The Optimisation of the Impact-Cumulation Water Cannon

by

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This work deals with the processes occurring during the firing of an impact-cumulation water cannon. The breakup of the impulsive water jets produced by such a device is explained in terms of the overpressure of the jets on emergence. Design criteria are identified to produce a fast, coherent jet from such a machine.

Pressure histories are shown in the nozzle of the water cannon, together with pressures caused by the impact of a fast moving piston on a water packet.

Measurements of the forces exerted by the water jets are given, for jets produced with air or with vacuum in the nozzle.

A method for finding the jet head velocity decay is introduced, together with results therefrom. A substantial decay in the jet velocity with distance from the nozzle is noted, for the basic design of water cannon.

The pressure distribution in lengths of straight tube attached to the end of the converging region of a nozzle is given, and shows a rapid and exponential fall from the high pressure generated in the nozzle.

The impulsive blast noise produced by the firing of the water cannon is quantified and is shown to be eliminated when the piston-water impact is cushioned.
Firstly, I should like to thank Dr D.G. Edwards of the Mechanical Engineering Department of the University of Surrey for his friendship and guidance throughout the course of this work.

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a  Speed of sound of fluid
A  An Area
B  A constant
C  Speed of sound
Cp Specific heat
E  Youngs Modulus
F  Force
g  Gravitational Constant
h  Pressure Head
i  An integer number
j  Compression
k  Polar radius of gyration, a constant, bulk modulus
K  A constant
l  Length of water packet, length of collimator
L  Length of nozzle
Lr  Length ratio of nozzle
m  Mass
n  A constant
P  Pressure
P1 Atmospheric pressure
P2 Primary shock strength
q  Artificial Viscosity
r  radius
R  Area ratio of nozzle, Radius, Gas Constant
t  Time
u  A velocity
U  A Velocity
V  A velocity
W  An Area
x  A length
z  A length, cell width
b  A decay parameter, a constant, compressibility
γ  Gas constant
ρ  Density
ρ_s  Target Density
θ  An angle
ν  Poissons Ratio

Subscripts
mx  Maximum
o  Initial, undisturbed conditions
p  Refers to Piston
s  Refers to Solid
Water jets have been used for many years to cut solid material. The requirement for cutting is that the water velocity should be of sufficient magnitude for the stagnation pressure of the jet to be greater than the compressive strength of the material. Over the past few years pump technology has advanced to the extent that, with the aid of pressure intensifiers, continuously rated pressures of 400 MN/m² are not uncommon, (14,86,90,91). This means, for example, that the hardest rocks may be cut using water jets, (granite, for example, has a compressive strength of 220 MN/m²)

Water jets are not subject to the wear of conventional cutters, for example diamond saws, however the power required for unit mass of material removed is greater (54,56), than for conventional methods of cutting and so apart from some specialised applications, see for example references (42) and (111), continuous jet cutting is not widely used at present.

The present study is concerned with the breaking of material using single shots of water, taking advantage of the high pressures generated by the impact of water on a target. Various means of producing such impacts have been identified. A continuous jet, for example, produced with a deliberate variation in pump pressure, will separate into individual slugs of water at long distance from the nozzle (89,108). A continuous jet may also be interrupted repetitively (39,77) to give separate packets of water. The present work deals with the production of a single shot of water by an impact-cumulation water cannon. The velocity of water attainable using water cannons exceeds that possible using current pump technology. The damage caused to a target is related to the velocity by a power law. The damage is generally quantified
Cooley (20) indicates that damage is related to the pressure by a power approaching two, and thereby to the velocity of the jet by a power of four. Hammitt (49) gives an equation relating damage to the cube of the jet velocity. This means that the water cannon has a greater potential for causing damage to targets than the other means of producing pulsed jets described above.

(1.2) Characteristics of Impulsive Jets

(1.2.1) Fluid Mechanics of Impulsive Jets

The initial pressure developed by the impact of a jet of water on a target may be substantially greater than the stagnation pressure of the jet. This phenomenon was first noted as a mechanism for material damage by Cook (19) in work on the erosion of propellor blades. The analysis that he performed was given earlier by Gibson (44) and is shown in appendix A. This shows that the pressure, $P$, generated by the impact of water travelling at a velocity $V$, impacting on a stationary, rigid target is given by:

$$P = D C_o V$$  \hspace{1cm} (1.1)  

where $D$ is the density of the water and $C_o$ is the velocity of sound in water.

[This result was first derived by Saint-Venant (107) in 1867 in work on the impact of elastic bars, although he did not present this equation explicitly].

A comparison of the pressure generated by the impact of a jet and its stagnation pressure is shown in fig (1). It is clear that the impact pressure exceeds the stagnation pressure for velocities up to approximately 3000 metres per second.
higher than is currently possible with conventional pumps and intensifiers. At lower jet velocities than this the impact pressure substantially exceeds the stagnation pressure. The impact pressure, however, lasts for only a few microseconds until relief waves from the free boundaries have traversed the impact zone. The main consequence of this impulsive blow is that a compressive wave travels into the material. This wave causes cracking of the material and the high speed of the following water then opens these cracks and removes material. The impact pressure generated by water jet impact on a compressible target was first given by De Haller (30) and is discussed in section 1.2.3. The flow phenomena involved with the impact of water on targets have been the subject of much work done in the past. In early studies the emphasis was on the effect of droplet impingement, the rapid erosion of turbine blades and the effect on aircraft flying through rain providing the impetus for the work (65,66). Recently attention has been concentrated on the damage effects of pulsed jets, produced by water cannon or otherwise (24,77).

The jets produced by water cannons can move at a supersonic speed relative to the air, and are liable to produce a shock wave, and form a blast. The impulsive noise may therefore be quite substantial and can present a safety hazard. This is of particular importance if the cannons are to be used in an enclosed space, for example a mine.

The high speed jets produced by water-cannon are subject to large aerodynamic forces. When the jets emerge from the nozzle any disruption of the jet will be amplified by these forces, causing jet break up. It is necessary that the jets retain their initial high velocity and remain coherent over a reasonable working distance between the nozzle exit and the
can break up on emergence from the nozzle within a few nozzle diameters because of an internal disruption (41). It has also been shown that instability of the jet and subsequent breakup will occur some thirty nozzle exit diameters from the nozzle exit (41), so that this would seem to put a maximum limit on the effective cutting range. This will be discussed in section 8.3.1. It is this breakup of the jets which at present limits the cutting potential of the water-cannon. A major part of the present work, therefore, is aimed at discovering the causes of jet breakup and, by eliminating these, producing a more coherent jet.

