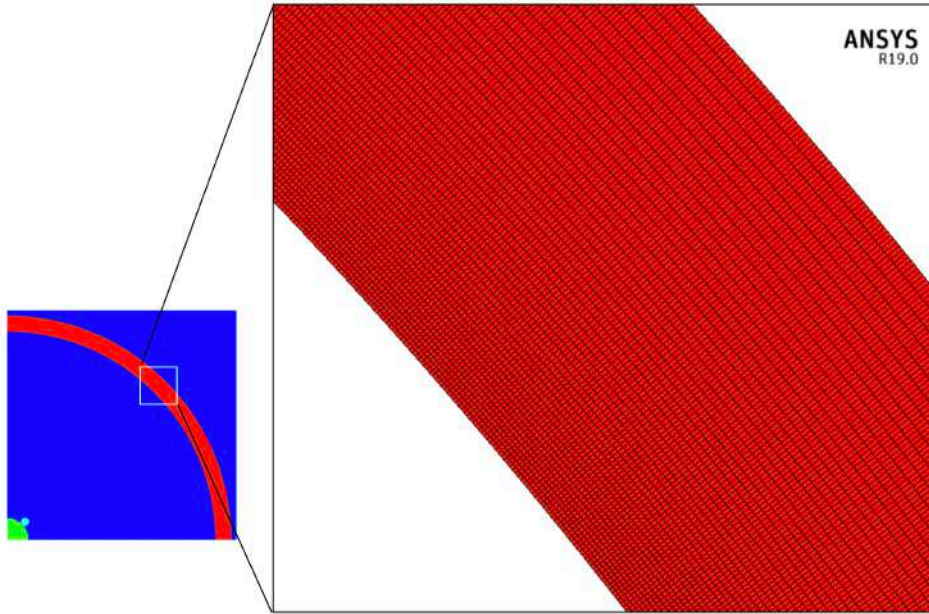


Figure 5.10: Magnified view of a section of the biased target mesh

Four different target mesh sizes ranging between 1mm and 0.1mm were tested. In addition to square elements, 1 additional simulation was run to test the effect of using biased mesh. Since only the inner face of the target cylinder was expected to be impacted and deformed by the bearing (no complete penetration of the target was observed), the elements towards the outer edge of the target did not need to be as refined as the elements towards the inner face. The mesh was biased such that their dimensions merged gradually (i.e. linear increase in element

size between the two mesh sizes, in the radial direction), as shown in Figure 5.10. The transition began at the inner face of the target cylinder (element size $0.1mm$), until it reached the outer edge (element size $0.5mm$). The mesh size across the circumference was kept constant at $0.1mm$ in the biased case.

Leakage is a numerical phenomenon whereby material represented in the Eulerian mesh travels through the Lagrangian mesh, as a result of the relative sizes of the Eulerian mesh and the Lagrangian part [78]. To ensure proper interaction between the air and target meshes, and to prevent leakage of gas products through the target [78], the maximum value of the target cylinder element size must not exceed the air mesh element size.

In addition to the target meshes, 3 different bearing mesh sizes were tested, each differing by a factor of two. The most coarse bearing mesh size tested was $0.5mm \times 0.5mm$ (i.e. 10 elements across the diameter, denoted as 10×10 in the graph). Not all possible combinations of bearing and plate mesh sizes were tested, as the simulation gave unphysical results for certain mesh combinations, where the target mesh was significantly more coarse than the bearing mesh. For example, the simulation run using the $0.125mm \times 0.125mm$ (i.e. 40×40) bearing mesh with $0.75mm$ target mesh element led to unphysical interaction where the bearing flattened completely before it could leave an indentation on the target mesh, and was discarded.

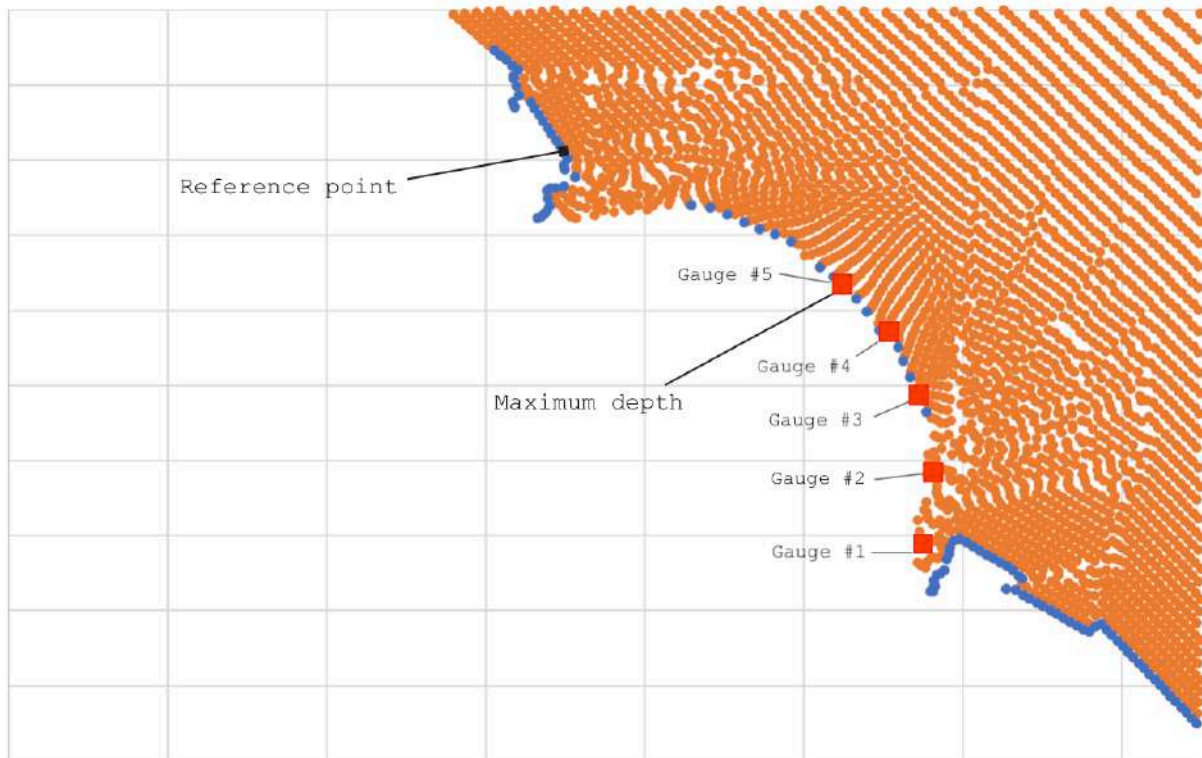


Figure 5.11: Crater nodal plot, showing the reference point for measurement of the maximum depth, as well as the five gauge points ($0.1mm$ biased mesh, 10×10 bearing mesh)

Once the target cylinder was impacted by the ball bearing, a crater was formed in the target.

In order to study the velocity and displacement along the crater, gauge points were placed on the impact site (shown in Figure 5.11). Data from the gauge points showed that the plastically deformed part eventually reached a period of oscillation within which plastic deformation had discontinued. Figure 5.12 shows a plot of the y -velocity versus time (left) and y -displacement versus time (right) for five gauge points along the crater surface in the $0.1mm$ biased mesh, 10×10 bearing mesh test, demonstrating the oscillations that occurred. It was also evident from Figure 5.12 that the gauge points reach an equilibrium position, which was considered to be the final depth of the crater.

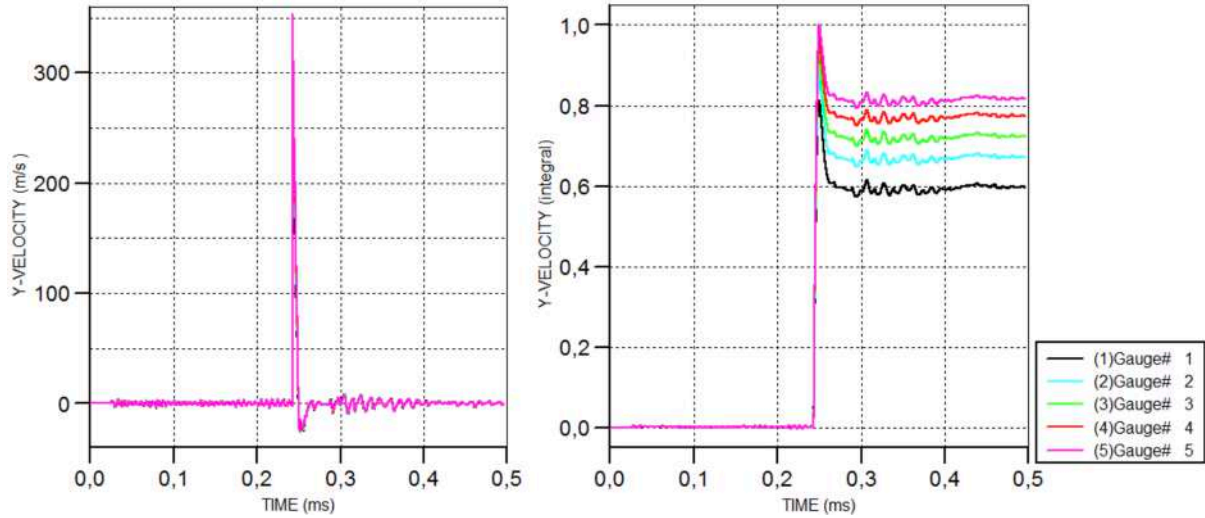


Figure 5.12: Plot of y -velocity versus time data (left) and y -displacement versus time (right) for five gauge points along the crater edge. The same trend is observed for the x -velocity.

It was also noted that the amount of oscillation after $t = 4ms$ was less than $0.01mm$. The crater depth was only recorded to the nearest $0.1mm$; therefore, the oscillation that occurred after $t = 4ms$ did not affect the final recorded crater depth. After a runtime of $4ms$, the simulation was stopped and the position of the nodes along the crater surface were exported from AUTODYN and plotted in Microsoft Excel to obtain the depth of the crater. An example of the crater nodal plot is shown in Figure 5.11. The crater depth was calculated by finding the difference in position (from the central axis of the cylinder) between the maximum depth and the reference point, which is defined as the nearest point adjacent to the lips of the crater at which the surface becomes level again.

The results of the mesh sensitivity study is summarised in Figure 5.13. The study showed that for the same target element resolution, the crater depth results varied less than 1% across different bearing mesh resolutions. This showed that the crater depths obtained were relatively insensitive to the bearing mesh resolutions, especially for target mesh sizes of $0.5mm$ and less. Target mesh resolution had a greater influence on the crater depth results. Simulations run with square meshes of $0.25mm$ or larger overestimated the crater depth by up to 104%. However, the simulated depth for target elements of $0.2mm$ and smaller were shown to approach the nominal

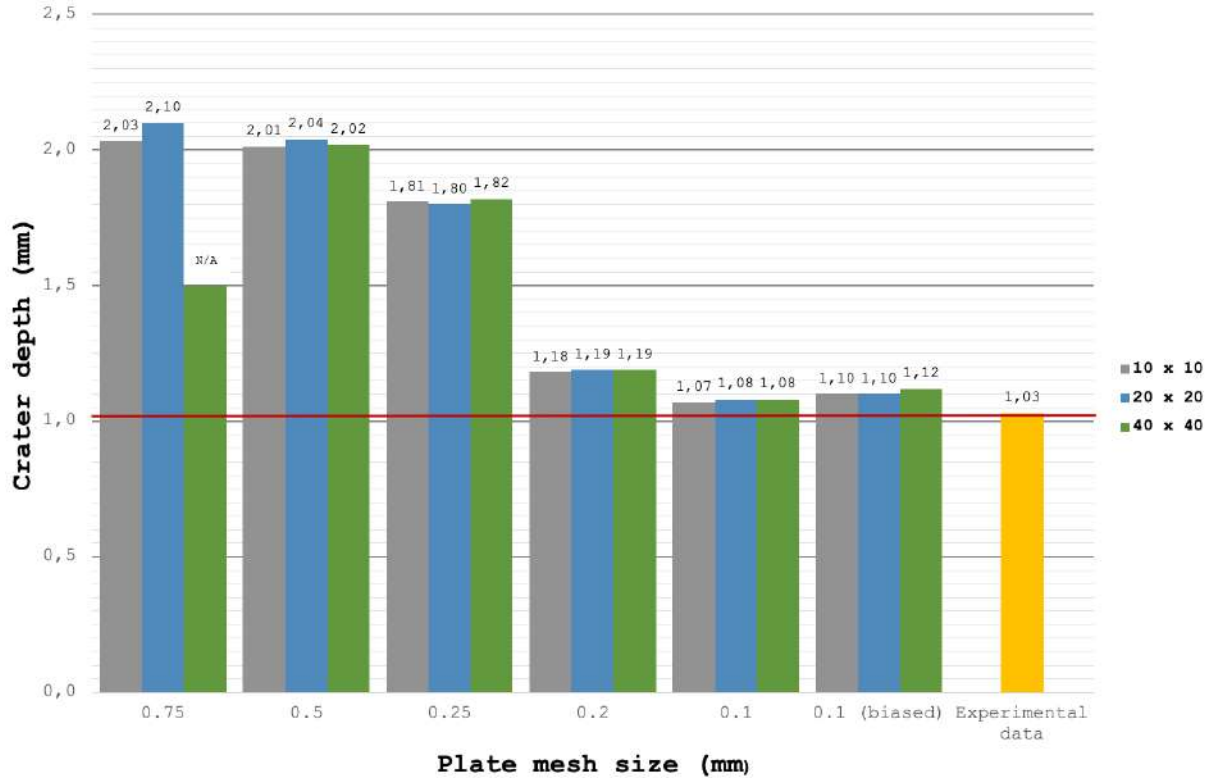


Figure 5.13: Graph showing the crater depths for different target mesh elements and ball bearing mesh elements

values obtained in the experiments. Mesh resolution of $0.2mm \times 0.2mm$, biased- $0.1mm$ and $0.1mm \times 0.1mm$ overestimated the crater depth by 14.5%, 6.9% and 3.8% respectively.

The simulation run with $0.1mm$ square target mesh, 10×10 -elements bearing mesh and $1mm$ air mesh resulted in a runtime of approximately 72 hours for $3.5ms$ of simulation time. The use of smaller elements would result in an excessively long runtime. One of the drawbacks of AUTODYN is the memory allocation issues when large number of elements are used in the mesh. Furthermore, the lack of parallel processing for all models discourages the use of very fine global meshes or extensively large models due to the considerable computational expense. Mesh resolutions finer than $0.1mm \times 0.1mm$ were therefore not considered.

A biased mesh of minimum element size $0.1mm$ was approximately 20% faster in terms of runtime and showed only 3% difference in crater depth error when compared to the $0.1mm$ square mesh. Refining the ball bearing mesh resolution increased the runtime significantly but produced negligible difference in simulated crater depth. Therefore, the $0.1mm$ biased target mesh and 10×10 -elements bearing mesh was chosen for the final blast test simulations.

The AUTODYN manual [78] recommends an Eulerian cell size smaller than the thickness of the Lagrangian part to avoid leakage. Since the thickness of the Lagrangian parts are $5mm$ and $10mm$ respectively, using Eulerian mesh of $1mm$ square elements meets this criterion.

5.5.5 2D Axial Plane Meshes

Based on the results of the mesh sensitivity analyses, the same mesh sizes were applied to the second numerical model which represented the experiment in the axial plane.

5.6 Simulation of Blast Load

The simulation of the blast load was achieved using two stages, namely detonation and unloading. The simulated target deformation was predominantly a result of bearing impact, rather than the blast load. Prior to impact, however, it is the blast load that determines the bearing behaviour, which is why mapping was not used in the initial stage.

5.6.1 Detonation Model

Detonation of the PE4 explosive, and the initial deformation and motion of the ball bearing was modelled in two separate planes, namely the radial plane and the axial plane. A schematic of the 2D numerical blast model in the radial plane is shown in Figure 5.2. Figure 5.3 shows the blast model in the axial plane. In both setups, air and the explosive were modelled using $1mm \times 1mm$ cells, and the ball bearings were modelled with 10 elements across the diameter. The Lagrangian target cylinder was modelled with $0.1mm \times 0.1mm$ square cells in the axial plane, and with $0.1mm$ biased mesh (described in section 5.5) in the radial plane. The clamped region of the cylinder was assigned a zero x -velocity and y -velocity boundary condition to represent a full encastre. The position and/or the number of bearings was varied for other loading scenarios. For consistency, the detonation model cell dimensions remained unchanged for all loading scenarios. Interaction of the explosive products, air, ball bearing and the target cylinder was accommodated by selecting the Automatic Langrange-Eulerian coupling option in AUTODYN. To capture the pressure wave propagation, 5 gauge points were placed in the air cell around the midpoint of the length/circumference of the cylinder inner wall (see Figures 5.2 and 5.3).

5.6.2 Unloading Model

5.6.2.1 Deactivation of Gas Products

Shortly before the bearing impacted the cylinder wall, gauge points were added onto the target cells in the vicinity of bearing impact area to monitor the crater node oscillation. Once the target craters have passed their peak depth and have begun to oscillate about their final displacement, the gases present in the model no longer affect the response of the target. The ball bearing velocities were shown to converge to a constant value almost immediately after it had impacted the target and bounced back, as shown in Figure 5.14. This implied that the gas products in the model no longer affected the response of the ball bearing after it impacted the target.

Therefore, to save computation time of the numerical model, the gas products were deactivated after the crater depth had reached its maximum and the ball bearing had entered its rebound trajectory. The simulation was terminated when 1) all the ball bearings had impacted the target and 2) the amount of oscillation about the final depth fell below $0.01mm$ for all target craters.

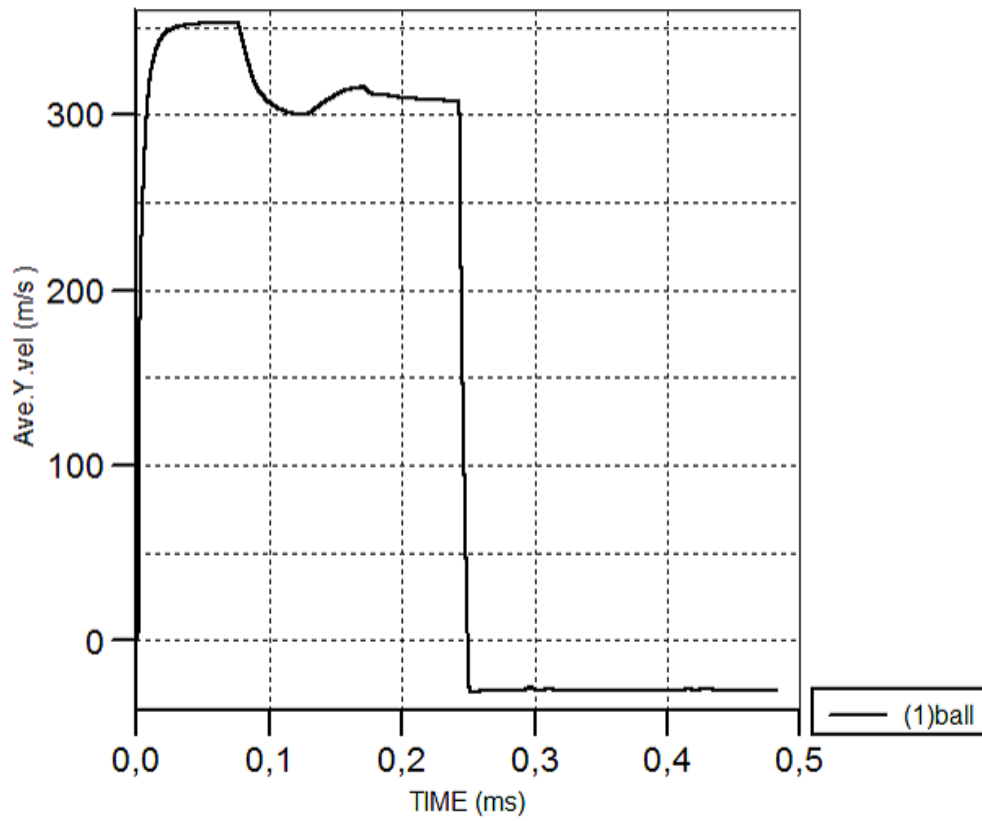


Figure 5.14: Plot of y -velocity versus time of the ball bearing in single bearing, 2D radial plane simulation

Since the crater depth was only recorded to the nearest 0.1mm , the oscillation that occurs beyond this point was considered to have no effect on the final recorded crater depth. The time taken to reach the termination criterion varied per simulation; the simulations were generally inspected $1\text{ms} - 1.5\text{ms}$ after all the ball bearings had impacted the target.

Chapter 6

Numerical Results I: Numerical Models

Results of the numerical simulations that are equivalent to the experiments are presented in this chapter. For the purposes of this chapter, 'target' refers to the target cylinder (refer to Figures 5.2 and 5.3 from Chapter 5). The inner edges of the target (i.e. the 2D equivalent of inner faces in 3D) are defined as Edge R-1 in radial configuration (as shown in Figure 6.1 a)) and Edges A-1, A-2 and A-3 in axial configuration (as shown in Figure 6.1 b)).

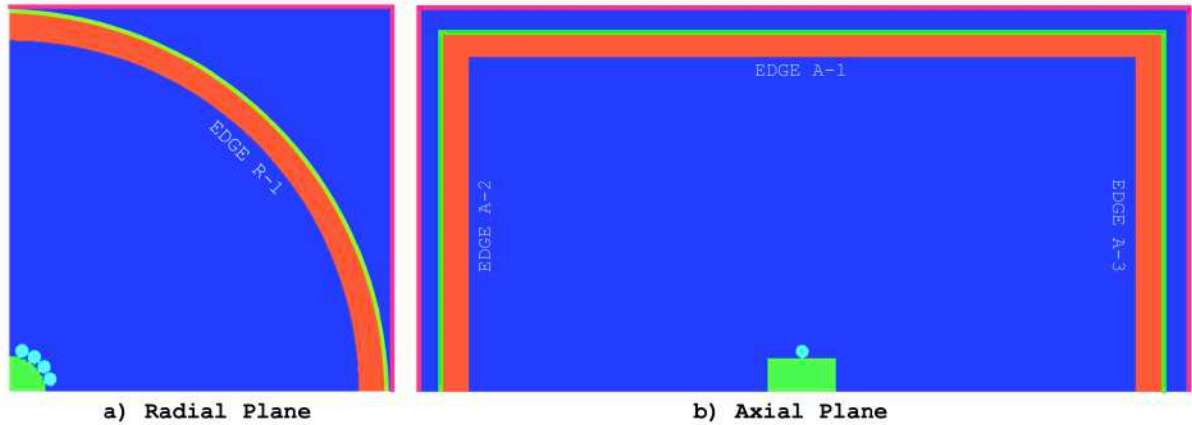


Figure 6.1: Naming convention for target geometry in a) radial configuration and b) axial configuration

The numerical results were benchmarked against the crater depths measured in the experiments. In the radial plane configuration, each ball bearing was equivalent to a single ball bearing, whereas each ball bearing in the axial plane configuration was equivalent to a row of ball bearings in the experiments. Therefore, in the radial plane simulations, the 4-ball bearings test was equivalent to a quarter-symmetric section of a row of ball bearings wrapped around the circumference of the cylinder. In the axial plane simulation series, the 1-ball bearing test, 2-ball bearings test and the 5-ball bearings test were equivalent to the 1-row, the 2-row and the 5-row experiment condition respectively.

6.1 Pressure Contours

6.1.1 Radial Plane Simulations

6.1.1.1 4-Ball Bearings Configuration

The transient pressure contours for the radial plane 4-ball bearings simulation, showing the interaction between the blast wave and the ball bearings, are presented in Figure 6.2. High pressures were developed around the centre of the cylinder as the blast wave expanded radially towards the target. The ball bearings acted as a barrier disturbing the initial blast wavefronts, which were no longer spherical (as was the case when there was no ball bearing.) The blast wave exhibited repetitive reflection back and forth after its initial impact with the target at $t = 0.035ms$, until the ball bearing impacted the target at $t = 0.145ms$. It was also observed that the blast wave propagation augmented or impeded the motion of the ball bearing, depending on the direction of the wave.

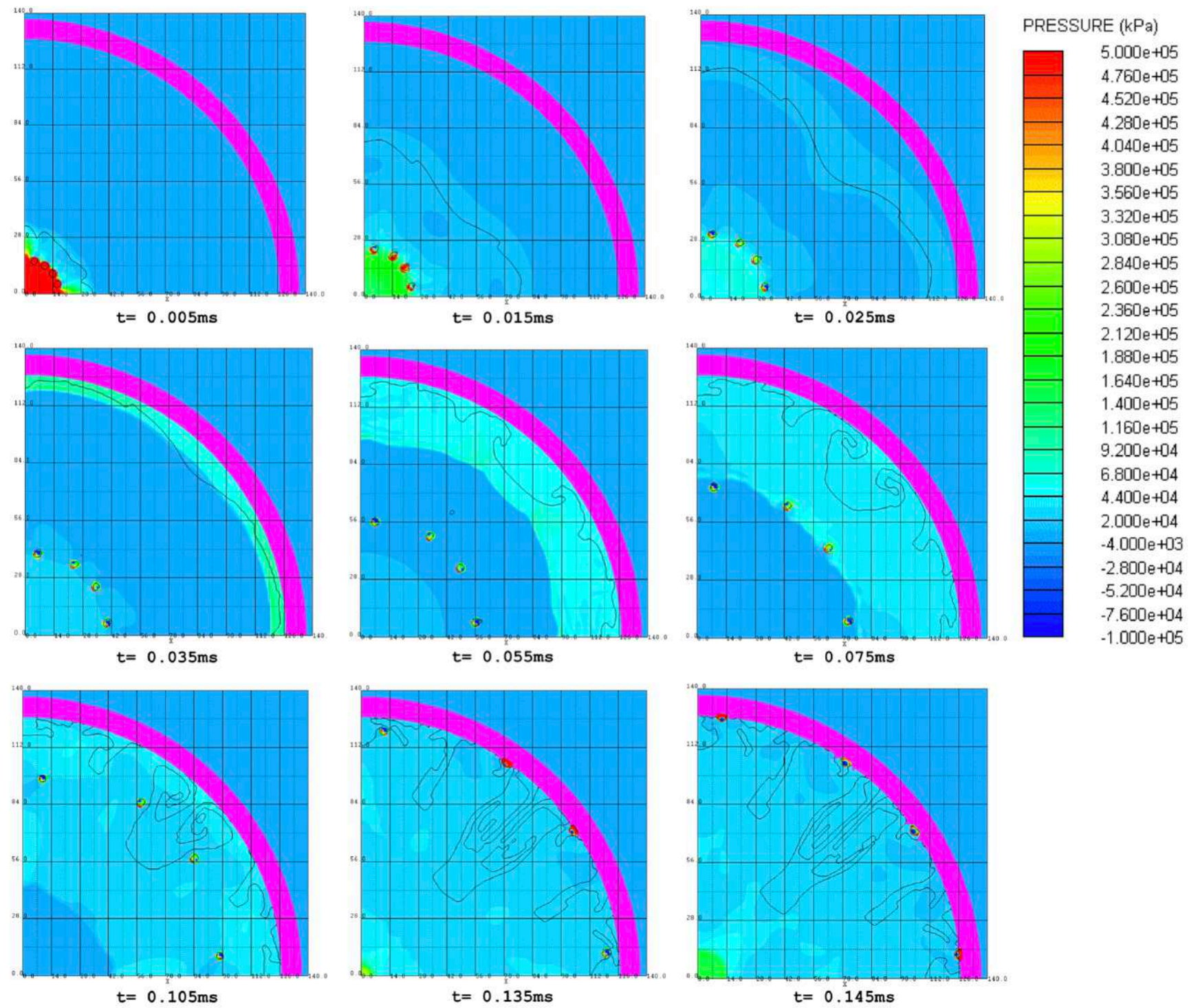


Figure 6.2: Pressure contour plots of radial plane, 4-ball bearings simulation ranging from $t = 0.005\text{ms}$ to $t = 0.145\text{ms}$

6.1.2 Axial Plane Simulations

6.1.2.1 Single-Ball Bearing Configuration

The transient pressure contours for the axial plane 1-ball bearing simulation, showing the interaction between the blast wave and the ball bearing, are presented in Figure 6.3. Similar to the radial plane simulation, high pressures were developed around the centre of the cylinder as the blast wave expanded. Similarly, the ball bearings acted as a barrier disturbing the blast wavefronts, modifying the shape of the wavefront. The pressure wave first impacted edges A-2 and A-3, before reflecting off edge A-1, as defined in Figure 6.1 a). The blast wave exhibited repetitive reflection back and forth, after its initial target impact at $t = 0.025ms$ until $t = 0.205ms$. The pressure wave interference at the corners resulted in the formation of high-pressure zones (around 5 times higher in magnitude when compared to adjacent wall pressure) as observed by Salvado [39] for a study on blast confined cubic boxes. It was also observed that the blast wave propagation augmented or impeded the ball bearing motion, depending on the direction of the blast wave.

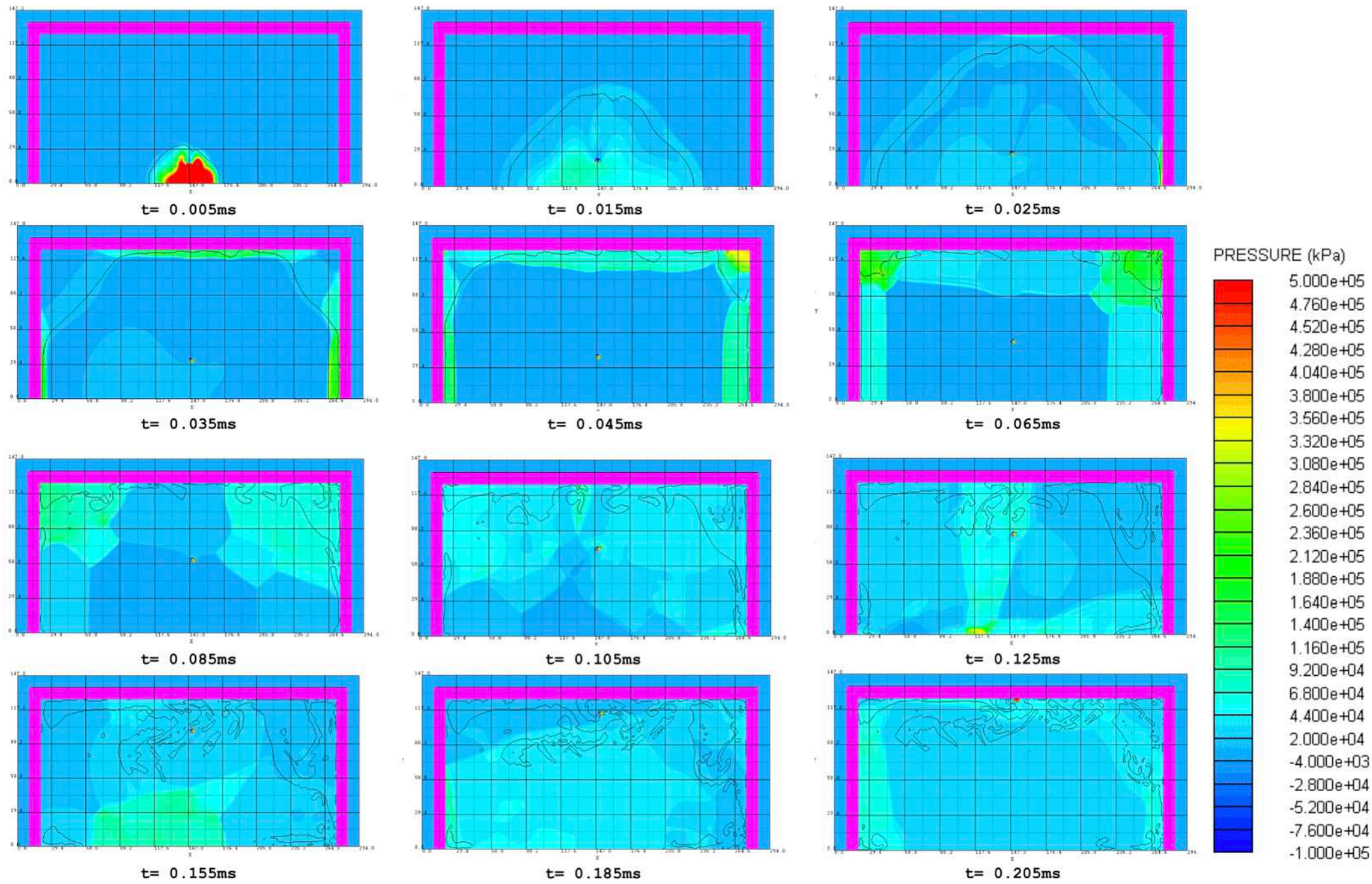


Figure 6.3: Pressure contour plots of axial plane, 1-ball bearing simulation ranging from $t = 0.005\text{ms}$ to $t = 0.205\text{ms}$

6.1.2.2 Multiple-Ball Bearings Configurations

The pressure contours for a series of simulations with 1, 2 and 5 ball bearings, compared with bare charge i.e. '0' ball bearing, in the axial plane at various points in time are shown in Figures 6.4 and 6.5. The differences in wavefront shapes and pressure contours under different configurations were more significant during the earlier stages of the simulation, between $t = 0ms$ - $0.015ms$. At $t = 0.01ms$, it was observed that the ball bearings acted as "barriers" and affected the overall shape of the pressure contour immediately after detonation. High-pressure zones were again observed in the corners, irrespective of the total number of ball bearings. The pressure towards the centre of the cylinder increased from $260MPa$ for 1-ball bearing test, to $980MPa$ for 5-ball bearings test. However, the pressure wave propagation slowed down as the number of ball bearings was increased, as observed at $t = 0.03ms$. At this point in time, the wavefront had reached edge A-1 under the 0, 1 and 2-ball bearings configurations. For the 5-ball bearings test, however, the wavefront had not reached the edge yet, implying that the wavefront was expanding at a slower rate compared to other configurations involving fewer ball bearings.

The pressure waves dissipated with time. The pressure wavefronts were shown to exhibit very similar patterns irrespective of the number of ball bearings involved, as can be observed at $t = 0.045ms$ in Figure 6.5. It was also observed that although the ball bearings caused disruption and turbulence, the global pressure response was not greatly affected by the presence of the ball bearing, since the area occupied by the ball bearing was very small compared to the overall occupied area of the cylinder.

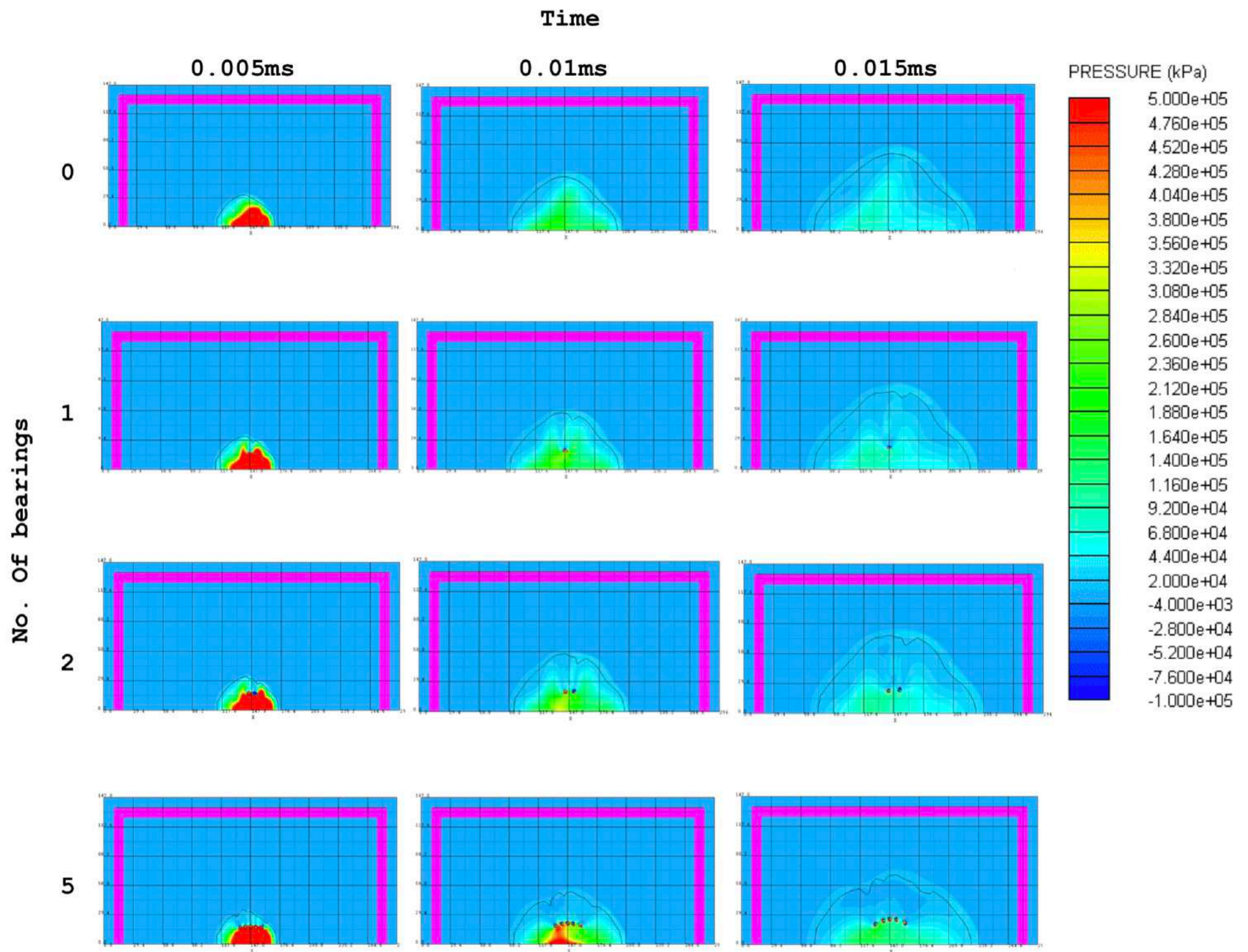


Figure 6.4: Series of pressure contour plots of axial plane, 1-, 2- and 5-ball bearings simulation ranging from $t = 0.005ms$ to $t = 0.015ms$

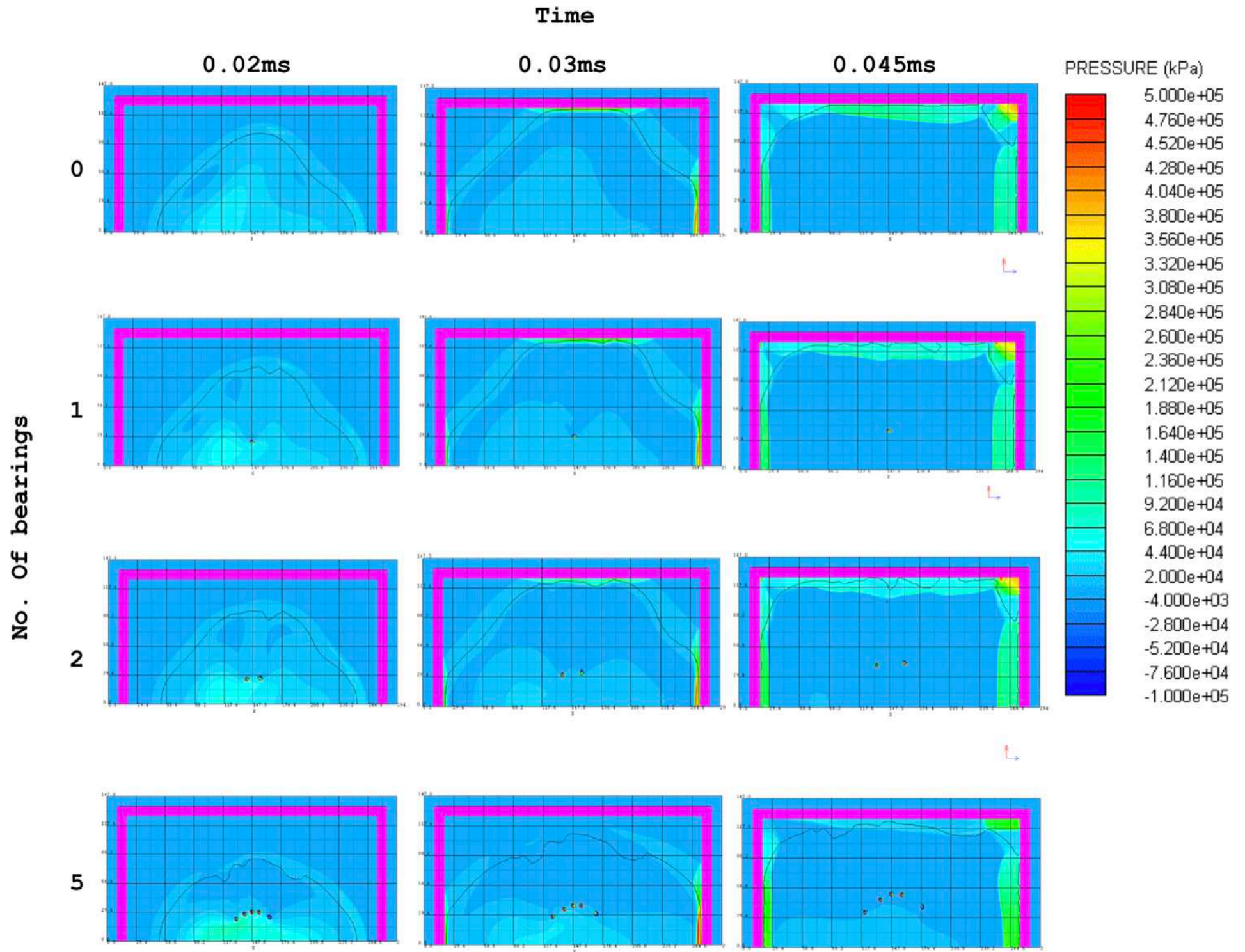


Figure 6.5: Series of pressure contour plots of axial plane, 1-, 2- and 5-ball bearings simulation ranging from $t = 0.02ms$ to $t = 0.045ms$

6.2 Velocity Profiles

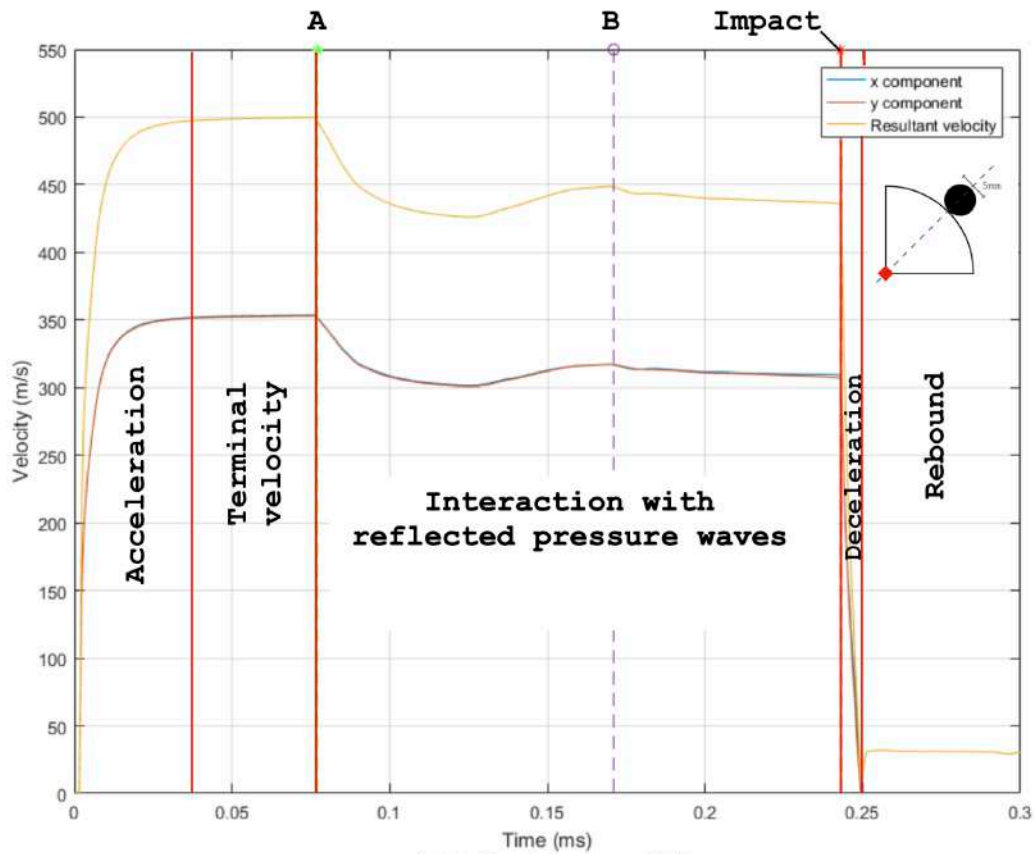
6.2.1 Definition of the General Velocity Profile

The velocity-time data for each ball bearing was exported from AUTODYN and plotted in MATLAB to investigate the maximum velocity gain and the velocity upon impact. The velocity profiles of the ball bearings from the different tests followed a general pattern consisting of six different stages, with the peak value and the duration of each stage differing per test configuration. The six different stages were identified as: initial acceleration, terminal velocity, pressure wave interaction, impact, deceleration, and rebound.

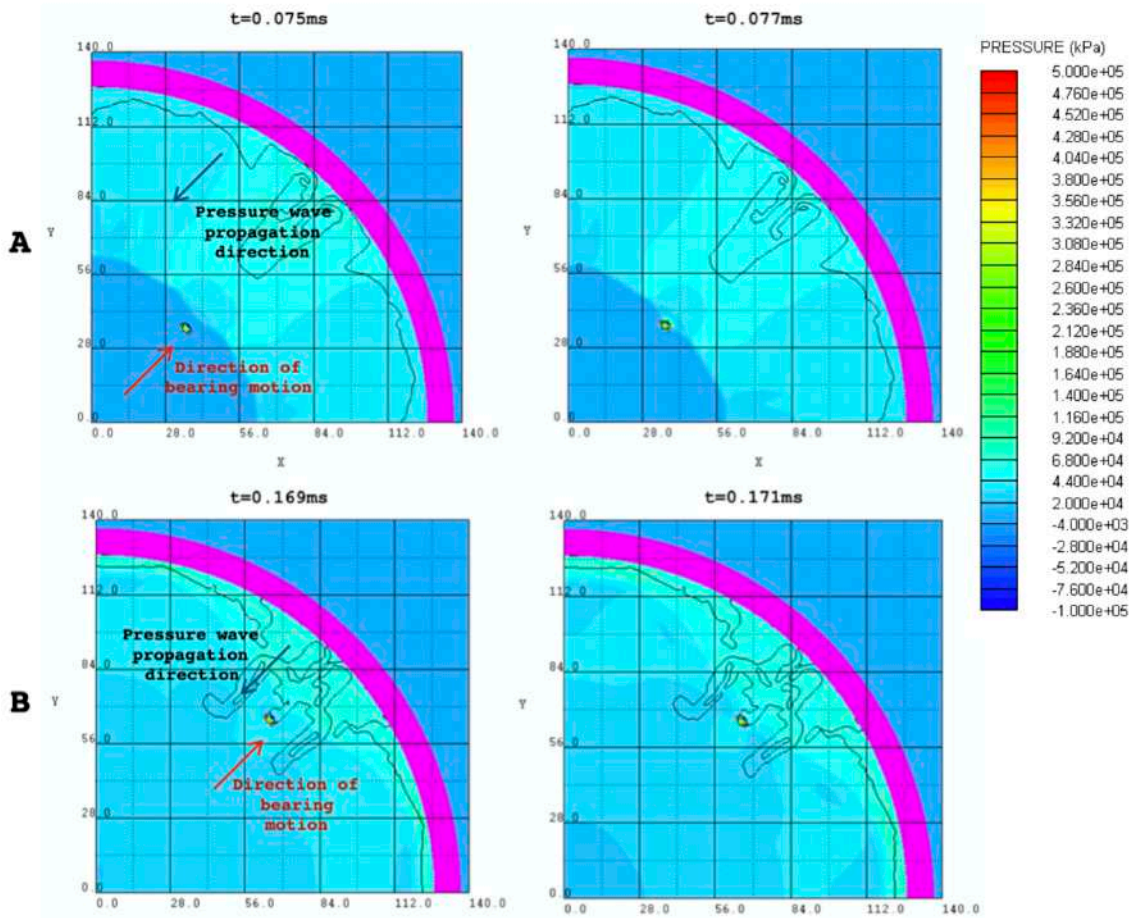
As an example, the x-velocity, y-velocity and resultant velocity profiles of the radial plane simulation with a single ball bearing located outside the charge at 45° from the x -axis is shown in Figure 6.6 a). Following charge detonation, the ball bearing rapidly accelerated (*initial acceleration stage*) over a very short period of time (typically around 0.01ms). The acceleration initially occurred at a linear rate with decreasing rate as the ball bearing approached its peak velocity. The ball bearing velocity then stabilised at its "terminal" velocity (*terminal velocity stage*), which corresponded to its peak velocity gain of 500m/s in Figure 6.6 a). The ball bearing maintained this peak velocity over a short period of time, until the blast pressure waves travelled back after being reflected off the target wall and interacted with the ball bearing (*pressure wave interaction stage*).

The blast wave propagation was shown to augment or impede the ball bearing motion, depending on the direction of the wave. The ball bearing travelled at a constant velocity until the reflected pressure wave hit the ball bearing in a direction opposite to the motion of the ball bearing, consequently decreasing the velocity. The time at which the pressure wave hit the ball bearing coincided exactly with the velocity drop labelled as A, as shown in Figure 6.6 b). The ball bearing velocity increased prior to point B, as the pressure wave was travelling in the same direction as the ball bearing. Around point B, the ball bearing entered a higher pressure zone in the vicinity of the target cylinder, and as a result of this impedance, the ball bearing velocity dropped again. The pressure wave interaction became more complex as the reflected waves interfered with each other, and the ball bearing velocity continued to drop until impact. The impact velocities were typically 10 to 20% lower than the peak value.

Post-impact, however, the pressure wave had dissipated significantly and did not affect the ball bearing velocity as significantly. After the ball bearing reached and impinged the target (*impact stage*) it decelerated rapidly at an initially linear rate (*deceleration stage*). The rate of deceleration also decreased as the ball bearing reached its final "rebound" velocity (*rebound stage*). The ball bearing was observed to bounce back after impacting the target.



a) Velocity profiles



b) Pressure contours at points A and B

Figure 6.6: Figure showing a) x -velocity, y -velocity and resultant velocity-time profiles of radial plane 1-ball bearing simulation, labelled with the six stages, and b) pressure contour plots showing pressure wave interaction with the ball bearing at points A and B

6.2.2 Radial Plane Simulations

6.2.2.1 4-Ball Bearings Configuration

Resultant Velocity Profile

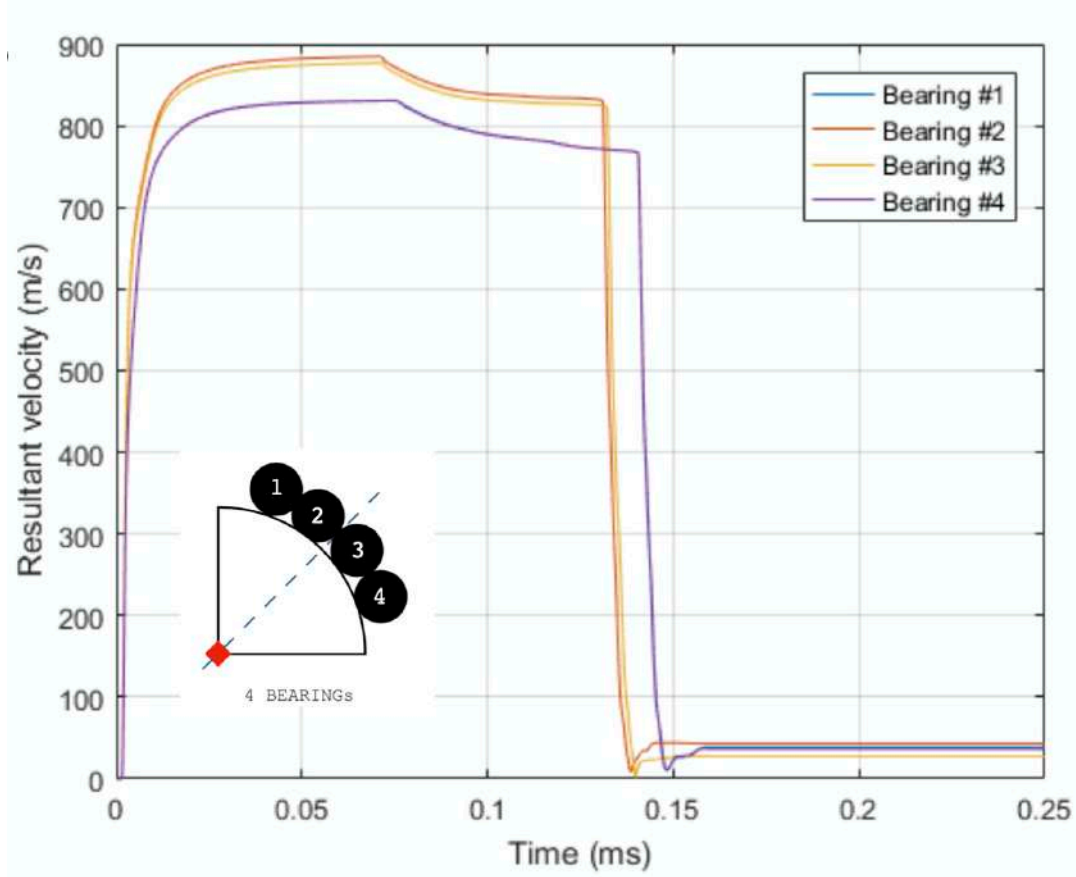


Figure 6.7: Resultant velocity-time profiles from radial plane 4-ball bearings simulation

The resultant velocity profiles for each ball bearing labelled #1 to #4 in the radial configuration are shown in Figure 6.7. The six stages of the resultant velocity (initial acceleration, terminal velocity, pressure wave interaction, impact, deceleration and rebound) were also present in all four velocity profiles of each ball bearing. The initial rate of acceleration were identical for the velocity of all the ball bearings immediately after detonation. However, the velocities diverged around $t = 0.005s$ after detonation. The profiles corresponding to ball bearings #2 and #3 were almost identical to each other, and the velocity profiles corresponding to ball bearing #1 and #4 were shown to completely overlap as well. The results showed that the ball bearing motion was symmetrical along the 45° line through the quarter-circle representing the charge (shown as a blue broken-line on the charge diagram in Figure 6.7).