(1.2.2) Water/Solid Impact Phenomena

Many experimental studies have been performed on the impact of water on solids over the past fifty years. Early studies took the form of inspection of the damage of various targets (38). It has only been in the past few years, with the advent of high speed computers and high-framing-rate cameras that the impact phenomena have been studied in detail.

Several numerical studies of the impact of liquid droplets with solid surfaces have been performed. These have featured a variety of droplet shapes. Hwang and Hammitt have studied the impact of spherical liquid drops in considerable detail, including the effect of a compressible target (60,61). Pidsley, (95), studied the impact of a liquid wedge on a rigid target, examining the effect of wedge angle on the maximum pressure. By reducing the wedge angle to zero he produced numerical results for a cylindrical impact. He examined the motion and the phenomena associated with the contact region.
the surface. The numerical results of Glenn (47), relating to the impact of a liquid cylinder, are pertinent to the present study. He presented an analysis in which he found that the radial jet along the target surface, caused by the relief waves travelling into the body of the jet from the sides, increased its velocity to two or three times the impact velocity. The wave motions in the cylinder and the start of the radial jetting are shown in fig (4). The initial radial jet velocity he gave as $2/\Pi$ times the initial velocity. He found that the initial compression was captured by the following rarefactions, which move faster, and that this took place in a relatively short time. Glenn also noted the possibility that the intersection of the relief waves in the centre of the cylinder would lead to cavitation.

(1.2.3) Impact Pressures

As previously noted the impact pressure generated by water on a target is given by:

$$P = \mathcal{D} C_o V$$  \hspace{1cm} (1.1)

This pressure is modified by the compressibility of the target, (30), to give the following expression:

$$P = \mathcal{D} C_o V/(1+\mathcal{D} C_o / \rho_s C_s)$$  \hspace{1cm} (1.2)

Where: $\mathcal{D}$ is the density of the target and $C_s$ is the speed of sound of the target

The pressure is also modified by the change in the speed of sound through water caused by the increased pressure and density which results from the impact (59). This increase in the speed of sound is given by:
Heymann, (51), gives a much simpler expression in an earlier paper which is sufficiently accurate for the present work, being within two percent of equation (1.3) for pressures up to 1000 MN/m²:

\[ C = C_0 + 2V \quad (1.4) \]

to give, altogether:

\[ P = \frac{\rho C V}{(1 + \rho C / \rho_s C_s)} \quad (1.5) \]

The effect of these various corrections to the incompressible impact pressure given by equation (1.1) is shown in figures (2) and (3), for the pressure due to the impact of an aluminium and a Nylon piston respectively, at various speeds. It will be noted that the Nylon, being relatively compressible, reduces the impact pressure by a factor of approximately two.

As noted previously the initial impact phase lasts only for the transit time of rarefaction waves from the sides of the water, which relieve the pressure to atmospheric. In the case of the impact of a cylindrical water packet on a target, lateral jetting will occur as the relief waves travel into the body of the water as fig (4) shows. The impact of a spherical drop or a liquid jet with a curved leading edge on a target will create a non-uniform pressure distribution, (67,106), with maximum pressures generated which are greater than the above theoretical maximum towards the outer diameter of the jet. This is due to the fact that the line of contact between the water and the surface of the impacted material moves away
There is, therefore, a compressed region bounded by the shock attached to the contact line. The wave diagram for this situation may be seen in fig. (5). Bowden and Field (10) first pointed out this phenomenon, indicating that high pressures could be associated with the contact point. Lateral jetting of the water is prevented until the speed of the contact point slows to the sonic velocity due to the increasing angle of the water surface. Fig (6) shows the wave motions in a spherical drop at this stage. Heymann (51) gave the maximum pressure produced by such an impact to be 2.8 times the pressure given in equation (1.1) above and cites indirect experimental evidence by Thiruvengadam (113), Thomas (114) and Jolliffe (68) in support. Hwang and Hammitt (61), however, found very little change in the maximum pressure from that given by the incompressible theory in their numerical analysis of the impact of a spherical drop. The numerical method of Hwang and Hammitt was criticised because of this result and for the choice of a non-slip boundary condition. The maximum impact pressure measured by Johnson and Vickers (67) and by Rochester and Brunton (106) was greater than that given by the incompressible theory for moderate speeds of impact, however the pressure distribution across the impacted surface differed in each case. The maximum pressure measured by Johnson and Vickers, using a strain-gauge technique, for a low speed jet impact (46 metres/second) was approximately 1.5 times the theoretical pressure. This was thought to be due to the curved leading edge of their jets, giving an impact similar to the spherical droplet as described above (50). Their plot of pressure distribution is shown in fig (7). In his recent numerical study of the impact of a spherical drop Pidsley (95)
contact points, and increased to approximately twice the incompressible value.

After the impact pressure phase the pressure will fall to the stagnation pressure:

\[ P = \frac{1}{2} D V^2 \]  

(1.6)

For the jets produced by a water cannon this pressure may well be high enough to cause further damage to the target.

(1.2.4) The Mechanisms of Impact Damage

Various investigators (28, 49, 102), have noted that many materials exhibit a threshold of water jet impact velocity below which no significant damage occurs. Such a threshold has been observed for both the velocity of impact of jets (29) and for the stagnation pressure exerted by continuous jets (89). The velocity of the water exiting from an impact cumulation water cannon varies with time, as noted experimentally in the present study, (see section 6.6), and by Edney (31), and as computed by Glenn (46) and Edwards and Welsh (36)). This decay of velocity varies according to the nozzle shape, all other parameters remaining constant. Thus a long nozzle produces a comparatively slow jet but with a longer running time. The combination of this variation of jet velocities with different nozzles, together with the damage threshold of materials means that nozzles should be selected according to the material being cut. Edwards and Welsh, (36), compared nozzle effectiveness using this concept by
denoting the 'useful' part of the nozzle discharge. The stagnation impulse may be found by evaluating the time-integral of the stagnation pressure at the nozzle exit for a prescribed period during jetting. Alternatively the summation may be halted at a prescribed jet velocity below which no damage will occur.