Ball bearings #1 and #4 had peak velocity values of approximately $830m/s$, whereas ball

bearings #2 and #3 had higher peak velocity values of approximately 880m/s . It should be noted that the reflected pressure waves passed the ball bearings at different times, as shown by the delayed velocity drop in ball bearings #1 and #4 as compared to ball bearings #2 and #3. This corroborated the observation that ball bearings #2 and #3 had higher peak velocities than ball bearings #1 and #4, and therefore would have travelled faster, reaching the reflected wavefront before ball bearings #1 and #4.

The percentage velocity drop caused by the wave-ball bearing interaction were in the range of 6 to 7 % for all four ball bearings. Ball bearing #2 impinged the target cylinder wall (edge R-1, as defined in Figure 6.1 a)) first, closely followed by ball bearing #3. There was a delay of approximately 0.01ms before ball bearings #1 and #4 reached the target as expected, as they were travelling at lower velocities compared to the other two ball bearings. The rate of deceleration post-impact were equal for all four ball bearings. Ball bearings #1 and #4 had the same rebound velocities, whereas the rebound velocities of ball bearings #2 and #3 were shown to diverge before stabilising.

X– and Y–velocity Profiles

Figures 6.8 and 6.9 show the x –velocity profiles and the y –velocity profiles of each ball bearing in the radial, 4-ball bearing simulation, respectively. The x – and y –velocity profiles followed a pattern very similar to the resultant velocity profiles with the peak value and the duration of each stage differing, with the exception of the x –velocity profile of ball bearing #1 and the y –velocity profile of ball bearing #4. Unlike the resultant velocity profiles, these two profiles exhibited a rapid fluctuation in velocity early on in the simulation. The x – and y –velocity profiles were shown to intercept the horizontal axis, indicating zero velocity and therefore a change in direction, before reaching a final rebound velocity in the direction opposite to initial ball bearing motion.

It should be noted that the ball bearing with the highest peak x –velocity also exhibited the lowest peak y –velocity. and vice versa. Looking at the x –velocity profiles corresponding to each ball bearing, it was observed that the peak x –velocity increased moving clockwise from ball bearing #1 to #4. The opposite was true for the y –velocity profiles, i.e. the peak x –velocity decreased moving clockwise from ball bearing #1 to #4. It was also observed that the x –velocity profile corresponding to ball bearing #1 was identical in shape to the y –velocity profile corresponding to ball bearing #4; the same applies to other pairs of ball bearings that are positioned symmetrically opposite to each other. In other words, the x –velocity profile corresponding to a ball bearing was identical in shape to the y –velocity profile of another ball bearing that is positioned symmetrically opposite relative to the 45° line through the quarter-circle representing the charge.

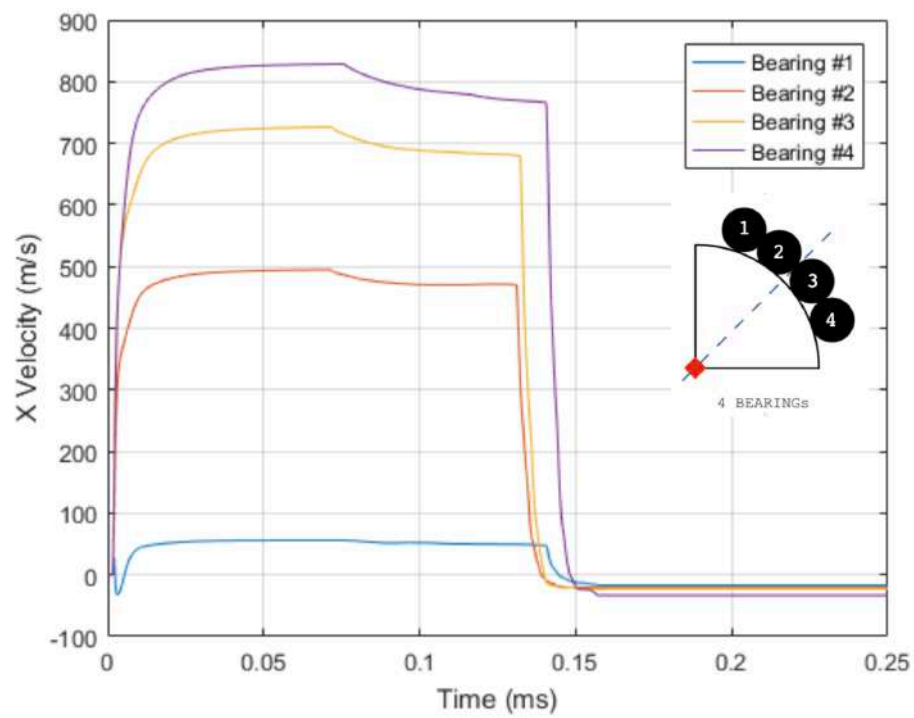


Figure 6.8: x velocity-time profiles from radial plane 4-ball bearings simulation

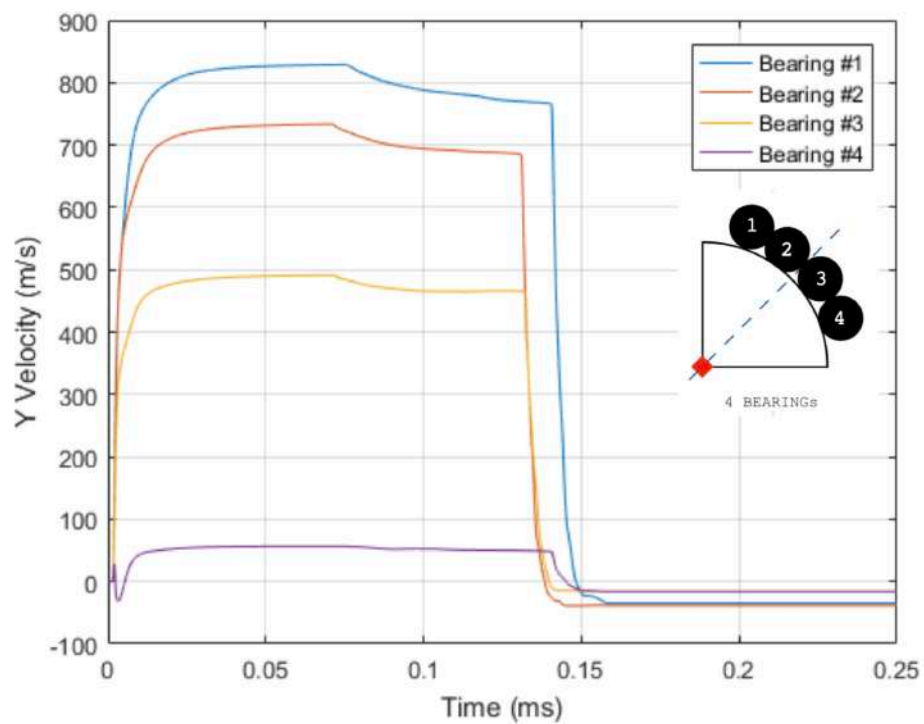


Figure 6.9: y velocity-time profiles from radial plane 4-ball bearings simulation

6.2.3 Axial Plane Simulations

6.2.3.1 1-Ball Bearing Configuration

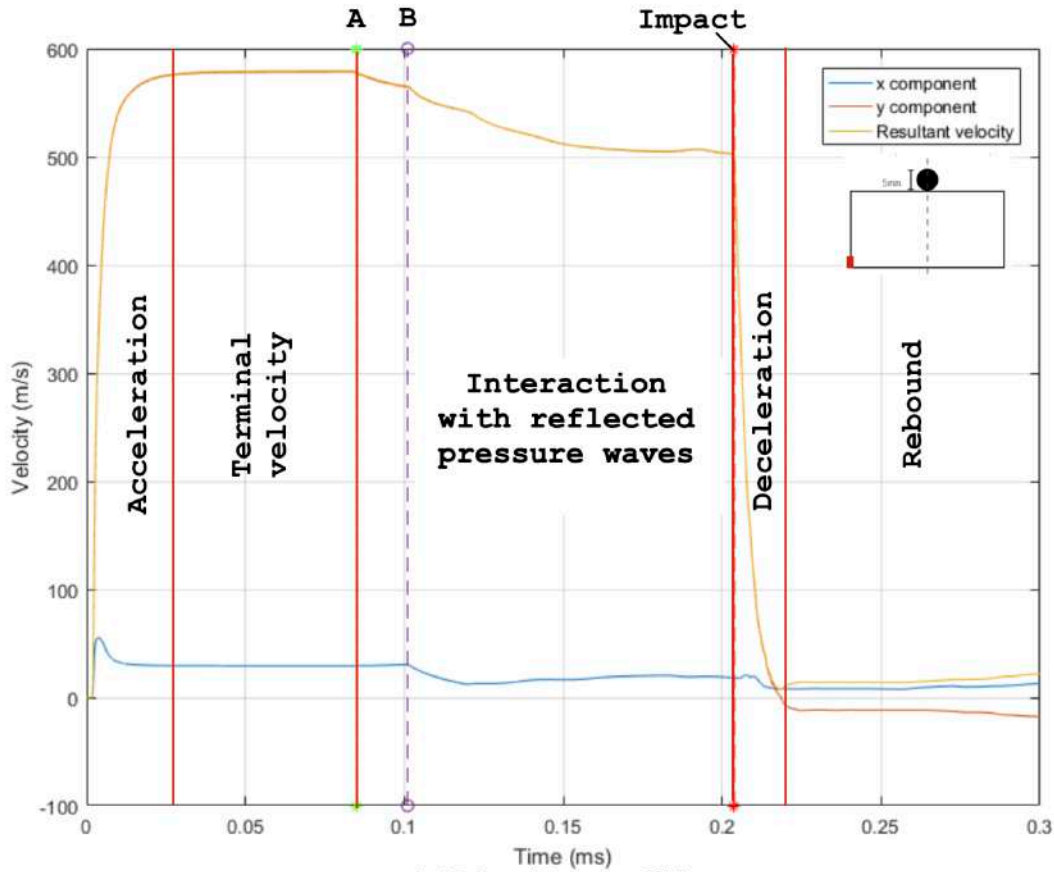
Resultant Velocity Profile

The resultant velocity profile of the ball bearing in the axial, 1-ball bearing simulation is shown in Figure 6.10 a). Following the charge detonation, the ball bearing rapidly accelerated over approximately $0.01ms$. The acceleration occurred at a linear rate initially, but exhibited decreasing rate as the ball bearing approached its peak velocity of $580m/s$. The blast pressure waves travelled back after being reflected off the target wall and interacted with the ball bearing again.

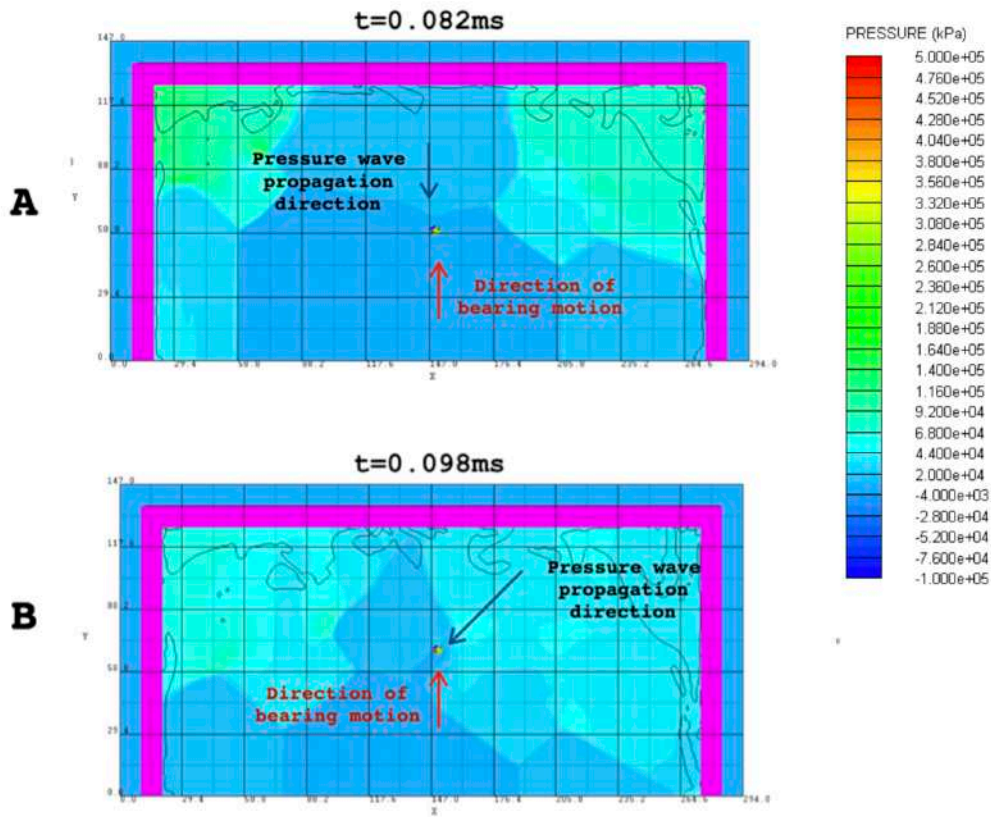
The ball bearing travelled at a constant velocity until the reflected pressure wave interacted with it in the direction opposite to the direction of motion of the ball bearing, hence causing the ball bearing to decelerate. The time at which the pressure wave interacted with the ball bearing coincided exactly with the velocity drop labelled as A, as shown in Figure 6.10 b). It was shown that the X velocity was not affected by the initial ball bearing-wave interaction, since the pressure wave was travelling in the y direction. At point B, the pressure wave diagonally reflected off the corner, and gained velocity in both x - and y -directions. This diagonally reflected wave hit the ball bearing (again, opposite to its positive direction of motion), causing the x velocity to drop (shown at point B). The pressure wave interaction became more complex as the reflected waves interacted with each other, as well as the ball bearing. The ball bearing velocity had dropped from $580m/s$ to $503m/s$ (13%) by the time of impact. Post-impact, however, the pressure wave had dissipated significantly and did not affect the ball bearing velocity as significantly. After the ball bearing impinged the target it decelerated rapidly, until the ball bearing reached its final velocity of $28m/s$.

X- and Y-velocity Profiles

The x - and y -velocity profiles of the ball bearing in the axial, 1-ball bearing simulation are also shown in Figure 6.10 a). The y -velocity component was significantly higher than the x -velocity component; the peak y -velocity was $578m/s$, whereas the corresponding x -velocity was $29m/s$ (only 5% of the y -velocity value). Therefore, the x -velocity component made an insignificant contribution to the resultant velocity whilst the y -velocity profile was almost identical to the resultant velocity in shape. It was also noted that the drop in x -velocity did not occur at the same point in time as the drop in y -velocity. This was due to the direction at which the reflected wavefronts impacted the ball bearing. The waves first reflected off edge A-1, and the reflected wave travelled predominantly in the y direction, interacting with the ball bearing first. The waves then reflected off the corners, which had both x - and y -velocity components followed and consequently interacted with the ball bearing again.



a) Velocity profiles



b) Pressure contours at points A and B

Figure 6.10: Figure showing a) the x -velocity, y -velocity and resultant velocity-time profiles of axial plane 1-ball bearing simulation, labelled with the six stages, and b) pressure contour plots showing pressure wave interaction with the ball bearing at points A and B

The y -velocity profile intercepted the horizontal axis, indicating a zero velocity followed by a change in direction when impacting the target and rebounding. The x -velocity component, on the contrary, was more unpredictable in its behaviour and did not have the same shape as the resultant velocity profile. The x -velocity profile did not intercept the horizontal axis, which implied that the ball bearing did not change direction of motion in the x -component.

The x -velocity plot showed a "spike" very early on in the simulation, when the velocity was initially shown to accelerate in the positive x -direction rapidly and decelerate suddenly before stabilising to a constant velocity value. This behaviour was most likely due to the interaction between the detonation wavefront and the ball bearing during the detonation process. Figure 6.11 shows the progression of the detonation wave through the cylindrical charge. Once the detonation was initiated, the reaction propagated through the charge from the detonation end towards the opposite end. At time $t = 0.002\text{ms}$, the detonation wave had reached the ball bearing. Since the charge to the right of the ball bearing had not yet been detonated, the high pressure created by the detonation wave to the left of the ball bearing was able to propel the ball bearing in the positive x -direction.

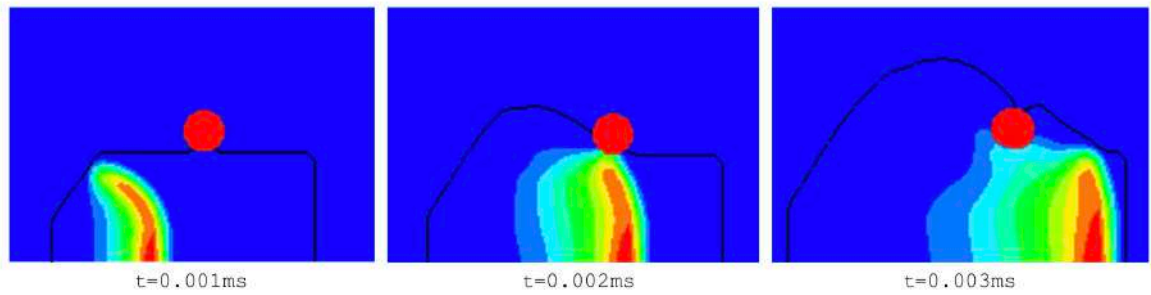


Figure 6.11: The progression of the detonation wave through the cylindrical charge and its interaction with the ball bearing (blue indicates low pressure, and red indicates high pressure)

However, the detonation process is a very rapid process with a short duration. Once the charge had been detonated completely, the source of propulsion had passed the ball bearing and the ball bearing stopped accelerating. König [47] stated that the accelerative drag due to axially outward component of release-wave gas flow over fragment was the driving factor behind the divergence of fragments in pre-formed fragments. Assuming that the ball bearings in this study are analogous to pre-formed fragments, the expansion of the rarefaction waves and the pressure of the dissipating gas products were the primary determining factors in the motion of the ball bearing once the charge was completely detonated. The approximately hemispherical shape of the wavefront determined the direction in which the ball bearing travelled, causing it to travel in the opposite direction to the initial motion, until the velocity stabilised to a constant value. The ball bearing was placed near the centre of the "hemispherical" wavefront and the ball bearing motion was predominantly in the positive y -direction with a comparably small x -component.

6.2.3.2 2-Ball Bearings Configuration

Resultant Velocity Profiles

The resultant velocity profiles for each ball bearing in the axial, 2-ball bearings simulation are shown in Figure 6.12. The initial rate of acceleration were identical for both velocity curves. However, the velocities diverged around $t = 0.005s$. Unlike in the radial plane simulation, the ball bearing velocities were not symmetrical along the vertical midplane of the rectangle representing the charge.

Ball bearing #1 had a peak velocity value of approximately $680m/s$, whereas ball bearing #2 had peak velocity value of approximately $750m/s$. It should be noted that the reflected pressure waves passed the ball bearings at different times, as shown by the delayed velocity drop in ball bearing #1 as compared to ball bearing #2. This corroborates the observation that ball bearing #2 had higher peak velocity than ball bearing #1 and therefore would have travelled faster, reaching the reflected wavefront before ball bearing #1.

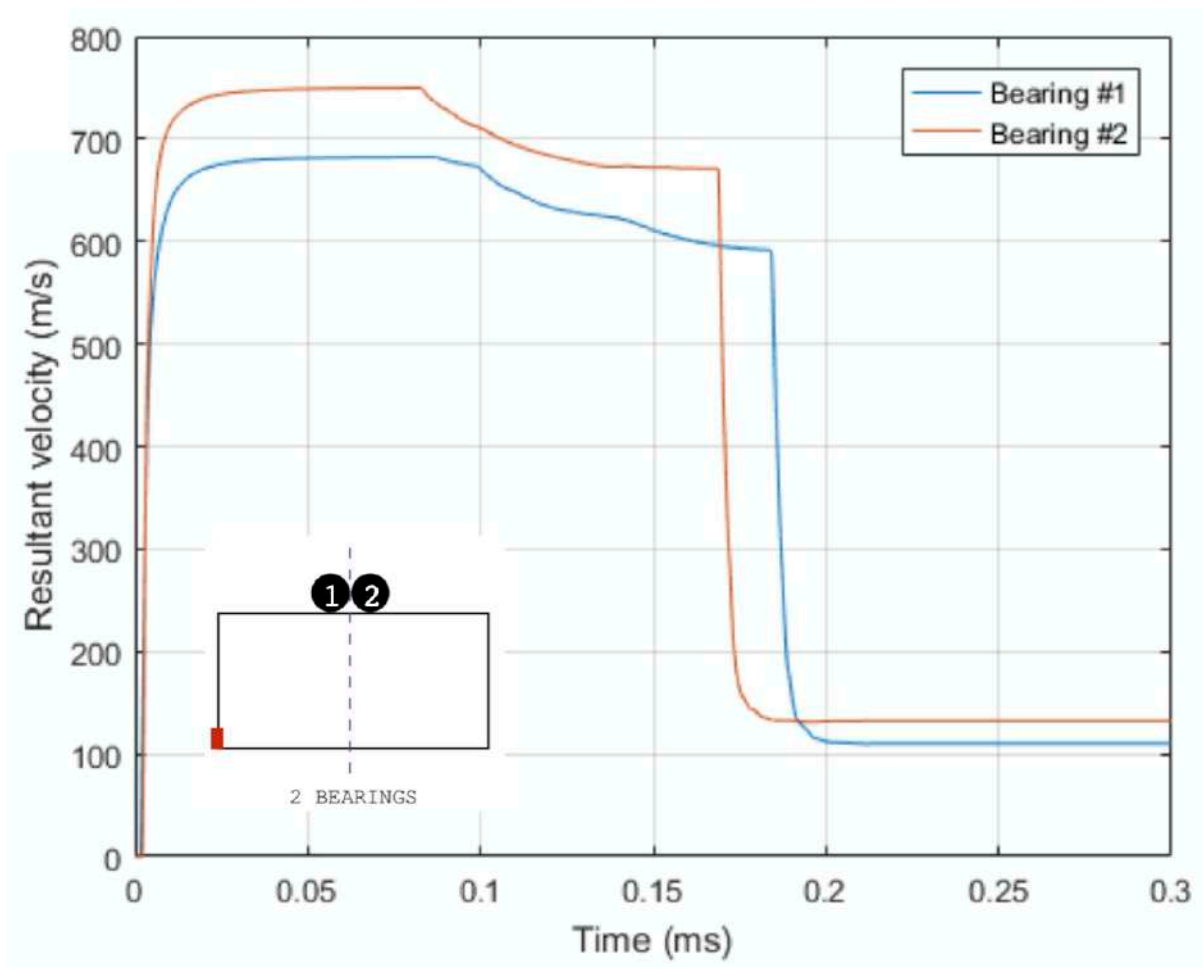


Figure 6.12: Resultant velocity-time profiles of axial plane 2-ball bearings simulation

The percentage velocity drop caused by the wave-ball bearing interaction was 13% for ball bearing #1 and 11% for ball bearing #2. Ball bearing #2 impinged the target 0.02ms before ball bearing #1 impacted the target. The rate of deceleration post-impact was equal for both ball bearings. The rebound velocity of ball bearing #2 stabilised at a constant rebound velocity value of 138m/s first, followed by ball bearing #1 stabilising to a lower rebound velocity value of 110m/s approximately 0.02ms later.

X- and Y-velocity Profiles

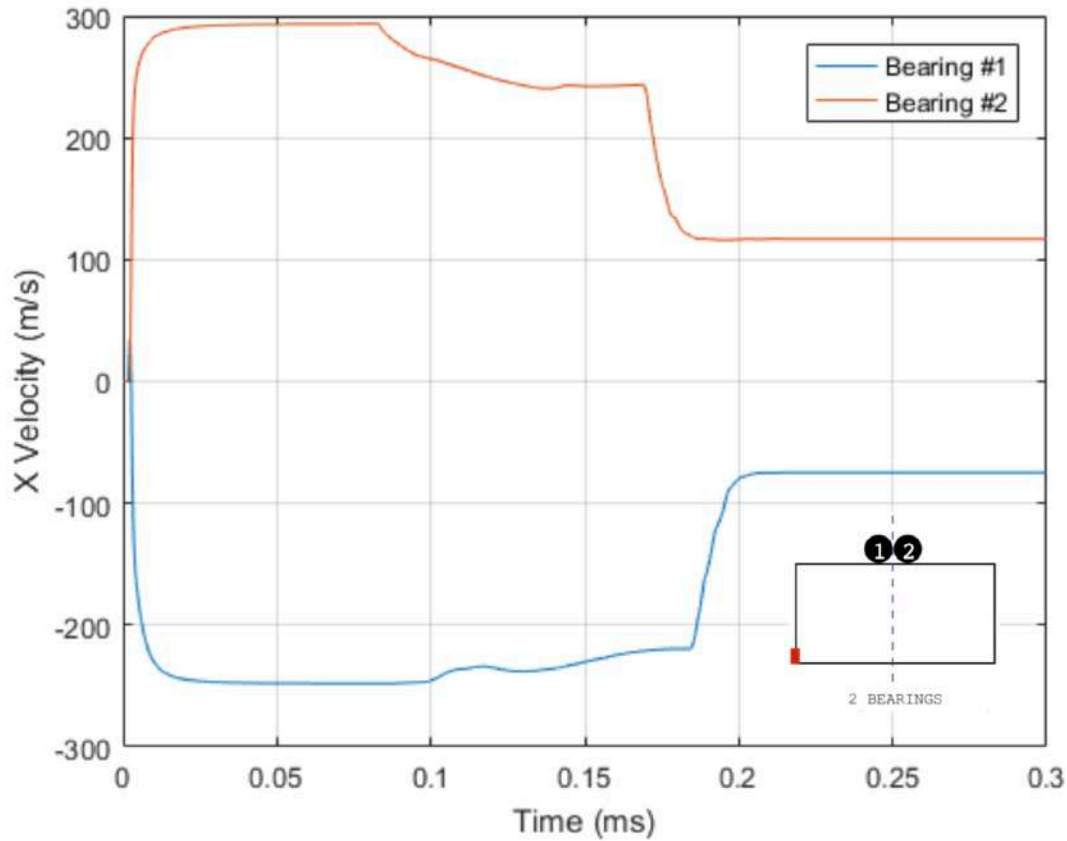


Figure 6.13: x velocity-time profiles of axial plane 2-ball bearings simulation

Figures 6.13 and 6.14 show the x -velocity profiles and the y -velocity profiles, respectively, of each ball bearing in the axial, 2-ball bearings simulation. The y -velocity components was significantly higher than the x -velocity components; the peak y -velocity of ball bearing #1 was 682m/s, and the x -velocity at that point in time was -248 m/s, which was only 39% of the y -velocity value. Similarly, the peak y -velocity of ball bearing #2 was 749m/s, and the x -velocity at the same point in time was -293 m/s, which was 43% of the y -velocity value. Therefore, the x -velocity component made a less significant contribution to the resultant velocity, and the overall shape of the resultant velocity profiles resembled the y -velocity profiles. The drop in absolute velocity of the ball bearing after its initial interaction with the reflected

wave was clear in the x -velocity profiles as well.

The drop in x -velocity occurred around the same point in time as the drop in y -velocity for both ball bearings, implying that the reflected waves that initially impacted the ball bearing had both x - and y -velocity components. The y -velocity profiles intercepted the horizontal axis, indicating zero velocity followed by a change in direction when impacting the target and rebounding, before reaching a final rebound velocity in the direction opposite to initial ball bearing motion. The x -velocity components, on the contrary, did not have the same shape as the resultant velocity profiles. Ball bearing #1 travelled in the opposite x direction to ball bearing #2, i.e. in the negative x direction. The x -velocity profiles did not intercept the horizontal axis, which implied that the ball bearings did not change direction of motion in the x -component. The x -velocity profile of ball bearing #1 exhibited a short spike in velocity early on in the simulation, similar to the behaviour shown in the x -velocity profile in Figure 6.10 a), for the same reasons as explained in the previous section.

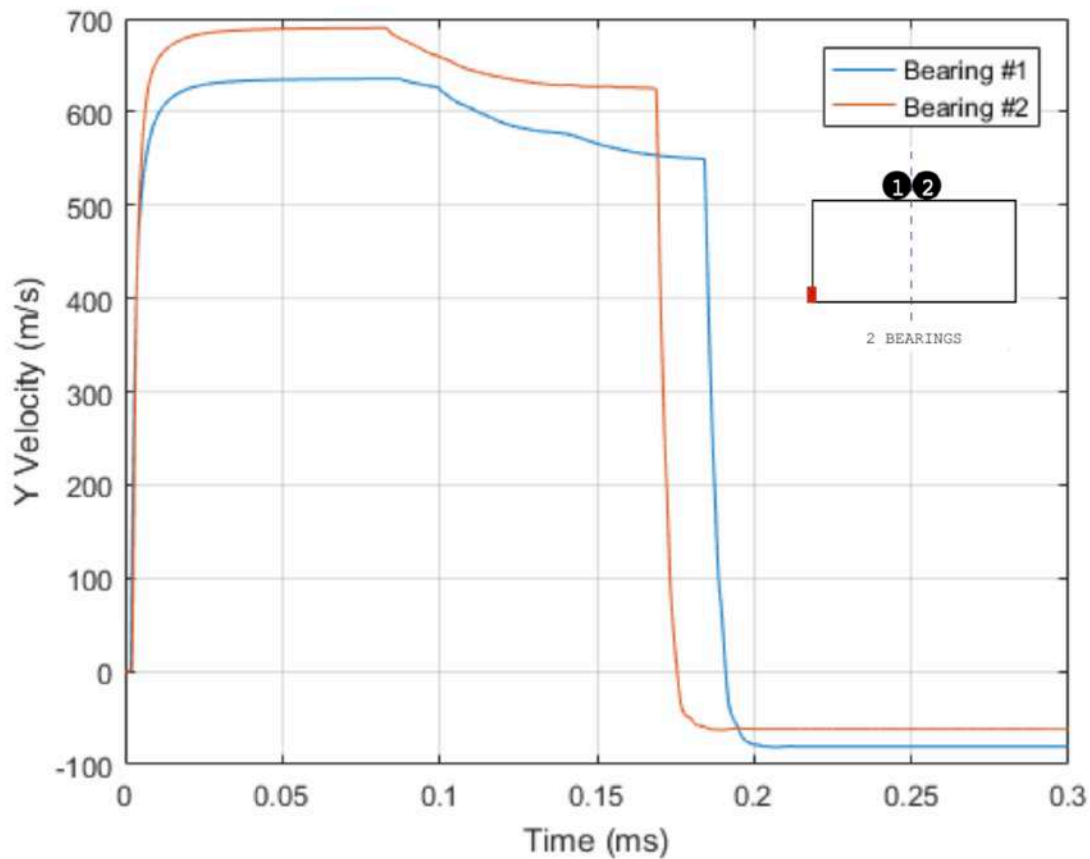


Figure 6.14: y velocity-time profiles of axial plane 2-ball bearings simulation

6.2.3.3 5-Ball Bearings Configuration

Resultant Velocity Profiles

The resultant velocity profiles for each ball bearing for the 5-ball bearings simulation in the axial plane are shown in Figure 6.15. The characteristics of the velocity-time profiles were similar, but the peak and impact velocity values and the time of impact were different in all cases. The velocities diverged around $t = 0.003s$. The profiles corresponding to ball bearings #3 and #4 were almost identical to each other, and the velocity profiles corresponding to ball bearing #2 and #5 were shown to overlap until $t = 0.075ms$. The resultant velocity profile of ball bearing #1 did not overlap with other profiles.

Ball bearings #3 and #4 had peak velocity values of $923m/s$ and $925m/s$, whereas ball bearings #2 and #5 had peak velocity values of $804m/s$ and $808m/s$. Ball bearing #1 had a peak velocity of $646m/s$. It should be noted that the reflected pressure waves passed the ball bearings at different times. The velocity drop occurred first with ball bearing #5, followed by ball bearings #3 and #4, and then ball bearings #1 and #2, with $0.005ms$ delays between each set. The order in which the velocity dropped implied that the reflected wavefront travelled from the right end of the charge towards the left end.

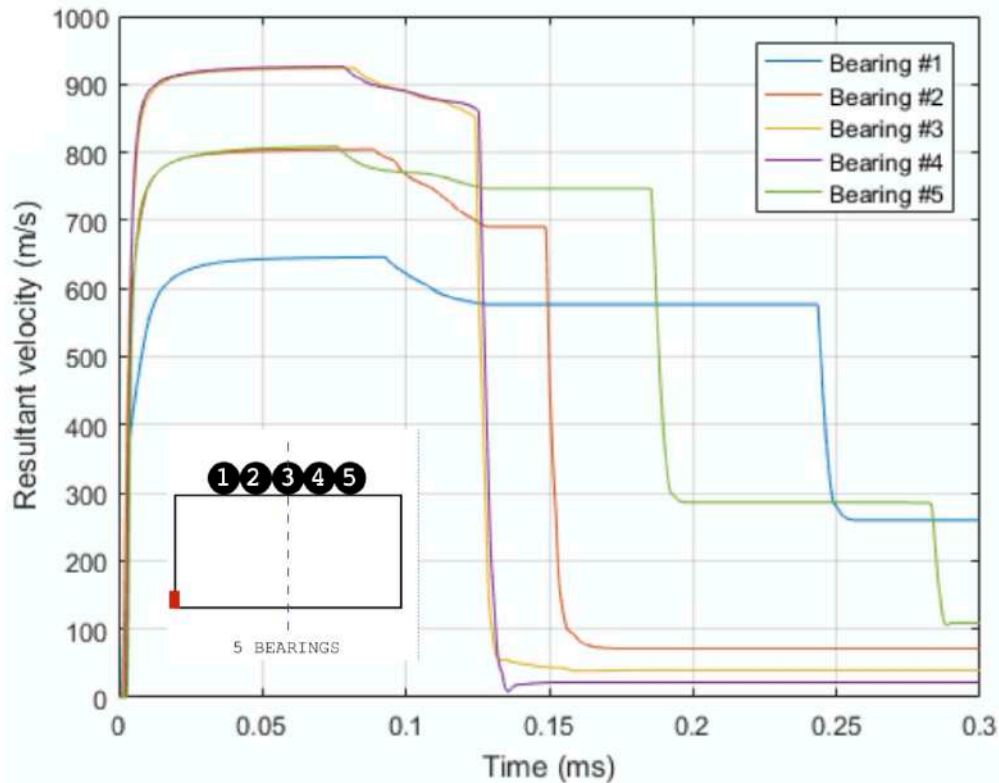


Figure 6.15: Resultant velocity-time profiles of axial plane 5-ball bearings simulation

The percentage drop in velocity caused by the wave-ball bearing interaction were as follows: 11% for ball bearing #1, 14% for ball bearing #2, 8% for ball bearing #3, 7% for ball bearing #4 and 8% for ball bearing #5. Ball bearing #3 impinged on the target cylinder wall (edge R-1) first, followed by ball bearing #4. After a delay of approximately $0.025ms$, ball bearing #2 also reached the target. ball bearing #5 hit the target after ball bearing #2, despite the fact that it was travelling at a higher velocity than ball bearing #2, showing that direction of motion was another factor that affects ball bearing-target interaction, as well as ball bearing velocity. Bearing #1 was the last ball bearing to impact the target edge A-1, at $t = 0.243ms$. Ball bearing #5 experienced a secondary rebound off edge A-3, $0.1ms$ after its initial rebound off edge A-1, matching the observation from experiments where craters were found on the cylinder cover plates and on the boss end.

The rate of deceleration post-impact were equal for all 5 ball bearings. Ball bearings #3 and #4 took a longer period of time for their velocities to settle to a constant value, compared to the other ball bearings. The rebound velocity of ball bearing #4 stabilised at a constant rebound velocity of $20m/s$ first, closely followed by ball bearing #3 stabilising to a slightly higher rebound velocity of $40m/s$. Ball bearing #2 stabilised to a rebound velocity of $72m/s$, followed by ball bearing #5 stabilising to $292m/s$. After ball bearing #1 had also reached its rebound velocity of $260m/s$, ball bearing #5 impacted the target again (on edge A-3), and underwent another rebound before settling to a new, lower rebound velocity of $108m/s$.

***X*– and *Y*–velocity Profiles**

Figures 6.16 and 6.17 show respectively the *x*–velocity profiles and the *y*–velocity profiles of each ball bearing in the axial, 5-ball bearings simulation. In ball bearings #2,#3 and #4 the *y*–velocity components was significantly higher than the *x*–velocity components; for example, the peak *y*–velocity of ball bearing #2 was $789m/s$, and the *x*–velocity at that point in time was $-248m/s$, which was only 20% of the *y*–velocity value. The *x*–velocity of ball bearings #3 and #4 were of even smaller percentages of their respective *y*–velocities. Therefore, in the case of ball bearings #2,#3 and #4 the *x*–velocity component made a less significant contribution to the resultant velocity, and the overall shape of the resultant velocity profiles of this set of ball bearings resembled the *y*–velocity profiles. The *x*–velocity components did not necessarily have the same shape as the resultant velocity profiles. In the cases of ball bearings #1 and #5, however, the *x*–velocities were comparable to the *y*–velocities; the *x*–velocity of ball bearing #1 was equal to 84% of its *y*–velocity value, and the *x*–velocity of ball bearing #5 was equal to 82% of its *y*–velocity value.

The *y*–velocity profiles intercepted the horizontal axis, indicating zero velocity and therefore a change in direction, before reaching a final rebound velocity in the direction opposite to initial ball bearing motion. The sudden increase in *y*–velocity observed around $t = 0.29ms$ in the *y*–velocity profile of ball bearing #5 was an indication of its secondary rebound off edge A-3.

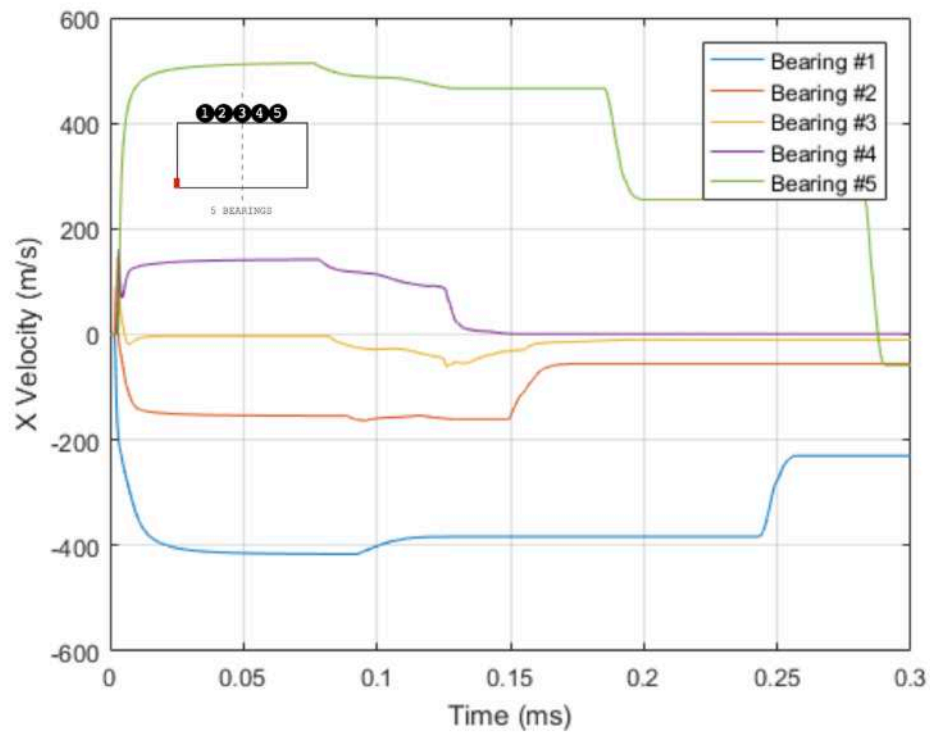


Figure 6.16: x velocity-time profiles of axial plane 5-ball bearings simulation

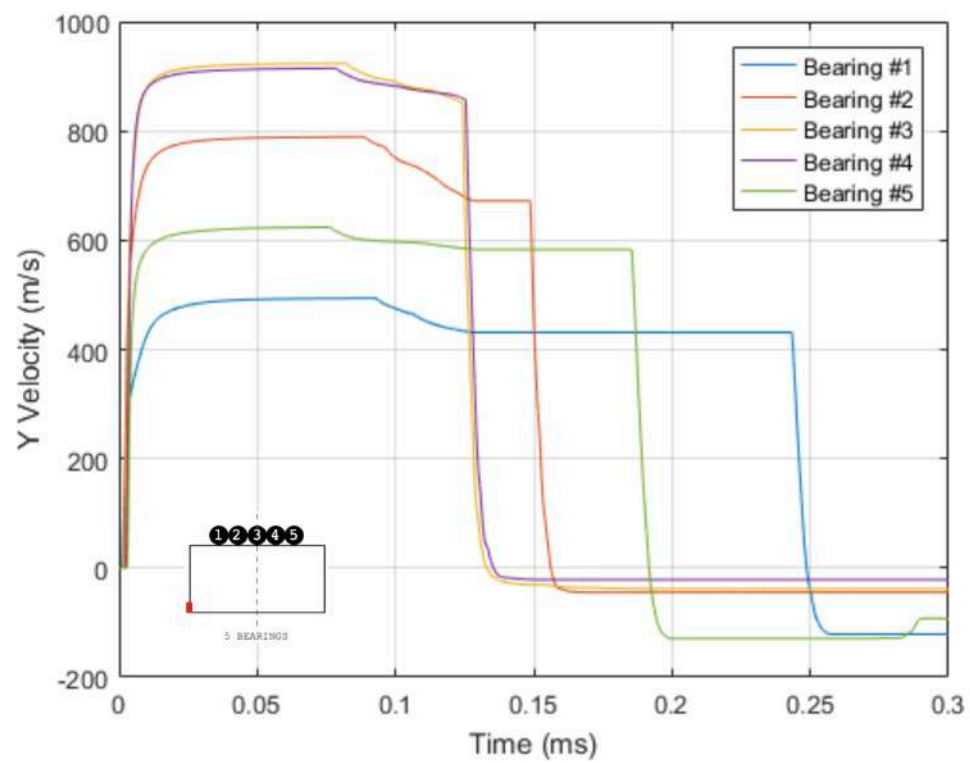


Figure 6.17: y velocity-time profiles of axial plane 5-ball bearings simulation

The peak velocity, the impact velocity and the rebound velocity of each profile in the x direction decreased from the rightmost ball bearing (#5) to the leftmost ball bearing (#1), i.e. ball bearing #5 had the highest velocities, and ball bearing #1 had the lowest (negative) velocity). It should be noted that ball bearing #3 had a very small x -velocity of near-zero, remaining below 10m/s throughout the majority of the simulation runtime, implying that ball bearing #3 moved predominantly in the y direction. Bearing #1 travelled in the opposite x direction to ball bearing #5, i.e. in the negative x direction. Similarly, ball bearing #2 travelled in the opposite direction to ball bearing #4. The x -velocity profiles did not intercept the horizontal axis, with the exception of ball bearing #5 after its secondary rebound, which implied that the ball bearings did not change direction of motion in the x -component. The x -velocity profiles corresponding to ball bearings #2,#3 and #4, a short and rapid spike in velocity was observed very early on in the simulation.

The drop in absolute velocity after interaction with the reflected waves was clear in the x -velocity profiles as well. As observed in the resultant velocity profiles, the velocity drop occurred first with ball bearing #5, followed by ball bearings #3 and #4, and then ball bearings #1 and #2, with 0.005ms delay between each set. The order in which the velocity drops occurred implied that the reflected wavefront travelled from the right end of the charge towards the left end. The x -velocity curves were not as smooth as the y -velocity curves, especially in the sections following the initial velocity drop. This was especially noticeable in the x -velocity curve for ball bearing #3.

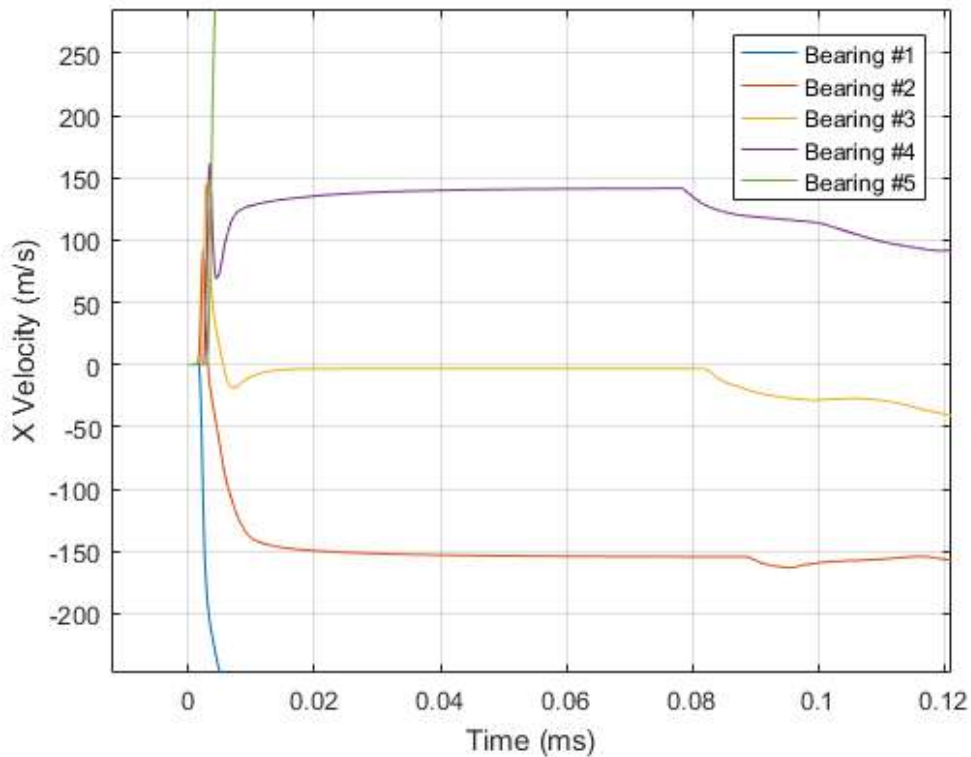


Figure 6.18: x velocity-time profiles of axial plane 5-ball bearings simulation zoomed in to show the "spikes" in x -velocity in detail

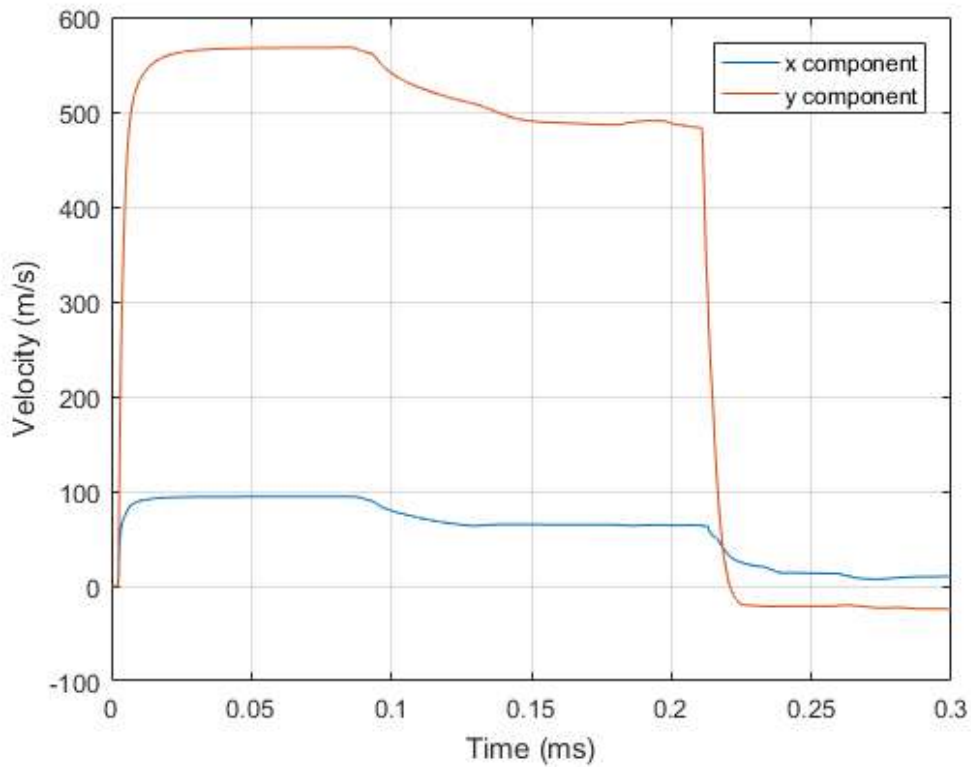


Figure 6.19: x - and y -velocity-time profiles of axial plane 1-ball bearing simulation, ball bearing x position = 152mm

The x -velocity plots exhibited "spikes" very early on in the simulation, with the exception of ball bearing #5, where the velocity was initially shown to accelerate in the positive x -direction rapidly and decelerate suddenly before stabilising to a constant velocity value. As explained in the previous section on axial plane, 1-ball bearing configuration, this behaviour was most likely due to the interaction between the detonation wavefront and the ball bearing during the detonation process.

Figure 6.18 shows the "spikes" in the x -velocity plots in more detail. The initial acceleration may be attributed to the initial interaction between the detonation front and the ball bearing, as discussed before. However, a slightly different behaviour was observed, particularly in ball bearings #3 and #4, where the ball bearing displayed another change in the direction of motion in the x -direction. Bearing #3 was placed at the same position as the ball bearing in the axial plane, 1-ball bearing simulation. Similarly, Figure 6.19 shows the x - and y -velocity components of the 1-ball bearing test, ball bearing position $x = 152\text{mm}$, the same position as ball bearing #4 in the 5-ball bearings test. Both ball bearings #3 and #4 exhibited different behaviour to their equivalents in terms of the x position of the bearing in the 1-ball bearing test. It was therefore evident that the total number of ball bearings involved in the test had an influence on the overall ball bearing motion. The drop in velocity from the initial peak increased as the number of ball bearings present to the right of the ball bearing increased. The ball bearings to the right of the ball bearing in question acted as a barrier impeding the

ball bearing motion, causing a reversal of direction of motion. However, once the charge was completely detonated, the expansion of the rarefaction waves was the primary source driving the ball bearing motion and the shape of the wavefront determined the direction in which the ball bearing travelled.

6.3 Crater Depth Experimental-Simulation Correlation

6.3.1 Comparison of Experimental and Simulation Crater Depths

The interaction between the target cylinder and the ball bearings was quantified by the depths of the craters formed in the cylinder. The numerical results were benchmarked against the crater depths measured in the experiments. For the following comparisons, the equivalent numerical conditions in the radial (4-ball bearings) and the axial planes (1-ball bearing, 2-ball bearings and 5-ball bearings) were correlated with the respective experimental results.

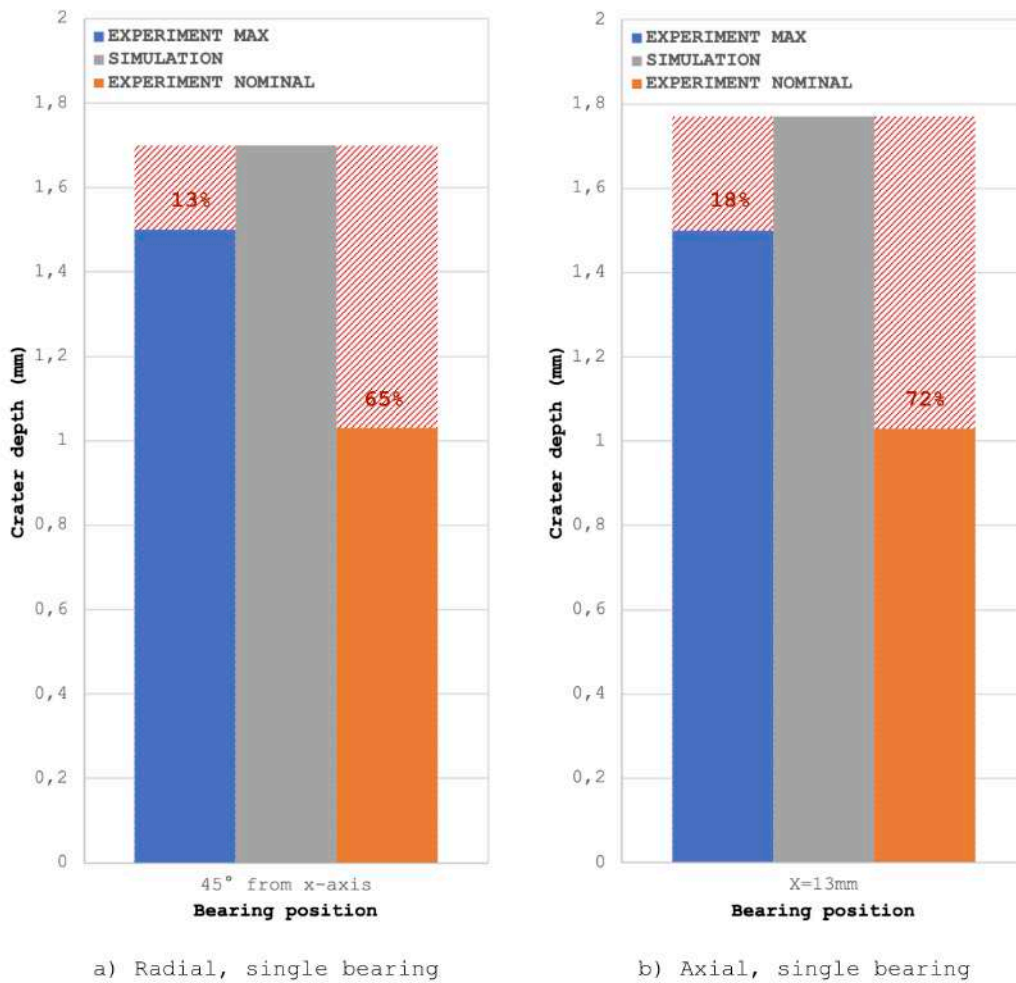


Figure 6.20: Comparison of experimental and simulation crater depths for single-ball bearing test conditions in a) radial plane and b) axial plane

Variation was observed in the experimental crater depth dataset under all the tested charged conditions. This implied that the crater depths were extremely sensitive to even a very small variation in ball bearing position and a number of other variables. It would also be very difficult to reproduce the same results even if the experiments were to be repeated, unlike the numerical models which would yield a consistent result every time. However, it must also be noted that

the experiments were aimed at modelling IEDs, which are extremely unpredictable in nature as well.

Figure 6.20 shows the comparison of the experimental crater depths to the simulated crater depths under the single-row charge condition. The crater depths from simulation were compared to both the nominal crater depth and the maximum crater depth obtained from the corresponding experiment, since a range of values as opposed to a single value was obtained experimentally. The simulations typically over-predicted the crater depth compared to the experiments. In the single-row tests, the simulation was shown to over-predict the depth by 13% and 18% in the radial plane and the axial plane respectively when compared to the maximum crater depth measured in the experiments. It was also noted that the simulated crater depth was higher in the axial plane than in the radial plane. This difference was most likely due to the difference in target geometry, i.e. the target was treated as a "plate" in the axial plane while in the radial plane, it was simulated as a cylinder, analogous to a "curved" plate.

The percentage difference was much higher when the simulation result was compared to the nominal experimental value, reaching up to a 72% difference. This reflected the high degree of variation observed in the experimental results, since the nominal depth was merely an average of the range of values obtained from experiments. The simulations overestimated the crater depths, especially at higher velocities. This was most likely due to the erosion of nodes. Element distortion was more severe at higher impact velocities, and therefore the degree of nodal erosion was also more severe. Since erosion in AUTODYN implies the deletion of eroded nodes [78], the erosion accounts for the overestimation of crater depths.

The multi-row test conditions were only possible to model in the axial plane. The comparison of the experimental and the simulated crater depths under 2-row and 5-row charge conditions is shown in Figures 6.21 a) and b), respectively. Generally, the percentage differences observed in the 2-row case were lower than the single-row case in the same plane. In the 5-row case, the simulation under-predicted the crater depths formed by ball bearings placed at the cylinder ends (indicated by the negative sign, in Figure 6.21 (b)). A large degree of over-prediction was observed in the craters formed by the three ball bearings in the mid-region.

The percentage difference in the experimental and simulation crater depths was much higher for the 5-row test condition than in the single-row or 2-row tests. For the middle-most ball bearing, the nominal-depth versus experimental depth difference reached up to 143%. The simulated depth values were closer in value to the experimental maximums than they were to the experimental nominal values, with the exception of the leftmost ball bearing, where the simulation under-predicted the experimental maximum by 45%. The larger percentage difference observed here matched the larger degree of variation in the 5-rows experimental data.

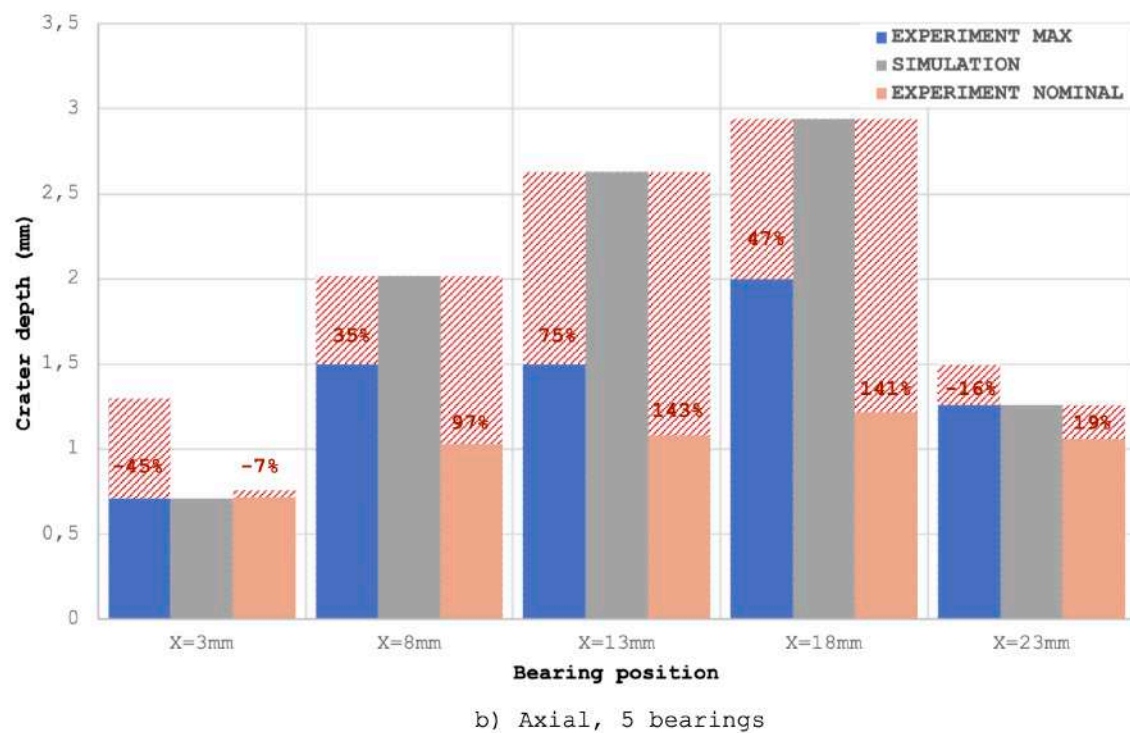
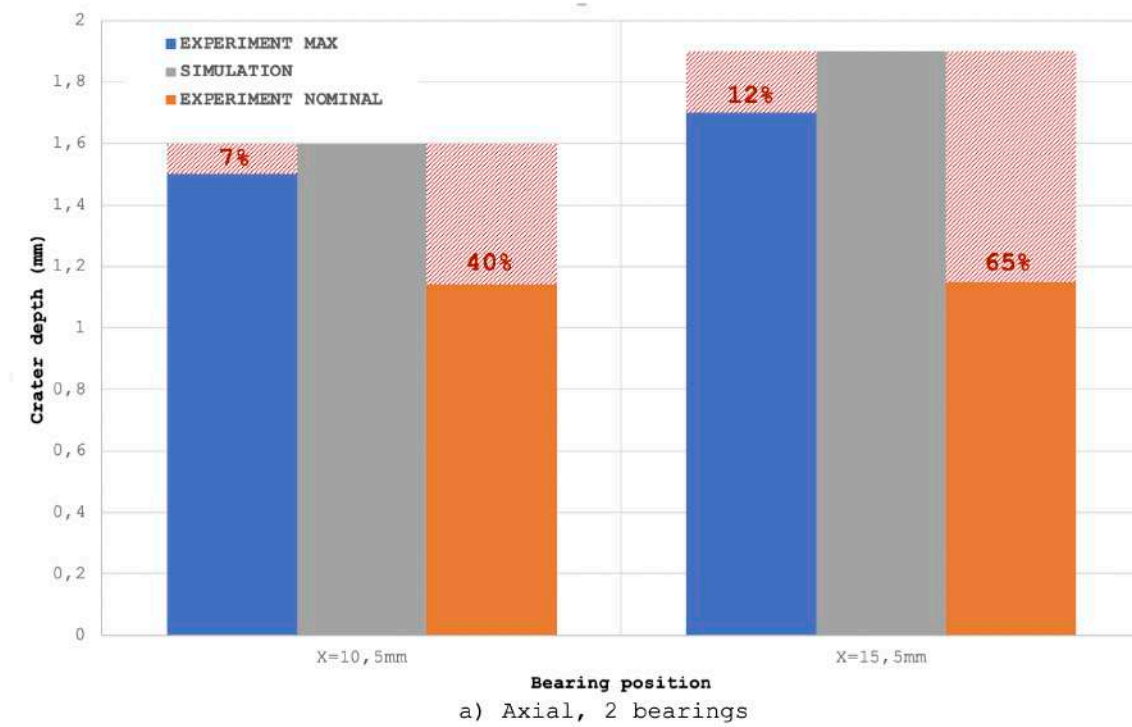


Figure 6.21: Comparison of experimental and axial-plane simulation crater depths for a) 2 rows and b) 5 rows test conditions

6.3.2 Relative Depth in the Axial Plane

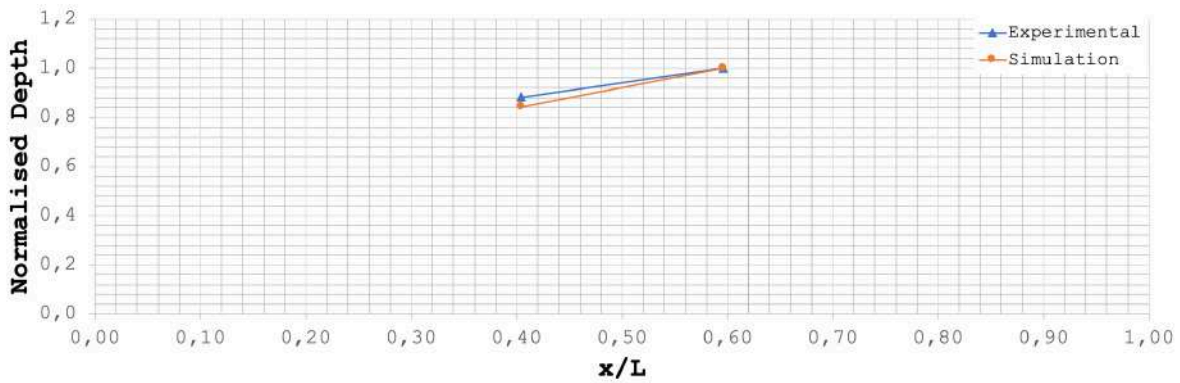


Figure 6.22: Plot of experimental and simulated normalised depth values vs. relative initial position of each ball bearing for 2-rows test condition

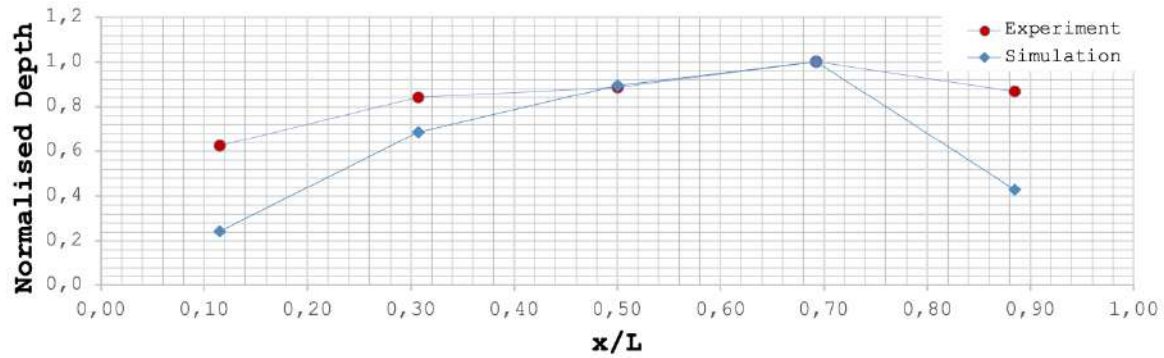


Figure 6.23: Plot of experimental and simulated normalised depth values vs. relative initial position of each ball bearing for 5-rows test condition

The normalised depth distribution profile was obtained by normalising each crater depth with respect to the corresponding maximum value, and plotting it against the relative initial position of the ball bearing. The experimental (maximum) and numerical depth distribution profiles for the 2-rows test and the 5-rows test are shown in Figures 6.22 and 6.23 respectively. In general, the numerical depth distribution showed fairly good correlation with the experimental results, exhibiting a similar pattern in the relative increase and decrease, as well as similarity in the position of the local peak. The distribution around the middle region of the 5-ball bearings profile showed very good correlation. However, significant difference was observed for the ball bearings positioned at the cylinder ends. This matched the earlier observation where the model was shown to under-predict the depth at the ends while the depths around the middle region were over-predicted.

6.3.3 Spatial Distribution of Crater Rows

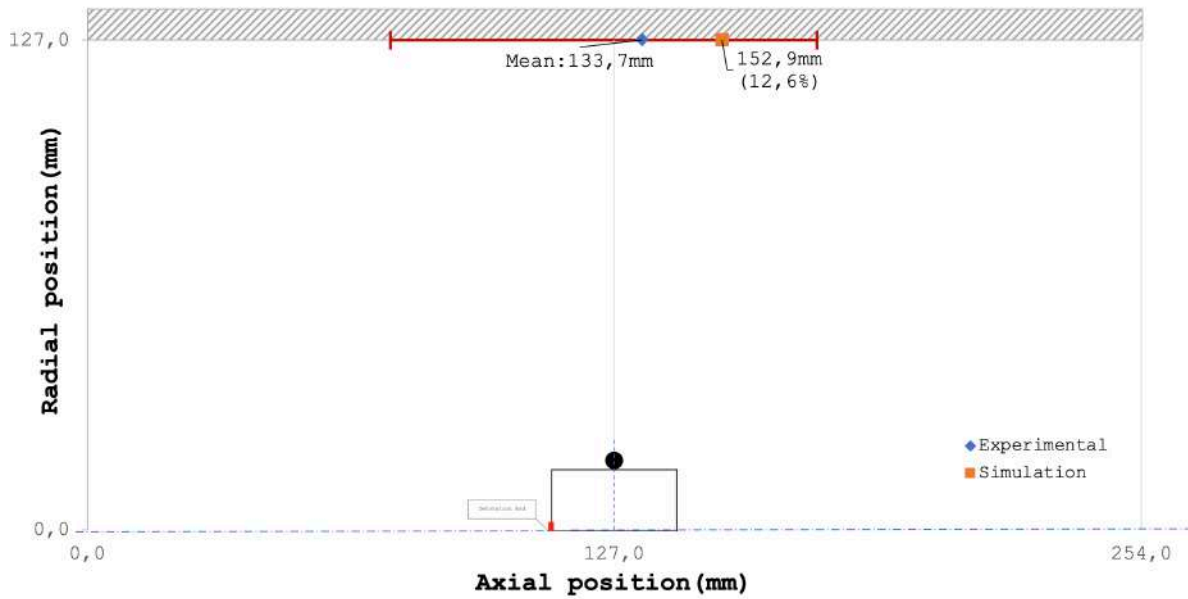


Figure 6.24: Axial distribution of the ball bearing rows from 1-row experiment and simulation. Blue data plot indicates the mean axial position of the row, the red error bars indicate the range of the axial positions. Value in parenthesis under the simulation data plot indicates the percentage error between the simulation position and the experimental row position.

The mean axial position of the ball bearing rows from the experiments and the simulations were plotted in Figure 6.24 (1-row test), Figure 6.25 (2-row test) and Figure 6.26 (5-row test). The red bars indicate the minimum and maximum range of observed axial positions from experiment. The ball bearing row axial position variation between the simulation and the experimental average ranged between 2.1% and 15.4%, with the exception of ball bearing #2 in the 5-ball bearing test, for which the percentage difference between the experimental average and the simulation was 24.1%. In the 1-row and 2-row tests, the ball bearing positions obtained from simulation fell within the range observed in the respective experiments. However, in the 5-ball bearings test, ball bearings #2, #3 and #5 fell outside the range obtained in the experiment.

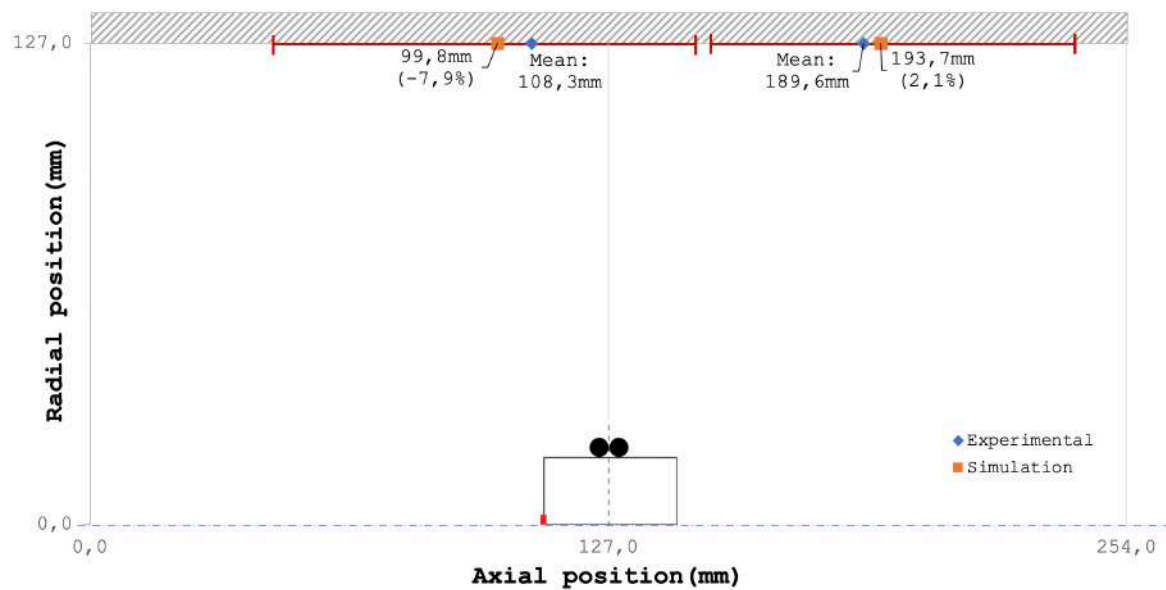


Figure 6.25: Axial distribution of the ball bearing rows from 2-row experiment and simulation. Blue data plot indicates the mean axial position of the row, the red error bars indicate the range of the axial positions. Value in parenthesis under the simulation data plot indicates the percentage error between the simulation position and the experimental row position.

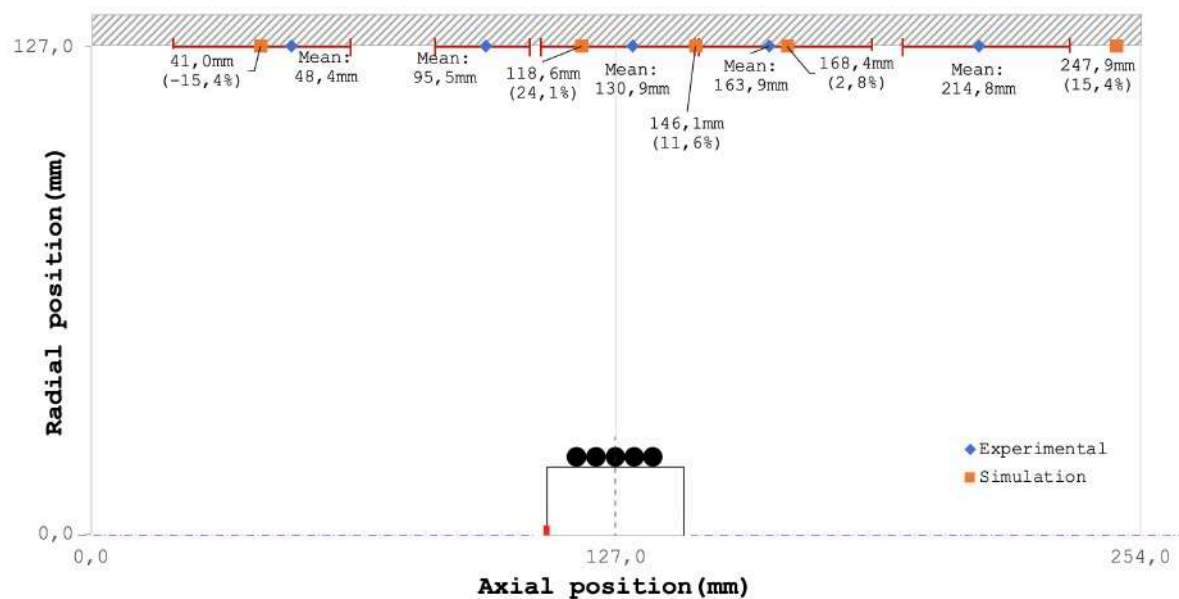


Figure 6.26: Axial distribution of the ball bearing rows from 5-row experiment and simulation. Blue data plot indicates the mean axial position of the row, the red error bars indicate the range of the axial positions. Value in parenthesis under the simulation data plot indicates the percentage error between the simulation position and the experimental row position.

6.3.3.1 Relative Distance between Crater Rows

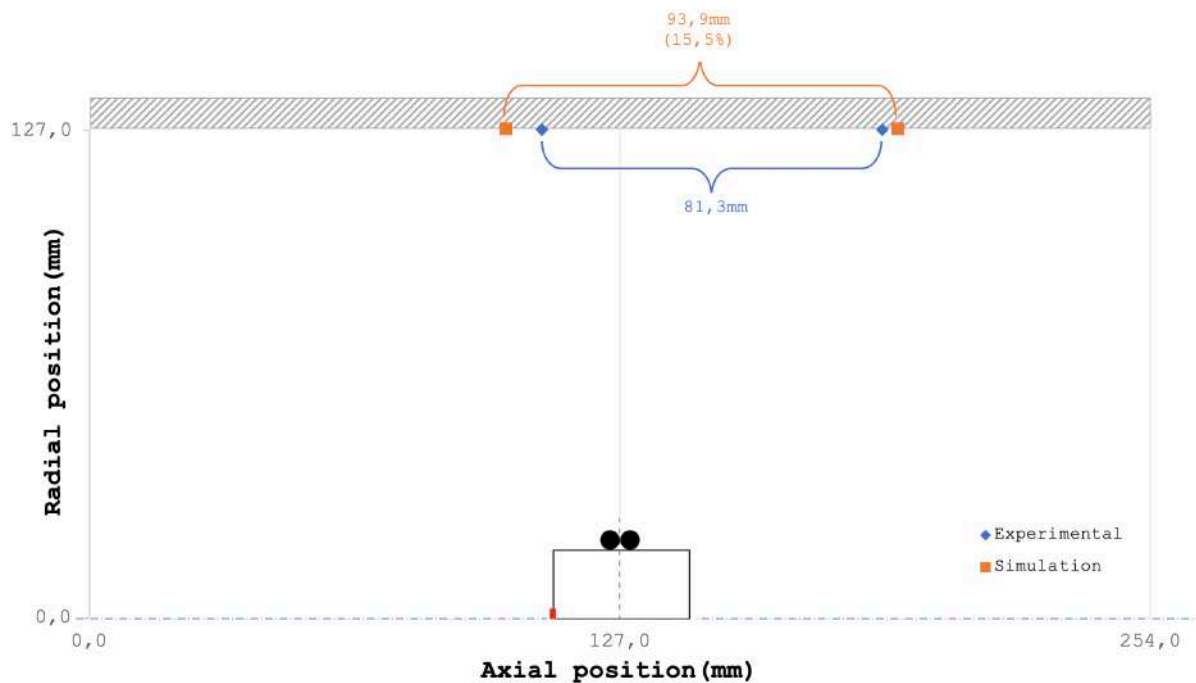


Figure 6.27: Plot of the relative distances between the adjacent ball bearings from the experiments and the simulations of 2-row test condition

The plot of the relative distances between the adjacent ball bearings from the experiments and the simulations were plotted in Figure 6.27 (2-row test) and Figure 6.28 (5-row test). In the 2-row test, the ball bearings were 15.5% further apart in the simulation than in the experiment. In the 5-row test, the ball bearings near the end were 56.2% - 64.3% further apart in the simulation than in the experiment. However, the ball bearing towards the middle were closer in relative distance in the simulation than in the experiment. Overall, the ball bearings were also shifted further from the detonation end in the simulation when compared to the experimental data.

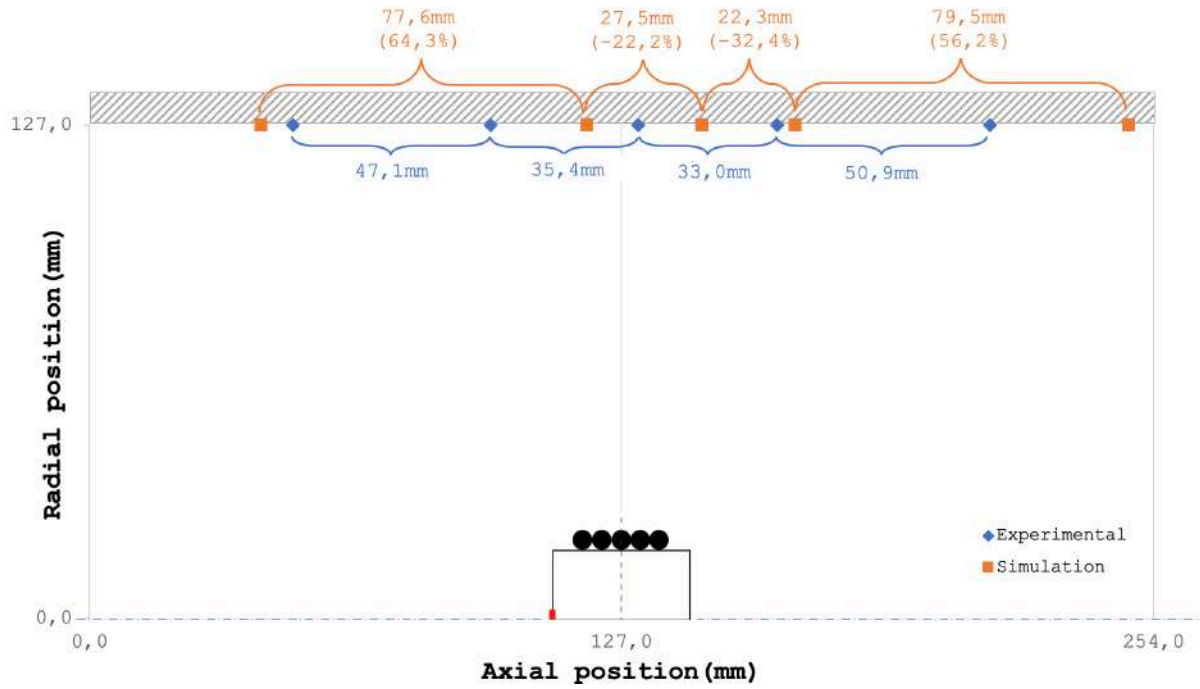


Figure 6.28: Plot of the relative distances between the adjacent ball bearings from the experiments and the simulations of 5-row test condition

6.3.4 Summary

The numerical models were validated against the average and maximum crater depth values obtained from previous experiments because there were significant variations (crater depths ranging from 0.5mm to 2mm) observed in the experiments. The model showed fairly good correlation with the maximum crater depths measured in the experiments. There was 13% to 18% difference between the simulated crater depth and the maximum experimental crater depth for the 1-row configuration. For the 2-rows configuration, the percentage differences ranged between 7% – 12%. Larger variation was observed in the 5-rows simulation. Good correlation were also obtained for the relative spatial distribution of the craters, i.e. the location where the ball bearings impacted the cylindrical structure. The numerical relative depth distribution showed fairly good correlation with the experimental results, exhibiting a similar pattern in the relative increase and decrease, as well as similarity in the position of the local peak. The differences between the numerical results and the experiments were expected as IEDs are extremely unpredictable in nature, and it would be difficult to reproduce the same results observed in the experiments.

Chapter 7

Numerical Results II: Parametric Study

For all figures in this section, the data label "Maximum/MAX" indicates the peak velocity gain of the specific ball bearing observed in the given simulation, while "Impact" indicates the velocity at which the ball bearing impinged the target for the first time. x/L refers to the relative initial axial position of the bearing. Similarly, y/R refers to the relative initial radial position of the bearing. The projection angles are measured relative to the positive y -axis, and the clockwise direction is taken as positive.

7.1 Radial Plane, Single Ball Bearing Simulations

7.1.1 Single Ball Bearing outside the Charge at Varying Positions

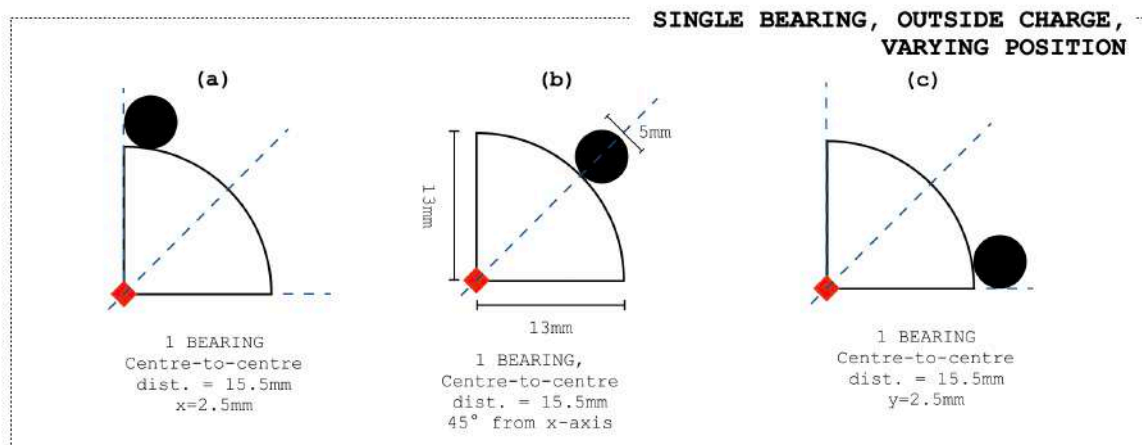


Figure 7.1: Radial plane, single ball bearing outside charge at varying positions setup

Table 7.1: Summary of results for single-ball bearing outside charge radial plane simulations

SINGLE BALL BEARING OUTSIDE CHARGE, VARIED POSITION							
Fig.	Angle from y -axis ($^{\circ}$)	Maximum velocity (m/s)	Maximum momentum ($10^{-3}kg.m/s$)	Projection angle $\theta(^{\circ})$	Impact velocity (m/s)	Impact projection angle $\theta(^{\circ})$	Crater depth (mm)
7.1 a)	8.0	628.6	93.0	28.9	561.6	29.9	1.2
7.1 b)	45.0	499.8	74.0	45.0	435.5	45.2	1.1
7.1 c)	82.0	628.8	93.1	61.1	561.7	60.1	1.2

The charge and ball bearing configuration for simulations involving a single ball bearing outside charge at varied position are shown in Figure 7.1. Table 7.1 lists a summary of the simulation results, including the angle of the central axis of the ball bearing to the y -axis in degrees, the maximum velocity and the projection angle of the ball bearing (measured from the y -axis) at maximum velocity, the impact velocity and the projection angle at impact, and the final simulated crater depth.

Three configurations were investigated, as shown in Figure 7.1. The ball bearing was either placed 1) adjacent to the y -axis (as shown in Figure 7.1 a)), or 2) such that its central axis coincided with the 45° line through the charge quarter circle (Figure 7.1 b)), or 3) adjacent to the x -axis (Figure 7.1 c)). When the ball bearing was adjacent to a x - or y -axis, the angle between its central axis and the x - or y -axis was 8° . The setup corresponding to row 1 of Table 7.1 will be referred to as 8° -configuration from this point onwards. Similarly, the test setup corresponding to row 2 and row 3 will be referred to as 45° -configuration and 82° -configuration respectively.

The maximum and impact velocities obtained under 8° -configuration and 82° -configuration were virtually equal, differing by 0.03% or less. The difference in projection angles obtained from the 8° -configuration and 45° -configuration was 16.1° - which was exactly equal to the difference in projection angles obtained from the 82° -configuration and 45° -configuration. Similarly, the differences in impact projection angles were also equal for both cases. It must also be noted that the crater depths were also equal for both the 8° -configuration and 82° -configuration. This showed that the 8° -configuration mirrored the 82° -configuration, as expected.

For both the 8° -configuration and the 82° -configuration, the ball bearing had travelled in a trajectory that deviated from its initial position by 21° by the time it had reached its maximum. Similarly, in both cases, the ball bearing had deviated by an additional 1° by the time of impact. Under the 45° -configuration, however, the amount of deviation was much less pronounced; the ball bearing had not deviated from its initial angular position by the time it had reached maximum velocity, and only deviated by 0.2° by the time of impact.

7.1.1.1 Pressure contours

The transient pressure contours for all configurations involving a single ball bearing outside charge at varied position are shown in Figure 7.2. High pressures were developed at the centre of the cylinder and charge as the blast wave expanded. The ball bearings acted as a barrier disturbing the blast wavefronts, which were no longer spherical. The pressure contours corresponding to the 8° -configuration mirrored the contours corresponding to the 82° -configuration about the $y = x$ line. The pressure contour at the 45° -configuration was symmetrical about $y = x$. In the early stages of the simulation, the shape of the wavefronts were distinctly different for each configuration. Figure 7.2 highlights the different stages at which the wavefronts exhibited distinct differences. However, as the blast wave continued to expand and reflected off the target repetitively after time $t = 0.04ms$ (similar to the behaviour shown in Figure 6.2, from $t = 0.035ms$ onwards), the wavefront shapes became more similar, eventually becoming indistinguishable.

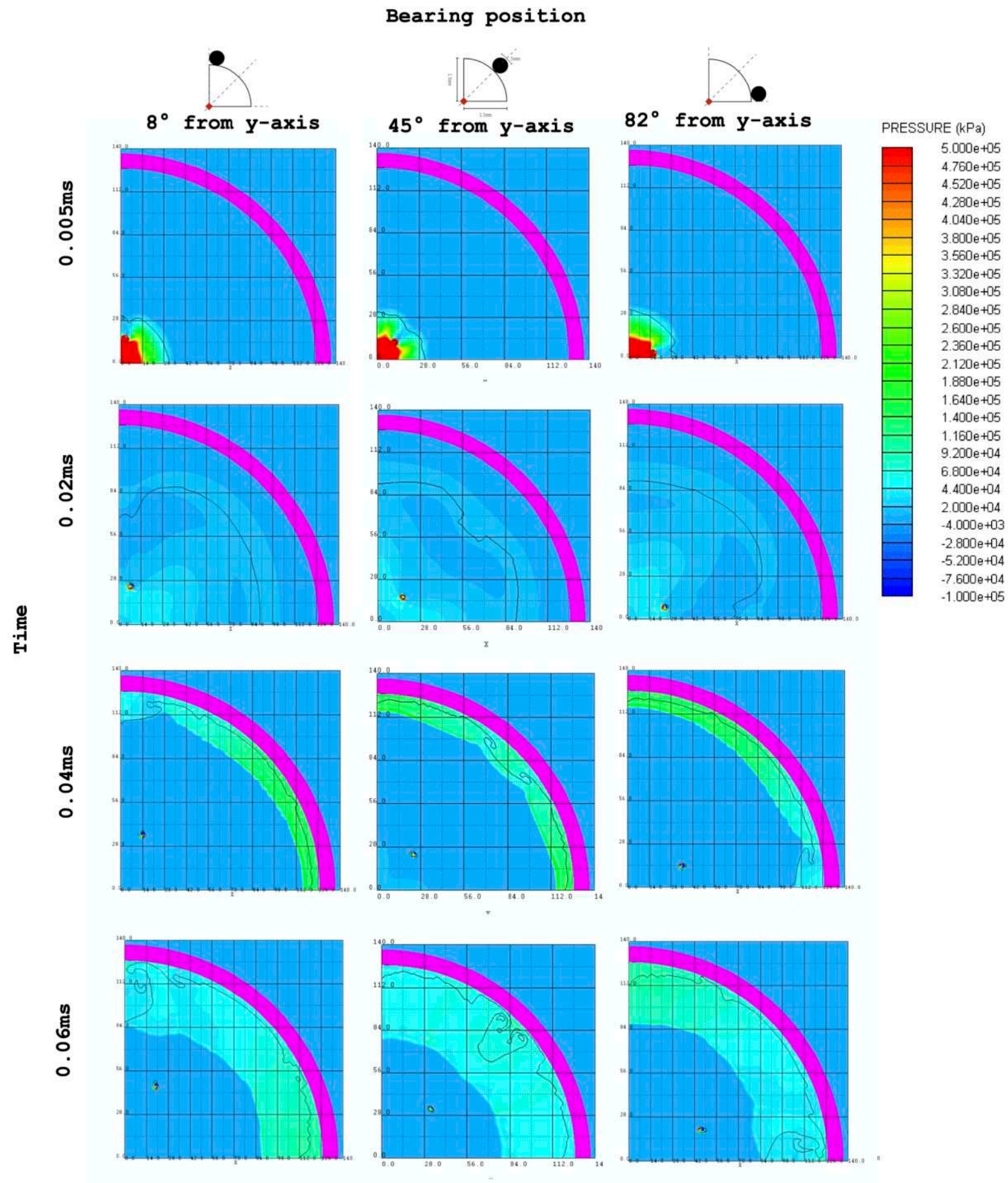


Figure 7.2: Pressure contour plots of radial plane simulation of single ball bearing outside charge at varied positions ranging from $t = 0.005ms$ to $t = 0.06ms$

7.1.1.2 Velocity profiles

Figure 7.3 shows the velocity-time profiles for all configurations involving a single ball bearing outside charge at varied positions. The shape of the velocity profiles were identical where the ball bearing was placed adjacent to either the x - or the y -axis, as expected, since the two configurations had virtually identical maximum and impact velocities. Under the 45° -configuration, the velocity profile exhibited a more dramatic decrease in velocity when the reflected pressure waves first interacted with the ball bearing. It was also observed that the ball bearing took a longer time to reach and impinge the target under the 45° -configuration, since the ball bearing had a lower maximum and impact velocity compared to the other two configurations. It also was noted that the percentage decrease from the maximum velocity to impact velocity were similar (ranging between 11 to 13%) regardless of ball bearing position.

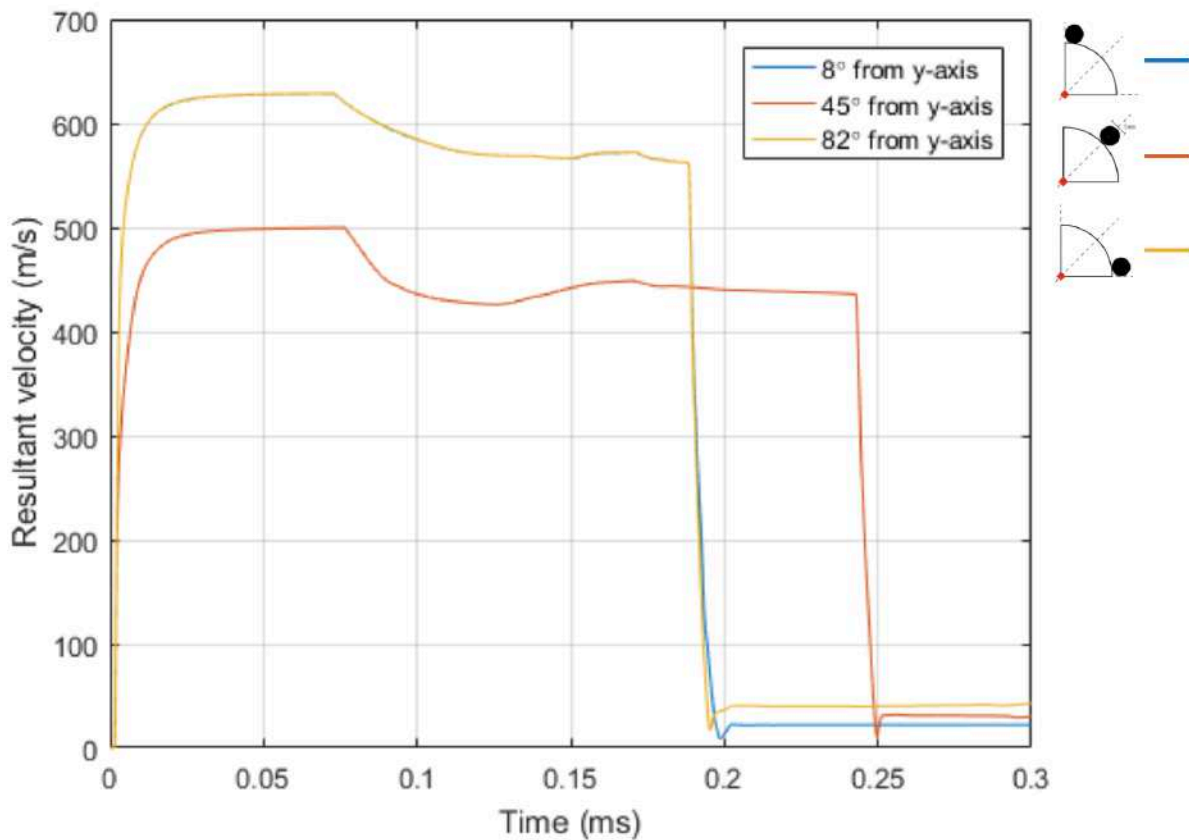


Figure 7.3: Graph of velocity vs. time graphs for tests with single ball bearing outside the charge at varied angles from y -axis in the radial plane

7.1.2 Single Ball Bearing inside the Charge at Varying Positions

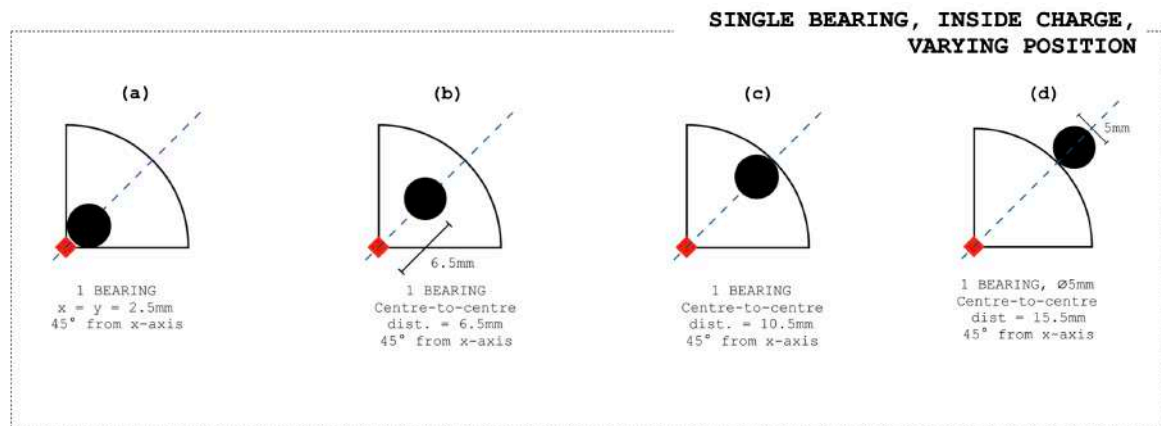


Figure 7.4: Radial plane, single ball bearing inside charge, varied position charge and ball bearing setup

Table 7.2: Summary of results for single-ball bearing inside charge radial plane simulations

SINGLE BALL BEARING INSIDE CHARGE, VARIED POSITION							
Fig.	Radial distance (h) (mm)	Maximum velocity (m/s)	Maximum momentum ($10^{-3}kg.m/s$)	Projection angle $\theta(^{\circ})$	Impact velocity (m/s)	Impact projection angle $\theta(^{\circ})$	Crater depth (mm)
7.4 a)	$\sqrt{2}(2.5)$	176.4	26.1	45.1	N/A	N/A	N/A
7.4 b)	6.5	337.4	49.9	45.0	276.6	45.1	0.9
7.4 c)	10.5	655.5	97.0	45.0	597.3	45.0	2.2
7.4 d)	15.5	499.8	74.0	45.0	435.5	45.2	1.1

The charge and ball bearing configuration for simulations involving a single ball bearing inside charge at varied radial position are shown in Figure 7.4. Table 7.2 lists a summary of the simulation results, including the ball bearing centre-to-centre distance (i.e. radial position), the maximum velocity and the projection angle of the ball bearing (measured from the y -axis) at maximum velocity, the impact velocity and the projection angle at impact, and the final simulated crater depth.

Four configurations were investigated, as shown in Figure 7.4. The ball bearing was placed such that its central axis coincided with the 45° line through the charge quarter circle, and the centre-to-centre distance between the centre of the quarter circle charge and the centre of the ball bearing was varied. The ball bearing was placed such that it was either 1) buried adjacent to the x - and y -axis (centre-to-centre distance $(h) = \sqrt{2}(2.5)mm$, as shown in Figure 7.4 a)), 2) buried inside the charge (Figure 7.4 b)), or 3) buried just inside the charge surface (Figure 7.4 c)), or 4) placed on the surface of the charge ($h = 15.5mm$, shown in Figure 7.4 d)). It was expected that the position of the ball bearing would affect the ball bearing velocity because the setup geometry was not symmetrical in the radial direction. As expected, the ball bearing velocities differed for all test configurations. The ball bearing velocities increased as the ball bearing radial distance increased (i.e. the ball bearing moved further away from the centre of the charge), and reached its maximum at $h = 10.5mm$, when buried just inside the charge and adjacent to the charge surface. The maximum crater depth ($2.2mm$) corresponded to the configuration with the highest impact velocity ($597.3m/s$). The projection angle of the ball bearing remained nearly constant at 45° (with a variation of up to 0.2° , or 0.44%), independent of the radial position of the ball bearing.

7.1.2.1 Pressure contours

The transient pressure contours at different points in time for all simulation configurations involving a single ball bearing inside charge at varied position are shown in Figure 7.5. The stages at which the wavefronts exhibit distinct differences are highlighted. As the ball bearing radial distance from the centre (h) increased, the disturbance in the pressure wavefronts became more pronounced. The ball bearing caused minimal disruption in the blast wavefronts when the ball bearing radial distance from the centre was small ($h \leq 6.5mm$), as evident in the nearly spherical shape of the wavefronts in the early stages of the simulation. When placed near the charge surface, the ball bearings acted as a barrier disturbing the blast wavefronts, which were no longer spherical. In the early stages of the simulation, it was observed that the ball bearing "split" the pressure waves, forcing the wavefront to travel around the ball bearing, as shown in the frames corresponding to $h = 10.5mm$ and $15.5mm$ at time $t = 0.02ms$. As the blast wave continued to expand and reflected off the target repetitively after time $t = 0.04ms$, the wavefront shapes became more similar, eventually becoming indistinguishable.

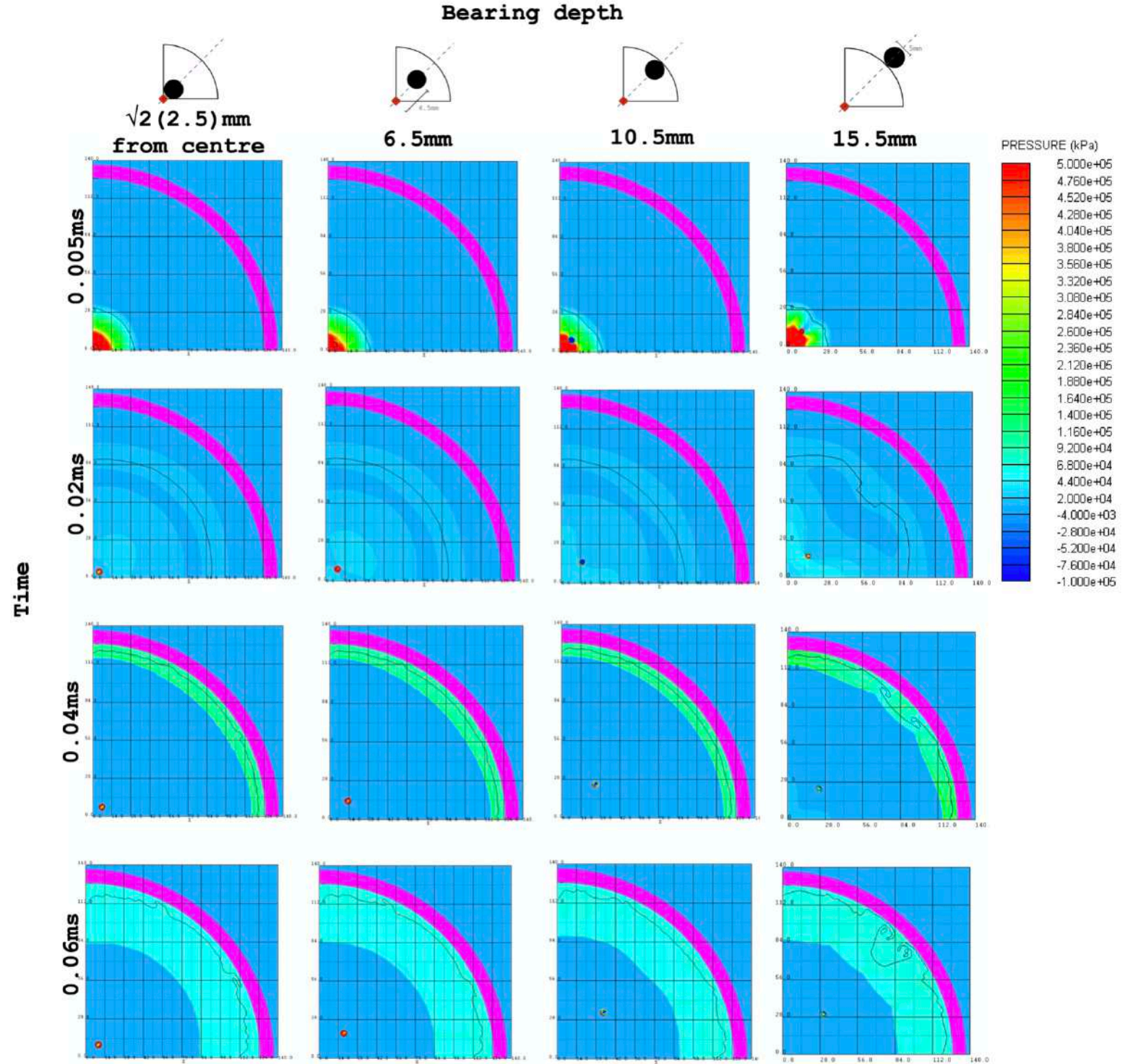


Figure 7.5: Pressure contour plots of radial plane simulation of single ball bearing inside charge at varied positions ranging from $t = 0.005\text{ms}$ to $t = 0.06\text{ms}$

7.1.2.2 Velocity profiles

Figure 7.6 shows the series of velocity-time graphs for tests with single ball bearing at varied radial distance from the centre. For test configuration $h = \sqrt{2}(2.5)mm$, the ball bearing velocity was the lowest (averaging around $100m/s$). As a result, the simulation required significantly longer runtime for the ball bearing to impact the target. Due to computational practicality, the simulation was not run to completion under this configuration, as indicated in the corresponding velocity-time graph with the label "Terminated before impact".