High speed impact of liquid has been observed (9) to cause the following types of material damage:

a) Circumferential surface fracture.

This is a brittle failure of material under high rates of strain, in which the material fractures a small distance, approximately one jet radius, from the impact area due to the neighbouring compression.

b) Subsurface flow and fracture.

Plastic flow is initiated at the region of maximum shear stress, which lies directly below the centre of the impact at a depth determined by the pressure distribution. Fractures are observed which are caused by the tensile stress tangential to the expanding compressive wave.

c) Plastic deformation of the surface.

This is caused by the pressure exceeding the material compressive strength, and is most readily observed in the impact of very high speed jets on metals. Ishlinsky (63) showed that the mean pressure required for plastic flow of this type is 2.5 to 3 times the compressive strength of the material. Stagnation pressures alone cannot give this required pressure (for metals) and so this is purely an impact phenomenon.

d) High speed radial flow of the liquid.

The speed of radial motion of a jet after impact is generally comparable to that of the impact velocity (47). Any
further water hammer pressures and is likely to shear the material. If the water has a curved surface, the radial flow may be far higher than the impact velocity, because of a 'shaped charge' effect, (47,62,66,95).

e) Reflection of compressive waves as tensile waves.

When a compressive wave in an elastic solid strikes a free boundary it is reflected as a tensile wave of the same magnitude. This tensile wave is more likely to cause damage to materials with a low tensile strength, concrete and rocks for example, than the original compressive wave. The reflection of tensile waves in thin targets can cause spalling of the target on the opposite side from the impact due to this mechanism (28).

These effects may be observed, to a greater or lesser degree, for all cases of water impact, supersonic or otherwise. Other mechanisms of material damage involve steady jets, whose stagnation pressure is well below the compressive strength of the material (102). For example a hydraulic wedging action was proposed by Rehbinder (103). According to this theory water will penetrate along grain boundaries and flaws and remove material by creating tension in the material and thereby failure. This mechanism is again more effective for materials with a low tensile strength and that are porous i.e. rocks. Another possible mechanism for the destruction of rocks was proposed by Evers et.al. (40). The model that was proposed involved the transient pressurisation of cracks in the rock by the passage of a jet over the surface. Air, trapped in the cracks, causes the pressure to fluctuate by first being compressed and then overexpanding. Subsequent recompression then takes place and the cycle continues. It was found theoretically that the pressure oscillations can
magnitude. This model was confirmed with experiments using 'dead-end' capillary tubes.

The degree of effectiveness of each individual damage mechanism depends on the strength properties of the target material. Granite, for example, is greatly affected by the reflection of compressive waves as tension at boundaries, whereas a softer rock, such as limestone, is affected to a lesser extent by this mechanism and can therefore suffer less overall damage as a result (118).

(1.3) Cavitation Aspects of Jet Cutting

Cavitation is relevant to the present study in several ways:

a) The study of cavitation damage by comparison with jet impact damage.

b) The production of transient pressures within the body of the water caused by the collapse and rebound of cavitation bubbles, which were themselves caused by the intersection of rarefaction waves.

c) Lastly, cavitation produced in an emerging, supersonic jet by the intersection of relief waves was proposed by Field and Lesser (41) as the principal cause of supersonic, impulsive jet breakup. This will be discussed further in section (8.3.1).

The collapse of a cavitation bubble can cause a considerable pressure rise in a region close to the collapse centre. Besant (2) and Rayleigh (101) gave the first analyses of the pressures generated by such a collapse. Parson and Cook (92) confirmed the importance of this effect in studies of the cavitation attack of propellor blades. From that date
which may be found in 'Cavitation' by Knapp et al. (70), and in a further book by Hammitt (50).

Water jet impact tests have been used for many years to evaluate the cavitation resistance of materials, based on the similarity of the damage caused by cavitation and water impact. Recent work using high speed photography, (69), and numerical techniques, (85), have confirmed the similarity: cavitation collapse in close proximity to a wall results in a high speed (100-600 metres/second) microjet directed towards the wall. This microjet, impacting on the wall, appears to be a major contributor to cavitation damage, (50).

Collapse of the bubbles creates high local pressures, but the pressure falls rapidly with distance. Re-expansion of the bubble then follows, caused by pressurisation of the trapped internal gas, which causes rebound of the bubble wall. This rebound will also create high pressures, however in this case the pressure is not as localised as the collapse pressure and a shock may be formed in the liquid, (55,64,96). Such shocks have been photographed by Kuttruff (71).

The impact of liquid on a target can cause cavitation. The numerical study of Hwang and Hammitt (62) predicted cavitation in a liquid drop impacting on a rigid target. Photographs showing such cavitation were taken by Brunton and Camus (13). In this case the cavitation is caused by the intersection of relief waves within the drop. These rarefaction waves are caused by reflection of the initial compression from the free surface, and by the intersection of the radial relief waves which relieve the high impact pressures. The latter waves cause the radial jetting of the drop.
There are three types of water cannon which have been subject to use either industrially or experimentally. They are distinguished by the positioning of the water in the nozzle prior to firing and the different means of imparting energy to the water. Fig (8) shows the three types, pressure-extrusion, impact-extrusion and impact-cumulation. In the pressure-extrusion water cannon both the nozzle and a portion of the barrel are full of water. A rigid piston in contact with the rear face of the water is propelled forwards by, for example, gas pressure, and ejects water ahead of it out of the nozzle. The velocity of the leading portion of the jet is low; the velocity of ejection increases progressively with time. It follows that in order to reach the head of the jet, the fastest fluid ejected must penetrate all the slower moving water ahead of it. This is naturally wasteful of kinetic energy and detrimental to cutting performance. Examples of industrial pressure-extrusion water cannons are those used for research into coal mining techniques in the U.S.S.R., (15).

The second type of water cannon is the impact-extrusion device. This has a piston which is allowed to accelerate before striking the water. The nozzle is full of water before the impact and again there is slow-moving water discharged before the high speed water. This is a major disadvantage for both the above types of water cannons as it has been found that even a very thin film of water on the target drastically reduces the impact pressure, (106). A full description of both an impact-extrusion and a pressure-extrusion device, is given by Mellors, Mohaupt and Burns (84).