The velocity increased as the charge position approached the surface of the cylindrical charge, and was shown to peak at $y = h = 10.5mm$, i.e. when the ball bearing was located just under the surface of the explosive charge. When the ball bearing was positioned outside the charge, the velocity was observed to decrease. It also was noted that the percentage difference from the maximum velocity to impact velocity did not remain constant regardless of ball bearing position. The percentage difference decreased from 18% to 8% between $h = 6.5mm$ and $h = 10.5mm$, and then increased to 12% at $h = 12.5mm$.

There was a distinct difference between the velocity profiles when the ball bearing was buried inside the charge, as compared to when it was placed outside the charge. When the ball bearing was placed outside the charge, the velocity profile showed a single instance of a major decrease in velocity.

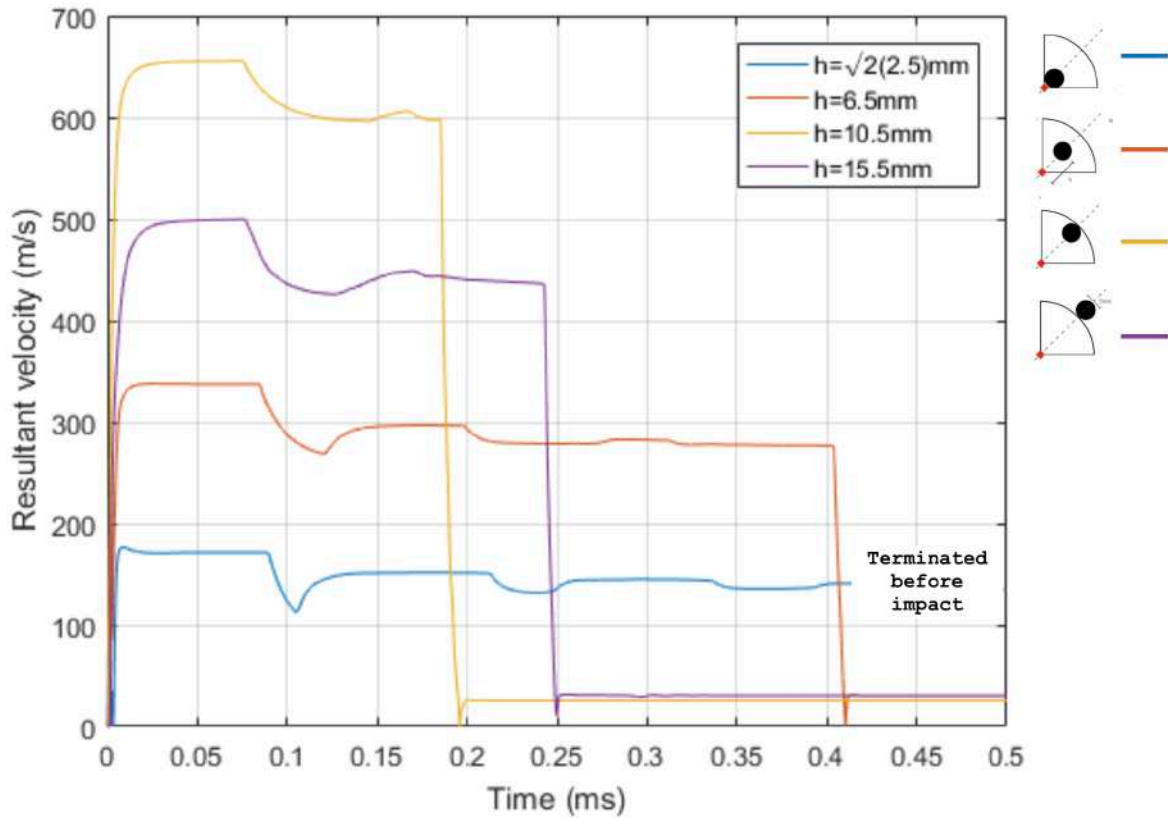


Figure 7.6: Graph of velocity vs. time for radial plane simulations of single ball bearing at varied radial distance from centre of charge

When the ball bearing was buried inside charge it was observed that the velocity profiles displayed a repetitive pattern of rise-and-fall. The frequency of the rise-and-fall cycle increased as the ball bearing distance from centre decreased (i.e. the ball bearing moved closer to the centre of the charge).

7.1.2.3 Velocity profile vs. Pressure contours

For the purposes of investigating the pressure wave and ball bearing interaction, a single ball bearing simulation in the radial plane at radial distance of 6.5mm from the centre was studied. Figure 7.7 shows that the ball bearing travelled at a constant velocity until the reflected pressure wave interacted with the ball bearing in the direction opposite to the direction of motion of the ball bearing, and the ball bearing velocity decreased. The time at which the pressure wave impacted the ball bearing coincided exactly with the velocity drop labelled as A. From point B onwards, the pressure wave began travelling in the same direction as the ball bearing, causing an increase in ball bearing velocity. The pressure wave reflected off the target and changed direction, once again travelling in the direction opposite to the direction of ball bearing motion. As a result of this impedance, the ball bearing velocity dropped again (point C). The pressure wave interaction became more complex as the reflected waves interfered with each other, and

the ball bearing velocity continued to drop until impact. Post-impact, however, the pressure wave had dissipated and did not affect the ball bearing velocity as significantly. The ball bearing velocity therefore settled to a nearly constant value post-impact.

When the ball bearing was placed closer to the centre, the rise-and-fall pattern in the velocity profile was more pronounced with an increasing frequency. This may be due to the fact that the ball bearing was closer and therefore more susceptible to the impact of the high-pressure concentration zone that was formed around the centre after detonation. The ball bearing also had a lower velocity when placed closer to the centre, and was therefore slower to move away from its initial position near the centre. The ball bearing remained in the higher-pressure zone for a longer time, as the magnitude of the pressure was the highest around the centre of the charge. Therefore, while the ball bearing remained in this zone, the pressure wavefront had a greater influence on the ball bearing velocity.

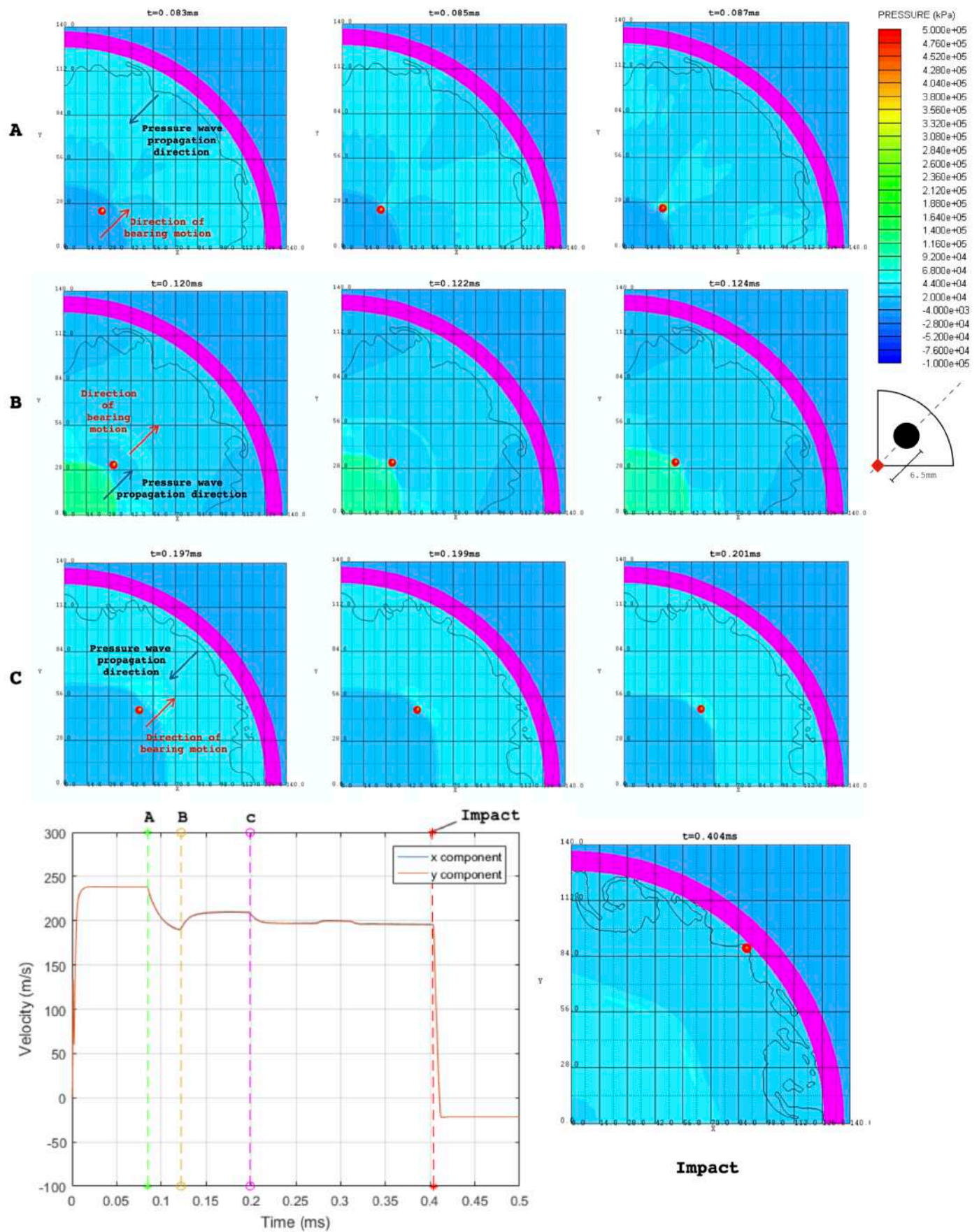


Figure 7.7: Comparison between the pressure contour plots and velocity-time profiles at various points in time in a radial plane simulation with single ball bearing inside charge at radial distance of 6.5mm

7.1.3 Single Ball Bearing outside the Charge at Varying Ball Bearing Diameters

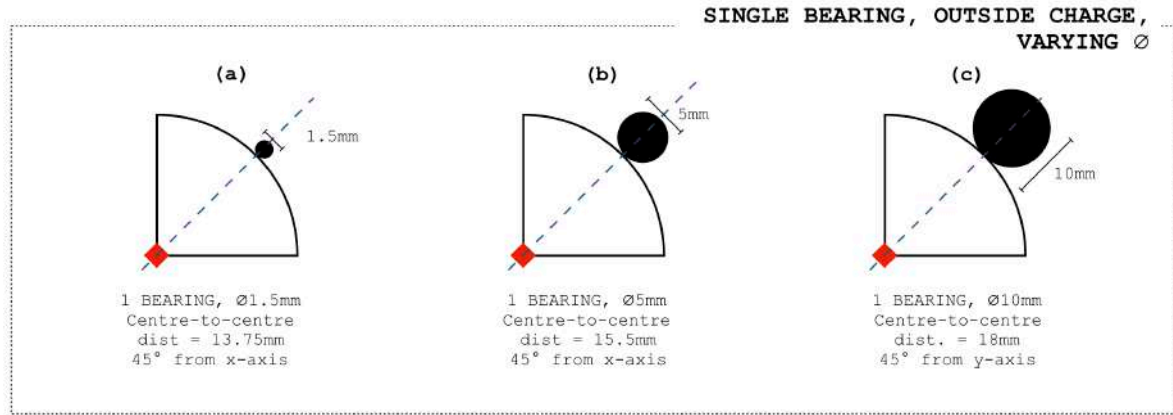


Figure 7.8: Radial plane, single ball bearing outside charge, varied ball bearing size charge and ball bearing setup

Table 7.3: Summary of results for single-ball bearing varied diameter radial plane simulations

SINGLE BALL BEARING VARIED DIAMETER							
Fig.	ball bearing diameter (d) (mm)	Maximum velocity (m/s)	Maximum momentum ($10^{-3}kg.m/s$)	Projection angle $\theta(^{\circ})$	Impact velocity (m/s)	Impact projection angle $\theta(^{\circ})$	Crater depth (mm)
7.8 a)	1.5	745.7	36.8	45.1	615.9	45.1	1.5
7.8 b)	5.0	499.8	74.0	45.0	435.5	45.2	1.1
7.8 c)	10.0	253.5	153.6	45.0	226.7	44.8	0.6

The charge and ball bearing configuration for simulations involving a single ball bearing outside charge at varied ball bearing diameter are shown in Figure 7.8. Table 7.3 lists a summary of the results including the ball bearing diameter, the maximum velocity and the projection angle of the ball bearing (measured from the y -axis) at maximum velocity, the impact velocity and the projection angle at impact, and the final simulated crater depth. The ball bearing was placed on the charge surface such that its central axis coincided with the 45° line through the quarter cylindrical charge. Three different diameters were investigated, as indicated in Figure 7.8. The ball bearing velocities were observed to decrease as the ball bearing diameter increased. The crater depth also decreased with an increase in ball bearing diameter. This implied that the crater depth was not solely dependent on the diameter of the projectile. The projection angle of the ball bearing at maximum velocity remained nearly constant at 45° (with a variation of up to 0.1° , or 0.22%). By time of impact, the projection angle had deviated up to 0.2° , or 0.44% .

Bearing diameter

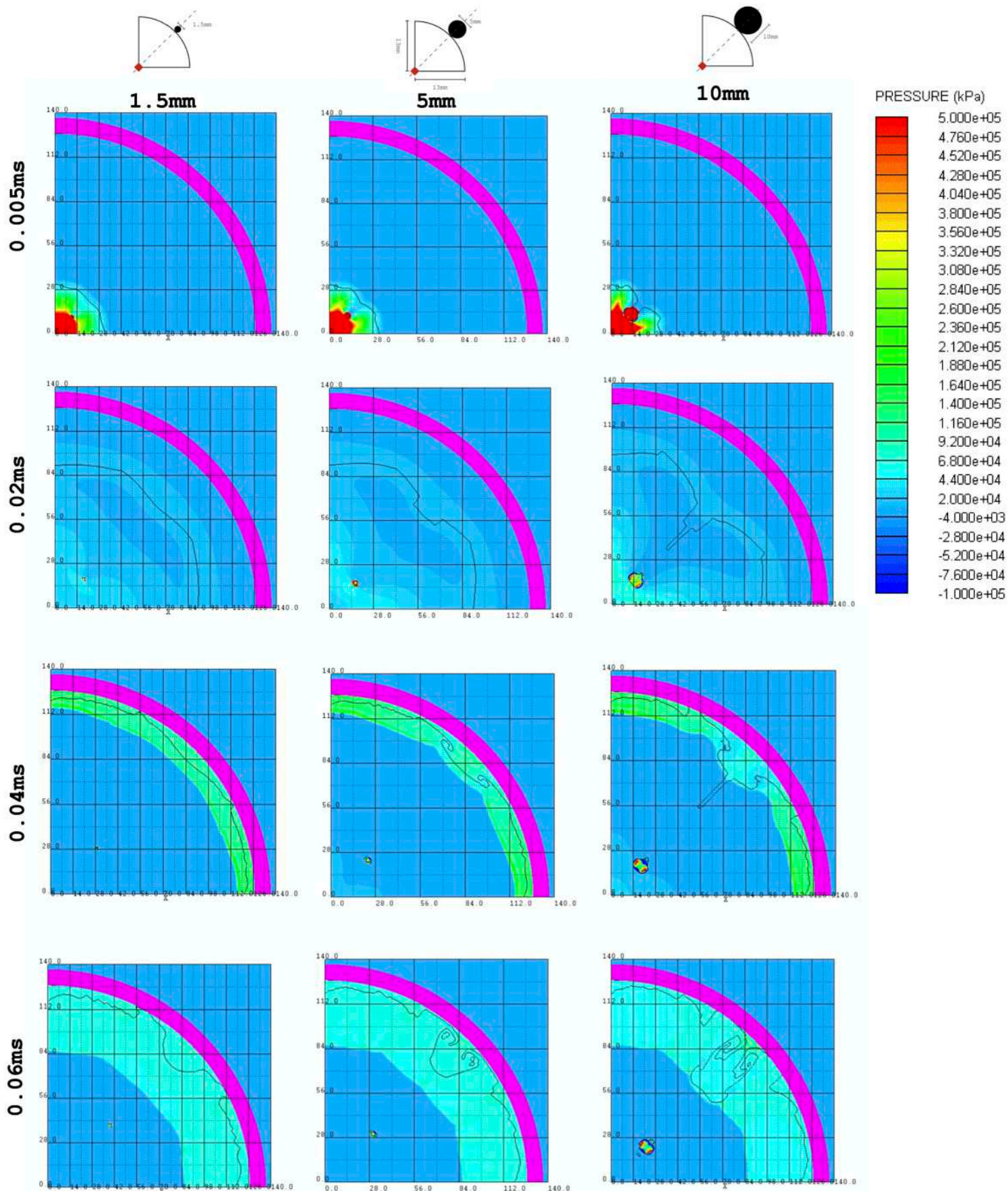


Figure 7.9: Pressure contour plots of radial plane simulation of single ball bearing of various diameter ranging from $t = 0.005ms$ to $t = 0.06ms$

7.1.3.1 Pressure contours

The transient pressure contours at different points in time for all simulation configurations involving a single ball bearing outside charge at varied ball bearing diameter are shown in Figure 7.9. The stages at which the wavefronts exhibit distinct differences are highlighted. As the ball bearing diameter increased, the disturbance in the pressure wavefronts became more pronounced and the wavefront was shown to lose its original spherical shape. The ball bearing with a $1.5mm$ diameter caused minimal disruption to the blast wavefront, evident in the nearly spherical shape of the wavefront in the early stages of the simulation. However, at $d = 10mm$ it was observed that the ball bearing "split" the pressure waves, forcing the wavefront to travel around the ball bearing. This showed that the degree of disturbance was proportional to the surface area occupied by the ball bearing compared to the overall simulated area. As the blast wave continued to expand and reflected off the target repetitively after time $t = 0.04ms$, the wavefront shapes became more similar, eventually becoming indistinguishable.

7.1.3.2 Velocity profiles

Figure 7.10 shows the series of velocity-time graphs for radial plane tests with single ball bearing at varied ball bearing diameters. There was a distinct difference between the velocity profiles as the ball bearing diameter was varied. It was noted that as the ball bearing diameter increased, the velocity profile became more irregular. The velocity (both maximum velocity and the impact velocity) decreased as the diameter of the ball bearing increased. The percentage difference between the maximum velocity and impact velocity decreased as the ball bearing diameter was increased, from 17.4% to 10.6%. This was to be expected, as the ball bearing diameter, and hence ball bearing mass and inertia also increased.

For ball bearing diameter $d = 1.5mm$, it could be observed that the ball bearing experienced a single major drop in velocity. The velocity profile remained relatively smooth afterwards, except for the slight increase in velocity shortly before impact. This may be due to the fact that the ball bearing was travelling very fast, and interacted with the pressure waves for a shorter amount of time. A lighter mass implied that the pressure wave interaction would have greater effect on the ball bearing motion. As the ball bearing diameter increased, the interaction time between the ball bearing and the pressure waves was prolonged. Therefore, there was an increased amount of interaction between the ball bearing and the pressure waves, causing more disruption in ball bearing motion and change in ball bearing velocity. Due to the increased surface area, larger ball bearings may also have experienced increased drag effects during flight.

For ball bearing diameter $d = 5mm$, the ball bearing velocity dropped, and then significantly increased once more when the reflected wavefront impacted the ball bearing in the same direction as its motion. Similarly, at ball bearing diameter $d = 10mm$, the ball bearing velocity profile was very irregular after its initial drop, implying that the ball bearing experienced more disturbance due to the pressure waves. The percentage initial drop in velocity decreased as the ball bearing mass increased.

It should be noted that smaller ball bearing may melt or shatter during the experiments because of its relative size and the blast loading effects. A melted fragment would have a greater effect on the target as it may penetrate the target as was observed by Chung Kim Yuen et al. [63] for the effects of charge encased in a metallic container on a steel target. However, failure of this nature was not implemented in the numerical simulation.

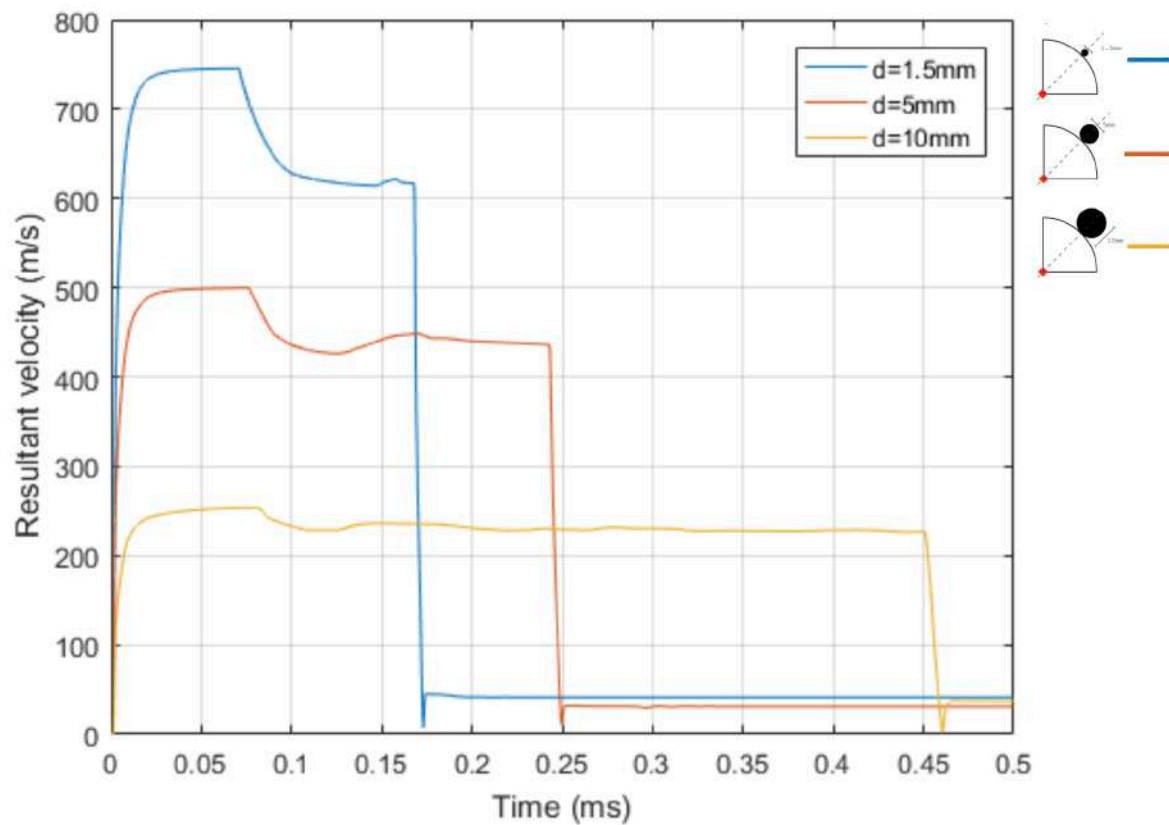


Figure 7.10: Graph of velocity vs. time for radial plane simulations of single ball bearing at varied diameter outside the charge

7.2 Radial Plane, Multiple Ball Bearings Simulations

7.2.1 Single, 2-, 3- and 4- Ball Bearings outside the Charge

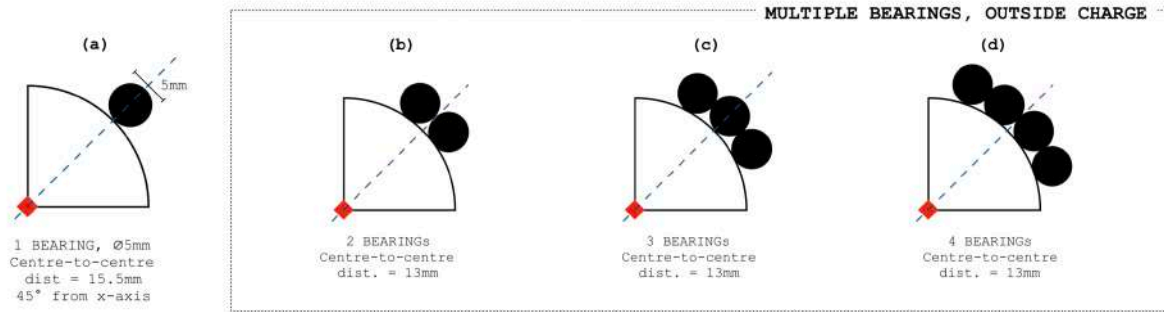


Figure 7.11: Radial plane, multiple ball bearing outside charge setup

Table 7.4: Summary of results for multiple-ball bearings radial plane simulations

Fig.	Total no. of ball bearings	Angle from y -axis ($^{\circ}$)	Maximum velocity (m/s)	Maximum momentum ($10^{-3}kg.m/s$)	Projection angle $\theta(^{\circ})$	Impact velocity (m/s)	Impact projection angle $\theta(^{\circ})$	Crater depth (mm)
7.11 a)	1	45.0	499.8	74.0	45.0	435.5	45.2	1.1
7.11 b)	2	37.0	553.0	81.8	25.1	487.0	24.4	1.1
		53.0	553.01	81.8	64.9	486.7	65.7	1.2
7.11 c)	3	29.0	713.6	105.6	8.2	654.4	7.4	1.4
		45.0	792.0	117.22	45.0	714.1	45.0	1.5
		61.0	713.5	105.6	81.7	654.3	82.5	1.2
7.11 d)	4	21.0	830.7	123.0	3.9	766.5	3.5	1.7
		37.0	884.8	131.0	34.0	829.2	34.5	1.7
		53.0	876.8	129.8	55.9	822.0	55.6	1.7
		69.0	830.5	122.9	86.1	766.3	86.5	1.7

The charge and ball bearing configuration for radial simulations involving multiple ball bearings outside charge are shown in Figure 7.11. Table 7.4 lists a summary of the results for the multiple-ball bearing, radial plane simulations, including the angle of the ball bearing central axis to the y -axis in degrees, the maximum velocity and the projection angle of the ball bearing (measured from the y -axis) at maximum velocity, the impact velocity and the projection angle at impact, and the final simulated crater depth. Four configurations were investigated, as shown in Figure 7.11. The ball bearings were placed such that 1) if there were odd number of ball bearings, the central axis of the middle-most ball bearing coincided with the 45° line through the charge, or, 2) if there were even number of ball bearings, such that the ball bearings were adjacent to the 45° line. The angle between the central axes of adjacent ball bearings was 8° .

The 1-ball bearing configuration, discussed in detail in Section 7.1, and was used as a benchmark for comparison for the other tests. Under 2-ball bearing test, the maximum and impact

velocities of both ball bearings were virtually equal, differing by 0.002%. The difference between the ball bearing initial angle from y -axis and the maximum velocity projection angles were equal (and opposite) for both ball bearings (11.9°). Similarly, the difference between the ball bearing initial angle from y -axis and the impact velocity projection angles were almost equal for both ball bearings (12.6° and 12.7° respectively). This showed that the ball bearings exhibited symmetrical behaviour about the 45° line (i.e. $y = x$ line). It must also be noted that the depths were also nearly identical ($0.1mm$ difference). Similarly, the ball bearings from the other configurations also exhibited symmetrical behaviour about the 45° line. Under both the 3-ball bearing and 4-ball bearing configuration, the angles of deviation from the initial position for the ball bearings were equal for ball bearings that were symmetrical about the 45° line. The maximum and impact velocity were also equal for ball bearings symmetrical about the 45° line in the 3-ball bearing configuration. It was noted that under the 4-ball bearing configuration, the velocities of ball bearings #2 and #3 (adjacent to the 45° line) were not equal; however, the percentage difference was less than 1%.

7.2.1.1 Pressure contours

The pressure contours for a series of simulations with 1, 2, 3 and 4 ball bearings are compared with bare charge (i.e. '0' ball bearing) in the radial plane at various points in time in Figure 7.12. The differences in wavefront shapes under the different configurations were more significant during the earlier stages of the simulation. At $t = 0.005ms$, it was observed that the ball bearings affected the overall shape of the pressure contour immediately after detonation. The pressure wave propagation slowed down as the number of ball bearings increased. At $t = 0.06ms$ it was observed that the pressure wavefront lagged further behind as the number of ball bearings was increased.

As the pressure waves dissipated, the pressure wavefronts were shown to exhibit very similar patterns independent of the number of ball bearings involved. It was observed that although the ball bearing caused disruption and turbulence, especially in the early stages of detonation, the global pressure response was not greatly affected by the presence of the ball bearing, especially as the simulation time progressed further. This was to be expected, since the area occupied by the ball bearing was very small compared to the overall simulation area. From $t = 0.06ms$ onwards, the wavefronts exhibited very similar shapes.

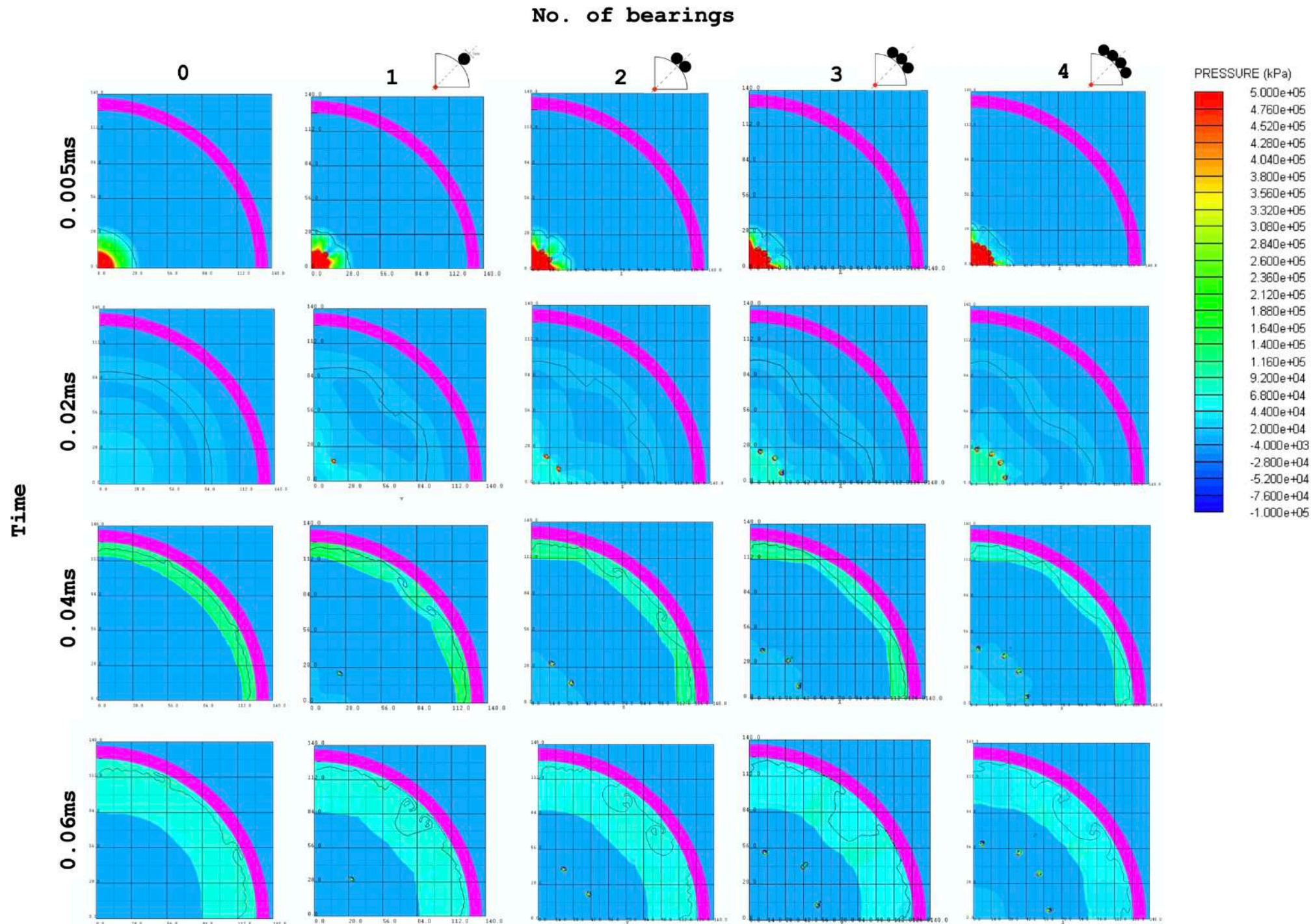


Figure 7.12: Pressure contour plots of radial plane simulation of multiple ball bearings outside charge at varied positions ranging from $t = 0.005ms$ to $t = 0.06ms$

7.2.1.2 Velocity profiles

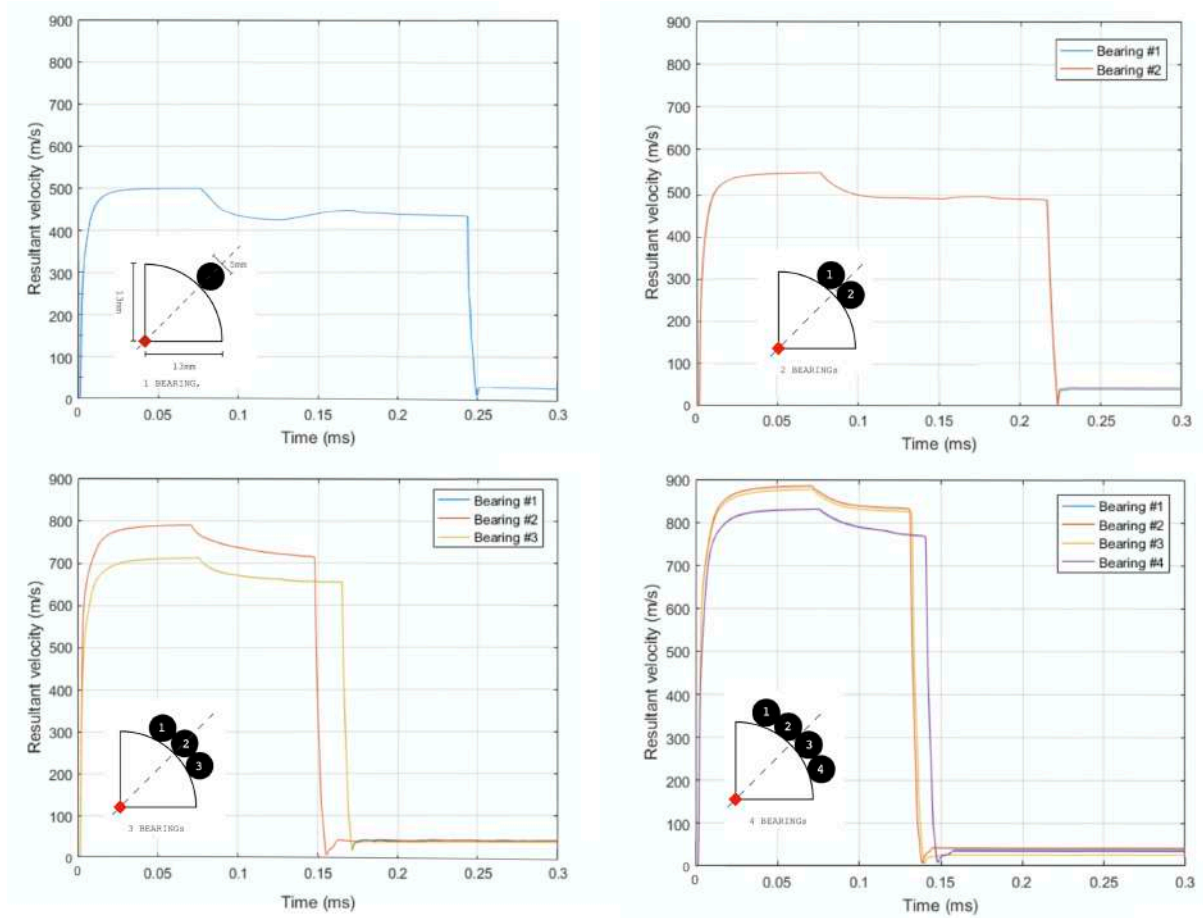


Figure 7.13: Graphs of velocity vs. time for radial plane simulations of multiple ball bearings outside charge

The resultant velocity profiles for each ball bearing in the radial, multiple ball bearing simulation are shown in Figure 7.13. The shape of the velocity profiles were identical for ball bearings positioned symmetrically to each other about the 45° line. For example, under the 2-ball bearing configuration the velocity profiles corresponding to ball bearings #1 and #2 were identical. Similarly, under the 3-ball bearing configuration, the velocity profiles corresponding to ball bearings #1 and #3 were similar in shape and magnitude. As the number of ball bearings increased, the velocity profile exhibited a smaller drop in velocity when the reflected pressure waves first interacted with the ball bearing. It was also observed that since the ball bearings had higher maximum and impact velocity as the number of ball bearings increased, the time taken for the ball bearing to impact the target was also reduced.

Figure 7.14 shows the plot of velocity vs. number of ball bearings. In a given simulation, the highest and the lowest value of the maximum velocity gain were recorded and denoted as "Highest Max." and "Lowest Max." in the figure. Similarly, the highest and the lowest value of the impact velocity were recorded and denoted as "Highest Impact" and "Lowest Impact".

This only applies to configurations where more than one ball bearing was involved. For the 1-ball bearing test, there was only one value and the highest and lowest values could not be distinguished.

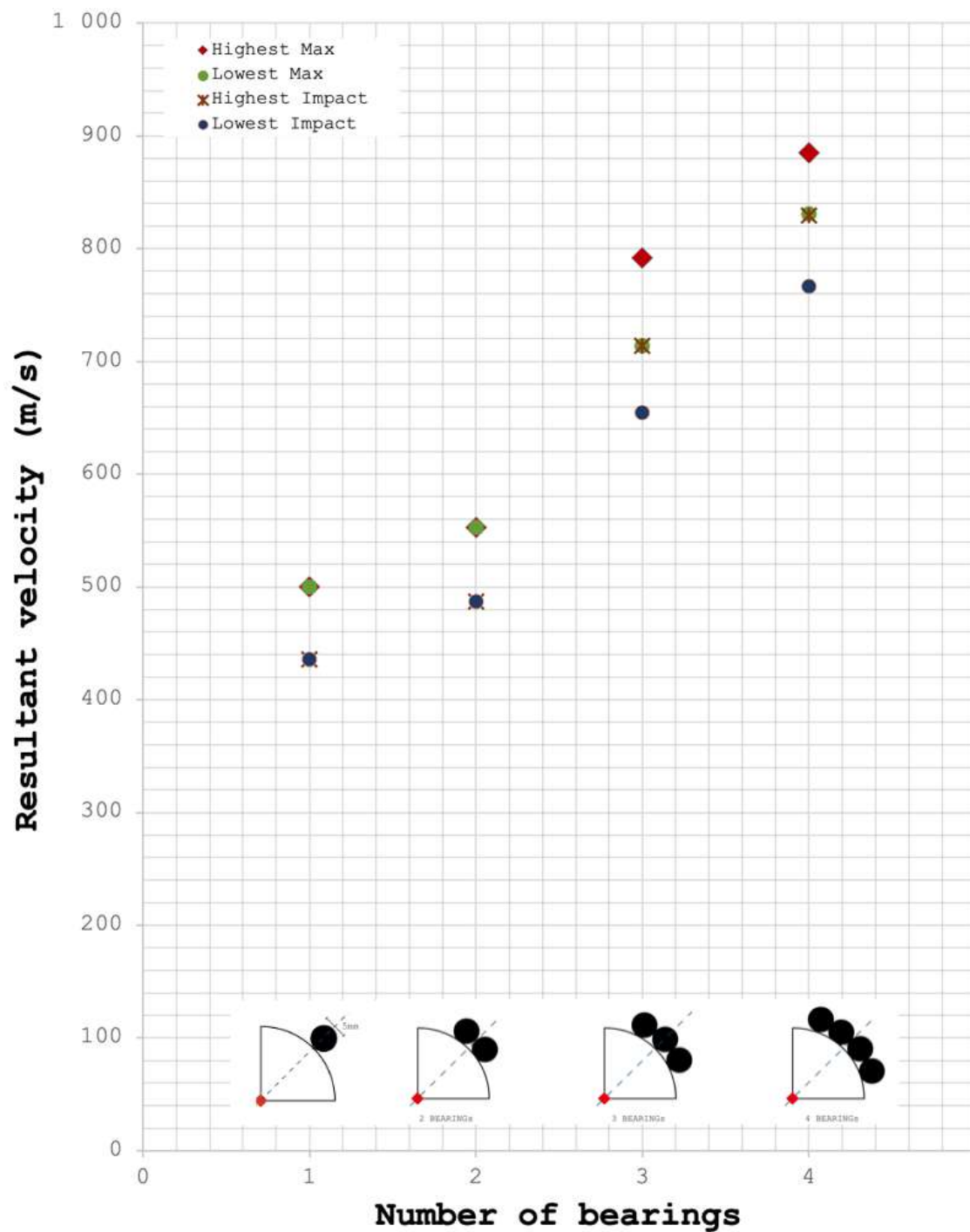


Figure 7.14: Series of velocity-time profiles for radial multiple-ball bearing simulations

The velocity increased as the number of ball bearings increased, and was shown to peak at 4-ball bearings. In the 2-ball bearings test, both ball bearings had the same maximum velocity

and impact velocity. However, in the 3-ball bearings and 4-ball bearings tests, there was a clear difference between the highest and lowest values. The lowest maximum velocity overlapped with the highest impact velocity.

In the current study, the ball bearings placed around the charge were comparable to discrete, "pre-formed" pieces. When discrete spherical bodies (i.e. ball bearings) were placed on a cylindrical body (in the radial plane), air gaps occurred in the circumferential direction. The area of these air gaps were much greater than for the pre-formed cubic fragments reported in literature. The energy and hence velocity loss was more significant for the ball bearings when compared to the pre-formed fragments. Since much of the energy from detonation escaped between the ball bearings and was not available to accelerate the fragments, the ball bearing velocities were lower, as reported by Taylor [45] and Anderson [48]. More "free space" and consequently more energy loss occurred when there were fewer rows of ball bearings (i.e. the energy loss was inversely proportional to the degree of confinement). Because of this phenomenon, the lowest velocity occurred in the single-ball bearing test, and the velocity was shown to increase as the total number of ball bearings increased.

7.3 Axial Plane, Single Ball Bearing Simulations

7.3.1 Single Ball Bearing outside the Charge at Varying X -Position

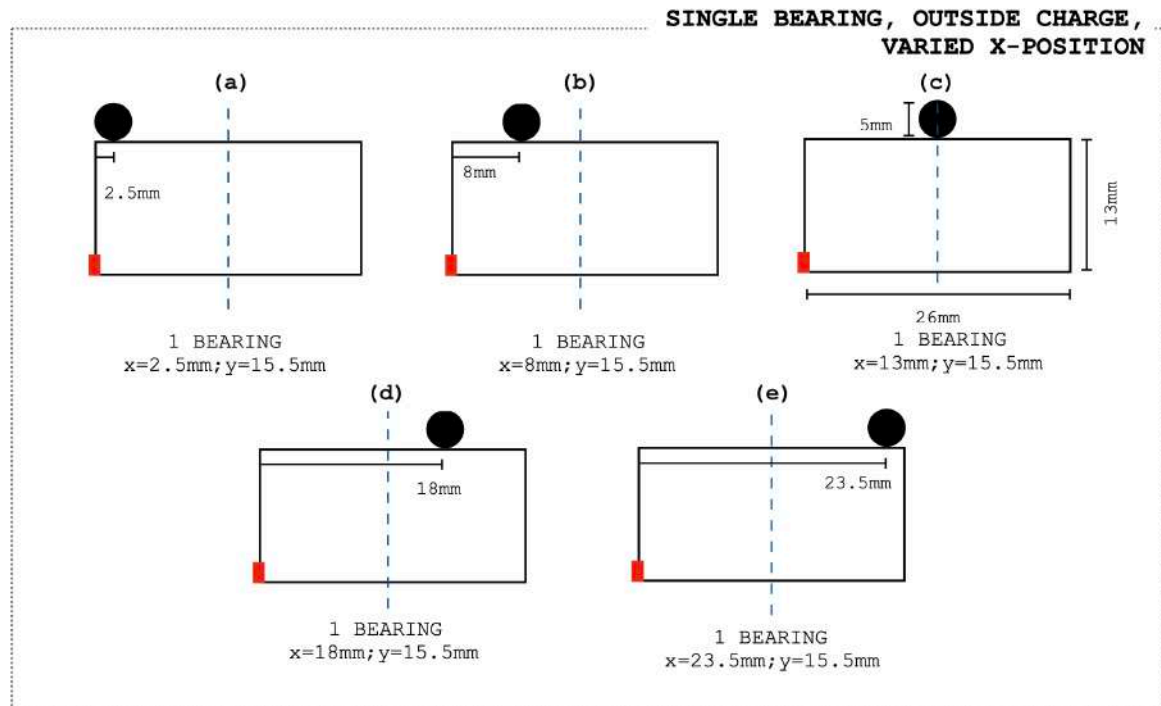


Figure 7.15: Axial plane, single ball bearing outside charge, varied x -position charge and ball bearing setup

Table 7.5: Summary of results for single-ball bearing outside charge, varied x -position axial plane simulations

SINGLE BALL BEARING OUTSIDE CHARGE, VARIED X -POSITION							
Fig.	Ball bearing x position (mm)	Maximum velocity (m/s)	Maximum momentum (10^{-3}kg.m/s)	Projection angle $\theta(^{\circ})$	Impact velocity (m/s)	Impact projection angle $\theta(^{\circ})$	Crater depth (mm)
7.15 a)	2.5	412.1	61.0	-23.8	326.0	-19.4	1.1
7.15 b)	8.0	527.5	78.1	-5.9	459.0	-6.2	1.5
7.15 c)	13.0	579.4	85.8	2.9	503.0	2.1	1.8
7.15 d)	18.0	575.6	85.2	9.4	486.9	7.5	1.4
7.15 e)	23.5	452.3	66.9	24.0	364.7	19.4	1.2

The charge and ball bearing configuration for axial plane simulations of a single ball bearing outside charge at varied axial position are shown in Figure 7.15. Table 7.5 lists a summary of the results, including the ball bearing x -position (i.e. ball bearing distance from the detonation end), the maximum velocity and the projection angle of the ball bearing (measured from the y -axis) at maximum velocity, the impact velocity and the projection angle at impact, and the final simulated crater depth.

Five configurations were investigated, as shown in Figure 7.15. The ball bearing y -position was fixed at 15.5mm , and the x -position was varied. It was expected that the position of the ball bearing would affect the ball bearing velocity as the setup geometry was not symmetrical in the axial direction, and the charge was detonated from one of the cylinder ends. As expected, the ball bearing velocities differed for all five test scenarios. The ball bearing velocities increased as the ball bearing moved further away from the detonation end, reaching a maximum at $x = 13.0\text{mm}$, i.e. when the ball bearing was placed in the middle of the rectangle representing the charge.

The ball bearing projection direction gradually shifted from -23.8° to 24° , as the ball bearing position shifted from the left to the right. At ball bearing position $x = 13.0\text{mm}$, the ball bearing projection angle was 2.9° , the ball bearing motion having shifted direction from the left to the right. The projection angle of the ball bearing was very similar or equal and opposite at the either ends. The maximum crater depth corresponded to the configuration with the highest impact velocity when the ball bearing was placed at $x = 13.0\text{mm}$.

7.3.1.1 Pressure contours

The transient pressure contours at different points in time for all simulation configurations involving a single ball bearing outside charge at varied axial position are shown in Figures 7.16, 7.17 and 7.18. High pressures were developed at the centre of the cylinder as the blast wave expanded. The ball bearings acted as a barrier disturbing the blast wavefronts. The shape of the wavefronts at time $t = 0.01\text{ms}$ were distinctly different for each configuration because of the presence of the ball bearing. However, as the blast wave continued to expand and reflected off the target repetitively, the wavefront shapes became more similar, eventually becoming indistinguishable.

The angle at which the ball bearing travelled was determined by the axial position of the ball bearing and therefore the direction at which the wavefront interacted with the ball bearing. The expanding hemispherical wavefront interacted with the ball bearing at different angles, propelling the ball bearing at different projection directions.

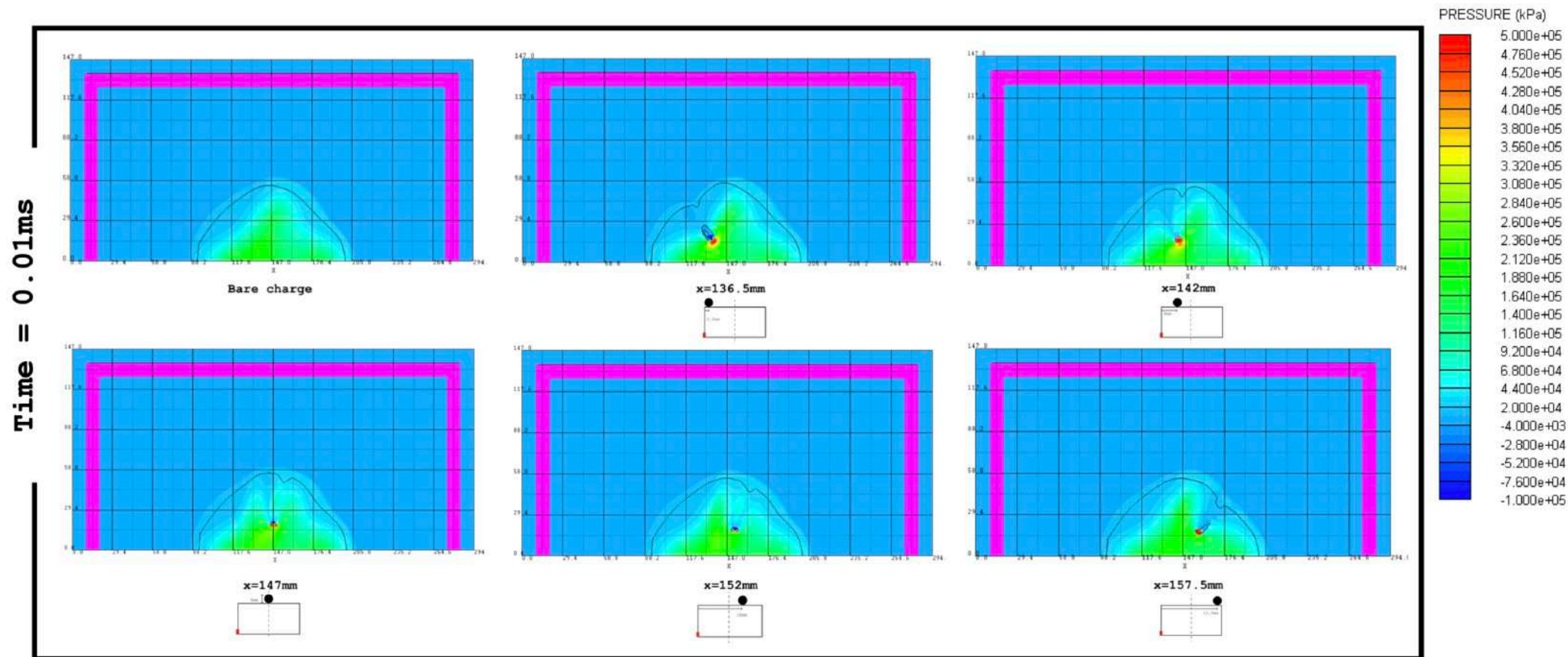


Figure 7.16: Pressure contour plots of axial plane simulation of single ball bearing outside charge at varied axial positions, at $t = 0.01ms$

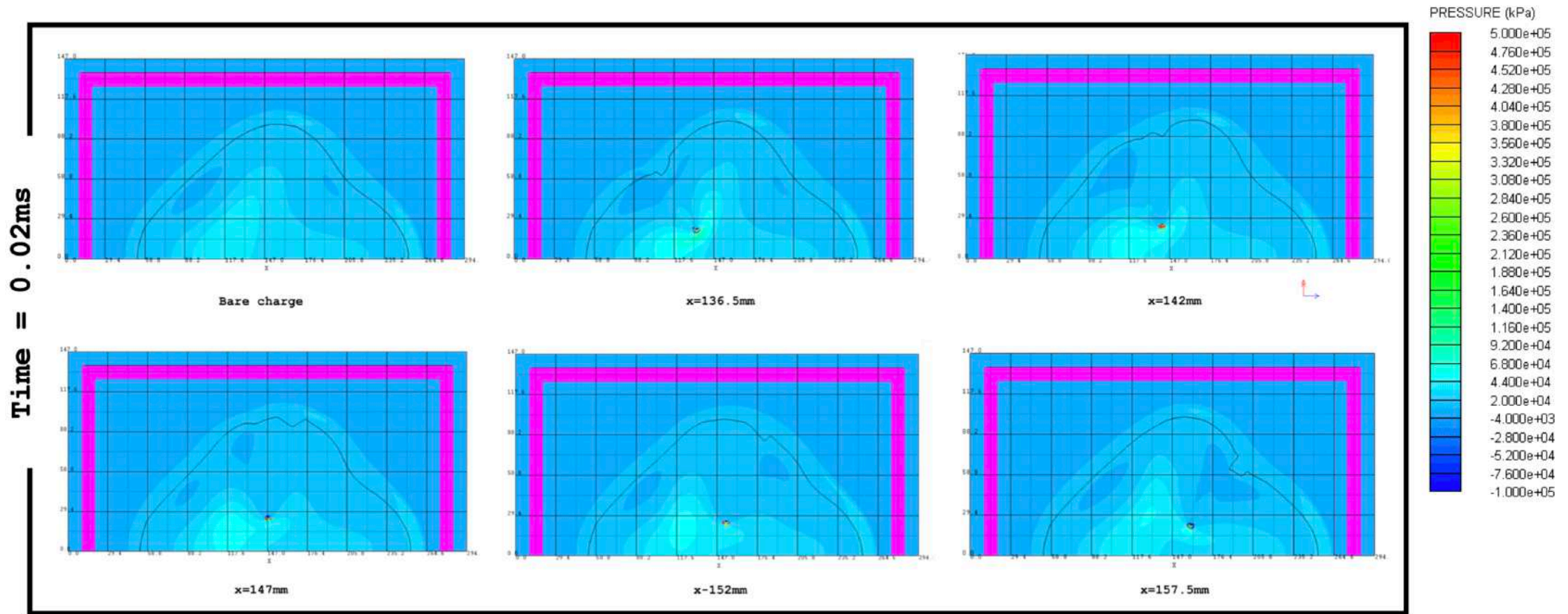


Figure 7.17: Pressure contour plots of axial plane simulation of single ball bearing outside charge at varied axial positions, at $t = 0.02\text{ms}$

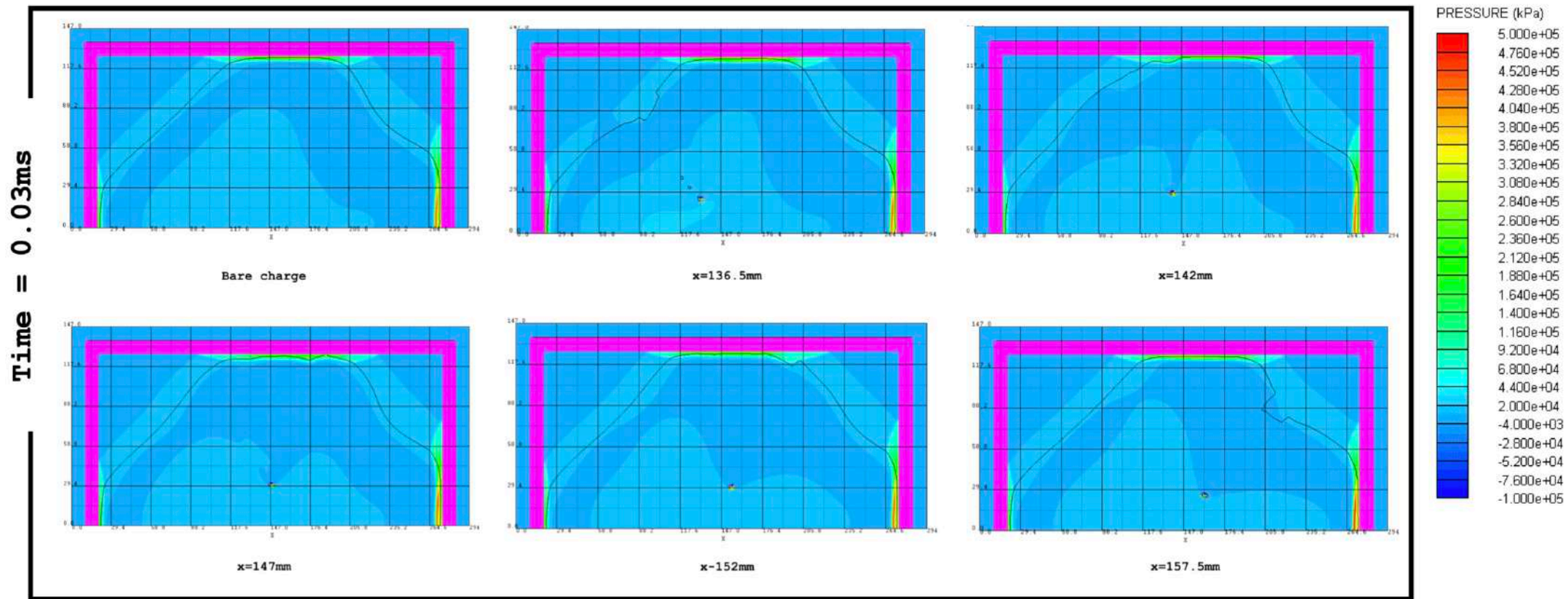


Figure 7.18: Pressure contour plots of axial plane simulation of single ball bearing outside charge at varied axial positions, at $t = 0.03ms$

7.3.1.2 Velocity profiles

Figure 7.19 shows the velocity-time profiles for all axial simulation configurations involving a single ball bearing placed outside charge at varied axial positions. There was a distinct difference between the velocity profiles with the change in ball bearing position. The ball bearing exhibited the lowest velocity gain when placed at either ends of the charge, and as a result, the time of impact was delayed. The percentage initial drop in velocity was also the highest when the ball bearing was placed at either end of the cylindrical charge. When the ball bearing was placed between $x = 8mm$ and $x = 18mm$, the velocity profile exhibited a more gradual decline from its maximum velocity to its impact velocity. The ball bearing velocity settled to a constant value, with the exception of $x = 13mm$ test, where the velocity increased slightly after impact. In all configurations, the impact velocity was lower than the maximum velocity by a constant amount. Even though the charges from both the experiments and the numerical models were partially confined rather than fully encased, the ball bearing was still shown to have a reduced velocity towards the charge end, exhibiting an "end effect" of casing fragments from encased charges as described by Charron [46] as a result of the rarefaction waves generated in the system.

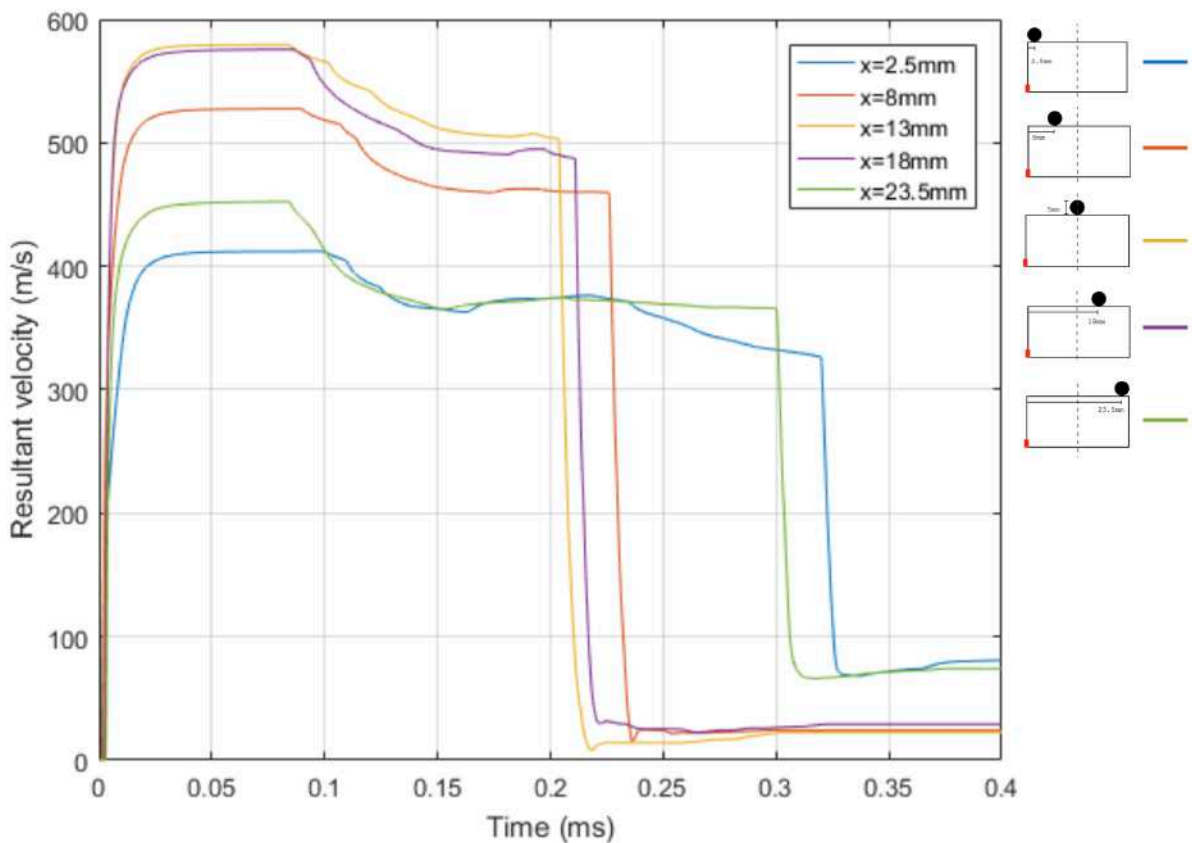


Figure 7.19: Graph of velocity vs. time for axial plane simulations of single ball bearing outside charge at varied axial positions

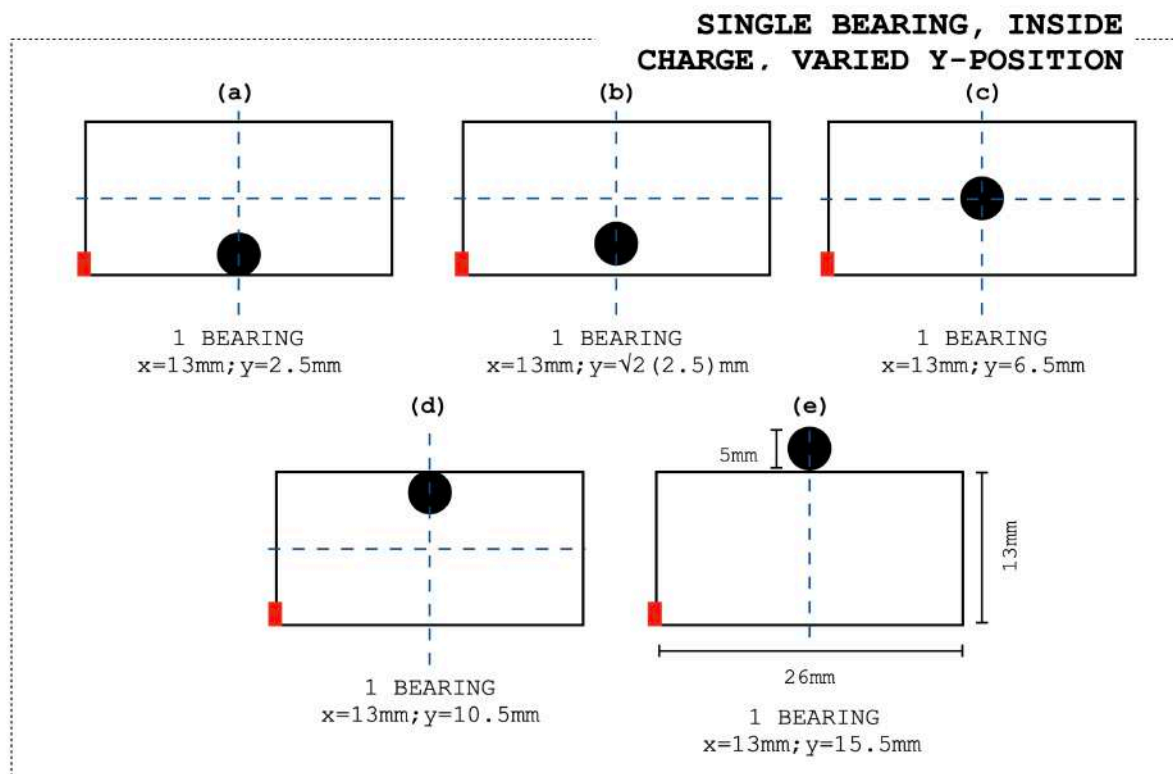
7.3.2 Single Ball Bearing inside the Charge at Varying Y -Position

Figure 7.20: Axial plane, single ball bearing inside charge, varied y -position charge and ball bearing setup

Table 7.6: Summary of results for single-ball bearing inside charge, varied y -position axial plane simulations

SINGLE BALL BEARING INSIDE CHARGE, VARIED Y -POSITION							
Fig.	ball bearing y position (mm)	Maximum velocity (m/s)	Maximum momentum (10^{-3}kg.m/s)	Projection angle $\theta(^{\circ})$	Impact velocity (m/s)	Impact projection angle $\theta(^{\circ})$	Crater depth (mm)
7.20 a)	2.5	151.5	22.4	-46.0	N/A	N/A	N/A
7.20 b)	$\sqrt{2}(2.5)$	167.0	24.7	-31.2	N/A	N/A	N/A
7.20 c)	6.5	333.1	49.3	-11.1	273.4	-16.2	0.8
7.20 d)	10.5	664.1	98.3	-0.3	597.1	-0.9	2.4
7.20 e)	15.5	579.4	85.8	2.9	503.0	2.1	1.8

The charge and ball bearing configuration for axial plane simulations of a single ball bearing inside charge at varied y -position are shown in Figure 7.20. Table 7.6 lists a summary of results, including the ball bearing distance from the central axis of charge cylinder (i.e. y -position), the maximum velocity and the projection angle of the ball bearing (measured from the y -axis) at maximum velocity, the impact velocity and the projection angle at impact, and the final simulated crater depth. Five different configurations were investigated. The ball bearing was placed 1) buried adjacent to the central axis of the charge (as shown in Figure 7.20 a)), 2) buried inside the charge (Figures 7.20 b) and c)), 3) buried just inside the charge surface (Figure 7.20 d)), or 4) placed on the surface of the charge (Figure 7.20 e)). ball bearing y -distance of $\sqrt{2}(2.5)mm$ was also investigated, to match the configuration used in the radial simulation.

The ball bearing velocity was observed to increase as the ball bearing y -distance increased (i.e. the ball bearing was placed further away from the centre of the charge), and reached its maximum at $y = 10.5mm$, when buried inside the charge and adjacent to the charge surface. The velocity decreased once the ball bearing was placed outside the charge-filled area. The ball bearing projection direction gradually shifted from 43.6° to 2.9° as the ball bearing y -distance increased. The maximum crater depth corresponded to the configuration with the highest impact velocity, as expected.

7.3.2.1 Pressure contours

The transient pressure contours at different points in time for all simulation configurations involving a single ball bearing inside charge at varied position are shown in Figures 7.21 and 7.22. As the ball bearing distance from the centre (y) increased, the disturbance in the pressure wavefronts became more pronounced. The ball bearing caused minimal disruption in the blast wavefronts when the ball bearing radial distance from the centre was small ($y \leq 6.5mm$), as was evident from the nearly spherical shape of the wavefronts in the early stages of the simulation. When placed near the circumferential edge, the ball bearings acted as a barrier disturbing the blast wavefronts, which were no longer "spherical". In the early stages of the simulation, it was observed that the ball bearing "split" the pressure waves, forcing the wavefront to travel around the ball bearing, as evident in the frames corresponding to $y = 10.5mm$ and $15.5mm$. As the blast wave continued to expand and reflect off the target repetitively, the wavefront shapes became more similar, eventually becoming indistinguishable.

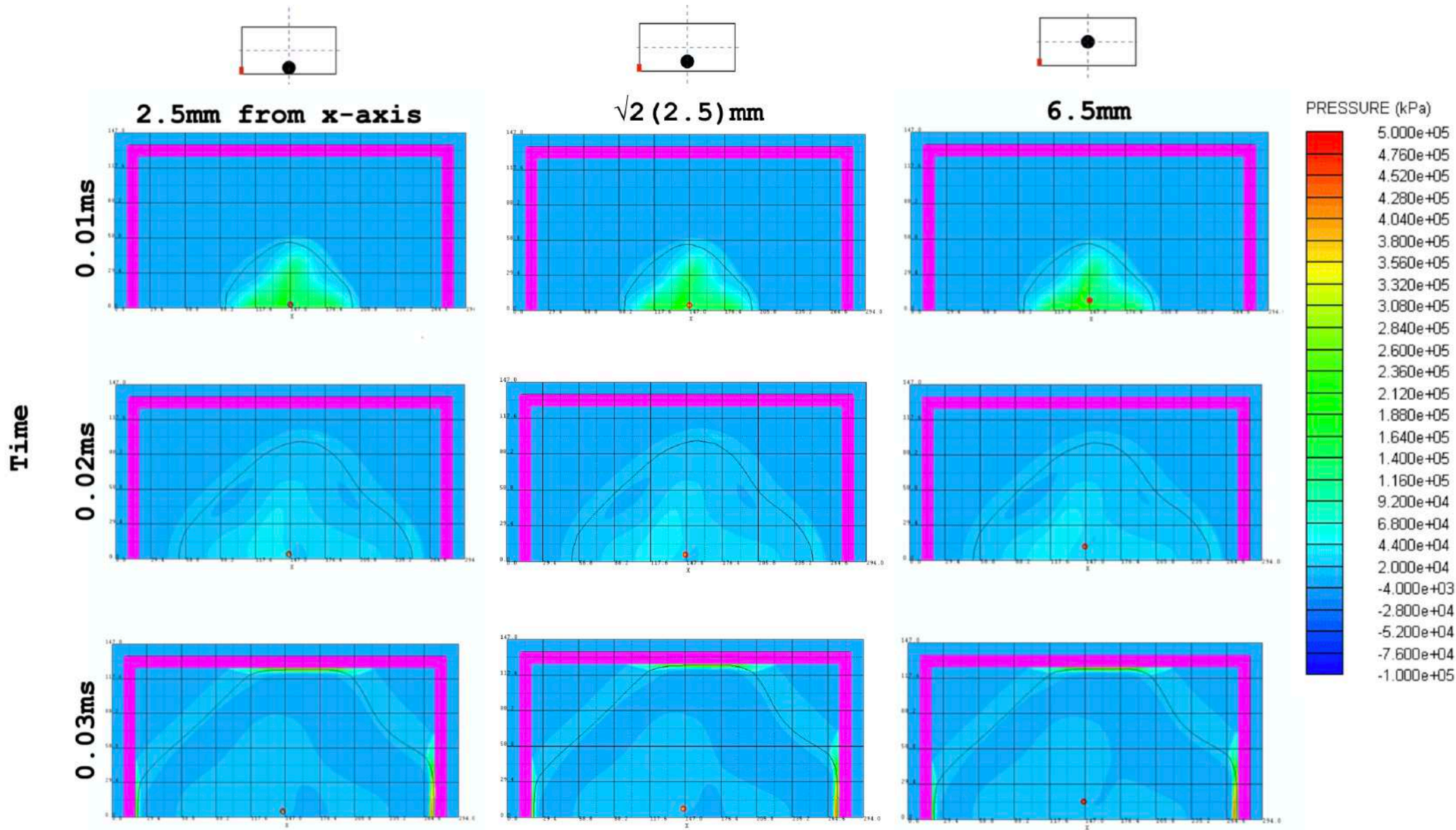


Figure 7.21: Pressure contour plots of axial plane simulation of single ball bearing inside charge at varied axial positions, ranging from $t = 0.01ms$ to $t = 0.03ms$

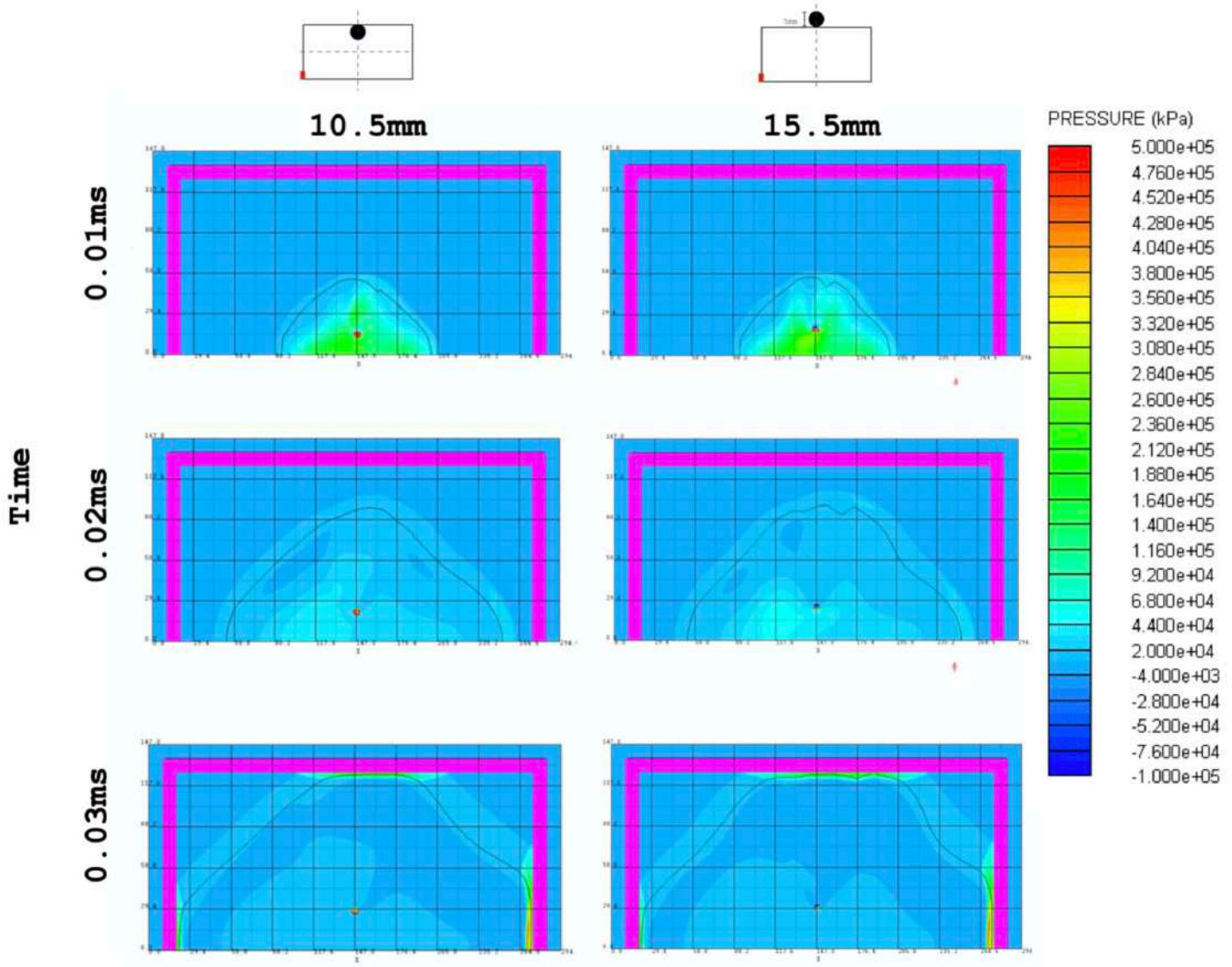


Figure 7.22: Pressure contour plots of axial plane simulation of single ball bearing inside charge at varied axial positions, ranging from $t = 0.01ms$ to $t = 0.03ms$

7.3.2.2 Velocity profiles

Figure 7.23 shows the series of velocity-time graphs for tests with single ball bearing at various y -distance from the centre. For test configurations $y = 2.5$ and $y = \sqrt{2}(2.5)mm$, the ball bearing velocity was low, averaging around $150m/s$. As a result, these simulations required significantly longer runtime for the ball bearing to impact the target. Due to computational practicality, the simulations were not run to completion under these configurations, as indicated in the corresponding velocity-time graph with the label "Terminated before impact".

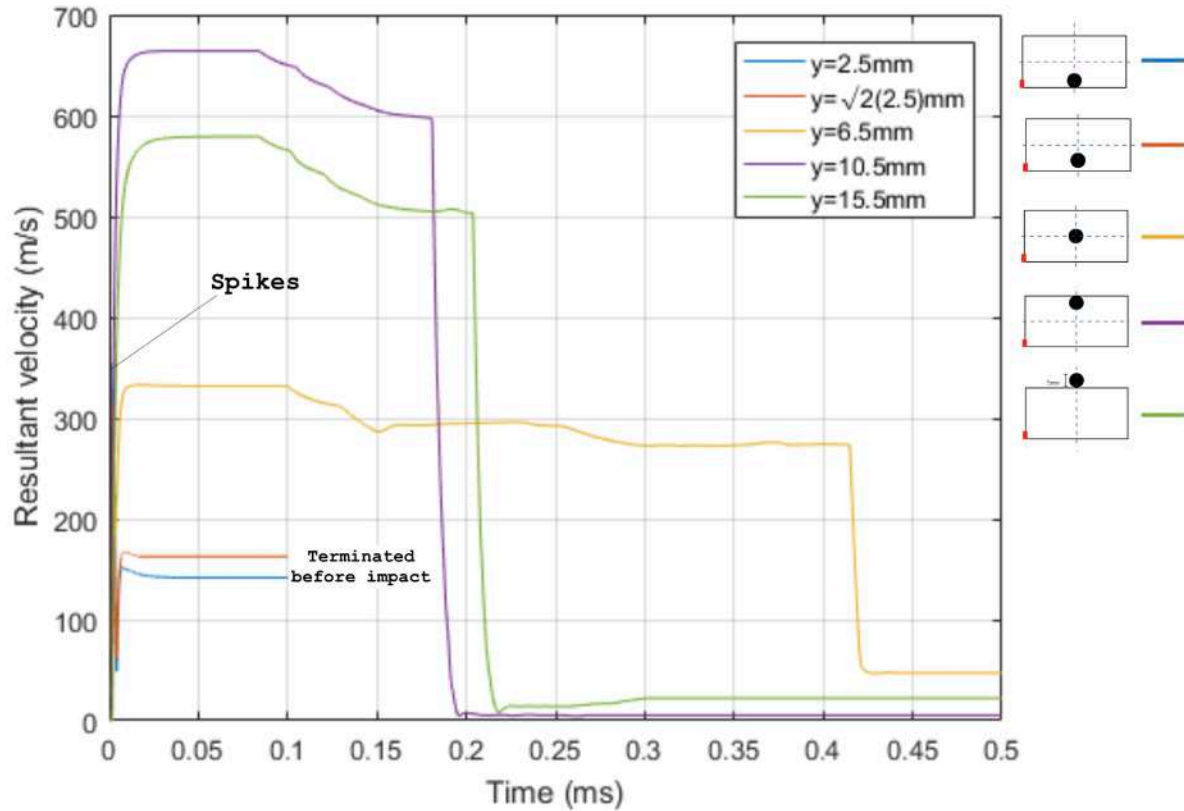


Figure 7.23: Graph of velocity vs. time for axial plane simulations of single ball bearing inside charge at varied axial positions

The velocity increased as the charge position approached the surface of the cylindrical charge, and was shown to peak at $y = 10.5\text{mm}$, i.e. buried just inside the surface. The velocity of the ball bearing decreased when it was positioned outside the charge. The higher velocity at $y = 10.5\text{mm}$ (than at $y = 15.5\text{mm}$) showed that the ball bearing's "depth of burial" inside the charge had an effect on the energy imparted onto the ball bearing.

There was a distinct difference between the velocity profiles when the ball bearing was buried inside the charge, compared to when it was placed outside the charge. When the ball bearing was placed near the central axis of the charge, a "spike" was observed in the velocity profile shortly after detonation, as labelled on Figure 7.23. The value of the "spike" was the highest, but was not recorded as the "maximum" velocity, as the ball bearing did not travel at this velocity. The magnitude of the "spike" decreased as the ball bearing moved further away from the charge centre. No "spike" in the velocity profile was observed when the ball bearing was placed adjacent to the charge surface.

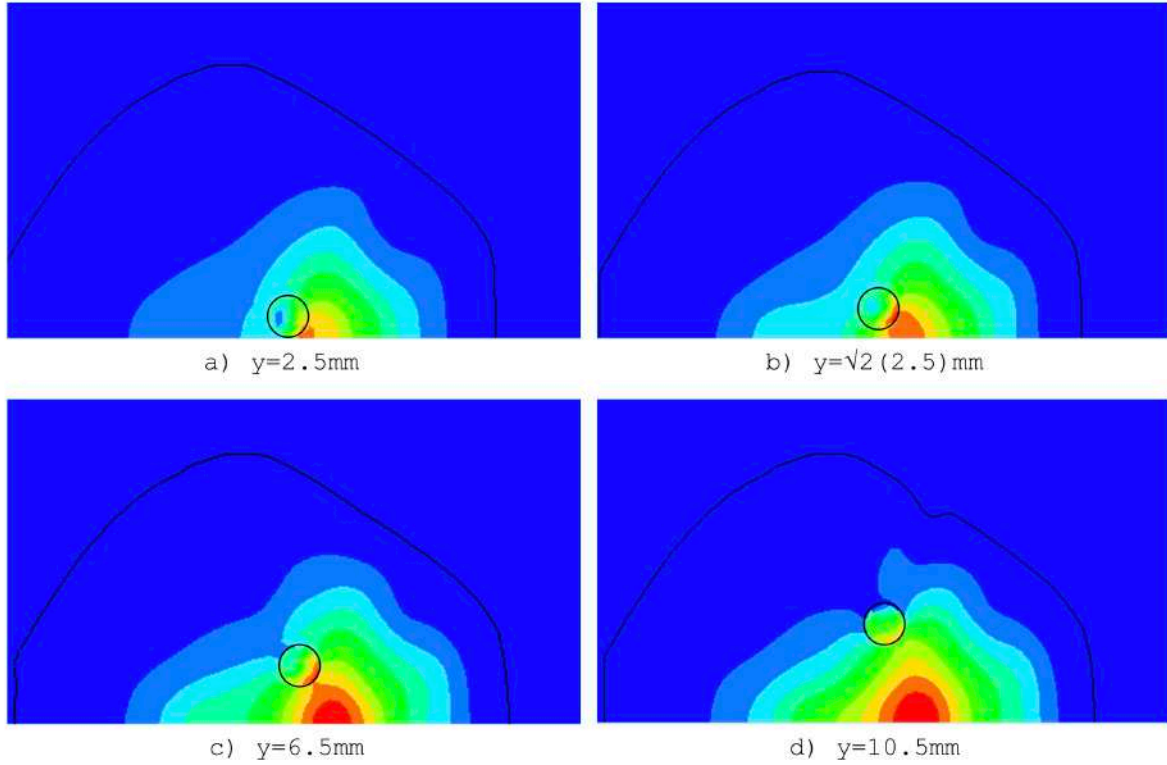


Figure 7.24: The ball bearing-detonation wave interaction at time $t = 0.004ms$ in the axial plane, single ball bearing inside simulations (blue indicates lower pressure, and red indicates high pressure; dark blue indicates atmospheric pressure)

The initial "spike" in velocity in the positive x -direction was most likely due to the interaction between the detonation shockfront and the ball bearing during the detonation process. A sudden change in direction of the ball bearing motion after the initial rise in velocity was also observed. Figure 7.24 shows the interaction between the ball bearing and the detonation waves at time $t = 0.004ms$ for different test conditions ($y = 2.5mm$, $y = \sqrt{2}(2.5)mm$, $y = 6.5mm$ and $y = 10.5mm$), when the charge had been detonated completely. When placed closer to the central axis of the cylindrical charge, the ball bearing was completely surrounded by the blast pressure waves in the middle of the cylindrical charge. As the ball bearing position shifted further up and closer to the surface of the cylindrical charge, the ball bearing was further away from the high-pressure zone towards the centre of the charge.

A lower pressure zone was observed on the left of the ball bearing, while a high-pressure zone was formed to the right. Therefore when the ball bearing was completely surrounded in the pressure waves, the pressure differential between the two ends may have caused the ball bearing to suddenly accelerate towards the lower-pressure end, thus causing the negative spike in velocity. However, once the charge was completely detonated, the expansion of the rarefaction waves and the fully detonated, dissipating gas products and its pressure became the dominant driving factor behind the motion of the ball bearing. The approximately hemispherical shape of the wavefront determined the direction in which the ball bearing travelled, until the velocity stabilised to a constant value.

7.3.3 Single Ball Bearing outside the Charge at Varying Ball Bearing Diameter

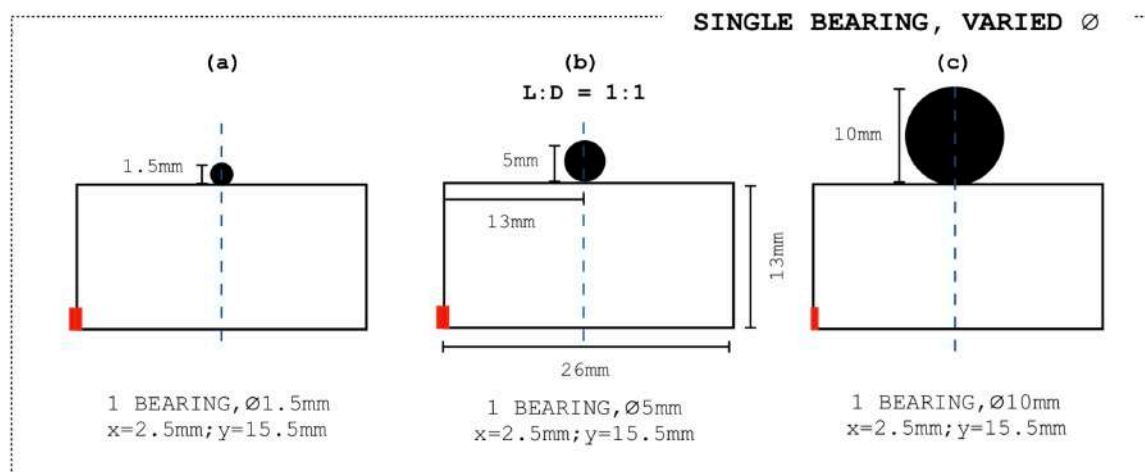


Figure 7.25: Axial plane, single ball bearing outside charge, varied ball bearing diameter charge and ball bearing setup

Table 7.7: Summary of results for single-ball bearing varied ball bearing diameter axial plane simulations

SINGLE BALL BEARING VARIED DIAMETER							
Fig.	ball bearing diameter (mm)	Maximum velocity (m/s)	Maximum momentum (10^{-3}kg.m/s)	Projection angle $\theta(^{\circ})$	Impact velocity (m/s)	Impact projection angle $\theta(^{\circ})$	Crater depth (mm)
7.25 a)	1.5	863.7	42.6	1.8	751.3	-0.1	1.1
7.25 b)	5	579.4	85.8	2.9	503.0	2.1	1.8
7.25 c)	10	315.0	190.9	2.1	282.2	0.3	2.6

The charge and ball bearing configuration for axial plane simulations involving a single ball bearing outside charge at varied ball bearing diameter are depicted in Figure 7.25. Table 7.7 lists a summary of results, including the ball bearing diameter, the maximum velocity and the projection angle of the ball bearing (measured from the y -axis) at maximum velocity, the impact velocity and the projection angle at impact, and the final simulated crater depth. The ball bearing was placed on the charge surface, such that its central axis coincided with the vertical mid-plane of the rectangle representing the charge. The ball bearing diameter was varied. Three different diameters were tested, as shown in Figure 7.25.

The ball bearing velocity was observed to decrease as the ball bearing diameter increased. However, the crater depth increased with an increase in ball bearing diameter, which was the reverse of the pattern observed in the radial plane simulations. This implied that the crater depth was not solely dependent on the velocity of the projectile, and that the target geometry affects the crater depth. The projection angle did not show a clear trend of increase or decrease

as the ball bearing diameter increased. The projection angle of the ball bearing was maximum when the ball bearing diameter was $5mm$, at 2.9° .

7.3.3.1 Pressure contours

The transient pressure contours at different points in time are shown in Figure 7.26. The stages at which the wavefronts exhibit distinct differences were highlighted. As the ball bearing diameter increased, the disturbance in the pressure wavefronts became more pronounced and the wavefront was shown to lose its spherical shape. The ball bearing caused minimal disruption to the blast wavefront at diameter $d = 1.5mm$, evident in the nearly spherical shape of the wavefront in the early stages of the simulation. However, at $d = 10mm$ it was observed that the ball bearing "split" the pressure waves, forcing the wavefront to travel around the ball bearing. This was to be expected, as the degree of disturbance was proportional to the surface area occupied by the ball bearing compared to the overall simulated area. As the blast wave continued to expand and reflected off the target repetitively, the wavefront shapes became more similar, eventually becoming indistinguishable.

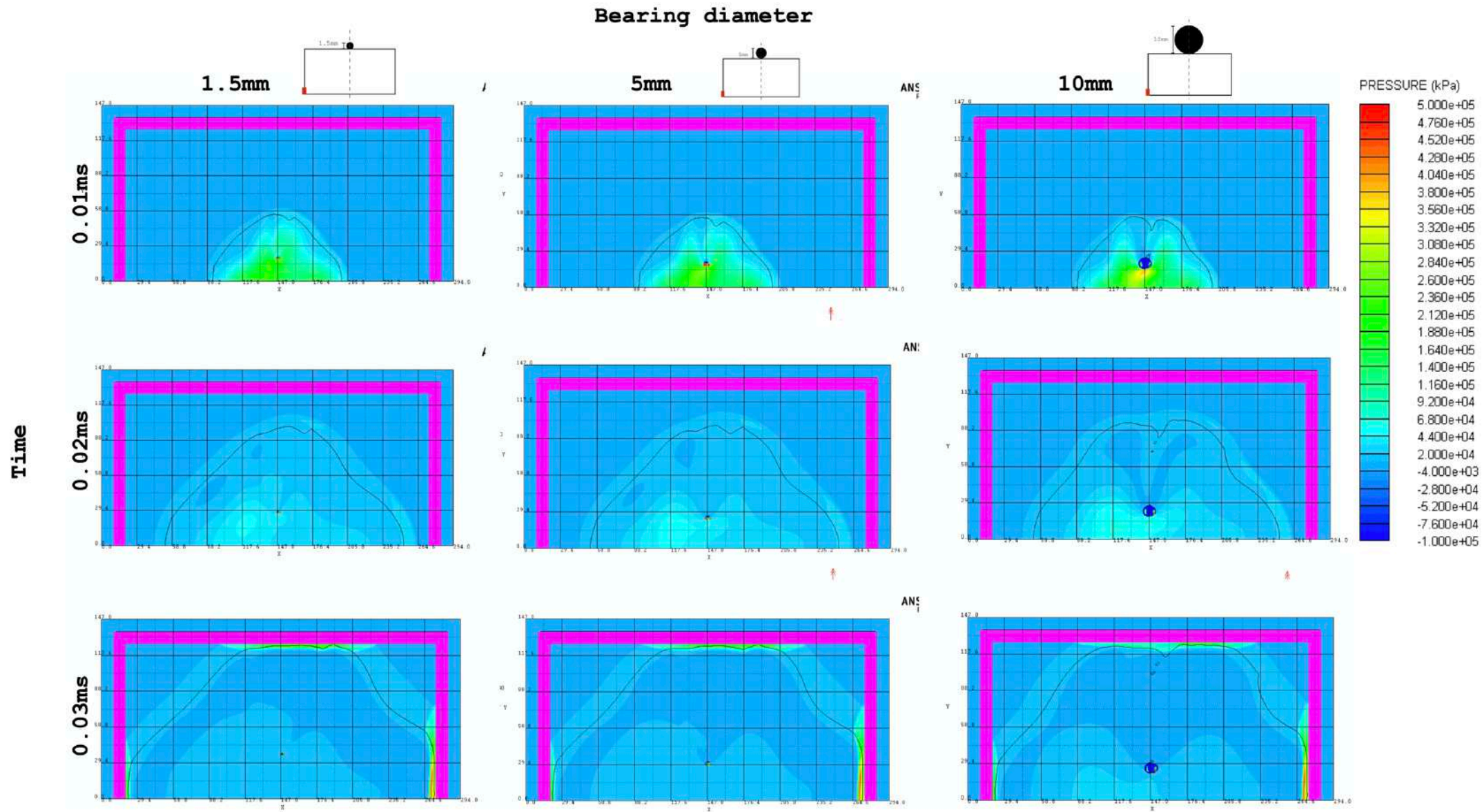


Figure 7.26: Pressure contour plots of axial plane simulation of single ball bearing outside charge at varied ball bearing diameters, ranging from $t = 0.01ms$ to $t = 0.03ms$

7.3.3.2 Velocity profiles

Figure 7.27 shows the series of velocity-time graphs for axial plane tests with single ball bearing at various ball bearing diameter. The velocity (both maximum velocity and the impact velocity) decreased as the ball bearing diameter increased. The percentage difference between the maximum velocity and impact velocity decreased as the ball bearing diameter was increased, from 13.1% to 10.4%. This was to be expected, as the ball bearing diameter differed by approximately ten times between the minimum and the maximum diameter. Therefore, ball bearing mass and inertia was also increased significantly.

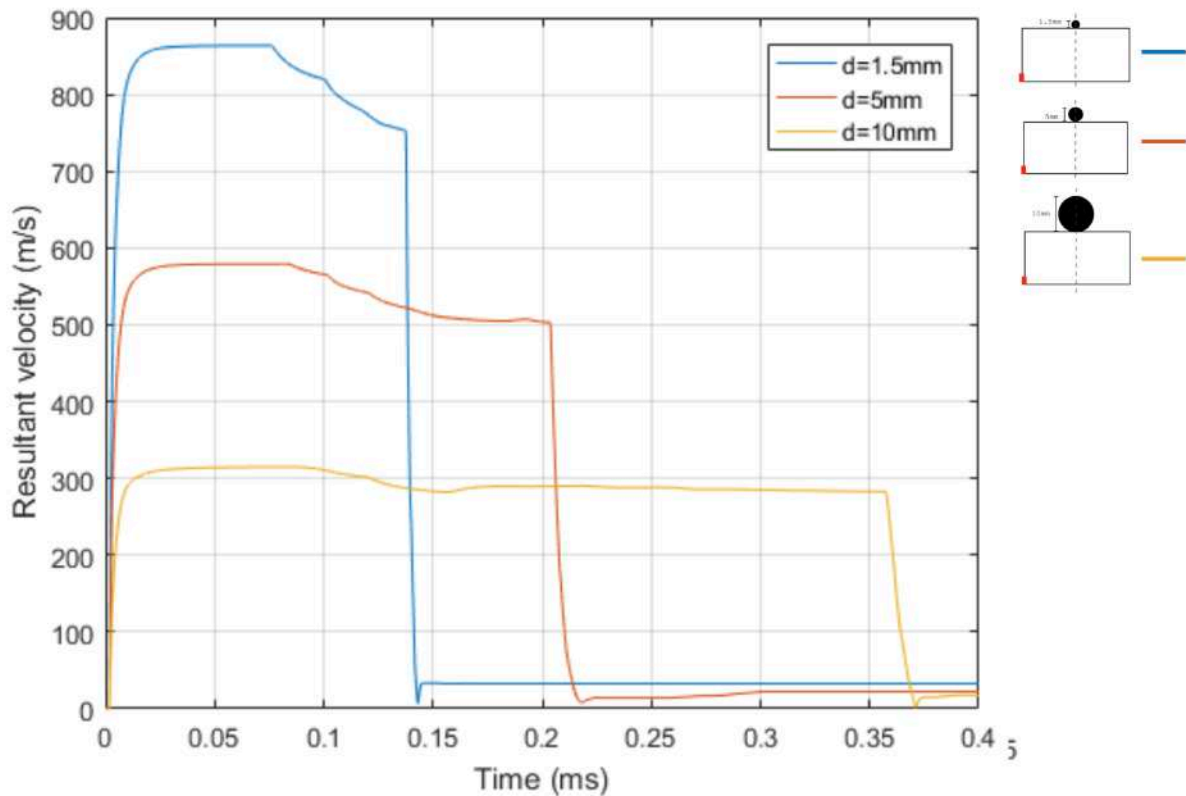


Figure 7.27: Graph of velocity vs. time for axial plane simulations of single ball bearing outside charge at varied ball bearing diameters

There was a distinct difference between the velocity profiles as the ball bearing diameter was varied. It was noted that as the ball bearing diameter increased, the velocity profile became more irregular. For ball bearing diameter $d = 1.5\text{mm}$, it could be observed that the ball bearing experienced a single major drop in velocity, and the velocity profile remained relatively smooth afterwards, except for the slight increase and consequent decrease in velocity shortly before impact. As explained in the radial plane section, this may be due to the fact that the ball bearing was travelling very fast, and interacted with the pressure waves for a shorter amount of time. The pressure wave interaction would have greater effect on the ball bearing motion for lighter ball bearings.

As the ball bearing diameter increased, the interaction time between the ball bearing and the pressure waves increased as well. In other words, there was an increased amount of interaction between the ball bearing and the pressure waves, causing more disruption in ball bearing motion and change in ball bearing velocity. At ball bearing diameter $d = 10mm$, two distinct "steps" were present in the velocity profiles where the ball bearing velocity dropped initially, stabilised for a short period of time, before decreasing further. This behaviour may be due to prolonged exposure of the ball bearing to reflected pressure waves.

7.3.4 Single Ball Bearing outside the Charge at Varying Aspect Ratios

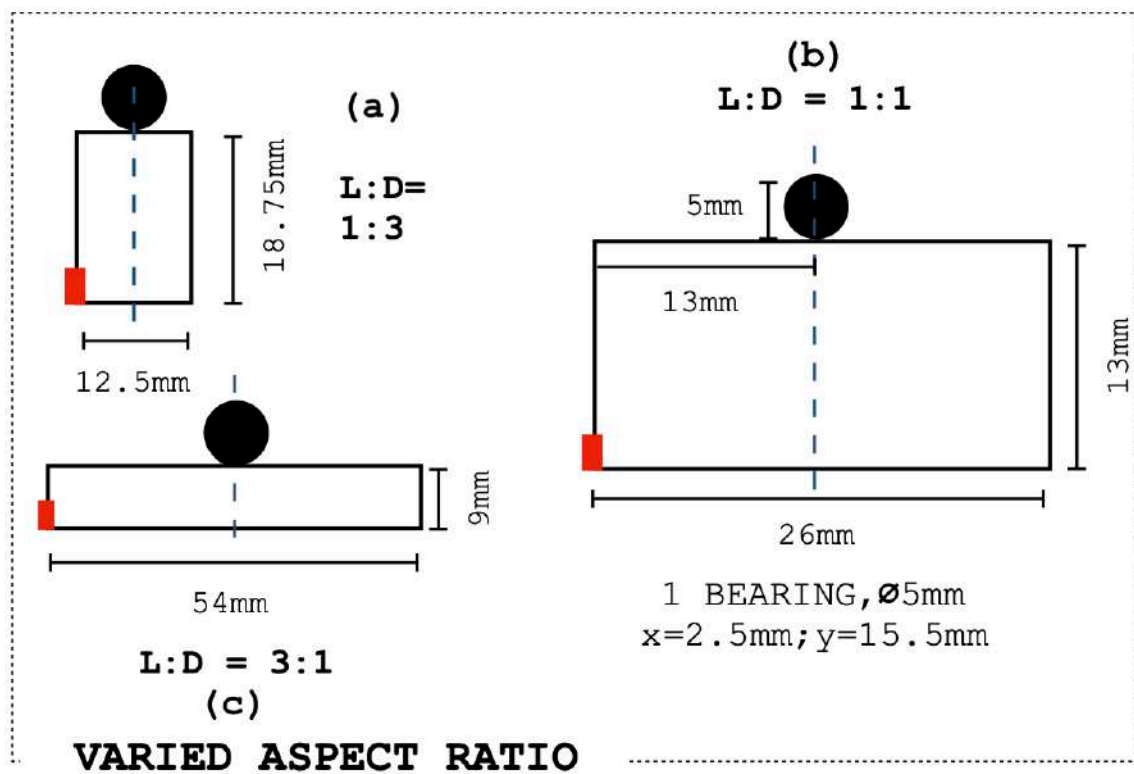


Figure 7.28: Axial plane, single ball bearing outside charge, varied charge aspect ratio charge and ball bearing setup

Table 7.8: Summary of results for single-ball bearing varied aspect ratio axial plane simulations

SINGLE BALL BEARING VARIED ASPECT RATIO (L/D)							
Fig.	Charge L/D	Maximum velocity (m/s)	Maximum momentum ($10^{-3}kg.m/s$)	Projection angle $\theta(^{\circ})$	Impact velocity (m/s)	Impact projection angle $\theta(^{\circ})$	Crater depth (mm)
7.28 a)	1/3	375.7	55.6	0.3	294.9	0.2	1.0
7.28 b)	1	579.4	85.8	2.9	503.0	2.1	1.8
7.28 c)	3	662.6	98.1	3.5	592.0	3.2	2.5

The charge and ball bearing configuration for axial plane simulations with a single ball bearing outside charge at varied charge aspect ratio are shown in Figure 7.28. Table 7.8 lists a summary of the results, including the aspect ratio (L/D), the maximum velocity and the projection angle of the ball bearing (measured from the y -axis) at maximum velocity, the impact velocity and the projection angle at impact, and the final simulated crater depth.

The ball bearing was placed such that its central axis coincided with the vertical mid-plane of the rectangle representing the charge. The ball bearing diameter was fixed at $5mm$. Three different aspect ratios were tested, as shown in Figure 7.28. The ball bearing velocities increased as the explosive charge aspect ratio (L/D) increased. The crater depth and the projection angles also increased with an increase in the ratio L/D . The projection angle of the ball bearing was maximum when the aspect ratio was equal to 3, at 3.5° .

7.3.4.1 Pressure contours

The transient pressure contours at different points in time are shown in Figure 7.29. The stages at which the wavefronts exhibit distinct differences were highlighted. The differences in wavefront shapes under different configurations were more significant during the earlier stages of the simulation. The pressure towards the centre of the cylinder increased significantly as the aspect ratio increased. The original wavefront differed in its aspect ratio depending on the charge aspect ratio. The wavefront shape was more slanted towards the right as the aspect ratio increased. A deeper "split" was observed in the wavefront under the test condition of $L/D = \frac{1}{3}$. The wavefront shape was more slanted to the left under the $L/D = 3$ condition.

As the pressure waves dissipated as the simulation progressed further, the pressure wavefronts were shown to exhibit very similar patterns independent of the aspect ratio involved. For example, the wavefronts impacted the target walls around the same areas independent of the test condition, as can be seen at $t = 0.03ms$.