The third type of water cannon, and the one to which the major part of the present work is directed, is the impact-cumulation water cannon. This device has a moving piston,
contained in a chamber directly upstream of the nozzle, the
nozzle itself being initially empty. The term "cumulation"
refers to the nozzle flow process: water, moving into the
converging nozzle generates compressive waves from the walls
of the nozzle. The compressions produce pressure increases
which are immediately relieved by expansions from the free
water/air interface. These expansions accelerate the water,
producing more compressive waves, and so on. This process is
shown diagrammatically in Figure (9). The result is that a
high pressure and velocity is generated in the leading portion
of the water packet.

Rhyming (105) has developed an analysis of the flow through
such a device, neglecting the compressibility of the water and
Glenn (48) has developed a finite difference code which can
model the flow in one dimension, with area change (see section
(2.2.1). Further developments of this code were carried out
by Edwards and Welsh (36) and included:
a) the incorporation of a realistic equation of state for
water based on the empirical data of Walker and Sternberg
(122),
b) the inclusion of a capability to model the impact of an
elastic, solid piston on the water packet.
c) the enhancement of computational stability by the
incorporation of an optional upwind-differencing scheme and
d) the provision of an improved artificial-viscosity
formulation, giving acceptable low smearing of shock fronts
coupled with enhanced stability in the flow behind the shock
compared to that achieved by earlier formulations. This
computer code formed part of a PhD study by Welsh (124), who
studied the effects of change of nozzle profile, position of
the water and water packet length on the cutting potential of
experimental confirmation of two of the predictions of the code, that of nozzle pressure and jet velocity. He found that the numerical results agreed closely with the experimental work. Edney (31) found a similar correlation between his experimental results and the numerical results of Glenn. Figure (10) shows the results of measurements of jet velocity by Edney, showing the correlation between the experimental and the computed results for a hyperbolic shape of nozzle. This study covered cases in which the nozzle initially either contained atmospheric air or was evacuated. Figure (11) shows an example of a computed nozzle pressure history for an exponential nozzle, indicating the rapid rise in pressure and velocity towards the end of the nozzle flow.

(1.5) **Industrial Uses of Continuous and Pulsed water jets**

The mechanisms of material breakage caused by a fast-moving jet of water are described in section (1.2.4). The industrial use of water cannons to date reflects the more effective modes of breakage, and is limited to rock and concrete cutting, (73,93,126). In this context, however, water cannons can be as efficient as conventional cutters (72,87,117). Continuous water jets have been used industrially for many years for cleaning purposes and for cutting soft materials. Increases in pump pressures have enabled an extension of the range of materials that can be cut using continuous jets. The cutting of coal, in particular, has attracted attention in recent years, both in terms of efficiency and safety. Using water jets in coal mining reduces the amount of respirable dust in the atmosphere and lessens spark hazard, (37,97). Hydraulic mines are in operation in Canada, the U.S.S.R, the U.S.A. and in Germany. Continuous water jets are used in these mines in
large-scale device with a high water-flow-rate. The standoff may be tens of metres and the excess water is used to transport the coal from the coal face. The second method uses much smaller jets of water close to the coal face. These jets cut the coal as they are traversed across the face. The volume of water used is insufficient to transport the coal which has to be moved using conveyor belts. This 'jet-cutter' system may also be used to augment mechanical cutters, (82,90). A full description of an hydraulic mine is given by Cooley (25). Coal cutting using pulsed water jets has been performed in the U.S.S.R. since 1959, although no full scale production mining has been attempted. Full size prototypes of multi-shot pressure-extrusion machines have been built, however, and trials carried out in mines. Chermensky (15) provides a full description of the work carried out and the results obtained. He reports that Tsyapko, (117), in a numerical comparison of coal breaking techniques, concluded that high-pressure pulsed water jets were more efficient than conventional pick cutters, pressure intensifiers and hydraulic monitors for medium and hard coal. Chermensky also reports that headings in sandstone may be performed with the pulsed jet devices at the same level of efficiency as mechanical breakers. He used a prototype machine and concluded that with the use of higher pressures the efficiency would be increased. This increase in efficiency with the use of higher pressures was also noted by Voitsekhovsky (120) in early studies of water cannons. During the course of this work Voitsekhovsky designed a nozzle which purported to give the best results for an impact-cumulation water cannon, (119). The length of the impacting piston was related to the area ratio of the nozzle, which was exponential in shape. The same shape of nozzle was
water cannon. A patent was filed by Cooley in 1969 (22) describing the design of a system for producing hypervelocity jets. This system consisted of an annular jet of water which was struck by a high-speed, shaped piston, travelling in the opposite direction. This piston produced the cutting jet by a combination of its own speed, impacting on the water, and a 'shaped charge' effect (3,8). The machine designed was never built, to the authors knowledge, and its proposed jet velocity of 'at least 9600 ft/second' has been equalled by water cannons using initially stationary water (31). Mohaupt et al., (87), have designed and constructed a pulsed jet device of the pressure extrusion type which has a multi-shot capability (10-20 shots per minute). They report that the economics of the pulsed jet system, compared with conventional hammers, appears to be favourable for breaking of concrete. A recent study by Labus (72) shows that a pressure-extrusion device is economically competitive with mechanical hammers for various uses, such as breaking concrete and asphalt roads.

The addition of abrasives to a steady water jet can greatly enhance its cutting potential, (1,53). Materials which are normally too hard for continuous water jets, such as metals, may be cut successfully using this method. The technique, which is only experimental at present, is to inject abrasives, such as sand or copper slag, into the water jet close to the nozzle exit. The abrasive particles are thus accelerated by the water and wear the target material through repeated impacts. The cutting ability of these systems, together with advantages such as no spark production, high speed of cutting and a cleaner cut than with conventional flame cutting techniques, gives the abrasive jet a good industrial potential.
coherent over long standoff distances, (57,127), however this is only true for continuous jets and no similar effect has been observed with pulsed jets.

Water jets can be used in many situations where it would be difficult to use conventional cutting techniques, for example in an explosive environment (42).