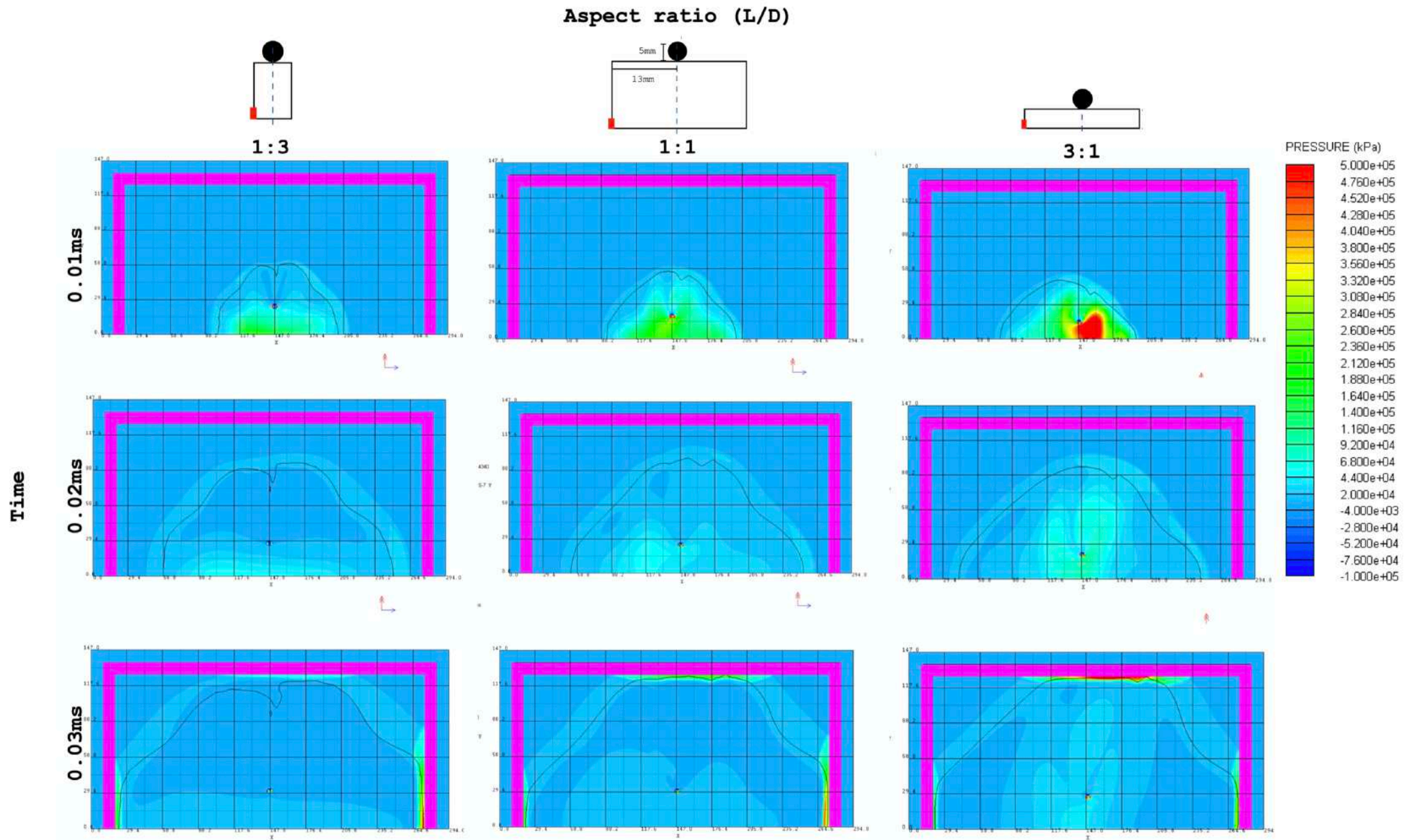


Figure 7.29: Pressure contour plots of axial plane simulation of single ball bearing outside charge at varied charge aspect ratios (L/D), ranging from $t = 0.01ms$ to $t = 0.03ms$

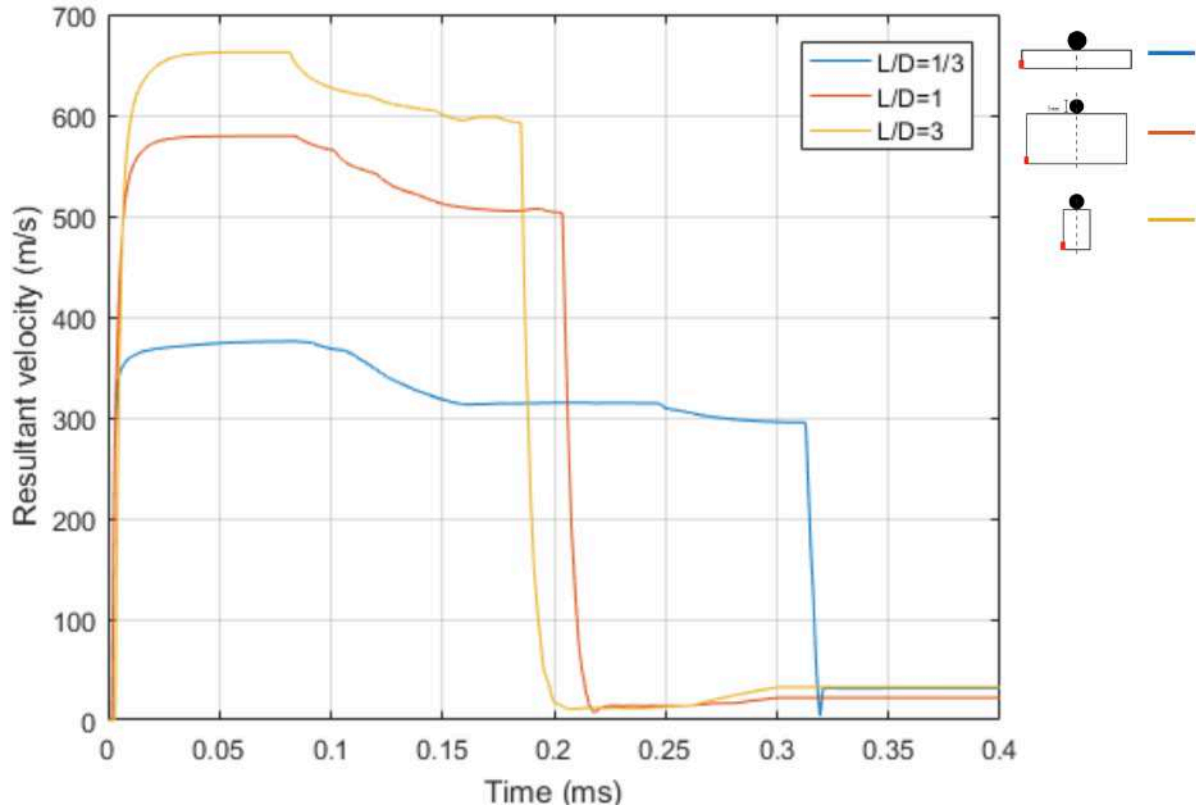


Figure 7.30: Graph of velocity vs. time for axial plane simulations of single ball bearing outside charge at varied charge aspect ratios

7.3.4.2 Velocity profiles

Figure 7.30 shows the series of velocity-time graphs for axial plane tests with single ball bearing at various charge aspect ratios (L/D). The velocity increased as the charge aspect ratio (L/D) increased. Increasing the length and therefore the ratio between the charge length L and the ball bearing diameter, increased the ball bearing velocity. This corresponds with the observation made by Zulkoski [50] that an increase in L/D with correspondingly decreased diameter resulted in an increase in energy even though weight and volume constraints remained the same, as a result of the end effects that become less of a factor on the total energy consideration for longer cylindrical charges.

At $L/D = 3$, it was observed that the ball bearing experienced a single, steep decrease in velocity with the velocity profile remaining relatively smooth afterwards until impact. This is due to the fact that the ball bearing was travelling very fast, and interacted with the pressure waves for a shorter amount of time. It should also be noted that the ball bearing velocity did not settle to a constant value, but instead increased, after impact. For $L/D = 1$, the velocity profile also exhibited a single drop in velocity, but the rate of the decrease was more gradual. At a lower aspect ratio, the ball bearing velocity was lower on average. Therefore there was an increased amount of interaction between the ball bearing and the pressure waves, causing more disruption in ball bearing motion and change in ball bearing velocity. At $L/D = \frac{1}{3}$, the ball

bearing velocity profile exhibited short and frequent oscillations after its initial drop, compared to the velocity profiles corresponding to higher L/D aspect ratios. This implied that the ball bearing experienced more disturbance during its motion, due to pressure waves. The percentage initial drop in velocity decreased as the aspect ratio L/D increased.

7.4 Axial Plane, Multiple Ball Bearings

7.4.1 Single, 2-, 3-, 4- and 5- Ball Bearings outside the Charge

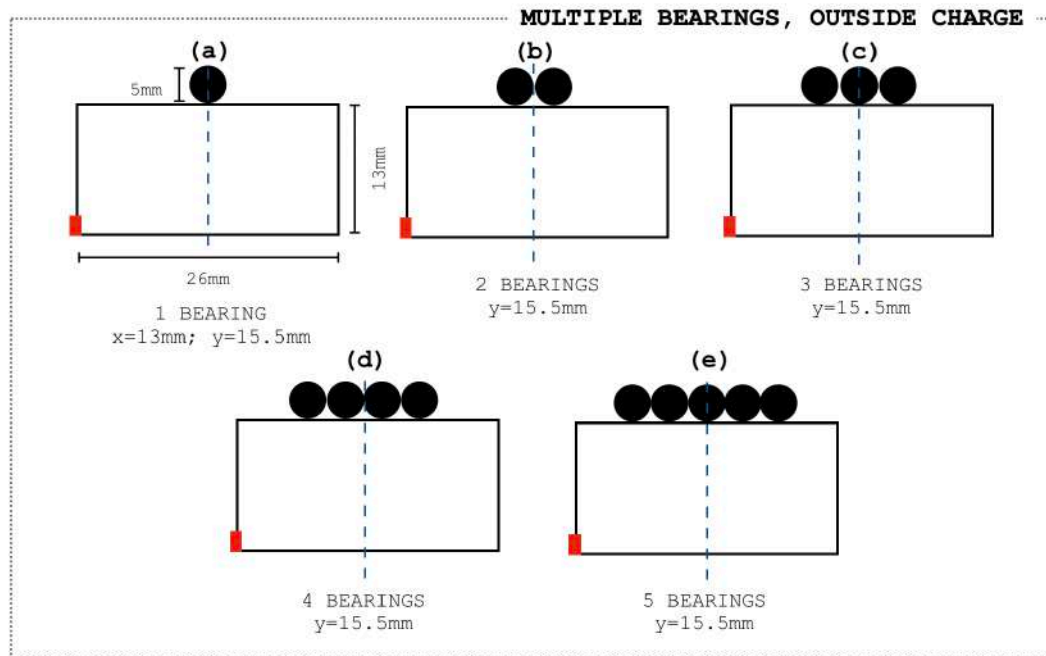


Figure 7.31: Axial plane, multiple ball bearings outside charge setup

Table 7.9: Summary of results for multiple-ball bearings axial plane simulations

Fig.	Total no. of ball bearings	Ball bearing x position (mm)	Maximum velocity (m/s)	Maximum momentum ($10^{-3}kg.m/s$)	Projection angle $\theta(^{\circ})$	Impact velocity (m/s)	Impact proj. angle $\theta(^{\circ})$	Crater depth (mm)	Crater depth from experiment (mm)	
									Maximum	Average
7.31 a)	1	13.0	579.4	85.8	2.9	503.0	2.1	1.8	1.5	1.0
7.31 b)	2	10.5	682.2	101.0	-21.3	591.7	-21.9	1.6	1.5	1.1
		15.5	749.8	111.0	23.0	670.4	21.2	1.9	1.7	1.1
7.31 c)	3	8.0	830.5	122.9	-0.4	766.0	-2.4	1.9	-	-
		13.0	841.2	124.5	28.4	727.8	26.0	1.4	-	-
		15.0	710.4	105.1	-27.9	616.7	-28.2	1.3	-	-
7.31 d)	4	5.5	676.7	100.2	-33.1	586.8	-32.9	0.9	-	-
		10.5	813.4	120.4	-6.5	683.6	-7.0	1.7	-	-
		15.5	884.6	130.9	5.7	816.3	2.9	1.7	-	-
		20.5	837.6	124.0	34.0	687.9	31.2	0.7	-	-
7.31 e)	5	3.0	646.1	95.6	-40.1	576.9	-41.6	0.7	1.3	0.8
		8.0	804.2	119.0	-11.1	690.7	-13.4	2.0	1.5	1.0
		13.0	923.5	136.7	-0.2	850.4	-3.0	2.6	1.5	1.1
		18.0	925.5	137.0	8.8	861.3	5.8	2.9	2.0	1.2
		23.0	808.2	119.6	39.5	746.5	38.6	1.3	1.5	1.1

The charge and ball bearing configuration for axial simulations involving multiple ball bearings outside charge are shown in Figure 7.31. A summary of the results for the multiple-ball bearing, axial plane simulations is listed in Table 7.9, including the total number of ball bearings, the

x -position of each ball bearing, the maximum velocity, the projection angle of the ball bearing (measured from the y -axis) at maximum velocity, the impact velocity, the projection angle at impact, the final simulated crater depth, and the crater depth from experiment when applicable.

Five configurations were used, as shown in Figure 7.31. The ball bearing was either placed such that 1) the central axis of the middle-most ball bearing coincided with the mid-plane through the charge rectangle, or, 2) if there were even number of ball bearings, such that the ball bearings were adjacent to the mid=plane line. The ball bearing centre distance from the bottom edge of the charge rectangle was fixed at $15.5mm$.

For the 2-ball bearings configuration, the maximum and impact velocities obtained for the two ball bearings were not equal, as the charge was detonated from the left end of the cylinder. Similarly, the ball bearing projection angles at maximum velocity were not equal and opposite. This showed that the simulations did not necessarily display symmetrical behaviour about the charge midplane as expected. It must also be noted that the depths corresponding to the ball bearings were not identical either, both in experiment and in simulation. Similarly, other configurations did not display symmetrical behaviour about the charge midplane. The maximum and impact velocity values did not show a clear pattern of symmetry, unlike in the radial plane simulations.

The ball bearing velocities increased as the ball bearing moved further away from the detonation end until it peaked in the mid-region of the charge cylinder, and subsequently decreased. In the 3-ball bearings test, the middlemost ball bearing exhibited maximum velocity. However, in the 4-ball bearings and 5-ball bearings tests, the ball bearing adjacent to the rightmost ball bearing (i.e. ball bearing #3 in the 4-ball bearings test and #4 in the 5-ball bearings test) showed maximum velocity. The maximum crater depth corresponded to the configuration with the highest impact velocity in all test configurations. Under multiple-ball bearing conditions, the ball bearing projection direction gradually shifted from the left to the right as the ball bearing position shifted from the detonation end to far end.

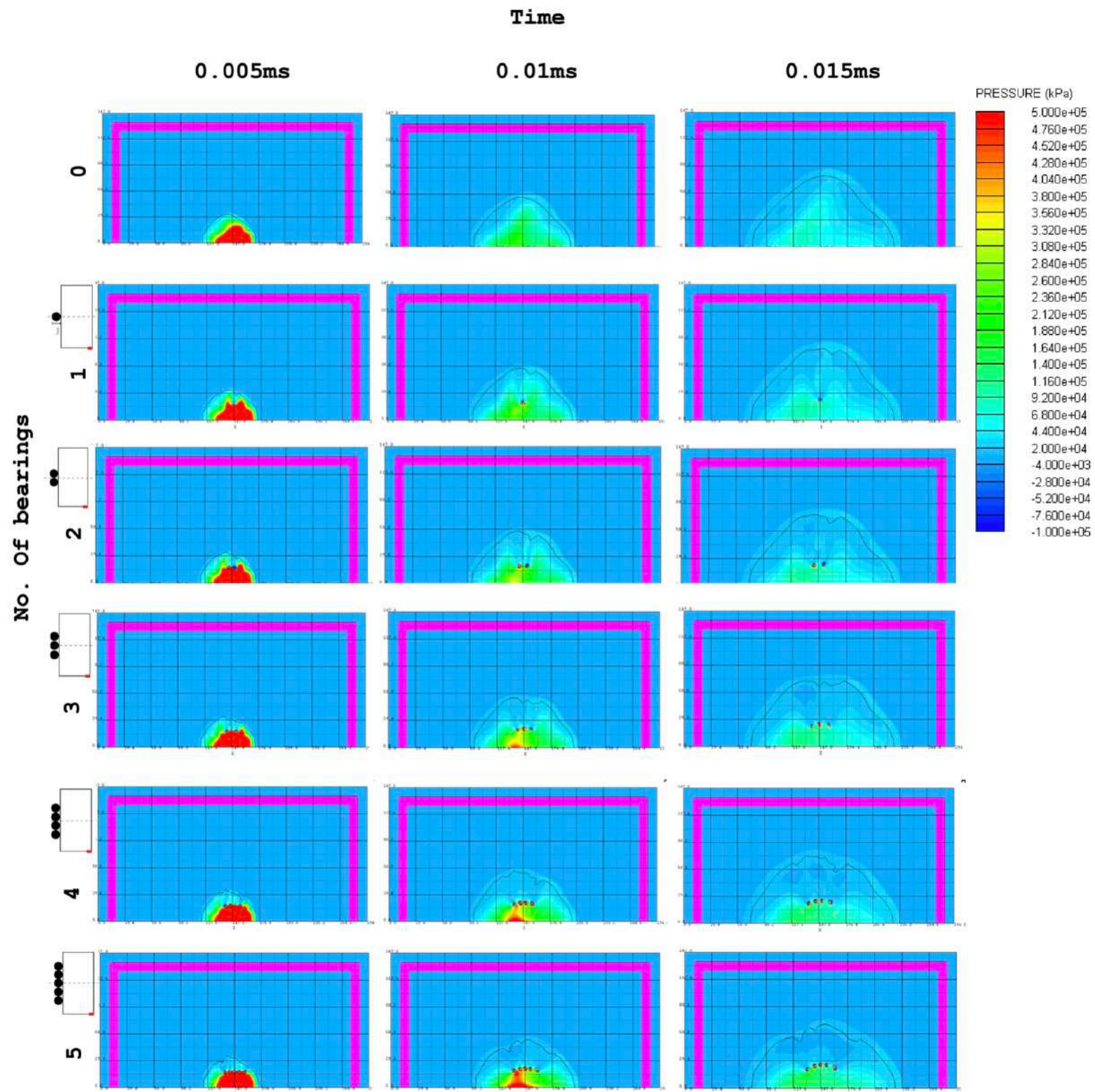


Figure 7.32: Pressure contour plots of axial plane simulation of multiple ball bearings outside charge at varied positions ranging from $t = 0.005ms$ to $t = 0.015ms$

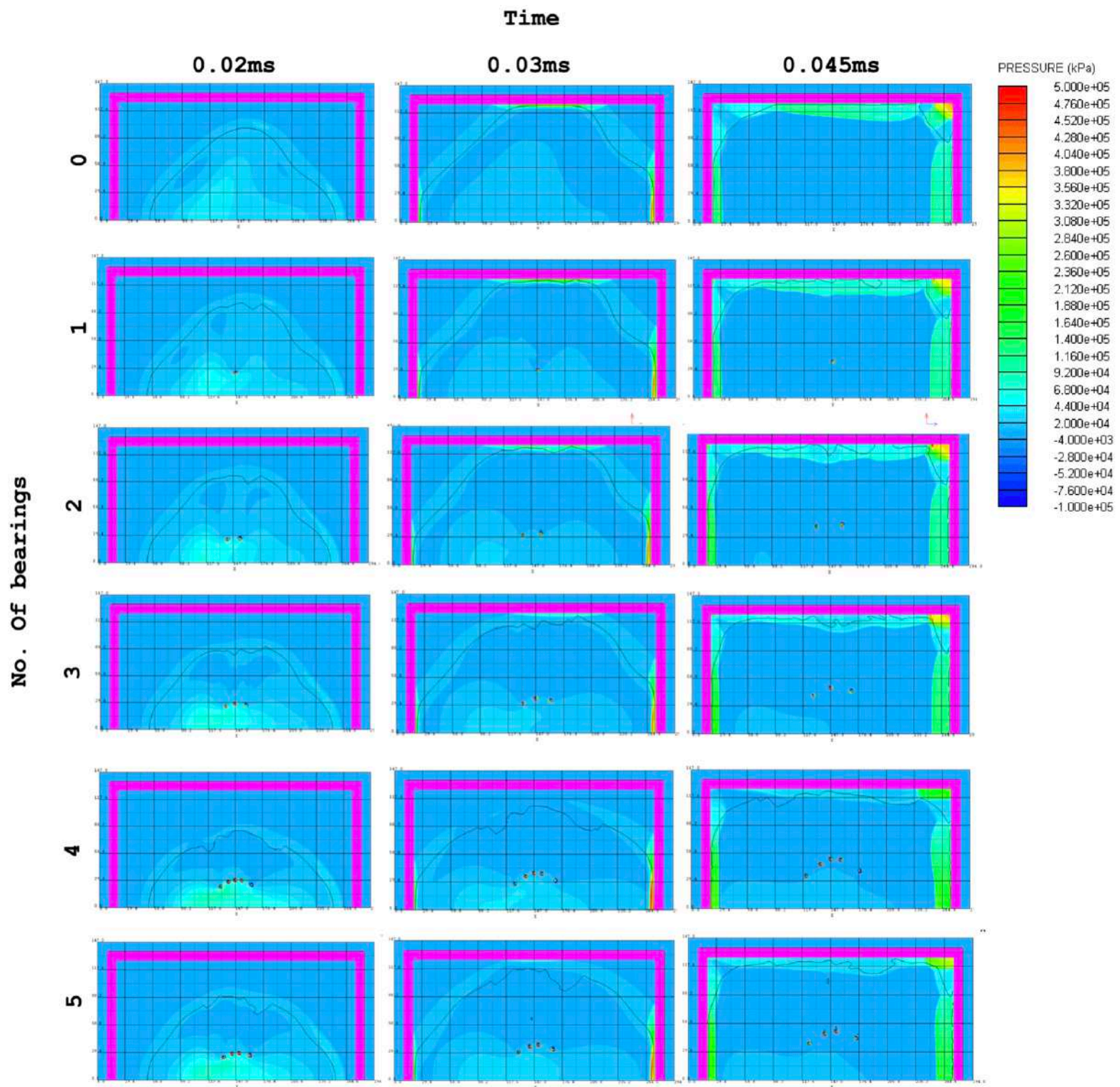


Figure 7.33: Pressure contour plots of axial plane simulation of multiple ball bearings outside charge at varied positions ranging from $t = 0.02ms$ to $t = 0.045ms$

7.4.1.1 Pressure contours

The pressure contours for a series of simulations with 1, 2, 3, 4 and 5 ball bearings, compared with bare charge (i.e. '0' ball bearing) in the axial plane at various points in time are shown in Figures 7.32 and 7.33. The differences in wavefront shapes and pressure contours under different configurations were more significant during the earlier stages of the simulation. At $t = 0.01ms$, it was observed that the ball bearings affected the overall shape of the wavefront immediately after detonation. The difference in wavefront shapes can be clearly observed at $t = 0.015ms$.

The pressure wave propagation slowed down as the number of ball bearings increased. The presence of a high-pressure zone towards the centre of the cylinder was still observed at $t = 0.01ms$ in simulations where 3 or more ball bearings were used, indicating that pressure dissipation occurred at a slower rate as the number of ball bearings increased. At $t = 0.03ms$, the wavefront had reached edge A-1 under the 0, 1, 2 and 3-ball bearings configurations. In the 4- and 5-ball bearings tests, however, the wavefront had not reached edge A-1 at that point in time, implying that the wavefront was expanding at a slower rate compared to other configurations involving fewer ball bearings. The pressure waves dissipated as the simulation progressed further. The pressure wavefronts were shown to exhibit very similar patterns independent of the number of ball bearings involved, as can be observed at $t = 0.045ms$. The ball bearing caused disruption and turbulence, especially in the early stages of detonation. The global pressure response was, however, not greatly affected by the presence of the ball bearing, especially as the simulation time progressed further. This was to be expected, since the surface area occupied by the ball bearing was very small compared to the overall simulation area.

7.4.1.2 Velocity profiles

The resultant velocity profiles for each ball bearing in the axial plane, multiple ball bearing simulation are shown in Figure 7.34. None of the velocity profiles overlapped with one another, unlike in the radial plane simulations. As the number of ball bearings increased, the ball bearings had higher maximum and impact velocity, taking less time to reach and impinge the target. The resultant velocity profiles for the 3-ball bearing and 4-ball bearing tests exhibited very similar trends to the 1-, 2- and 5-ball bearings tests. For example, the initial rate of acceleration were identical for all velocity curves very early on. However, the velocity profiles diverged around $t = 0.005s$. Unlike in the radial plane simulation, the ball bearing velocities were not symmetrical along the middle of the rectangle representing the charge. The rate of deceleration post-impact were equal for all ball bearings in a given test. The ball bearings generally settled to a constant value almost immediately after impact, with the exception of ball bearing #3 in the 4-ball bearings test, which displayed another jump in velocity before settling to a constant value.

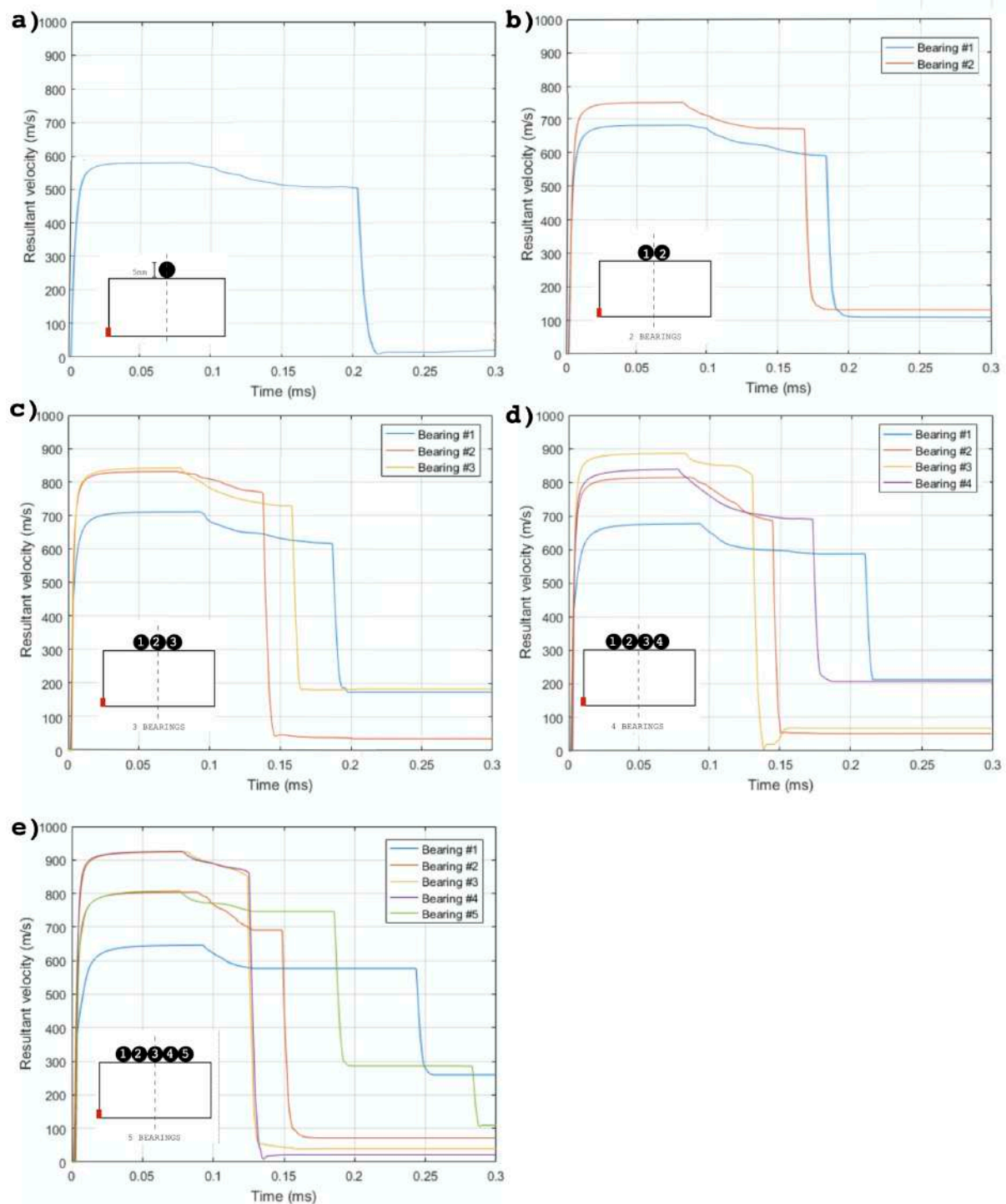


Figure 7.34: Series of velocity-time profiles for axial multiple-ball bearing simulations

As the number of ball bearings increased, the overall velocity gain of the ball bearings also increased. The ball bearings near the edge of the charge showed lowest velocity gain, and therefore took a longer period of time to impact the target. The order of ball bearing impact was however, not necessarily related to the magnitude of the ball bearing impact velocity. For example, ball bearing #4 from the 4-ball bearings simulation had a higher velocity gain than

beating #2 from the same test. Nevertheless, ball bearing #4 had impacted the target almost $0.025ms$ after ball bearing #2, indicating that the ball bearing velocity was not the sole factor in determining the ball bearing impact time. The projection angle and direction were additional determining factors in ball bearing interaction with the target. Projection angle and direction resulted in different length of travel to impact.

In the multiple ball bearing test series, multiple velocity values were obtained from each test. The maximum and the minimum value (for the peak and the impact velocity) observed in each test were recorded and plotted against the total number of ball bearings in in Figure 7.35 for the different tests. There was a clear difference between the highest and lowest velocities from 2-ball bearing test onwards. The data points corresponding to the highest peak and impact velocity showed an increasing trend. Therefore increasing the total number of ball bearings resulted in a higher "peak velocity" observed in the system as a whole. The velocity increased as the number of ball bearings increased, and was shown to peak at 5 ball bearings.

In all cases, the endmost ball bearings exhibited the lowest peak and impact velocities, and the lowest peak and impact velocities were shown to reach a maximum at 3-ball bearings test and decreased as the total number of ball bearings increased further. This trend was similar to the pattern observed in the single ball bearing, varied axial (x) position simulations. This may be due to the fact that the number of ball bearings increased, the "endmost" ball bearings were shifted closer to the charge ends. This observation confirmed that the axial position of the ball bearing has a predictable effect on the ball bearing velocity, independent of the total number of ball bearings involved.

In the current study, the ball bearings placed around the charge were comparable to discrete, "pre-formed" pieces. When the ball bearings were placed on a cylindrical body (in the axial plane), air gaps occurred between the ball bearings in the axial direction, allowing much of the energy from detonation to escape between the ball bearings. When the charge was less confined, less energy available to accelerate the ball bearings and the ball bearing velocities were lower. Therefore, the lowest velocity occurred in the single-ball bearing test, and the velocity was shown to increase as the number of ball bearings was increased.

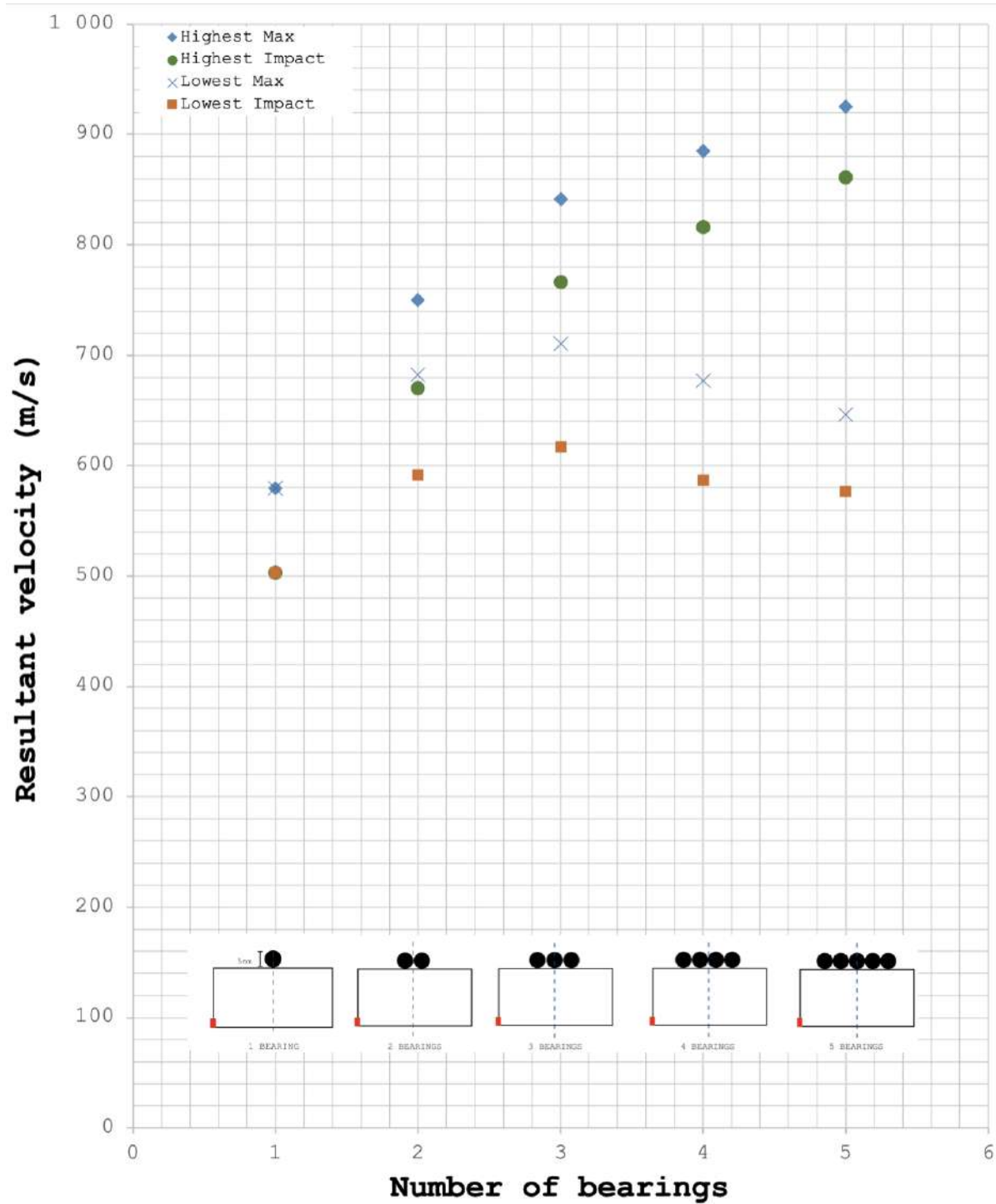


Figure 7.35: Graph of velocity vs. total number of ball bearings results for tests with multiple ball bearings outside the charge

7.5 Summary

7.5.1 Effect of Placement of Ball Bearings

Placement outside the Explosive Charge

In the radial plane configurations, when the ball bearing was placed adjacent to either the x - or the y -axis, the two configurations mirrored each other and had virtually identical maximum and impact velocities. Under the 45° -configuration, the ball bearing had a lower maximum and impact velocity compared to the other two configurations and took a longer time to reach and impinge the target. Under both the 8° -configuration and the 82° -configuration, the ball bearing had deviated from their initial position by 22° by the time of impact. Under the 45° -configuration the ball bearing deviated by 0.2° from its initial position by the time of impact. In the axial plane configurations, the ball bearing maximum and impact velocities increased as the ball bearing moved further away from the detonation end, and reached its maximum when the ball bearing was placed in the middle of the rectangle representing the charge. In all configurations, the impact velocity was lower than the maximum velocity by a constant amount. The ball bearings exhibited lower velocity towards the charge end, similar to the "end effect" of casing fragments from encased charges. The ball bearing projection direction gradually shifted from -23.8° to 24° as the ball bearing position shifted from the left to the right.

The recorded impact velocities were higher in the radial plane simulations than in the axial plane simulations. However, the crater depths were significantly higher in the axial plane simulations (up to $0.6mm$ higher) despite the difference in impact velocity, most likely due to the geometry of the target. The maximum crater depth corresponded to the configuration with the highest impact velocity in both the radial and axial planes.

Placement inside the Explosive Charge

In both the radial and axial plane configurations, the velocity increased as the charge position approached the surface of the cylindrical charge, and was shown to peak at $y = h = 10.5mm$, i.e. when the ball bearing was located just under the surface of the explosive charge. When the ball bearing was positioned on the charge surface, its velocity decreased. In the radial plane simulations, the projection angle of the ball bearing remained nearly constant at 45° independent of the radial position of the ball bearing. In the axial plane simulations, however, the ball bearing projection direction did not remain constant but gradually shifted from -43.6° to 2.9° as the ball bearing y -distance increased.

When the ball bearing was buried inside the explosive charge, i.e. $y = h \leq 10.5mm$, the recorded impact velocities and crater depths were similar in both the radial plane simulations and the axial plane simulations. The maximum crater depth corresponded to the configuration with the highest impact velocity in both the radial and axial planes.

7.5.2 Effect of the Size of the Ball Bearing

The ball bearing velocities decreased as the ball bearing diameter increased, in both the radial and axial plane configurations. In the radial plane simulations, the crater depth also decreased with an increase in ball bearing diameter. However, the crater depth increased with an increase in ball bearing diameter in the axial plane simulations. This implied that the target geometry affects the crater depth. In the radial plane, the projection angle of the ball bearing at impact remained nearly constant at 45° . The projection angles at impact hardly deviated from their initial projection angle of 0° , but did not show a clear trend of increase or decrease as the ball bearing diameter increased in the axial plane simulations.

7.5.3 Effect of the Aspect Ratio of Explosive Charge

The ball bearing velocities increased as the explosive charge length-to-diameter aspect ratio (L/D) increased. The crater depth and the projection angle at impact also increased with an increasing L/D ratio.

7.5.4 Effect of the Total Number of Ball Bearings

The overall maximum and impact velocities of the ball bearings increased as the number of ball bearings increased, in both the radial and axial plane configurations. More "free space" and consequently more energy loss occurred when there were fewer ball bearings. It was also noted that the pressure wave propagation slowed down as the number of ball bearings increased.

In the radial plane simulations, the ball bearings exhibited symmetrical behaviour about the 45° line (i.e. $y = x$ line). However, in the axial plane configurations the simulations did not display any symmetrical behaviour about the charge mid-plane due to the direction of detonation. In the axial plane simulations, the ball bearing velocities increased as the ball bearing moved further away from the detonation end until it peaked in the mid-region of the charge cylinder, and subsequently decreased again. The endmost ball bearings exhibited the lowest peak and impact velocities.

Chapter 8

Discussion

8.1 Crater Depth

8.1.1 Ball Bearing Velocity vs Crater Depth

In this section, the crater depths recorded in the simulations were plotted against their respective impact velocities and the Fish-Summers equation [74], the JSC equation [75] and the semi-infinite model equation [75] (henceforth collectively referred to as the NASA equations and listed for ease of reference) were also plotted on the same set of axes for comparison.

Fish-Summers Equation

$$t_t = K_1 m^{0.352} V_n^{0.875} \rho^{\frac{1}{6}} \quad (\text{Equation 2.20})$$

JSC Equation

$$p = 5.24 d^{\frac{19}{18}} B H^{-0.25} \left(\frac{\rho_p}{\rho_t} \right)^{0.5} \left(\frac{V_n}{C_s} \right)^{\frac{2}{3}} \quad (\text{Equation 2.21})$$

Semi-Infinite Equation

$$P_\infty = K_\infty m^{0.352} \rho_m^{\frac{1}{6}} V_n^{\frac{2}{3}} \quad (\text{Equation 2.22})$$

Radial Plane Simulation Data

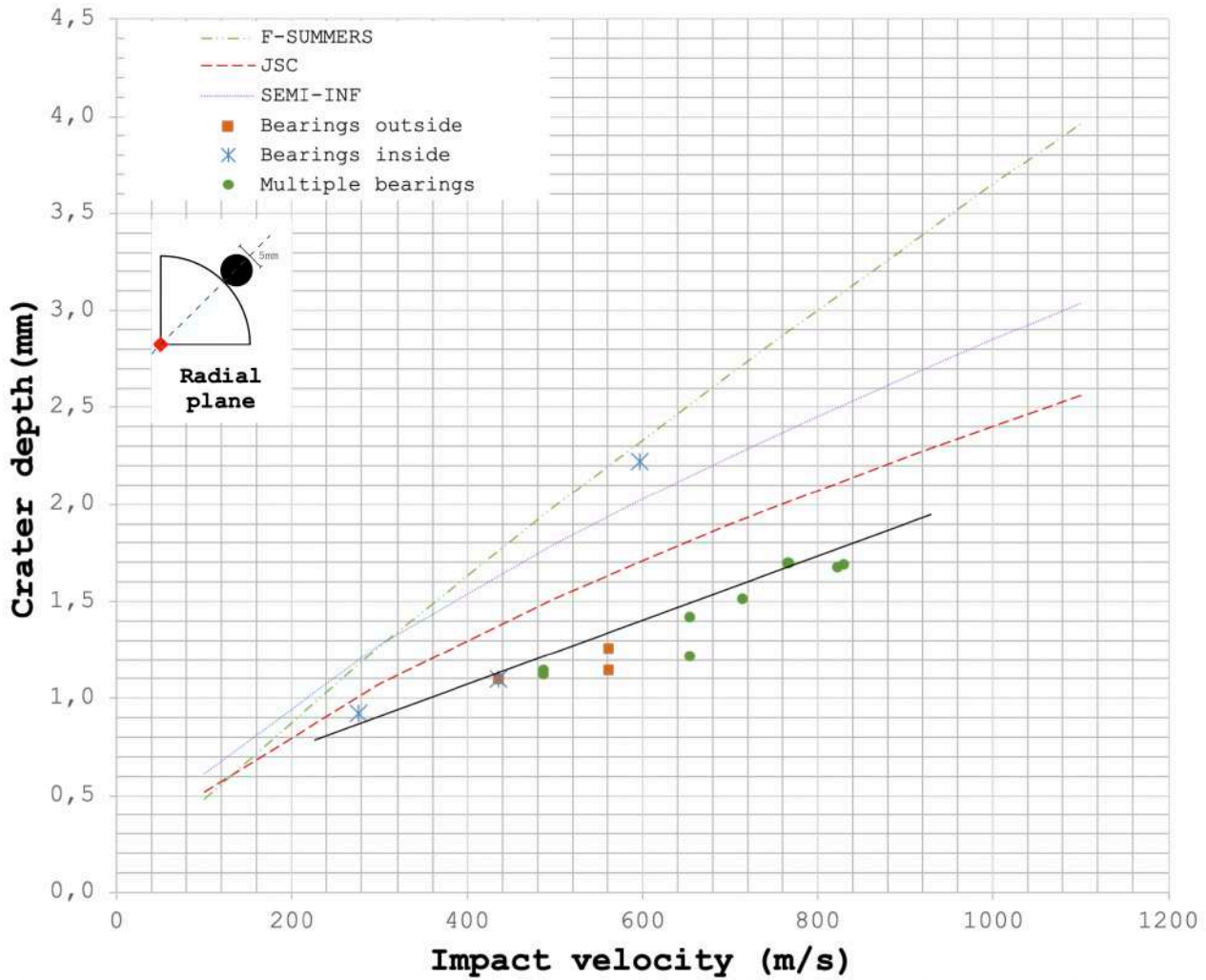


Figure 8.1: Crater depth vs. Impact velocity for all data points in the radial plane simulations, plotted on the same set of axes as the NASA equations

Crater depths recorded in the radial simulations are plotted against their respective impact velocities as shown in Figure 8.1. The results exhibited an increase in crater depth with increasing velocity. The gradient for the radial-plane model curve was similar to that of the JSC equation curve. However, the radial-plane data was shown to under-predict the crater depths for the given velocity when compared to the equations. This may be due to the fact that the cylinder surface was curved. Figure 8.2 shows a schematic 2D diagram of a circular (i.e. spherical in 3D) projectile impacting a curved surface (i.e. radial "slice" of a 3D cylinder). The projectile was assumed to impinge the surface with a velocity vector that coincided with the vertical axis (the vectors are shown on the diagram as red arrows). The size of the projectile in the diagram was exaggerated for demonstrative purposes. Although the vector that coincided with the centre of the projectile was "normal" to the contact surface, the other velocity vectors were at non-90° angles to the target surface. This was analogous to oblique (non-ideal) impact. Goldsmith [69]

reported that the velocity required to penetrate or perforate a target of given thickness was higher when the projectile impact was at an angle. Therefore, in the oblique impact case, higher velocity was required to achieve the same crater depth as normal impact; the reverse was also true, and oblique impact would produce a shallower crater at a given velocity compared to normal impact.

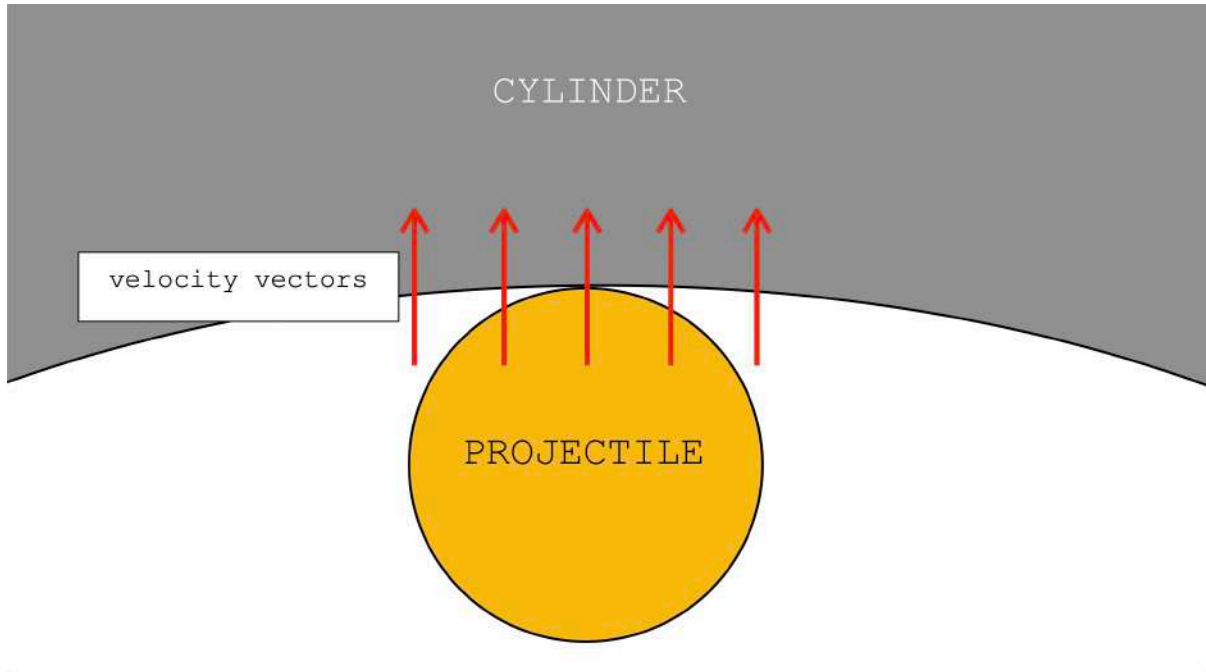


Figure 8.2: Schematic of a circular projectile impacting a curved surface (The size of the projectile in the diagram is exaggerated for demonstrative purposes.)

Axial Plane Simulation Data

Crater depths recorded in the axial simulations are plotted against their respective impact velocities and shown in Figures 8.3 and 8.4. The data points from single-ball bearing axial plane simulations, as well as the data points from ball bearings with a projection angle (θ) of less than 15° in multiple-ball bearing axial plane simulations are shown in Figure 8.3. The data points reproduced a similar trend to the JSC equation and the semi-infinite equation curves. Despite a few outliers, most of the data points fell within the range of the analytical solutions. It should be noted that the NASA equations were designed for spherical projectile impact on plates. This trend was in agreement with the initial geometrical assumption made in the axial plane simulations, where the target was essentially treated as a plate.

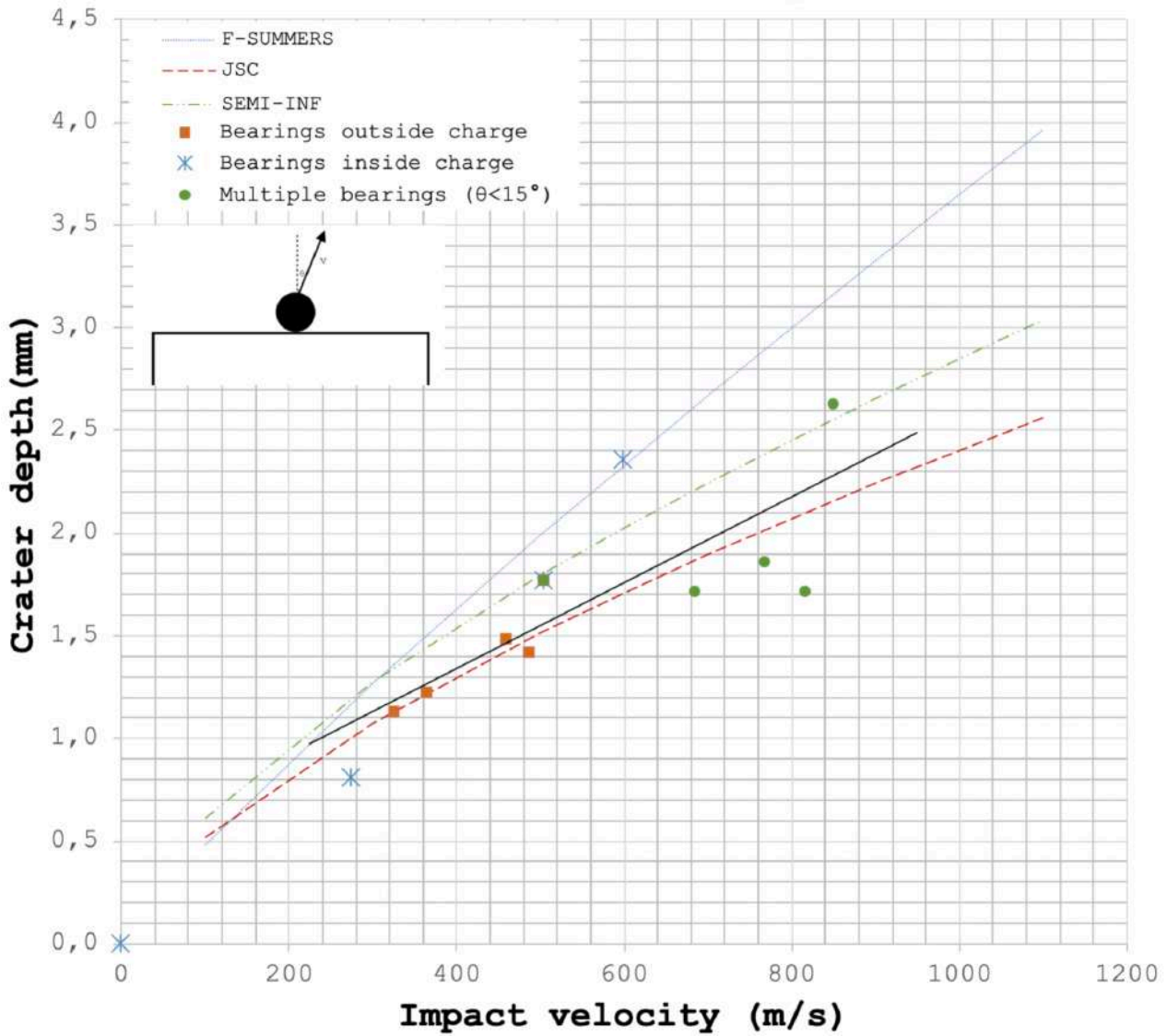


Figure 8.3: Crater depth vs. Impact velocity of all data from ball bearings with a projection angle (θ) of less than 15° in the axial plane simulations, plotted on the same set of axes as the NASA equations

Figure 8.4 shows only the data points corresponding to the multiple-ball bearings test condition in the axial plane. The results for the multiple-ball bearings tests produced a steeper gradient than the three NASA equations showing that the simulations under-predicted the crater depths at lower velocities, while over-predicting the crater depths at higher velocities compared to the NASA equations. It must be noted that the NASA equations assumed: 1) an ideal collision condition i.e. normal impact of a purely translating projectile against a stationary object [69], and 2) non-deformable projectiles [73]. Neither of these assumptions were valid in the current study, as the ball bearings exhibited significant deformation during blast, and the projection angle θ , measured from the axis normal to the surface, reached up to 40 degrees in the most severe case. The results confirmed that the NASA equations could not accurately predict the crater depths under the non-ideal conditions. Various studies on target damage from highly oblique hypervelocity impacts of steel spheres, such as those described by Finnigan [70] and

Orphal et al. [89] have shown that equations designed for ideal impact could not be used when the striker impinged the target at angularity.

The degree of scatter observed in the radial plane simulation data was not as severe as the axial case. This may be due to the fact that the variation in projection angle of the ball bearings was much smaller. While the projection angles in multiple-ball bearings axial plane tests ranged between -40° and 40° , the variation was less in the radial case, not exceeding $\pm 10^\circ$ in the severest of cases.