An industrial impact-cumulation water cannon has yet to be built, although computer results (124) indicate that, for a given pressure developed in the nozzle, the impact-cumulation process produces a faster jet than the impact-extrusion device.
(2.1) Nozzle Flow Analysis

An account of the various analytical solutions to the nozzle flow of pulsed jets is given by Welsh (124). The analysis most relevant to the present work is the incompressible, unsteady analysis of Rhyming (105), which dealt with a packet of water moving with a uniform speed entering a converging nozzle. The three main parameters investigated in this study were the area ratio of the nozzle, the water packet to nozzle length ratio and the shape of the nozzle. It was found that the maximum jet velocity was inversely proportional to the square-root of the length ratio, thus a longer nozzle produces a slower jet, all other factors remaining constant. The area ratio was also found to have a significant effect on the maximum jet velocity, which was found to be proportional to the sum of the square-root of the area ratio and the logarithm of the area ratio. This advantage, however, was found to be offset by the increase in nozzle pressure, which varied as the sum of the area ratio and its logarithm. Rhyming also found that in order to obtain the same performance for a short or a long water packet, then the initial energy of the fluid packets must be the same, i.e. for the same initial velocity a longer water packet is beneficial. The analysis was performed in parallel with a compressible, numerical study by Glenn which is discussed in section 2.2.1. A further case considered was one in which the water is set into motion by the impact of a solid piston. A simplified point-mass representation of the piston was used as the basis for calculation of the momentum transfer to the water. The impact was assumed to be a 'rigid body'
particularly with regard to the maximum pressure developed in the nozzle. The jet velocity, however, was overpredicted by fifteen percent compared to the compressible case. The possibility of cavitation in the water packet was mentioned, though no attempt to describe the effects was included. Formulae were presented giving both the maximum pressure and the jet velocity for various nozzles as a function of time. Certain of these formulae are relevant to the present study and are given in appendix B. These formulae are used later, in chapter 5, to compare the theoretical results due to Rhyming with the experimental results of nozzle pressure and jet velocity.

(2.2) Computation of the Internal flow

(2.2.1) Computer Models of Water-packet and Nozzle Flow

The analysis of Rhyming of the flow through an initially empty nozzle of a packet of water poses a dilemma for a numerical computation of the problem. His analysis shows that the velocity and pressure rise to the maximum value in a small, leading-edge portion of the water packet, towards the end of the nozzle flow. An Eulerian model, for example the Fluid in Cell code (43), where the fluid flows through a stationary grid, cannot deal accurately with the changing velocities near the nozzle exit. It is more convenient in the present problem to align the grid with the fluid packet, thereby using the Lagrangian approach (58). This method, however, suffers from the disadvantage that the entire region of interest, that is near the nozzle exit, is covered by only a few cells. The first numerical analysis to avoid these difficulties was presented by Glenn (48). His model was based
described as a 'leaky-lagrangian' scheme in that some of the fluid is allowed to traverse the boundaries of a lagrangian mesh. The water packet is rezoned each time-step to keep the total number of cells the same, allowing for the extension of the length of the water packet as it moves through the nozzle. Mass and momentum terms are adjusted for the converted cells.

An artificial viscosity, as first introduced by Von Neumann and Richtmeyer (121), is used to enable the code to deal effectively with shocks. This keeps the numerical oscillations to a minimum by allowing an additional diffusion to act in regions of steep pressure gradient. An example of the use of the artificial viscosity is shown in figure (12), taken from page 49 of Pidsley's thesis. The formulation of the artificial viscosity which gives the best compromise between the oscillations and the spreading of the shock is generally found by trial and error. It depends on the strengths of the shocks being developed, the number of cells and the time step used. The first numerical code developed by Glenn dealt with the flow of a water packet, initially moving, through an empty nozzle. Improvements to this code included adding the effect of air in the nozzle (78), and impacting the stationary water with a moving piston to set it in motion. The air in front of the water was handled by introducing a 'fast rezone' procedure to the code. This procedure altered the time-step of the computations in order to correctly model the motion of the gas and re-established the time-step once this had been achieved. It was found that the effect of the air in front of the water reduced the maximum velocity only slightly. The computed results are briefly summarised at the end of this section.

The impact of a moving piston on the stationary water was
by Welsh, using two, different, computational models. The first model of piston-water impact used in this study, found the piston-water interface velocity by examining the effect of pressure waves in the water and in the piston, (see appendix C). The rear water cell was then set to this velocity. The piston was treated as a solid body in the later stages of the flow to avoid unnecessary complications. Numerical instabilities were encountered, thought to be the inability of the code to handle shock waves generated by the piston-water impact. An artificial viscosity was then added and refined which reduced these oscillations to a 'satisfactory' level, as described above. In order to negate entirely the effects of these oscillations a second impact model was developed. The piston was treated as a rigid body, ignoring the effects of wave motion in the piston on the pressure being generated in the water. The numerical calculations were started when the incident shock had reached the leading edge of the water packet, the whole water packet then being said to be in uniform motion. With short pistons the starting time was altered to the time of return of the first wave through the piston.

The numerical codes developed by Glenn and by Welsh are one-dimensional in nature, that is the properties of each cell do not vary in the radial direction. The flow, however, as indicated in section 1.4, is patently two-dimensional, with compressive waves from the walls of the nozzle accelerating the bulk of the fluid in the centre. The errors involved are, however, minimised by the fact that early in the nozzle flow the fluid is moving only relatively slowly and, towards the end of the nozzle flow, the reduction in area of the nozzle allows the compression waves to travel rapidly to the centre.
indicated that the computer codes are sufficiently accurate to be used with confidence to predict the jet velocity and the pressures in the nozzle.

The codes, however, cannot be used to predict the breakup of the jet outside the nozzle nor do they deal with the possible oscillations in impact chamber pressure due to radial vibrations of the piston or motion of the walls of the impact chamber. An example of the development of the nozzle pressure, as predicted by the computer code, is given in figure (13), showing the pressure history for the conical nozzle used in the present study. The slow initial growth of pressure is apparent, together with the very rapid increase in pressure near the nozzle exit. A secondary pressure peak is also shown. A computer generated plot of the maximum pressure with time for the short exponential nozzle is shown in figure (14). The secondary pressure peak was confirmed experimentally in the present and previous work, see section (5.2). It was thought, (24,124), that the secondary pressure peaks were generated by further impacts of the piston on the water. This will be further discussed in section (5.2.2).