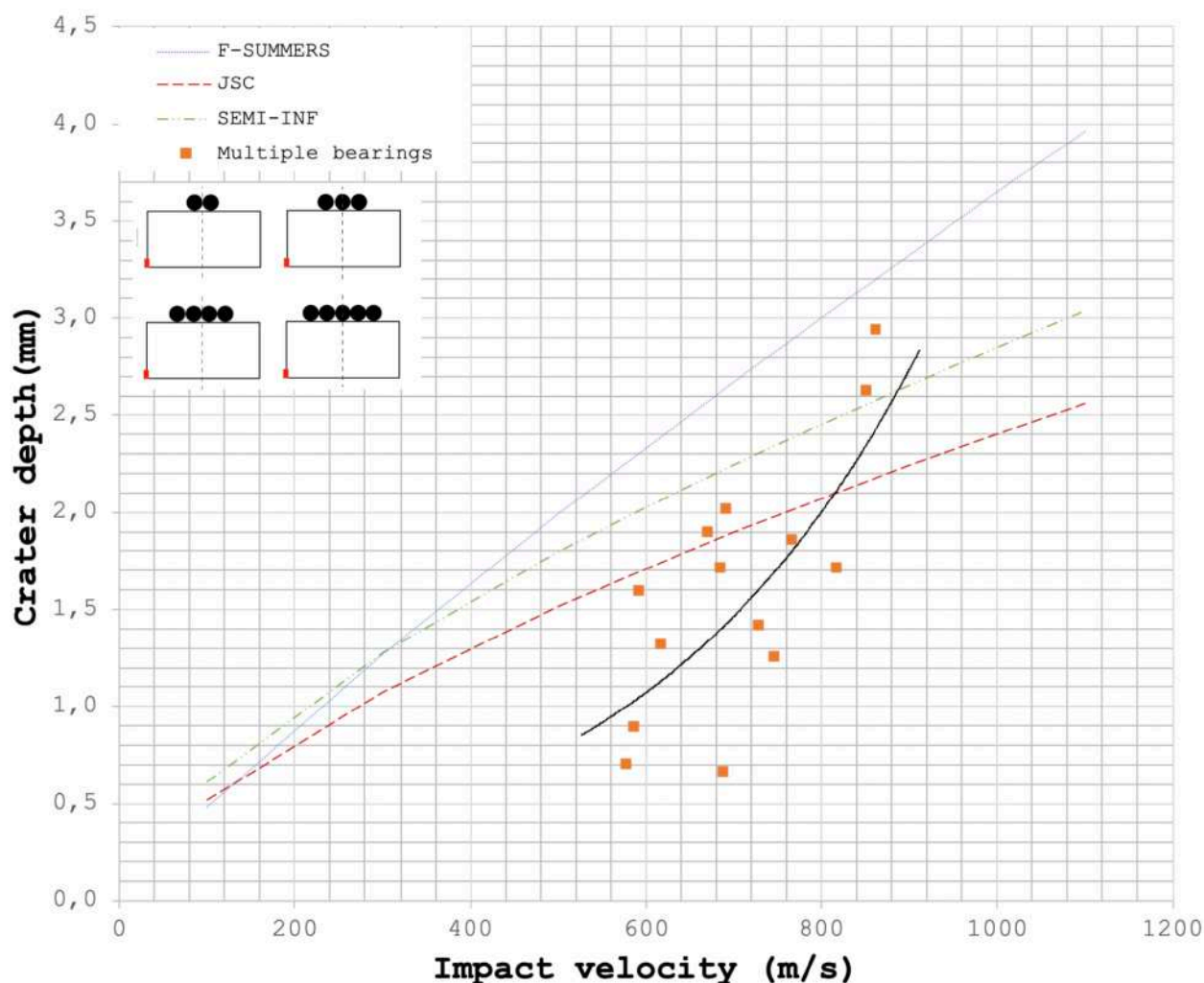


Figure 8.4: Crater depth vs. Impact velocity of all data from the axial plane, multiple-ball bearings simulations plotted on the same set of axes as the NASA equations

8.2 The Distribution of Ball Bearing Velocity in the Axial Direction

Due to the geometry of the setup, the axial distribution of velocity can only be analysed in the axial plane simulations. The effect of the axial position of the ball bearing on the charge can be investigated by 1) studying a single ball bearing placed at varying positions, and 2) by studying the behaviour of each ball bearing in cases when multiple ball bearings are placed on the charge simultaneously.

8.2.1 Comparison to Modified Gurney Formula

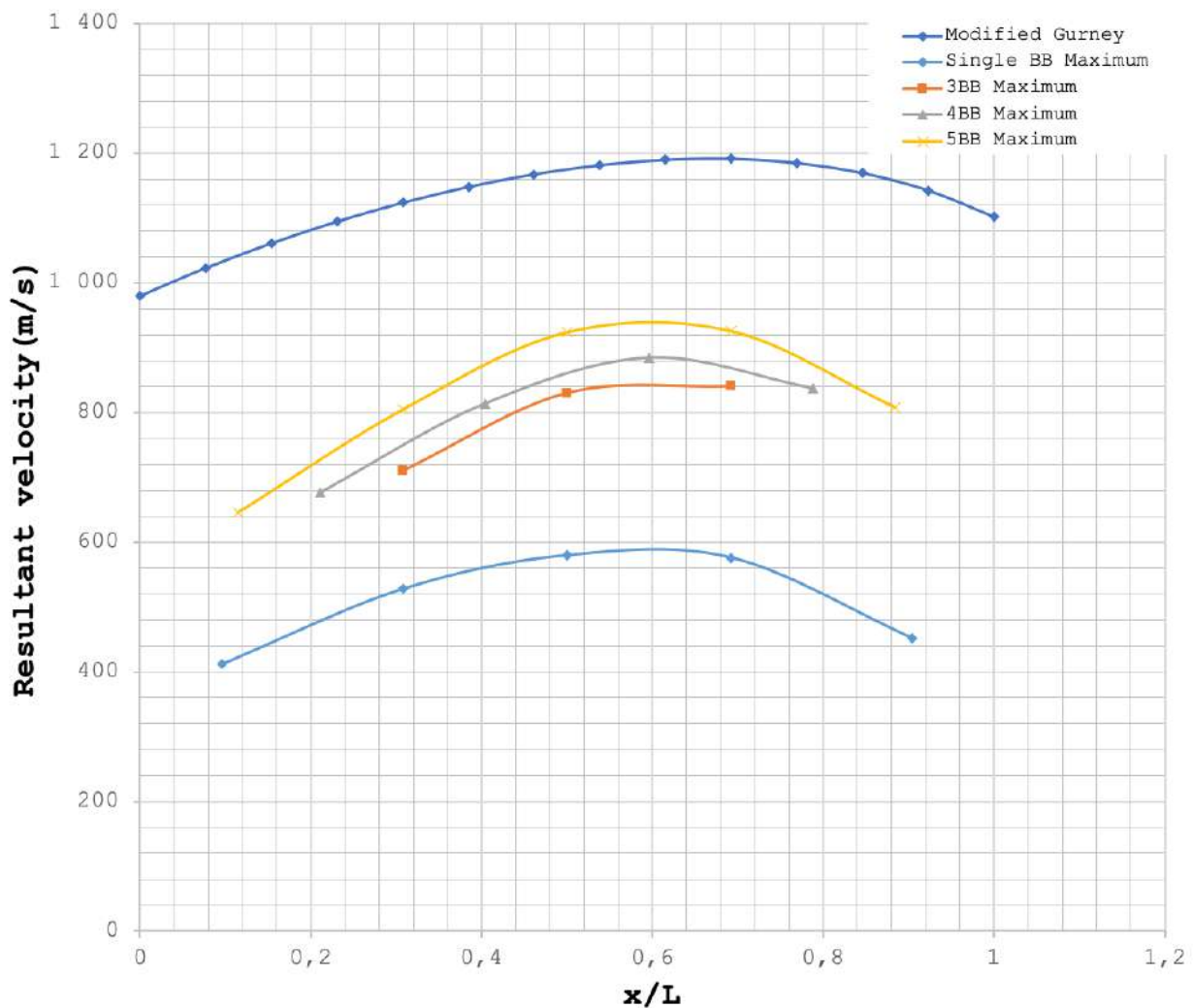


Figure 8.5: Plot of velocity vs. relative initial position of single and multiple-ball bearings axial plane simulations and modified Gurney formula by Huang et. al [51]

To compare the results of single- and multiple-ball bearing series with the analytical solution, the absolute and normalised velocity distributions of the test results were plotted against the relative initial ball bearing position (x/L) as shown in Figure 8.5. The modified Gurney formula (Equation 2.11) by Huang et al. [51] was plotted on the same set of axis. Among the possible values of casing mass M (corresponding to 3-, 4- or 5-rows of ball bearings), the casing mass that produced the lower bound of Gurney velocity predictions i.e. M equivalent to the mass of 5 rows of ball bearings, was used, as all velocity results from the simulations were lower than the theoretical Gurney velocity.

$$V_0 = (1 - 0.361e^{-1.111\frac{x}{d}})(1 - 0.192e^{-3.03\frac{(L-x)}{d}})\sqrt{2E}\sqrt{\frac{C/M}{1 + 0.5C/M}} \quad (\text{Equation 2.11})$$

It must be noted that due to the difference in the degree of charge confinement, the numerical models cannot be directly compared to the model used by Huang et al. [51]. However, the results showed that the behaviour of cylindrical charges partially confined with ball bearings was similar to that of encased charges to a degree. Since no analytical models are readily available to describe "partial charge confinement", the Gurney equation and modified versions thereof were used as a benchmark.

It was observed that the simulated velocity was significantly less than the velocity predicted by the modified Gurney formula. Since the modified Gurney formula was intended for predicting the fragment velocity of encased charges. The charges tested in the current study were partially confined with ball bearings. Because ball bearings incorporated into charge are analogous to "pre-formed fragments", the energy transferred from the detonation products to the ball bearings would be substantially reduced when compared to a continuous casing. For example, Pearson [55] showed that the velocity reduction for pre-fragmented case components compared to fragments from solid cylinders approached 20 %. More "free space" and consequently more energy loss occurred when there were fewer rows of ball bearings (i.e. the energy loss was inversely proportional to the degree of confinement). Because of this phenomenon, the lowest velocity occurred in the single-ball bearing test, and the velocity was shown to increase as the number of ball bearings was increased, as evident in Figure 8.5.

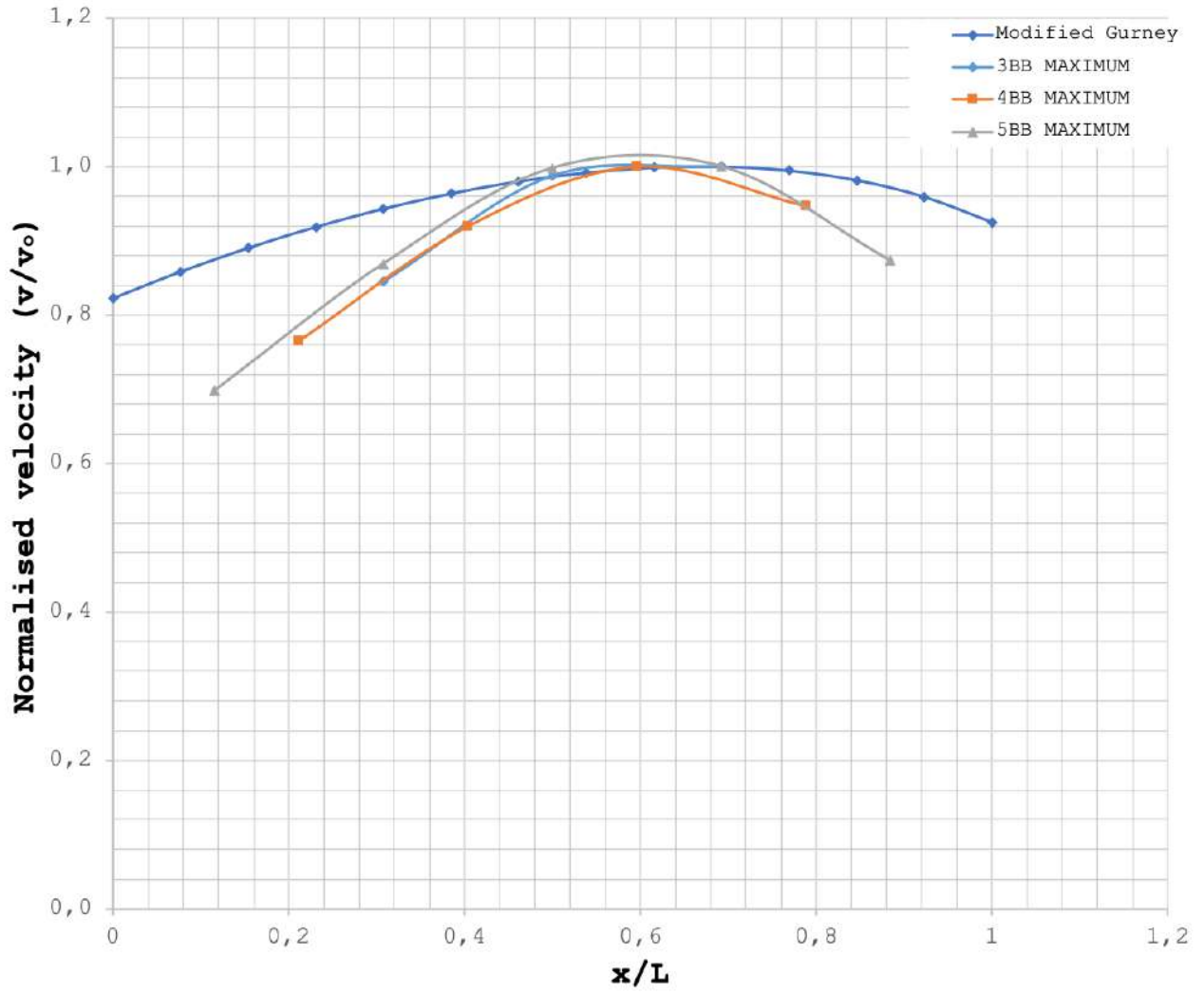


Figure 8.6: Normalised maximum velocity vs. relative initial position of multiple-ball bearings axial plane simulations and modified Gurney formula by Huang et al [51]

Figures 8.6 and 8.7 show the normalised velocities from multiple-ball bearings axial plane simulations, where the velocities of each simulation have been normalised with respect to their corresponding maximum value i.e. V_0/V_{max} . The figure shows that the normalised velocity curves for the multiple-ball bearing tests all coincided into an inverse quadratic function which exhibited a steeper curve than the normalised version of modified Gurney formula by Huang et al. (Equation 2.11) [51]. This implied that the ball bearing velocity loss towards the charge cylinder ends was more severe with multiple ball bearings than in the Gurney prediction model. Even though each of the multiple-ball bearings tests had distinctively different velocity values, the normalised function exhibited very similar shapes. The similarity in shape of the curves indicated that values of V_0/V_{max} were relatively independent of the total number of ball bearings in the charge, and may be expressed as a function of the ratio x/L .

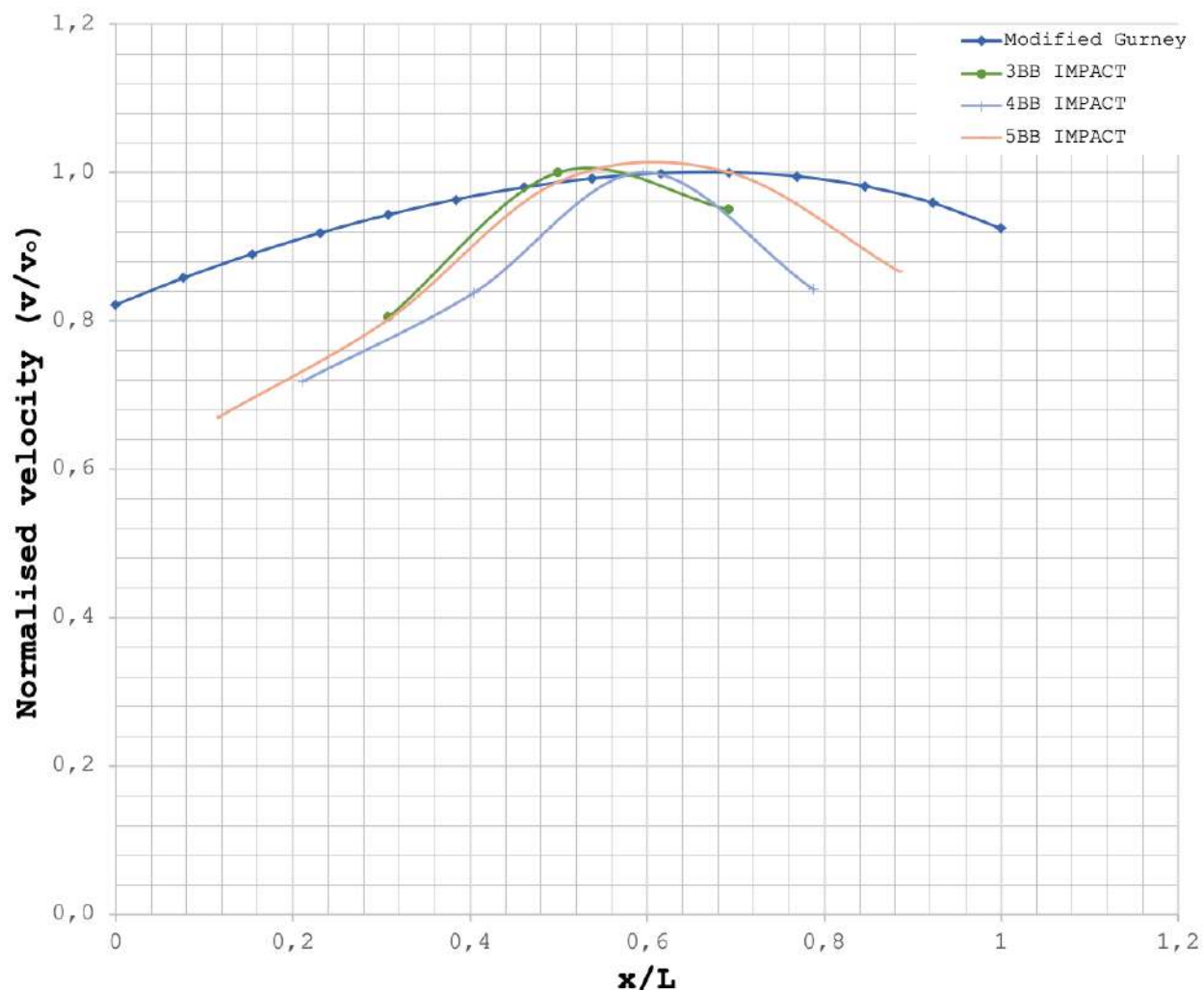


Figure 8.7: Normalised impact velocity vs. relative initial position of multiple-ball bearings axial plane simulations and modified Gurney formula by Huang et. al [51]

The normalised impact velocity curves are plotted in Figure 8.7. Although the gradients towards the curve ends showed similarities, the curve maxima did not occur at the same x/L values and the width of the curves also showed variation. The difference between Figures 8.6 and 8.7 implied that the ball bearing motion and velocity became less predictable during the window between its initial peak and its impact, i.e. when the ball bearings interacted with the reflected blast waves.

8.3 Projection Angle of Ball Bearings in the Axial Plane

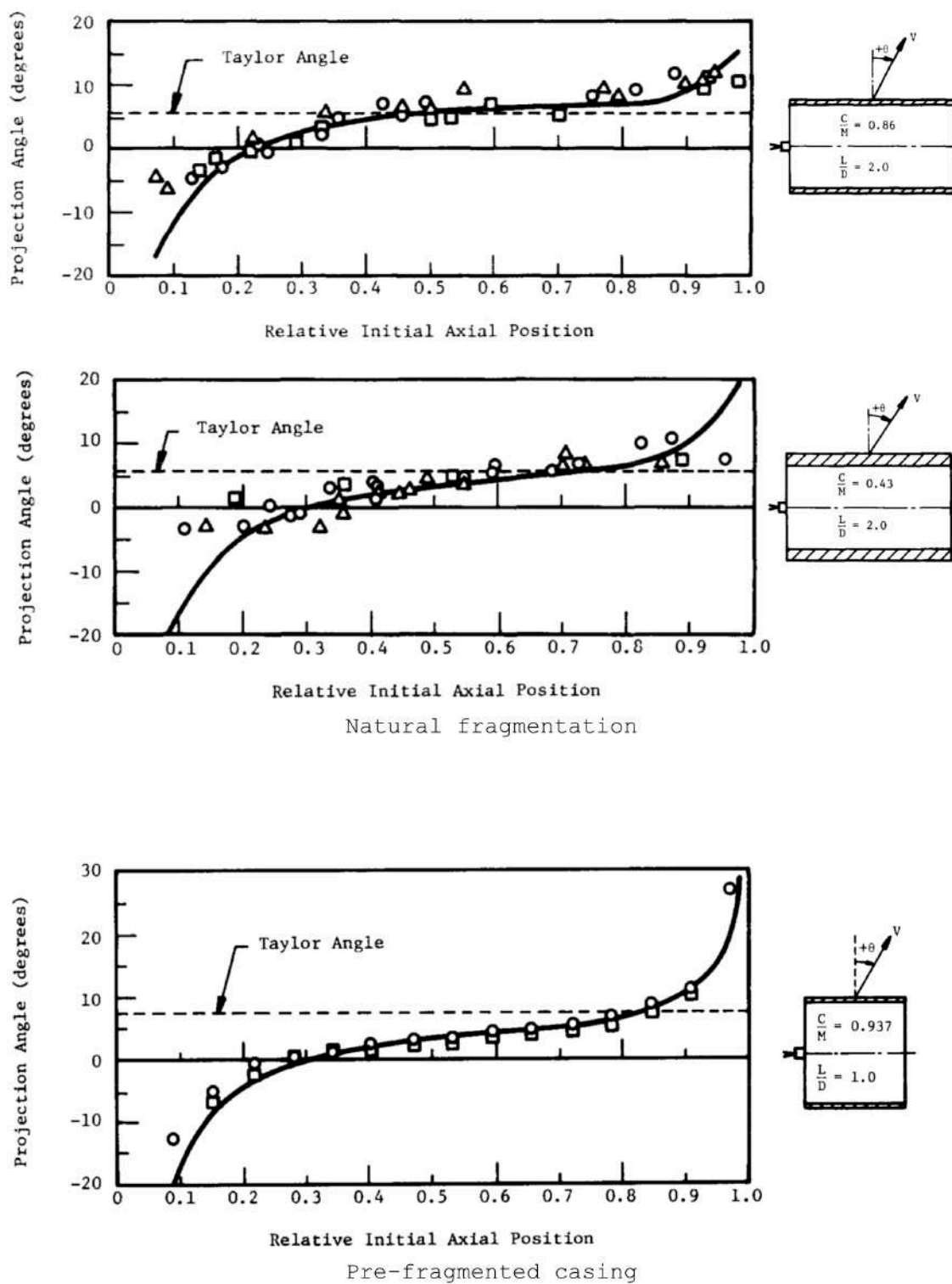


Figure 8.8: Fragment projection angle vs. relative initial axial position curves for encased charges of various L/D and C/M factors, from Anderson's test data [48]

The direction of motion of each ball bearing was captured by the means of their projection angle (θ), defined as the angle between the velocity vector and the y -axis (positive in the clockwise direction). Anderson [48] showed that the projection angle versus relative initial axial position (x/L) curves shared the same general shape, as shown in Figure 8.8. The gradients towards the charge ends varied depending on the aspect ratio (L/D) and charge-to-casing-mass ratio (C/M) values. For comparison, the projection angle curve for fragments from a cylindrical charge of $L/D = 1$ from the experiments by Anderson [48] was also included in Figure 8.9, with a set of plots of projection angle versus relative initial position of single and multiple-ball bearings from axial plane simulations. This particular curve was chosen as the test had the same L/D ratio as the charges tested in the current study.

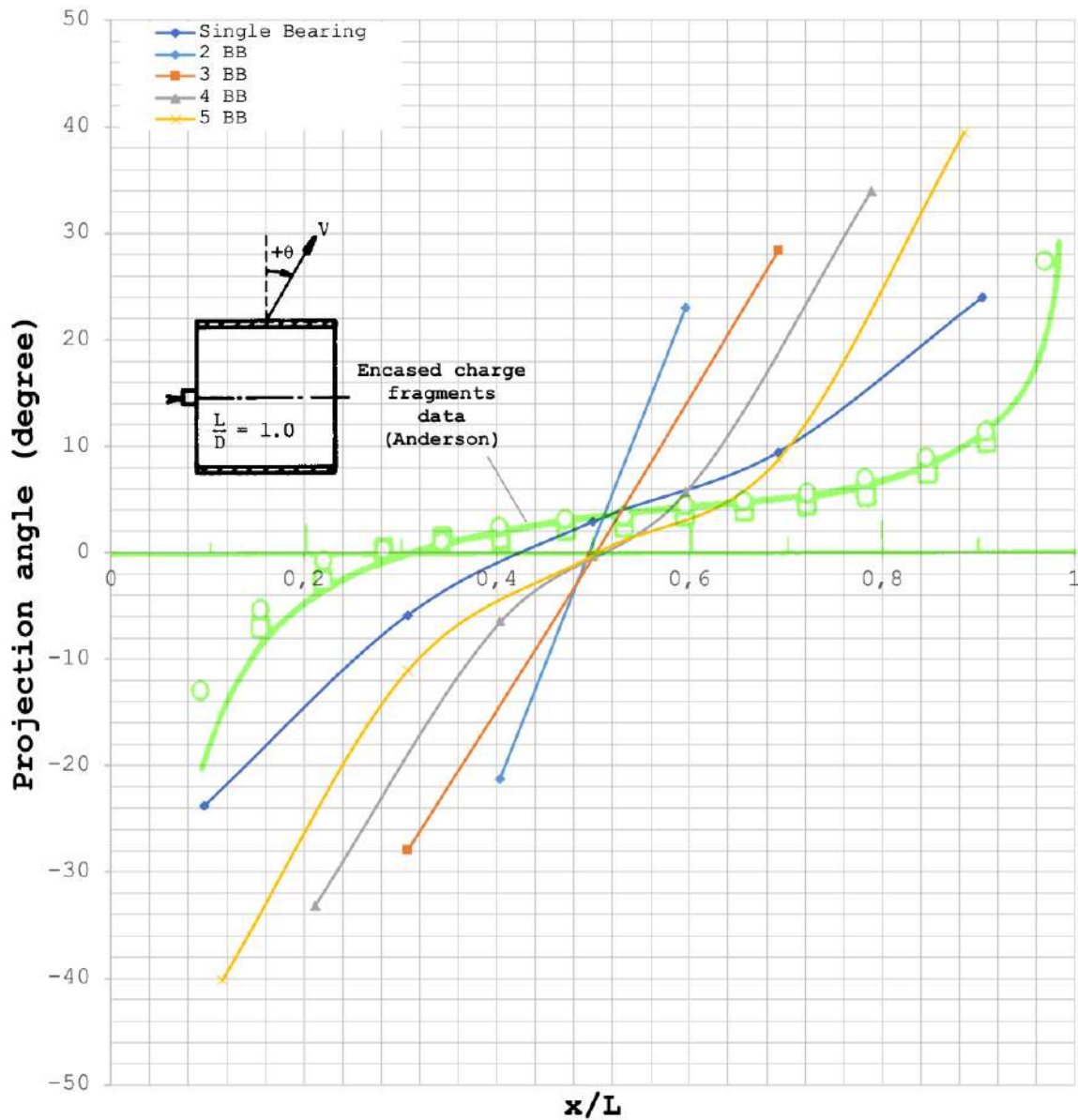


Figure 8.9: Projection angle vs. relative initial position of multiple-ball bearings axial plane simulations

The plot labelled as "Single bearing" in Figure 8.9 showed the projection angles of a single ball bearing at different axial (x) position on charge. The plot had a different shape and gradient to the results from the multiple-ball bearings test series. This implied that the axial position of the ball bearing was not the sole factor affecting the projection angle, and the total number of rows of ball bearings also had an influence on the projection angle.

The projection angle curves corresponding to the multiple-ball bearings series were shown to overlap and "pivot" around the point $x/L = 0.5$. As the number of ball bearing increased i.e. as the charge confinement approached complete encasement, the data was shown to approach the shape of the projection angle data for encased charge fragments. The gradient around the mid-section of the curve approached a gradient of approximately zero, although the point of inflection remained fixed at $x/L = 0.5$. It was also noted that the multiple-ball bearings simulation curves were positioned lower than the experimental curve.

8.4 Comparison between the Radial and Axial Plane Simulations

Contrary to the axial velocity distribution, the radial distribution of velocity can be analysed from both the radial and axial plane simulation sets. In this section, y/R refers to the relative initial radial position of the ball bearing in the axial plane. In the radial plane, the relative initial radial position of the ball bearing is also referred to as h/R . Note that in a 3D cylinder, y is equal to h (refer to the charge diagrams in Figure 8.10).

8.4.1 The Distribution of Ball Bearing Velocity in the Radial Direction

8.4.1.1 Ball Bearing Velocity in the Radial and Axial Planes

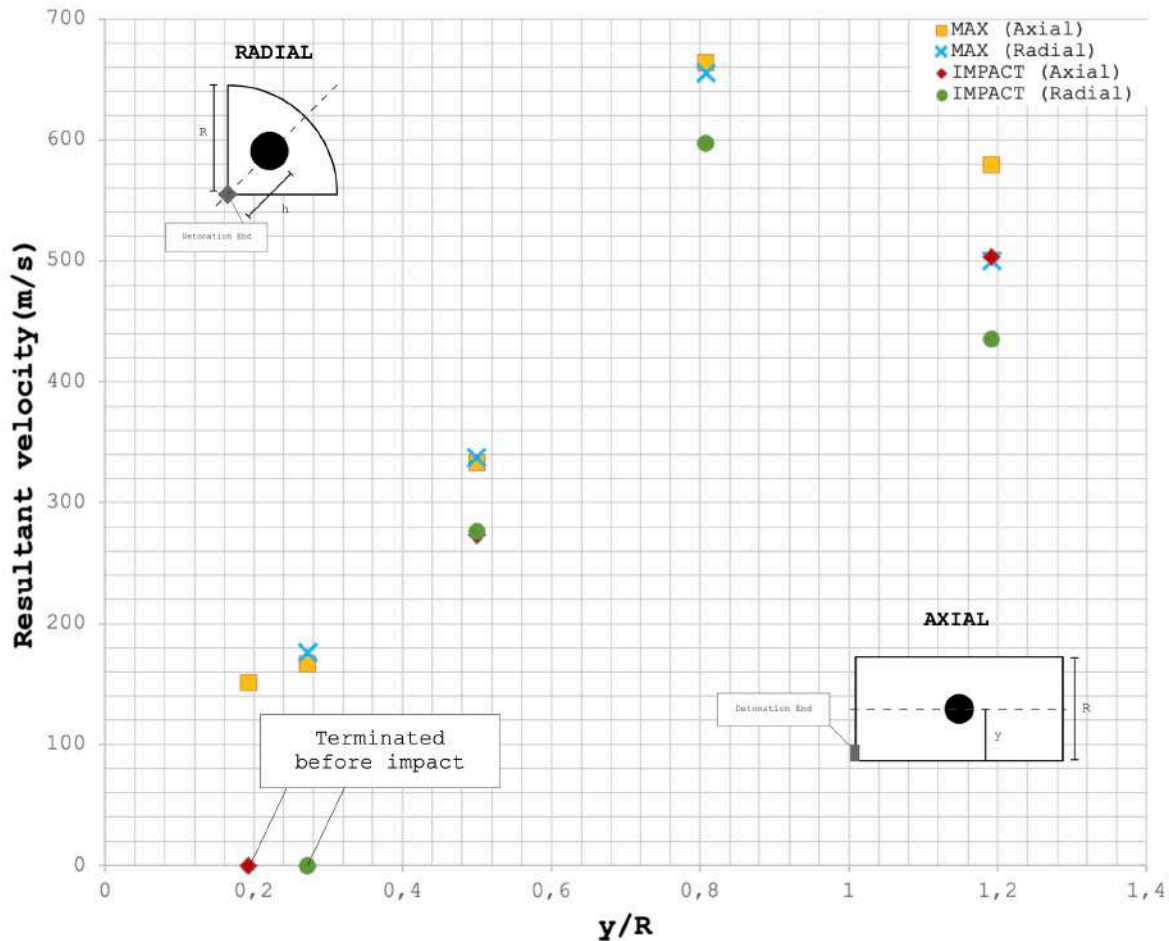


Figure 8.10: Graph of velocity vs. total number of ball bearings results for tests with single ball bearing inside charge at various y (i.e. h) positions

The plot of resultant velocity versus relative initial radial position in both the axial and radial plane is shown in Figure 8.10. The predicted velocities in the radial and the axial plane simu-

lations were approximately equal for $x/L < 1$, i.e. when the ball bearing was buried inside the charge. However, the axial plane simulation exhibited a higher peak and impact velocity than the radial plane simulation for $x/L > 1$ (i.e. ball bearing positioned outside the charge). The general shape of the curves were the same for both the radial and the axial plane results. The velocity increased as the ball bearing position approached the surface of the cylindrical charge, and was shown to peak at $y = h = 10.5\text{mm}$. When the ball bearing was positioned outside the charge, the ball bearing velocity decreased.

The increase in the ball bearing velocity with increasing y/R may be explained by considering the shape of the shock wavefront, as described by Logan [18]. The r and z components of the velocity vector at a point on the shock front at angle θ and radius of curvature σ shown in Figure 8.11 are defined as $\omega(\theta)$ and $u(\theta)$. Approximating the ball bearing from the current study as a discrete "fragment", velocity vector from Figure 8.11 can be drawn to coincide with the ball bearing centre to show the incidence direction of detonation wave. The incidence directions of detonation wave for ball bearings positioned at various y positions in the axial plane are summarised in Figure 8.12.

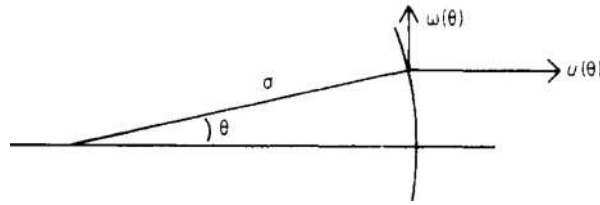


Figure 8.11: Geometry of the shock front depicted by Logan [18]

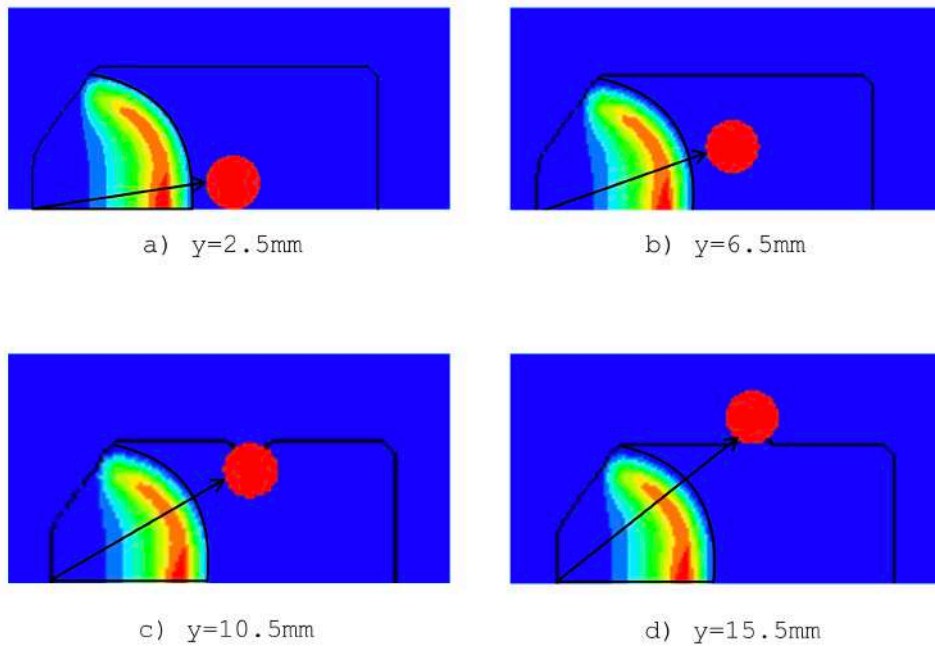


Figure 8.12: Pressure contour plots showing the incidence direction of detonation wave for ball bearings positioned at various y positions in the axial plane

From Figure 8.12 it is evident that the angle of incidence was solely dependent on the y -position of the ball bearing, as the x -position of the ball bearing was kept constant. As the ball bearing was placed closer to the charge surface, the incidence angle also increased. Therefore, the numerical simulation results showed that the ball bearing velocity increased with the incident angle of the detonation wave.

Li et al. [53] showed that the Gurney equation modification factor proposed by Huang et al. [51], $f(x)$, is comprised of a rarefaction factor, $h(x)$, and an incidence angle term, $g(\alpha)$, where α is the wave incidence angle at a given point in the casing. In other words, $f(x) = h(x).g(\alpha)$. It was shown that $g(\alpha)$ increased as the incidence angle α increased. Given that the rarefaction term $h(x)$ remains constant, an increase in $g(\alpha)$ leads to an increase in the overall fragment velocity.

The numerical results presented in Figure 8.12 showed that when the ball bearing was buried inside the explosive charge, the velocity of the ball bearing increased as the incidence angle increased. However, the ball bearing velocity decreased when it was positioned outside the charge despite the increase in y -position and in the incidence angle. This showed that the wave incidence angle was not the sole factor that had an influence on the ball bearing velocity. According to the model presented by Li et al. [53], the velocity of a fragment (and therefore the ball bearing) was influenced by both the incidence angle and the rarefaction wave effects. Therefore, the change in ball bearing position from "buried" to "exposed", may lead to a decrease in the rarefaction term, $h(x)$, leading to an overall decrease in velocity despite the increase in incidence angle.

8.4.1.2 Multiple Ball Bearings at Different Radial Positions

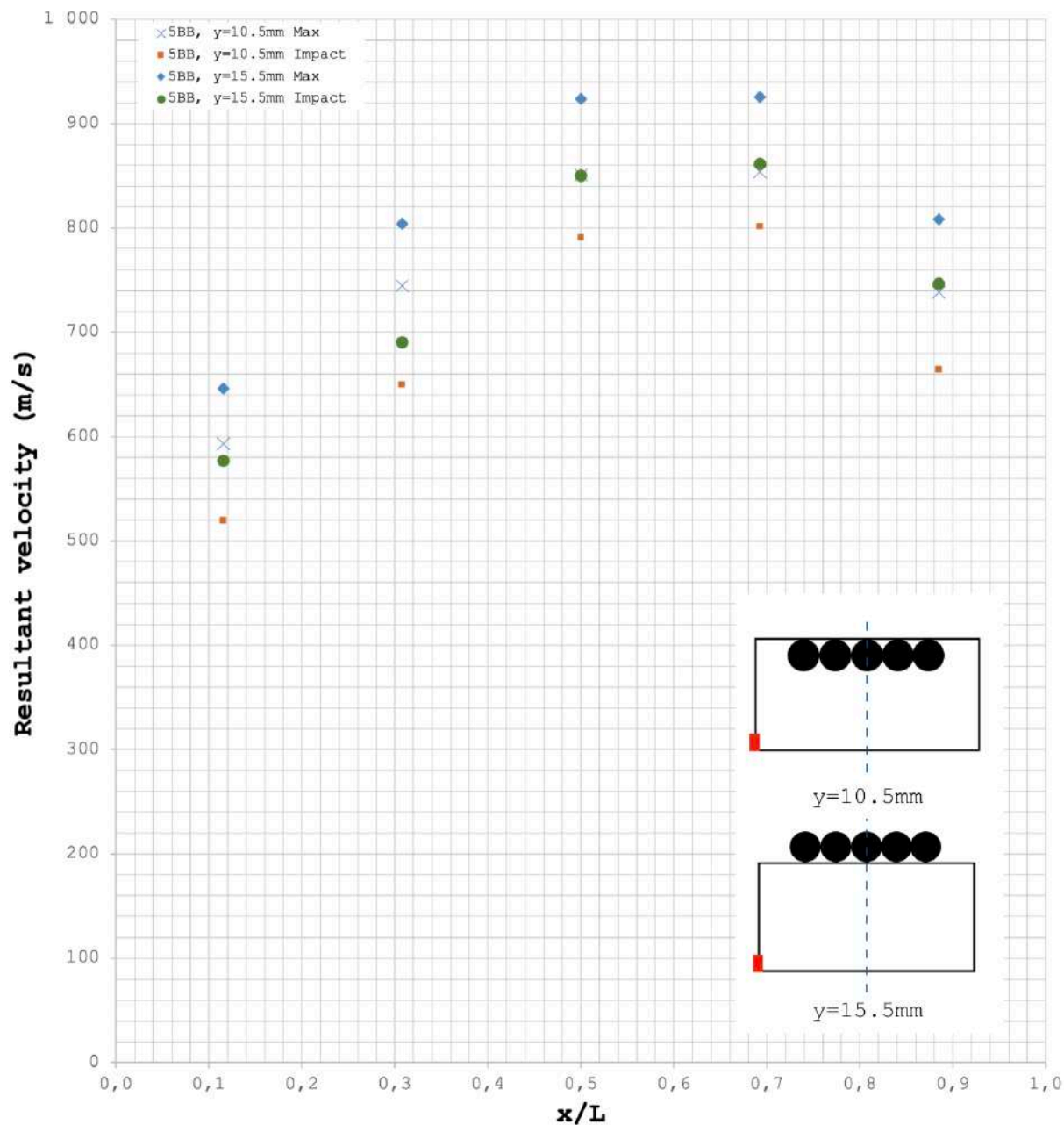


Figure 8.13: Plot of velocity vs. relative initial position of each ball bearing in axial plane tests with 5 ball bearings at positions $y = 10.5mm$ and $y = 15.5mm$

To further investigate the combined effect of varying the radial position of the ball bearings when there are multiple ball bearings involved, another simulation was conducted with 5 ball bearings at position $y = 10.5mm$ (ball bearings located just under the surface of the explosive charge). The ball bearing resultant velocities were compared with the results from the simulation of 5 ball bearings located at $y = 15.5mm$ (ball bearings placed on the surface of the explosive), as shown in Figure 8.13. Although the ball bearing velocity was shown to peak at radial position $y = 10.5mm$ in the single ball bearing test set, when the total number of ball bearings was

increased to 5 ball bearings, the ball bearing velocities were higher at $y = 15.5mm$ than at $y = 10.5mm$. This further confirmed that the total number of ball bearings involved has a significant influence on ball bearing behaviour, even when combined with other factors such as change in radial position. Increasing the total number of ball bearings incorporated in the charge had a greater effect on the rarefaction term, $h(x)$, when compared to the single-ball bearing scenario.

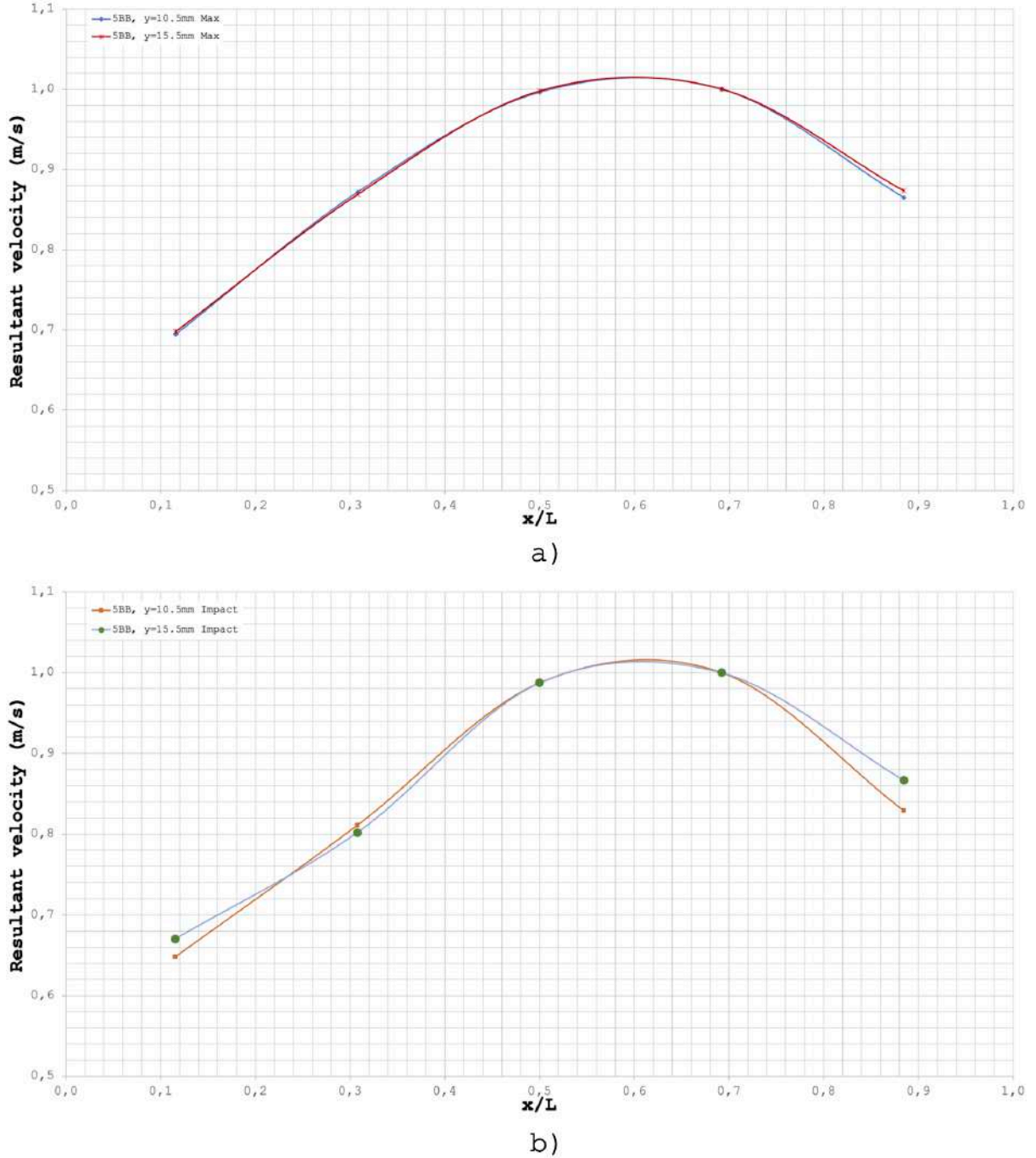


Figure 8.14: Normalised a) ball bearing maximum velocity vs. relative initial position and b) ball bearing impact velocity vs. relative initial position, in axial plane tests with 5 ball bearings at positions $y = 10.5mm$ and $y = 15.5mm$

No significant difference was, however, observed in the "shape profile" of the relative velocity distribution. Figure 8.14 a) shows that the shape profiles for the maximum ball bearing velocities of the two cases overlapped almost completely. This implies that the velocity vs. relative initial position plot of the $y = 10.5mm$ test was essentially identical in shape to the plot corresponding to the $y = 15.5mm$ test. Altering the radial position of the set of ball bearings resulted in a uniform, global decrease in ball bearing velocity, and all the ball bearings involved in the test experienced the same amount of decrease. A similar trend was observed in the shape profile for the impact ball bearing velocity, as shown in Figure 8.14 b). The shape profiles diverged slightly towards either end of the charge.

8.4.2 The Effect of Varying Ball Bearing Diameter

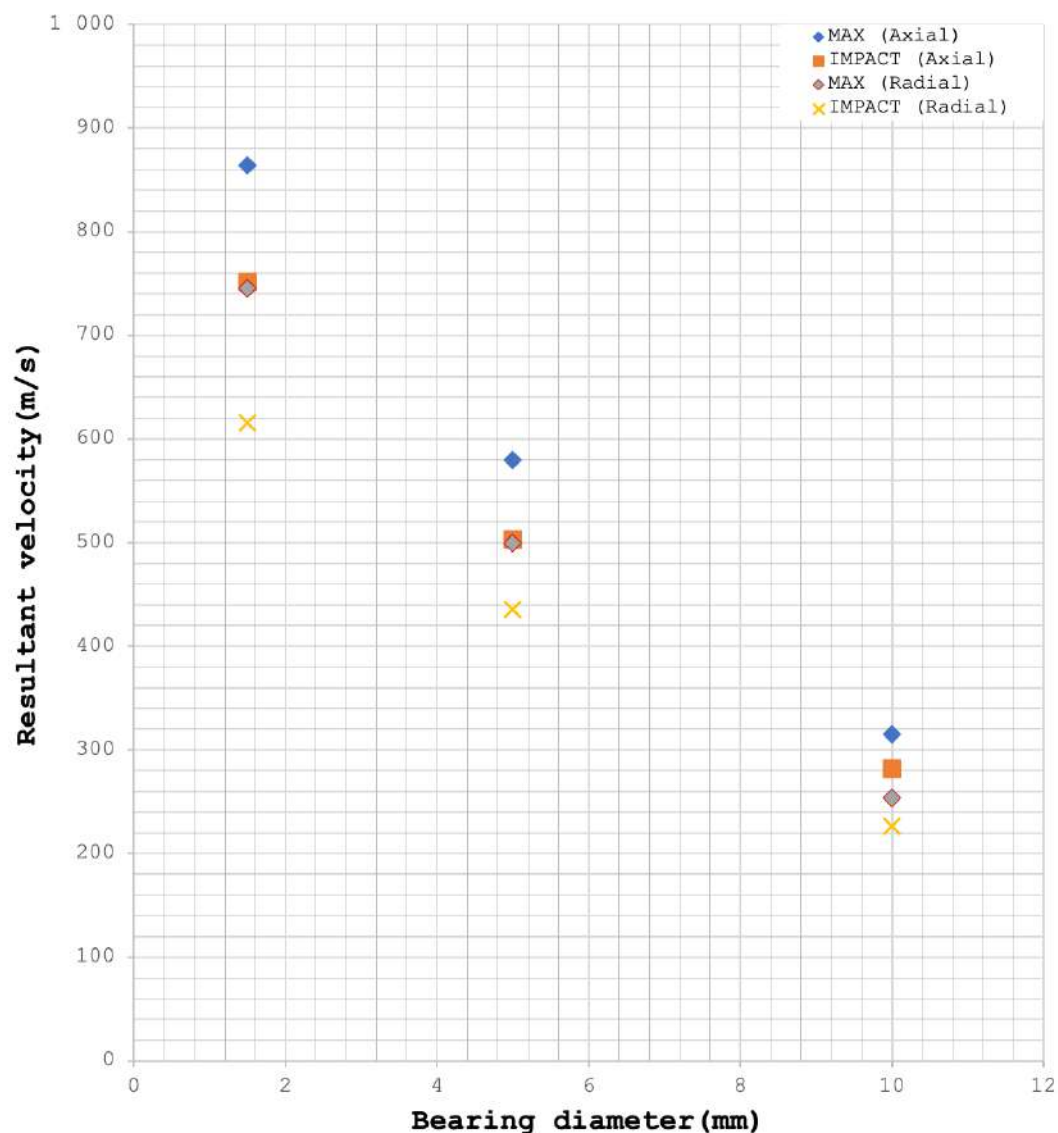


Figure 8.15: Graph of velocity vs. ball bearing diameter in axial and radial plane simulations

The resultant velocity vs. the ball bearing diameter data from the parametric study investigating three different ball bearing diameters in both the axial and radial plane, is shown in Figure 8.15. The axial plane simulation estimated a higher peak and impact velocity than the radial plane simulation for all given ball bearing diameters. The velocity decreased as the ball bearing diameter decreased. The general trend of the data was the same for both the radial and the axial plane results. At ball bearing diameters of 1.5mm and 5mm , the axial plane simulation showed similar impact velocity results to the radial plane simulation. For the 10mm diameter simulation, the axial plane impact velocity was higher than the radial plane maximum velocity.

8.4.3 Comparison of Velocities in Axial and Radial Simulations

In the radial plane simulation, the 4 ball bearing case was the only way to simulate a full "row of ball bearings" placed around the cylindrical charge. However, in the radial plane, it was impossible to distinguish whether there were more than one rows. In the axial plane, each ball bearing represented a row and multiple rows could therefore be simulated. The axial simulation results showed that the total number of rows of ball bearings did affect the overall ball bearing and target response. Therefore, the fact that the radial plane simulation could not distinguish between whether there were multiple rows or not may have been an issue when comparing the two simulations.

8.5 Pressure Profiles

8.5.1 Axial vs. Radial Comparison

Figure 8.16 shows the pressure vs. time graphs obtained from simulation of bare, 22g cylindrical charge detonated in 1) the axial plane and 2) the radial plane configuration. The pressure was measured at a gauge point located on cylinder circumference or length midpoint, as shown in Figures 5.2 and 5.3. The pressure-time curve showed similar characteristics of the Friedlander pressure curve for blast loads with a sharp increase in pressure at the time of arrival and decaying exponentially thereafter.