(2.2.2) Discussion of Computed results

The conclusions arrived at by computer studies, (35,48,78,124), are as follows:

a) The jet velocity decay at the nozzle exit plane is directly related to the ratio of the water packet length to the nozzle length. Short nozzles, for a given area ratio and water packet length, produce the fastest jets and the most rapid velocity decay at the nozzle exit.
maximum jet velocity is noted with increasing nozzle length. The smaller the length ratio the greater is the pressure developed inside the nozzle.

c) The 'stagnation impulse' is directly related to the water-packet length, such that the longer the water packet the greater the stagnation impulse.

d) The maximum velocity and the 'stagnation impulse' are both enhanced if the water packet velocity is increased. The gain in velocity and 'stagnation impulse' is greater, for the same nozzle pressure, than variations produced by altering the nozzle area ratio.

e) The addition of a collimator to the end of a nozzle results in an increase in the maximum velocity at the nozzle exit plane and reduces the overpressure of the water at the time of discharge, this overpressure being said to cause the rapid breakup of supersonic jets (see section 8.3.1). Thus the addition of a collimator will improve jet coherence for such jets.

f) A nozzle profile which features a large rate of change of area at the entrance to the nozzle gives a greater 'stagnation impulse' for the same nozzle area ratio, due to a lower decay rate of the jet velocity at the nozzle exit. There is also a decrease in the maximum pressure developed in the nozzle. This agrees with, and expands on, the result of Glenn who noted that the efficiency fell with increasing area ratio. Glenn also noted that the effect of a large rate of change of area at the beginning of the nozzle was detrimental to the
g) The best performance is obtained using a moving packet of water, termed a 'fluid piston', with solid pistons improving to this performance level with increasing length. Glenn's study also came to the conclusion that a 'fluid piston' was more desirable than a solid piston. The solid piston uses some of its kinetic energy in compressing the water on impact, giving a slower jet velocity than a fluid piston using an equal amount of water, moving at the same velocity. The solid piston also calls for a strong impact chamber; the 'fluid piston' does not.

h) The tensile strength of water has a negligible effect on water cannon performance.

The numerical study of Locher which included the effects of air in the nozzle arrived at the following conclusions:

a) The reduction in the peak discharge velocity due to the presence of air in the nozzle as opposed to a vacuum is five to seven percent. This is for an hyperbolic nozzle for both a 'fluid piston' and a metal piston. This result was experimentally confirmed by Edney who found no major difference in jet velocity between jets produced in a vacuum or in air.

b) The air in front of the water is heated to a very high temperature, approximately 6000 K. This could affect the use of water-cannons for mining because of possible ignition of flammable gases. No experimental evidence of these high temperatures has been forthcoming, however. Cooley (23) suggested that the very high temperatures predicted would lead
The velocity measurement system employed in the present work, described later, measures light cut off by the jet, using a sensitive photomultiplier. Any flash of light would lead to a notable output from the photomultiplier. This was not observed.

c) A strong shock is formed inside the nozzle, indicating the potential of such a device as a shock tube driver. Later work confirmed this potential, a device having been constructed and tested (75).

The basic aim of the project is:

TO IMPROVE THE INDUSTRIAL POTENTIAL OF AN IMPACT-CUMULATION WATER CANNON

Subsidiary aims identified in order to do this are:

(1) TO IMPROVE WATER JET SPEED
(2) TO PRODUCE A COHERENT JET
(3) TO REDUCE THE NOISE GENERATED BY THE CANNON

These aims are to be achieved:

(1) BY COMPARING THE IDEAL AND EXPERIMENTAL BEHAVIOUR OF THE FLOW PROCESSES
(2) BY IDENTIFYING REASONS FOR LOSS OF PERFORMANCE
(3) BY CORRECTING AND ADAPTING THE WATER-CANNON TO REDUCE THE FACTORS CAUSING LOSS OF PERFORMANCE
(3.1) Description

The impact-cumulation water cannon used in this study was modified from the existing University of Surrey water cannon, (124). A diagram of the basic water cannon is shown in figure (15). Essentially the water cannon comprises three sections: a piston launcher, an impact chamber and a nozzle. The piston launcher consists of a gas chamber and a barrel, separated by a thin 'Mylar' diaphragm. The barrel is bolted to the gas chamber, the joint being sealed with an 'O'-ring. Inserted in the barrel, and free to slide in it, is a piston, which is a solid metal or Nylon cylinder of diameter 76.2 millimetres, (three inches), and which may be of various lengths. This type of device is thus known as a 'free-piston water cannon'.

The gas chamber is pressurised progressively with nitrogen from a storage bottle until the diaphragm bursts. The bursting pressure may be controlled by changing the number of Mylar sheets used for the diaphragm. It was found that the bursting pressure was predictable using this method, to within approximately five percent of the total pressure. Bursting pressures of up to 0.69 MN/m² (100 p.s.i.) were employed. The piston is accelerated along the barrel under the influence of this expanding gas until it strikes a stationary packet of water. The effective length of the barrel is extended by the piston-velocity module and has a total length of 1.3 metres. The piston-velocity module has tapped holes along its length suitable for instrumentation to measure the piston velocity. The instrumentation used will be discussed in section 4.2.2..

The water is contained in the impact chamber between two diaphragms and is positioned immediately upstream of the nozzle. The impact chamber itself is a thick-walled steel
by the impact of the piston on the water. It has a length of 152.4 mm (6 inches), a bore of 76.2 mm (3 inches) and an outer diameter of 203.2 mm (8 inches). Tappings for pressure transducers are provided at three points along the length of the chamber, see section 4.2.1. The water required is supplied by means of a gravity feed via a tapped hole at the bottom of the chamber, the displaced air being expelled from a hole at the top. Visual inspection of the water-filled chamber showed that there was no air trapped inside using this method of filling. The impact chamber is bolted to the barrel of the water cannon and to the nozzle holder. The nozzle itself is screwed into the nozzle holder such that the transition from the bore of the impact chamber to the bore of the nozzle is smooth. Two 'O'-rings seals prevent leakage of water through the joints between the barrel and impact chamber and the impact chamber and the nozzle. The water which is discharged from the nozzle is caught and its energy dissipated in a catchment tank designed by Welsh which is not shown on the diagram.

Modifications to the basic cannon described above included the provision of an air gap between the barrel and the impact chamber to allow for spillage of the air from in front of the piston, a vacuum chamber into which the jet could be fired and a pneumatically operated separation of the impact chamber from both the nozzle holder and the barrel. The inclusion of the air gap allowed free spillage of the air ahead of the piston, with the aim of obtaining a flat, sharp impact of the piston on the water rather than a cushioned and more gradual momentum transfer, as occurred in the system used previously (see section 5.1.1). A diagram of the system incorporating the air gap is shown in fig (16). This provides a more practical
prior to firing. The air gap technique was used previously by Cooley (24), although in this case the gap was much larger, the piston actually being in free flight before impact.

The vacuum chamber was fitted to the front of the water cannon such that jets could be fired into a vacuum. The vacuum chamber may be seen in plate (1).

The impact chamber/nozzle assembly was modified to allow pneumatic clamping and retraction of the impact chamber so as to facilitate and speed up the changing of the diaphragms. This arrangement may be seen in plate (1).

(3.2) Operating Procedure for the Water Cannon

The Mylar diaphragms are first inserted to hold the water in the impact-chamber, and the surfaces bolted together or brought together pneumatically. The impact-chamber is then filled with water and the valves closed. Having ensured that there are no leaks of water, a piston is then inserted into the barrel, close to the gas chamber. A number of Mylar diaphragms are then placed between the gas chamber and the barrel, and these sections bolted together. The instrumentation is then reset.

As a safety precaution the laboratory is vacated to fire the cannon. This is necessary both because of the noise generated by the cannon and because of the high pressures generated in the impact chamber and the nozzle.

After each firing of the water cannon the barrel and the nozzle are cleaned, with particular attention being paid to small fragments of Mylar diaphragm broken from the main piece of diaphragm. These fragments were thought by Welsh to be the major cause of run-to-run variations in jet velocity and
Periodically the pressure transducers are removed and the grease column renewed (see section 4.2.1). If photodiodes are used for the piston velocity measurement, see section 4.2.2, then the glass windows are also cleaned periodically.

Occasionally the bore of the piston velocity module is honed to remove traces of rust. The barrel, being stainless steel, does not require this treatment.
(4.1) **Principal Objectives of the Experiments**

The existing University of Surrey water cannon (124) was modified as part of the present project, so as to facilitate its use and to shorten the down-time between firings.

The previous arrangement featured a demountable container for the water used for each firing. The container could be recharged with water only by unscrewing the nozzle and sliding the container out along the axis of the cannon. This proved to be a very laborious and slow process. A considerable saving in preparation time was achieved through the use of the redesigned layout shown in figure (17) and plate (1). Diaphragms are placed between the impact chamber/barrel and the nozzle holder/impact chamber surfaces, the water cannon being moved on its support trolley. The surfaces are then bolted together. Thin (0.1 mm) 'Mylar' diaphragms were normally used at the water packet/nozzle interface and either thin (0.2 mm) aluminium or Mylar diaphragms at the barrel/water packet interface.

The impact end of the water-packet was modified to include an air gap between the barrel of the water cannon and the water-packet as previously mentioned in section 3.1. A diagram of this arrangement may be seen in figure (16).

Pistons were manufactured from Nylon, aluminium and mild-steel in lengths of 50, 100, and 200 mm. Piston material densities and speeds of sound are shown in Table (1). In addition, trials were performed to evaluate the feasibility of using hollow water-filled pistons, or pistons which propelled the packet of water along the barrel into the nozzle.

Three nozzles were used in the present study, two having an exponential wall profile and the other a conical shape.
could be changed by unscrewing them from the nozzle holder. Straight parallel collimators of various lengths, (see Table (3)), could be attached to the front of the conical nozzle.

The principal experimental studies made during the present work were as follows:

a) Nozzle, collimator and impact-chamber wall pressures,
b) Piston velocity,
c) The velocity of the head of the high speed jets,
d) Photographic study of the emerging jets.
e) The blast generated by the emerging jet.
f) The force exerted by the jet on a target.
g) The motion of the walls of the impact chamber.

The various transient phenomena associated with the firing of an impact-cumulation water cannon are found to occur over a variety of time scales which, though short, are spread over some two orders of magnitude. For example, the piston velocity is measured by timing the final 10 centimetres of travel before impact. At a speed of 50 m/s this gives a two-millisecond timing interval. This was conveniently recorded by means of a writing oscilloscope, (a Medelec type FOR-4). However a much faster means of recording is necessary for the measurement of the impact phenomena, which last for only a few hundred microseconds, and have rise times of the order of microseconds. A storage oscilloscope and camera were used in some of the first tests but it was found to have only a limited resolution and had no pre-trigger facility. A transient recorder was used for the measurement of the impact phenomena, (A Physical Data type 515). This was a multi-channel recorder with a pre-trigger facility, capable of 0.5 microsecond sampling. The records could be displayed on an oscilloscope for immediate viewing and output to an analogue
apparatus for the measurement of jet velocity, described later, was recorded on a second transient recorder, (a Datalab type DL 905). This had a maximum 0.2 microsecond sampling rate with a pretrigger facility. The use of a second transient recorder was necessitated by the time delay, of between two to four milliseconds, between the measurement of the impact pressure and the discharge of the jet, and the limited time envelope granted by the recorders which were being run at a high speed.

(4.2) Experimental Measurements

(4.2.1) Nozzle, Collimator and Impact-chamber Pressures

Past experience (27,31) has revealed the necessity to protect piezo-electric pressure transducers from direct exposure to the flow in the nozzle of a water cannon. The provision of a short column of grease between the pressure-sensitive face of the transducer and the nozzle surface (28,124) has been found to be both effective and not detrimental to measurement accuracy (124). Accordingly the technique was retained for the present measurement of pressure in the nozzle and the impact chamber. Measurements of pressure were taken at three points along the length of the impact chamber, as shown in figures (18) and (19). Wall pressures were measured in the short exponential nozzle, using a transducer in the position indicated in figure (20). Measurements of pressure were also taken at various positions in the collimators attached to the end of the conical nozzle. A diagram of the collimator pressure-measurement points is shown in figure (21), the dimensions being given in Table (3).