In both the radial plane and axial plane simulations, the "secondary shock" phenomenon (as defined by Karpp et al. [34]) was evident in the multiple occurrences of "peaks" after the main shock was initially reflected off the target wall. The peaks were caused by the main shockwave being reflected back and forth by the high density air-explosive interface. Multiple subsequent loading pulses were created, which progressively decreased in magnitude. The multiple loading pulses can be observed in Figure 8.16 a), which shows the gas pressure near the cylinder wall. The rest of the shockwave was reflected from the centre and impinged on the wall again, creating repeated pressure loading. The oscillation in the pressure profile was due to the contraction and expansion of the shockwave off the cylindrical wall. In the axial case, the "double peaking" [34] was again observed in the second series of peaks, as shown in Figure 8.16 b).

In general, the axial plane simulation resulted in a higher pressure peak compared to the radial plane simulation. The reflection time points also differed, despite both simulation setups having an equal radial distance between the charge and the target. For example, the second pressure peak occurred around $t = 0.095ms$ in the axial plane simulation, while in the radial plane, it only occurred around $0.16ms$.

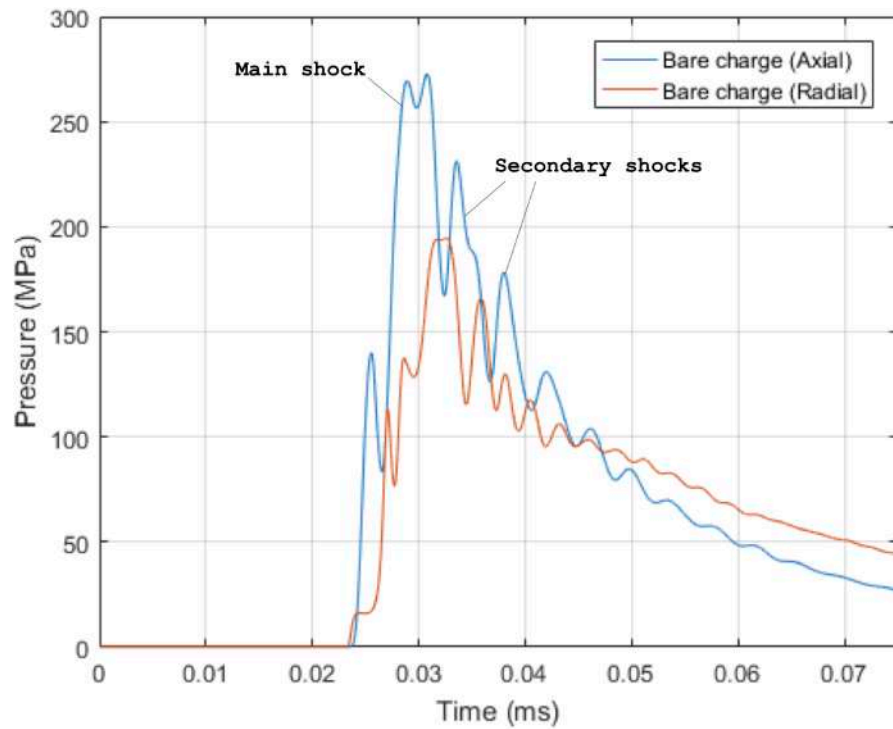
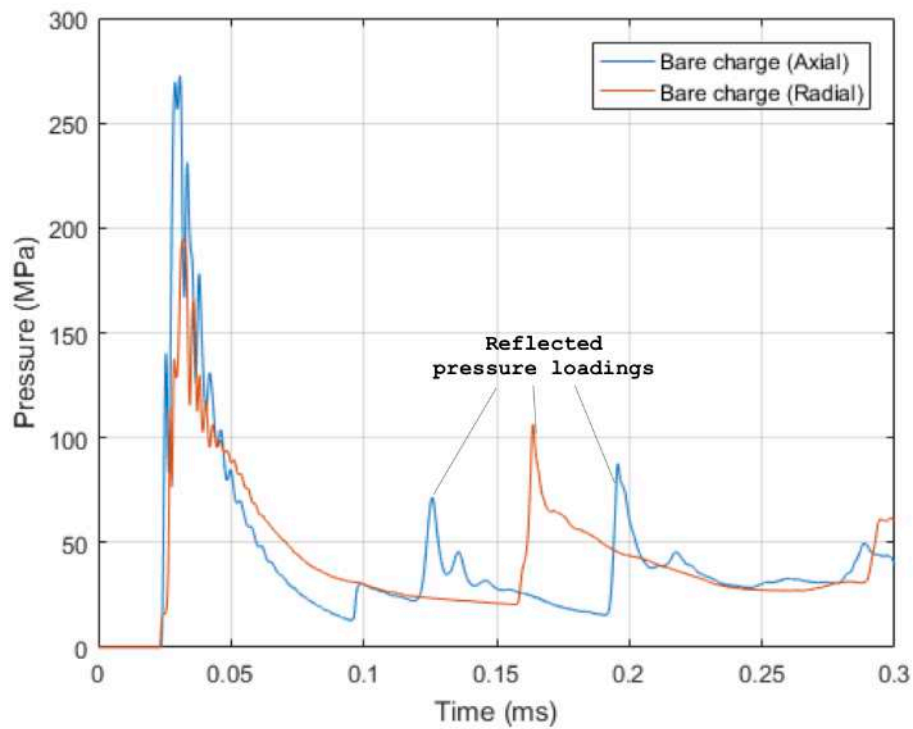
a) $t=0.075\text{ms}$ b) $t=0.3\text{ms}$

Figure 8.16: Pressure-time history comparison obtained from simulation of bare, 22g cylindrical charge in axial and radial planes

8.5.2 Effect of "Encasement"

Radial Plane Simulations (0-, 1- and 3-Ball Bearings)

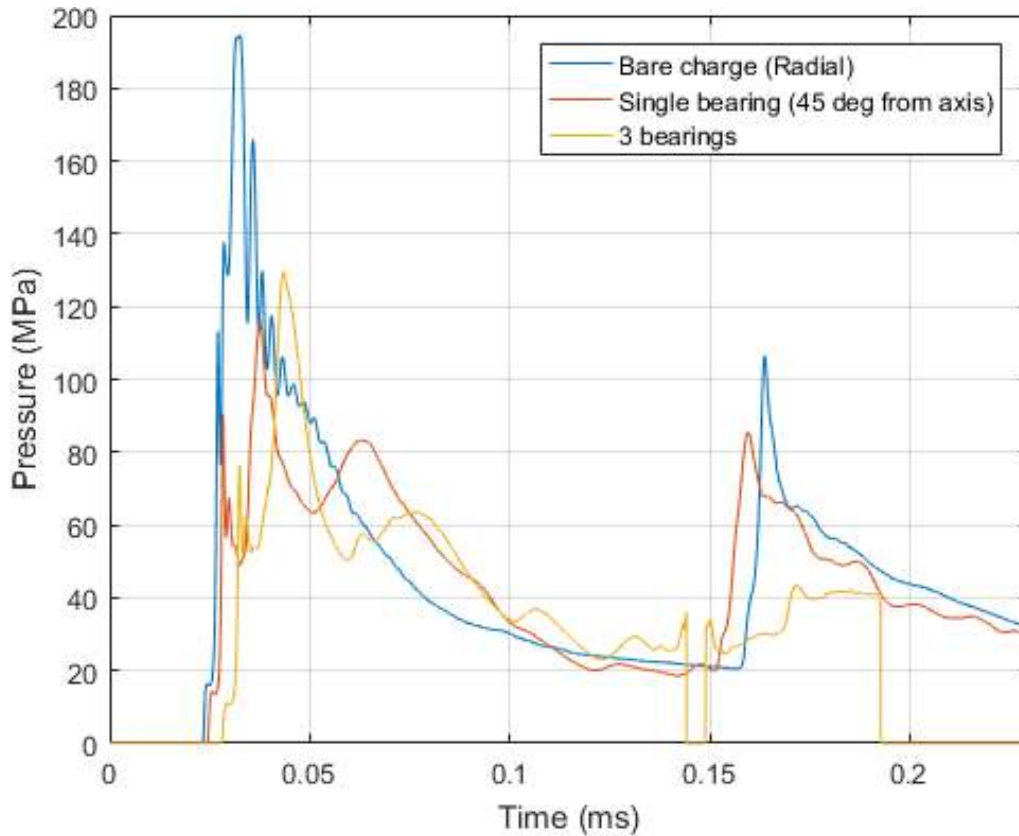


Figure 8.17: Pressure-time history comparisons for simulations with varied number of ball bearings in the radial plane

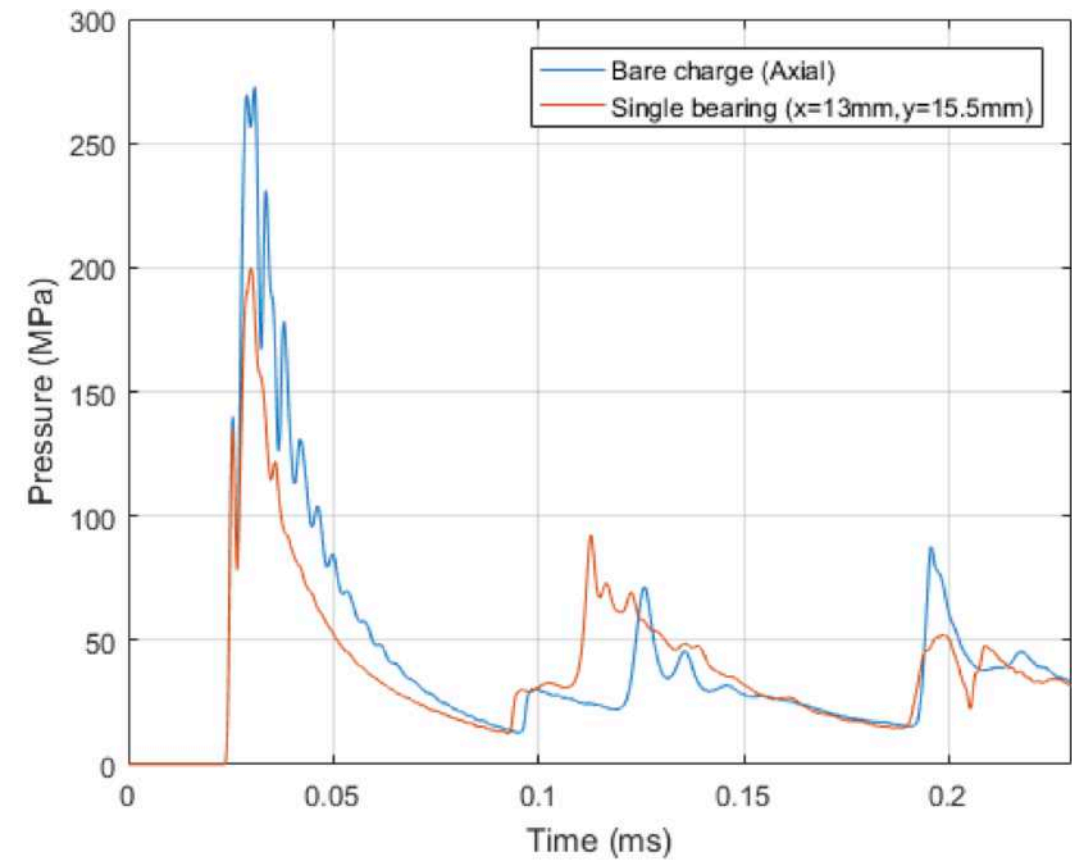
Figures 8.17 and 8.18 show the pressure vs. time plots of bare charge simulation compared to the single and multiple ball bearings simulations in the radial and axial plane, respectively. In both radial and axial planes, the observed initial pressure peak was higher in the bare charge simulation. This was to be expected, as blast generated by a cased charge is typically weaker than a bare charge of the same charge mass [59], and the presence of the ball bearings was analogous to a "casing" with gaps between the pre-formed pieces.

Overall, the pressure profiles exhibited the same trend of the typical Friedlander curve for blast load, with a sharp initial increase and decay followed by repeated pressure wave reflection. With the presence of an increasing number of ball bearings, the curve exhibited more irregular oscillations. The initial pressure peak of the bare charge simulation had the highest magnitude. The pressure peak observed in the 3-ball bearings simulation was, however, higher than that corresponding to the single ball bearing simulation. (Figure 8.17) This showed that in the radial plane, the increase in number of ball bearings was not necessarily related to a decrease in blast magnitude.

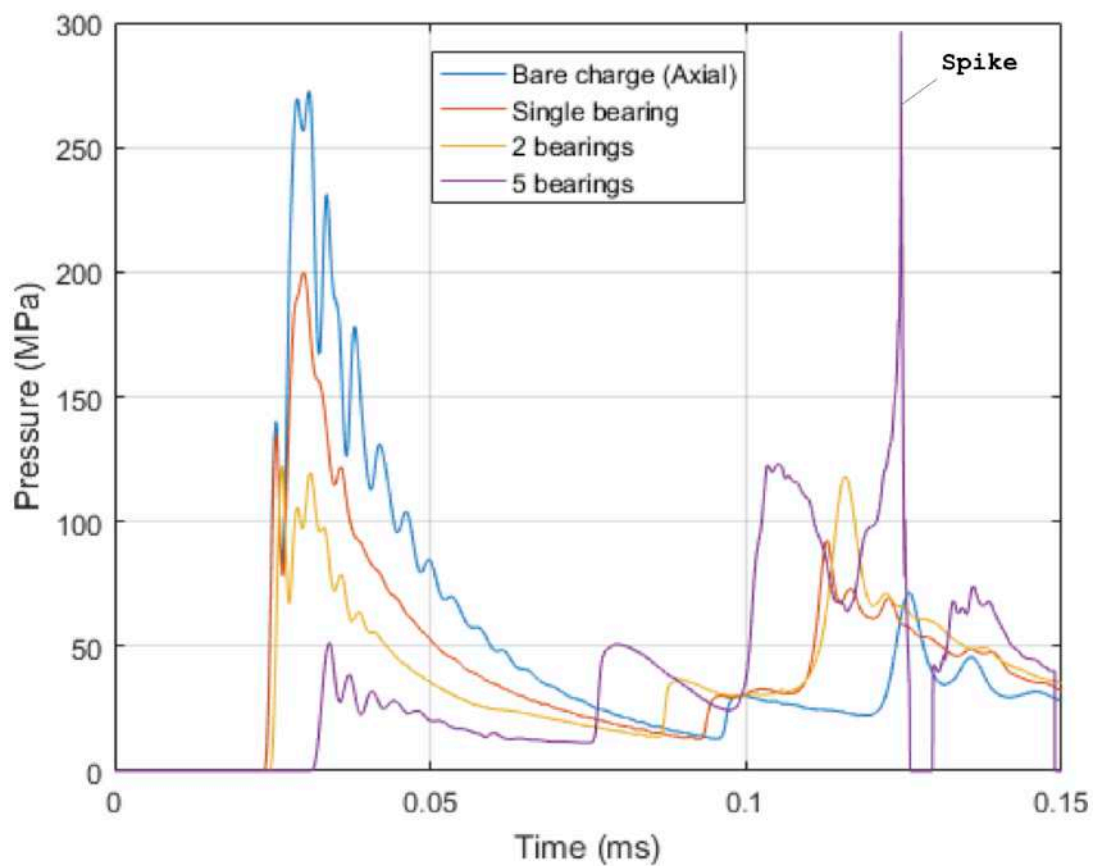
In the single ball bearing simulation, the second pressure peak occurred earlier than in the bare charge simulation. The pressure profile corresponding to the 3 ball bearings simulation did not show a clear second pressure peak within the duration of $t = 0.23ms$. The sudden drop in pressure to zero shown before $t = 0.15ms$ and around $t = 0.2ms$ indicated that the ball bearing was moving through the space occupied by the pressure gauge, causing the recorded pressure to read as a zero value.

Axial Plane Simulations (0-, 1-, 2- and 5-Ball Bearings)

Figure 8.18 a) shows that the reflected pressure loadings occurred at around same points in time for both the bare charge and single ball bearing simulations. However, as the number of ball bearings increased, the presence of ball bearings led to a general decrease in initial reflected pressure as per encasement theory. As shown in Figure 8.18, the peak pressure observed during the first peak decreased as the total number of ball bearings increased. It was also observed that as the total number of ball bearings increased, the second pressure peak occurred earlier and also increased in magnitude, as evident in Figure 8.18 b). In the 5 ball bearings simulation, the second pressure peak was larger than the initial peak. The 5 ball bearings simulation exhibited a sudden major increase in pressure prior to dropping to zero (due to a ball bearing moving across the gauge point). This "spiking" behaviour may be a numerical error rather than an indication of a physical phenomenon.



a) Single bearing



b) Multiple bearing

Figure 8.18: Pressure-time history comparisons for simulations with varied number of ball bearings in the axial plane

8.5.3 Effect of Varying Charge Length-to-Diameter Ratio

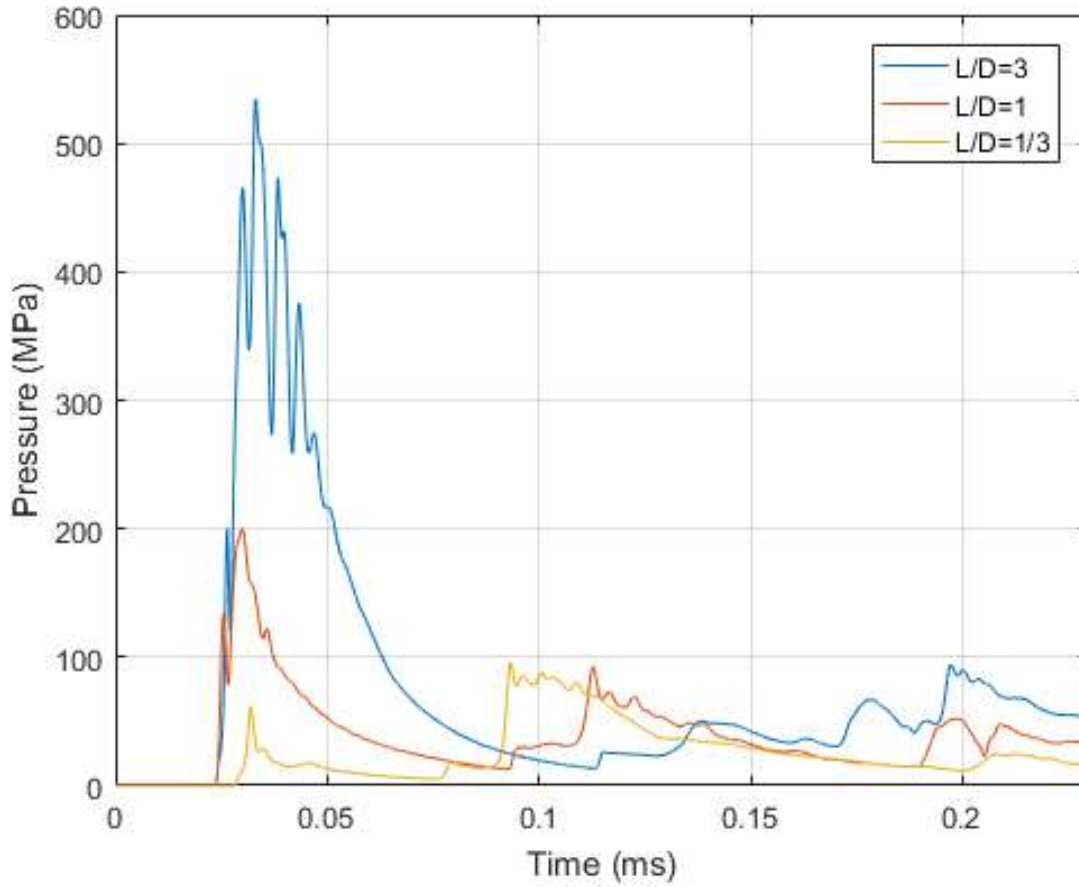


Figure 8.19: Pressure-time history comparisons for simulations of varied length-to-diameter (L/D) ratios

Figure 8.19 shows the pressure-time history comparisons for simulations of varied length-to-diameter (L/D) ratios. It was evident from the pressure profiles that an increase in L/D led to an increase in magnitude of the observed pressure peak, as observed by Hu et al. [38], who stated that more energy from the blast was directed radially for higher L/D values.

Karpp et al. [34] reported that varying the size of the explosive charge resulted in a different pressure pulse shape, and at a specific charge diameter, the second “peak” from air-explosive interface reflection may be larger than the initial peak. In Figure 8.19 it was observed that at $L/D = \frac{1}{3}$, the second pressure peak was distinctly larger than the initial peak. Up to this point, the subsequent reflected pressure intensity after the initial impact was generally lower than the first blast (with the exception of the 5 ball bearings simulation in the axial plane).

Chapter 9

Conclusion

This project set out to numerically investigate the the effect of incorporating foreign objects into explosive and its subsequent influences on the response of a target structure. The numerical study was aimed at studying how ball bearing incorporated into a cylindrically shaped charge in different configurations interacted with blast and its subsequent effects onto a cylindrical target. The effects of different factors such as the number and size of ball bearings incorporated inside the explosive charge and the manner in which the ball bearings were placed with respect to the charge together with the aspect ratio (length-to-diameter ratio) of the charge size were also investigated.

9.1 Validity of the Numerical Models

The numerical models were developed in ANSYS AUTODYN. 2D numerical simulation techniques, based in two different planes, were used to simulate the velocity distribution of ball bearings encasing the cylindrical explosive charge. The numerical models were validated against the average and maximum crater depth values obtained from previous experiments because there were significant variations observed (crater depths ranging from $0.5mm$ to $2mm$) in the experiments. The model showed fairly good correlation with the maximum crater depths measured in the experiments. Larger variations were observed when the number of ball bearings increased, however. Good correlation was also obtained for the relative spatial distribution of the craters, i.e. the location where the ball bearings impacted the cylindrical structure. The differences between the numerical results and the experiments were expected as IEDs are extremely unpredictable in nature. It would be difficult to reproduce the same results observed in the experiments. Repeat tests in the experiments would show more variations.

9.2 Velocity of Ball Bearings

Comparison with Empirical/Analytical Solutions

The ball bearing velocity-crater depth correlation was compared to empirically/analytically obtained projectile velocity equations reported as the Fish-Summers equation [74], the JSC equation [75] and the semi-infinite model equation [75]. In the radial plane simulations, the numerical model predicted a maximum crater depth of 1.7mm at an impact velocity of 829m/s , compared to the crater depths of 3.1mm (Fish-Summers), 2.1mm (JSC) and 2.5mm (Semi-infinite) calculated from the projectile velocity equations at the given velocity. The results from the simulations carried out in the radial plane under-predicted the crater depths for the given velocity when compared to the equations, but correlated well with the empirical/analytical solutions in terms of the trend of the data. The target cylinder surface was curved and under compression, which implied that a higher stress (and therefore force) was required for the impinging projectile to form a crater of the same depth as in a plate target of the same material.

Similarly, the axial plane numerical model predicted a maximum crater depth of 2.9mm at an impact velocity of 857m/s , compared to calculated crater depths of 3.2mm (Fish-Summers), 2.2mm (JSC) and 2.6mm (Semi-infinite). Axial plane simulation data points showed good correlation with the equations under the limited condition that the projection angle of the ball bearing was near-perpendicular to the charge surface. The results confirmed that the NASA equations could not accurately predict the crater depths under the non-ideal conditions.

The Distribution of Ball Bearing Velocity in the Axial Direction

The results of the numerical simulations were also verified against reported equations for fragments and pre-formed fragments, which are based on experimental data which indicated a non-uniform velocity distribution along the cylinder axis. Overall, there was a good agreement between the 2D model and the experimental measurements, including the distribution of the lower velocity values near the cylinder edges. It was shown that the behaviour of cylindrical charges partially confined in ball bearings was similar to that of encased charges to a degree.

It was observed that the simulated velocity was significantly less than the velocity predicted by the modified Gurney formula by Huang et al. [51]. Since the Gurney formula was intended for predicting the fragment velocity of encased charges and the charges tested in the current study were partially confined by ball bearings, the predicted Gurney velocity was higher than the numerical results. The normalised velocity curves for the multiple-ball bearing tests all coincided into an inverse quadratic function, which exhibited a steeper curve than the normalised version of modified Gurney formula by Huang et al. [51]. This implied that the ball bearing velocity loss towards the charge cylinder ends was more severe with multiple ball bearings than in the Gurney prediction model.

The Distribution of Ball Bearing Velocity in the Radial Direction

The numerical results showed that when the ball bearing was buried inside the explosive charge, the velocity of the ball bearing increased as the incidence angle increased. However, the ball bearing velocity decreased when it was positioned outside the charge despite the increase in y -position and in the incidence angle. This showed that the wave incidence angle was not the sole factor that had an influence on the ball bearing velocity. According to the model presented by Li et al. [53], the velocity of a fragment (and therefore the ball bearing) was influenced by both the incidence angle and the rarefaction wave effects. Therefore, the change in ball bearing position from "buried" to "exposed", may lead to a decrease in the rarefaction term, leading to an overall decrease in velocity despite the increase in incidence angle.

Although the ball bearing velocity was shown to peak at radial position $y = 10.5mm$ in the single ball bearing test set, when the total number of ball bearings was increased to 5 ball bearings, the ball bearing velocities were higher at $y = 15.5mm$ than at $y = 10.5mm$. This further confirmed that the total number of ball bearings involved has a significant influence on ball bearing behaviour, even when combined with other factors such as change in radial position. Increasing the total number of ball bearings incorporated in the charge had a greater effect on the rarefaction term when compared to the single-ball bearing scenario.

9.3 Effect of the Placement of Ball Bearings

Placement outside the Explosive Charge

In the radial plane configurations, when the ball bearing was placed adjacent to either the x - or the y -axis, the two configurations mirrored each other, as expected. Under the 45° -configuration, the ball bearing had a lower maximum and impact velocity compared to the other two configurations. In the axial plane configurations, the ball bearing maximum and impact velocities increased as the ball bearing moved further away from the detonation end, and reached its maximum when the ball bearing was placed on mid length of the charge. In all configurations, the impact velocity was lower than the maximum velocity by a constant amount. The ball bearings exhibited lower velocity towards the charge end, similar to the "end effect" of casing fragments from encased charges.

The recorded impact velocities were higher in the radial plane simulations than in the axial plane simulations. However, the crater depths were significantly higher in the axial plane simulations (up to $0.6mm$ higher) despite the difference in impact velocity, most likely due to the geometry of the target. The maximum crater depth corresponded to the configuration with the highest impact velocity in both the radial and axial planes.

Placement inside the Explosive Charge

In both the radial and axial plane configurations, the velocity increased as the charge position approached the surface of the cylindrical charge, and was shown to peak at $y = h = 10.5mm$, i.e. when the ball bearing was located just under the surface of the explosive charge. When the ball bearing was positioned on the charge surface, its velocity decreased. When the ball bearing was buried inside the explosive charge, the recorded impact velocities and crater depths were similar in both the radial plane simulations and the axial plane simulations. When the ball bearing was placed near the central axis of the charge, a "spike" was observed in the velocity profile shortly after detonation. A sudden change in direction of the ball bearing motion after the initial rise in velocity was also observed as a result of the interaction between the detonation shockfront and the ball bearing during the detonation process.

9.4 Effect of the Size of the Ball Bearing

The impact and maximum velocities of the ball bearing decreased as the ball bearing diameter was increased, in both the radial and axial plane configurations. In the radial plane simulations, the crater depth also decreased with an increase in ball bearing diameter. However, the crater depth increased with an increase in ball bearing diameter in the axial plane simulations. This implied that the target geometry affects the crater depth. In the radial plane, the projection angle of the ball bearing at impact remained nearly constant at 45° . The projection angles at impact deviated very little their initial projection angle of 0° , but did not show a clear trend of increase or decrease as the ball bearing diameter increased in the axial plane simulations.

9.5 Effect of the Total Number of Ball Bearings

The overall maximum and impact velocities of the ball bearings increased as the number of ball bearings increased, in both the radial and axial plane configurations. More "free space" and consequently more energy loss occurred when there were fewer ball bearings. Therefore the velocity increased with the total number of ball bearings. It was also noted that the pressure wave propagation slowed down as the number of ball bearings increased.

In the radial plane simulations, the ball bearings exhibited symmetrical behaviour about the 45° line (i.e. $y = x$ line). However, in the axial plane configurations the simulations did not necessarily display symmetrical behaviour about the charge mid-plane due to the direction of detonation. In the axial plane simulations, the ball bearing velocities increased as the ball bearing moved further away from the detonation end until it peaked in the mid-region of the charge cylinder, and subsequently decreased again. The endmost ball bearings exhibited the lowest peak and impact velocities.

The direction of motion of each ball bearing was measured by the means of their projection

angle (θ), defined as the angle between the velocity vector and the y -axis (positive in the clockwise direction). As the number of ball bearing increased i.e. as the charge confinement approached complete encasement, the data was shown to approach the shape of the projection angle curve for fragments from a cylindrical charge of $L/D = 1$ from the experiments by Anderson [48].

The inclusion of ball bearings in the charge resulted in an overall decrease in peak pressure, and the percentage decrease increased with the total number of ball bearings. Charge covered in rows of ball bearings acted similar to encased charges, especially to charges with pre-fragmented casings. With the presence of an increasing number of ball bearings, it was observed that the pressure profile exhibited small but frequent oscillations as a result of interaction with the ball bearings. In the axial plane simulations, it was also observed that as the total number of ball bearings increased, the second pressure peak occurred earlier and also increased in magnitude.

9.6 Effect of the Aspect Ratio of Explosive Charge

The overall maximum and impact velocities of the ball bearings increased as the explosive charge length-to-diameter aspect ratio (L/D) increased. The crater depth and the projection angle at impact also increased with an increase in the ratio L/D . It was found that an increase in L/D led to an increase in magnitude of the observed pressure peak, implying that more energy from the blast was directed radially for higher L/D values. It was also observed that at $L/D = \frac{1}{3}$, the second pressure peak was distinctly larger than the initial peak, whereas in other tests the subsequent reflected pressure intensity after the initial impact was generally lower than the first blast.

Chapter 10

Recommendations

- More experiments should be conducted to expand the crater depth data set so that the relation between the charge condition and crater depth can be isolated.
- For improved validation of the numerical model against the experimental data, other experimental methods to measure the velocity of the ball bearings during blast should be investigated.
- Future experiments should be improved so that it can be made more repeatable.
- The setup should be modelled in 3D to eliminate the issue of the discrepancy between the radial and axial plane simulations. A 3D numerical model will allow for improved capture of the target geometry and response.
- A damage and failure model should be included to produce representative simulation results for tests which exhibit bearing failure.
- Erosion criteria should be investigated further, to model the target crater depths more accurately.
- Other projectile geometries should be tested/simulated so as to model other types of shrapnel found in IEDs, such as nuts and bolts.
- Other target geometries, such as cuboid boxes, should be tested/simulated, to isolate the relation between the target geometry and crater depth.

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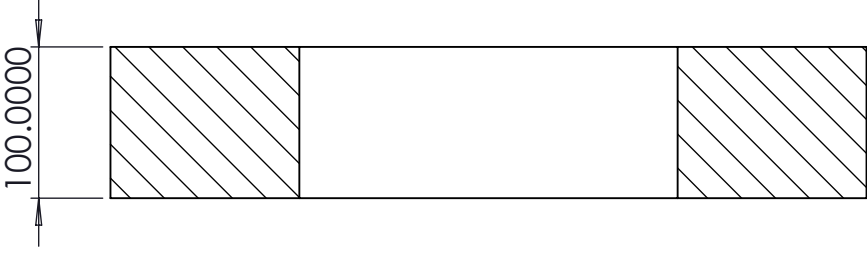
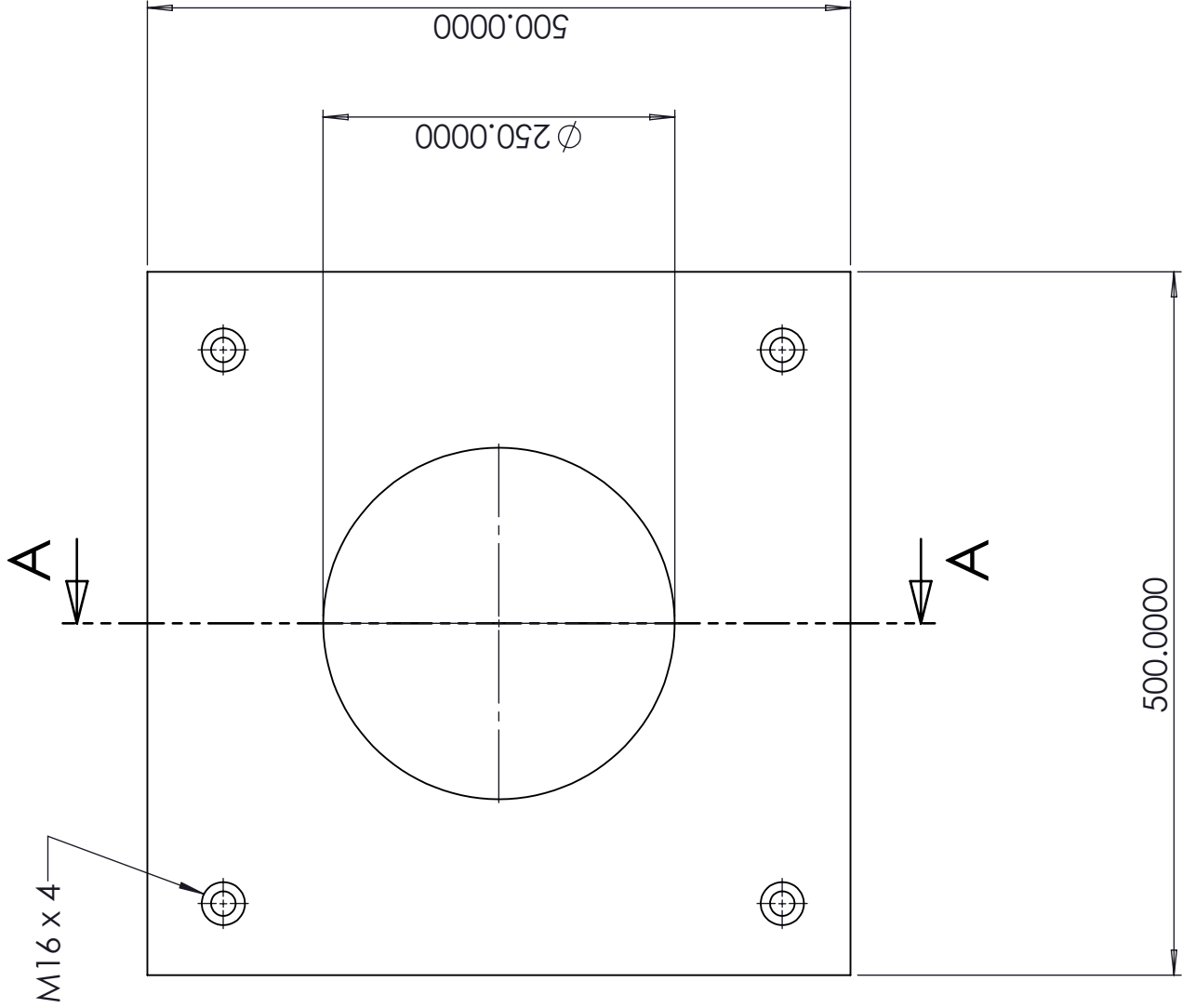
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Appendix A

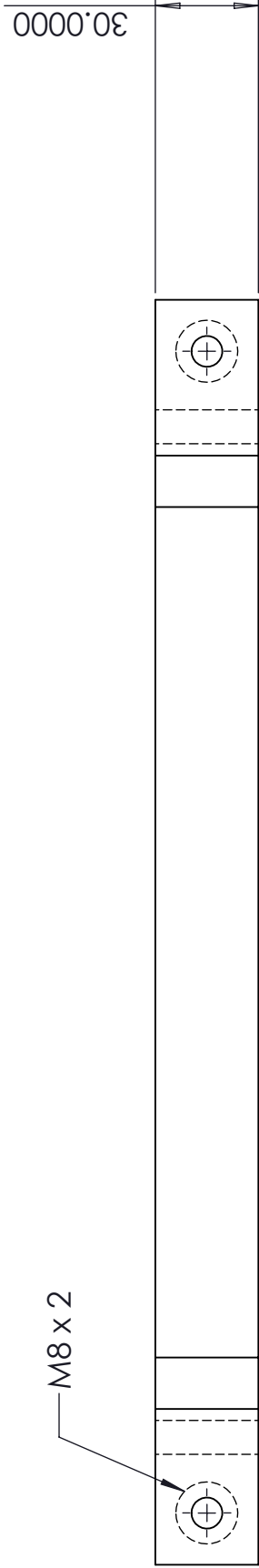
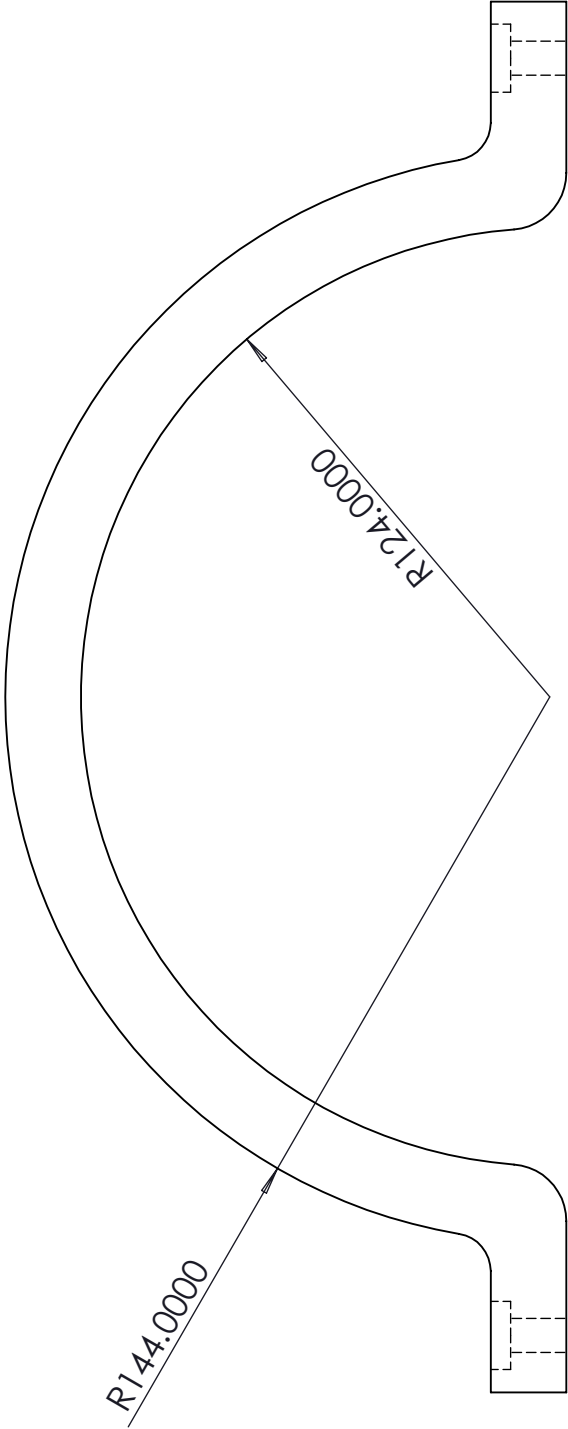
Machine Drawings



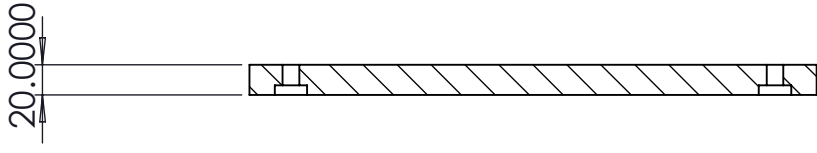
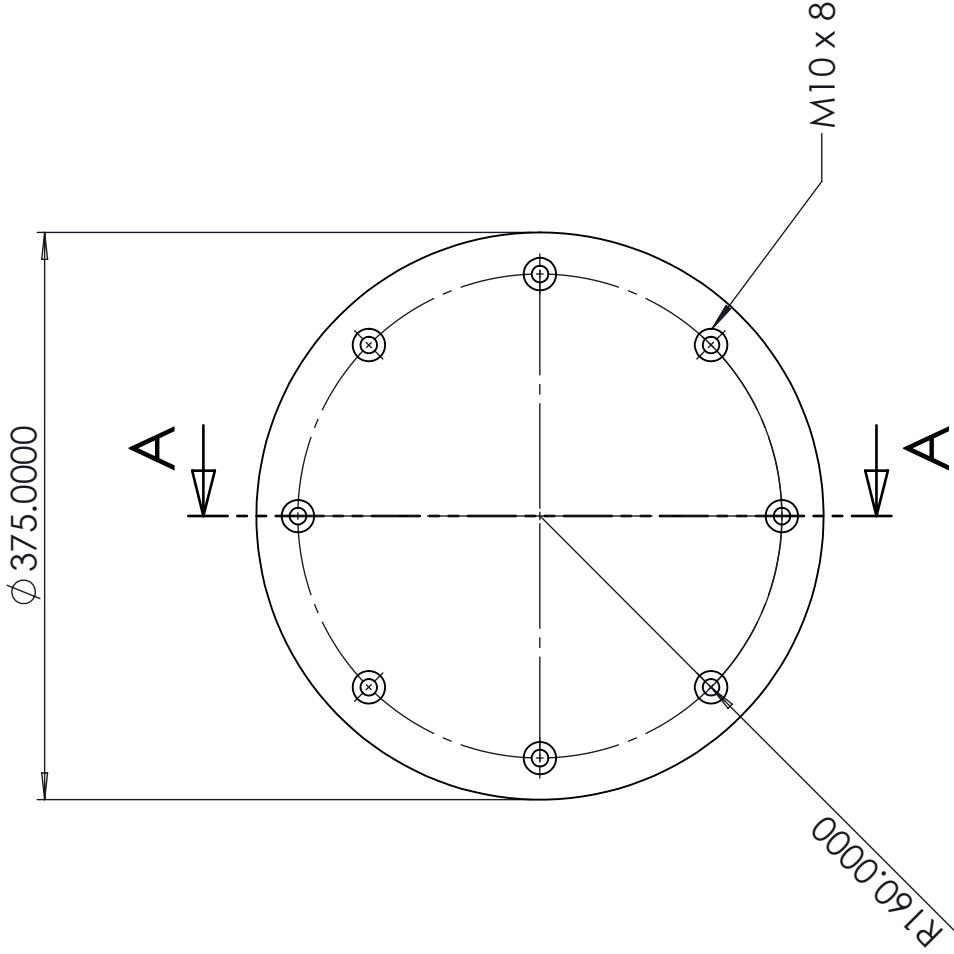
SECTION A-A

SCALE 1 : 5

A4 Landscape	University of Cape Town Department of Mechanical Engineering				
	Title: backing plate				
Quantity:	Part Finish	Date:	Scale:	of	
		8/14/2017	1:10	Sheet1	1
Material:	Drawn By:		Drawing Number		



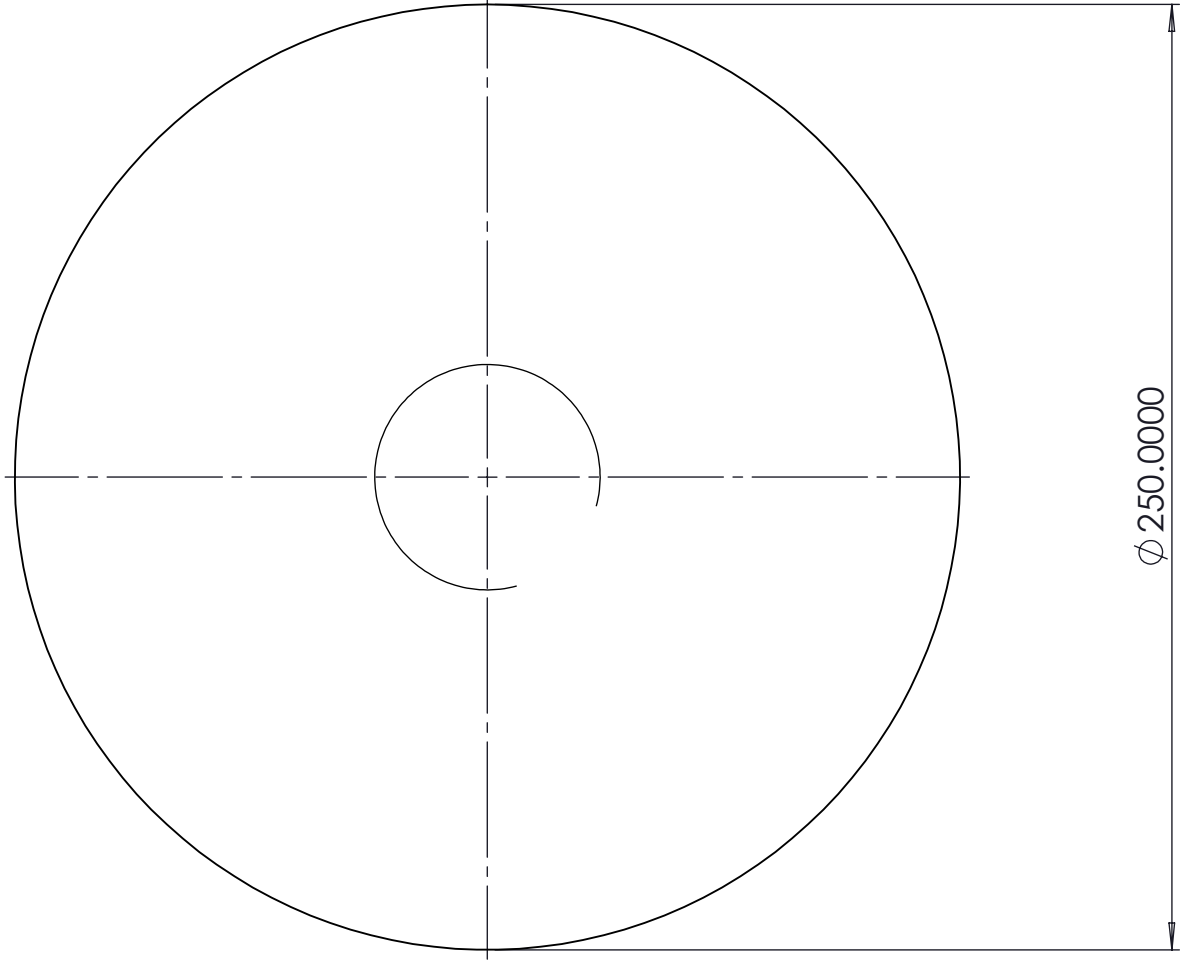
A4 Landscape	University of Cape Town Department of Mechanical Engineering				
	Title: clamp				
Quantity:	Part Finish	Date:	Scale:	of	
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Material:	Drawn By:				
	Drawing Number				



SECTION A-A

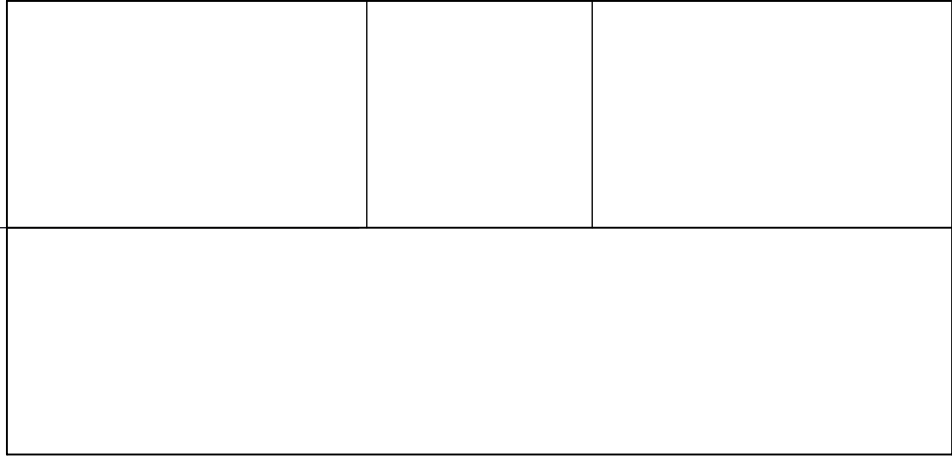
University of Cape Town				
Department of Mechanical Engineering				
A4 Landscape	Title: pipe cover			
	Quantity:	Part Finish	Date: 8/14/2017	Scale: 1:5
Material:			Drawn By:	Sheet1 1
				Drawing Number

60.0000

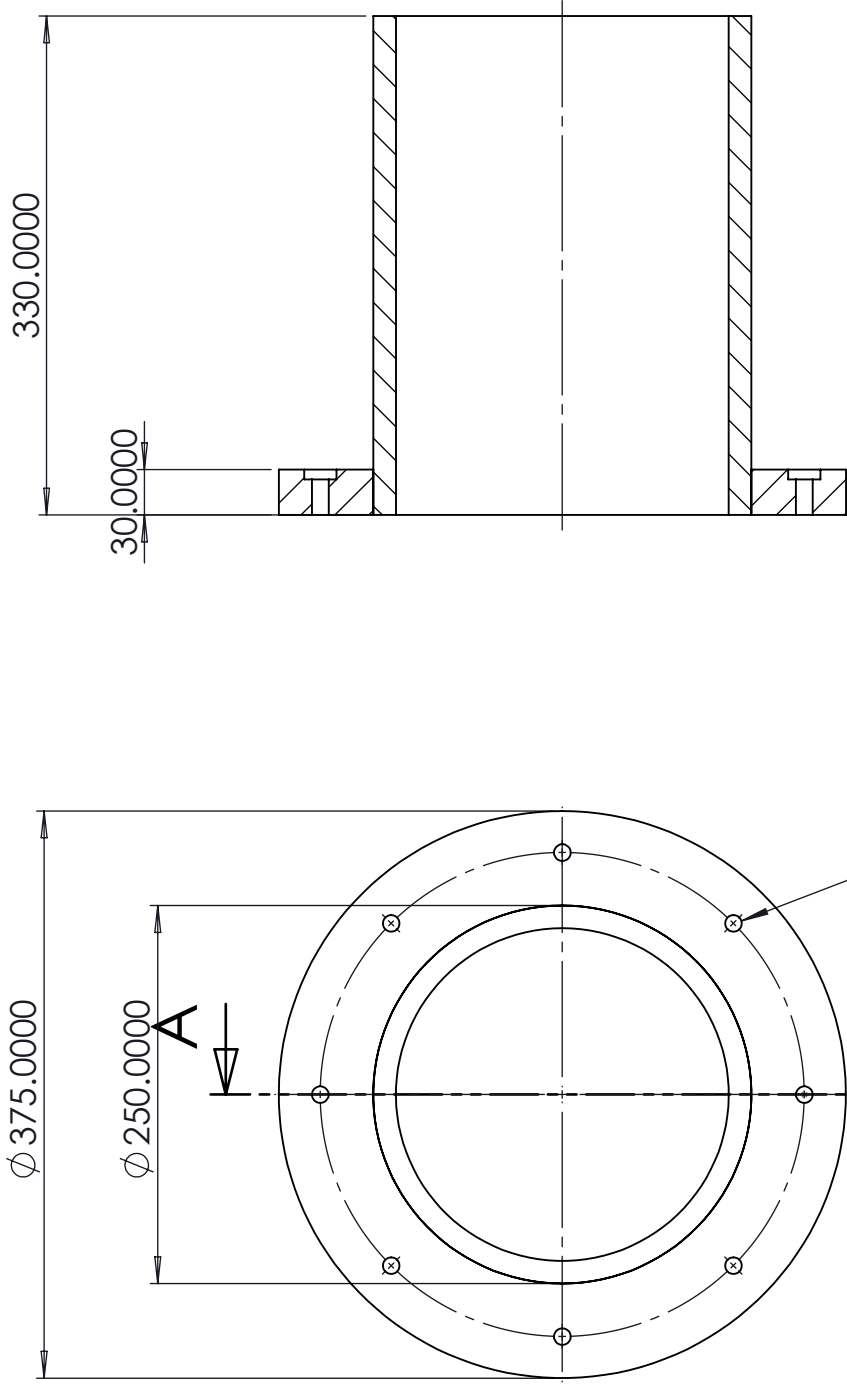


Ø 250.0000

120.0000



A4 Landscape	University of Cape Town Department of Mechanical Engineering			
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Quantity:	Part Finish	Date:	Scale:	of
Material:		8/14/2017	1:5	Sheet1
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SECTION A-A SCALE 1 : 5

A4 Landscape	University of Cape Town Department of Mechanical Engineering			
 Assembly Drawing	Title: pipe			
	Scale: 1:10	Date: 8/14/2017	Sheet1 1	of 1
	Drawn By: Drawing Number			

Appendix B

Photos of Test Cylinder Specimens Post-blast



TEST 2



TEST 3



TEST 4





TEST 5



TEST 6



TEST 7