The transducers used were all of the piezo-electric type
They provided a pressure-dependent charge output which was fed to standard charge amplifiers, (Kistler type 5007). The latter gave an output voltage which was coupled directly to the transient recorder. The accuracy of the pressures measured depended mainly on the scaling of the traces from the X-Y plotter and was approximately 2 percent of the maximum pressure recorded. The grease used to protect these transducers was Apeizon 'M' type vacuum grease.

(4.2.2) **Piston velocity measurement**

Two methods were employed to measure the piston velocity. The system in use at the outset of the present study relied on the successive interruption of a pair of light filaments. These traversed the barrel through small glass windows and entered a pair of photodiodes. After repeated use it was found that the windows became dirty and the system then performed erratically. In order to improve the triggering and the response of the system, a magnetic sensor, (Radio Spares type RS 304-166), was then used. The mounting and position of this sensor is shown in figure (16). The pistons which were non-magnetic were fitted with two mild steel bands a known distance apart. The motion of the steel past the sensor gives a step rise in output voltage. In order to further improve the measurement of the piston velocity the output of the magnetic sensor could be recorded by the transient recorder which, with its pre-trigger facility provided superior records to the writing oscilloscope. The output traces from the diodes and the magnetic sensor are shown in figure (22). The piston velocity could be measured to 3 percent by this method.
Initially the jet-head velocity was measured using the system developed by Welsh. This system involved the cutting off of light, by the intrusion of the jet, from two photodiodes placed a known distance apart. The output from each photodiode was recorded on separate channels of the transient recorder. An example of the photodiode outputs is shown in figure (23). In practice, the system proved unsatisfactory in that the start of the record of a given photodiode signal was invariably ill-defined, making it difficult to time the point of cutoff of light. This factor limited the resolution of the system, runs having to be discarded as the traces were so poor. In order to improve the accuracy of the measurement and to give an indication of the change in the jet head velocity with distance the system shown in figure (24) was developed. A point source of light was reflected from a concave mirror to obtain a parallel beam of light some 8 inches in diameter. This beam of light was directed across the path of the jet onto a plate through which a series of holes has been drilled. The holes were positioned a known distance, 25.4 mm, apart, in a straight line, in line with the axis of the nozzle. The resulting filaments of light were directed via a plane mirror onto another concave mirror. A photomultiplier, (Mullard type 150cvp), was positioned at the focus of this mirror to collect the light from all the holes. As the jet emerged from the nozzle it sequentially obscured the filaments of light, giving a stepped output from the photomultiplier. The output signal was recorded on the fast transient recorder and was displayed on an oscilloscope and/or plotted.

The light source used gave a 50 Hz a.c. ripple to the
study this was of no consequence. Background light was kept to a minimum in order to reduce the signal to noise ratio. The mirrors had to be adjusted carefully to focus the light on the sensitive area of the photomultiplier and were rechecked before each firing of the water cannon. This method proved capable of illustrating the decay in jet head velocity as the jet receded from the nozzle exit, with an accuracy of approximately 5-10 percent. Output signals from the photomultiplier and the results of the measurements are discussed in section 5.4.

(4.2.4) Photography of the Jets

Schlieren photographs were taken of the jets produced by the water cannon using the system shown in figure (25). For a typical run the laboratory was darkened and the camera shutter opened. An argon arc spark light source was then triggered at the required point and time by the passage of the jet, by means of the impact of the jet on a force-transducer/striker arrangement, shown in figure (26). The striker used was a match stuck to the top of the load washer using Apeizon 'Q' compound or plasticene. The striker was used in order to protect the force transducer and its cable from any direct impact of the jet. The force transducer used was a Kistler type 901A load washer. This system was found to give a reliable triggering signal, with pictures produced at the correct stage of the flow process. Triggering from a pressure transducer placed inside the nozzle was found to be unsatisfactory owing to run-to-run variation in the speed of the jet. The amplified load washer signal triggered the second
triggered the spark source after a small, preset, time delay. The load washer arrangement was mounted independently of the water cannon in order to prevent pretriggering due to vibration.

(4.2.5) **Blast Measurements**

Measurements were taken of the atmospheric air-blast produced by the water cannon in order to quantify the noise generated by this type of device. The technique and apparatus used was that described by Edwards and Smith (33). A diagram of the apparatus is shown in figure (27). The base of the mounting apparatus is a microphone stand. This was adapted to hold a thin tube (6mm diameter O.D.) of length approximately 0.7 metres. Attached to the end of the tube is the blast gauge itself, the connecting cables passing through the bore of the tube. The design of the miniature blast gauge used is shown in figure (28). The pressure transducer acting as the sensing element of the blast gauge is an Entran EPF-200-50 pressure gauge with a maximum range of 50 p.s.i.. It is connected to the Bryans transient recorder via a Fylde type 359TA amplifier. Tests have shown that this gauge will faithfully follow the pressure variations of a blast and that it has only a small effect on the incident shock (33). It is, however, subject to a natural oscillation frequency of less than 50 kHz. To counter the effect of vibrations the recording from the blast gauge is first stored on the transient recorder and then analysed on a computer. A fast Fourier analysis of the recording may be performed and the noise eliminated by removal of unwanted frequency bands and
(4.2.6) Jet Force Measurement

In order to evaluate the cutting potential of the impact-accumulation water cannon, measurements of the force that the water jets induced on a target were carried out. This was performed using the system shown in figure (29). The jet is allowed to impinge on a flat metal plate mounted on a load washer, giving a direct reading of force. The plate diameter and the standoff distance may be changed to indicate the spread of the jet. This arrangement is incapable of responding sufficiently rapidly to give the impact pressure, equation (1.1), which lasts for only a few microseconds, but it will follow the 'quasi-static' stagnation pressure caused by the impingement of the high speed jet. The load washer used was a Kistler 901A force transducer with a type 5001 amplifier giving a direct volt/Newton reading. The load washer used was mounted independently of the water cannon to prevent spurious readings due to vibration.

The frequency response of the load washer/plate combination was measured by impacting a long steel bar on the plate. The load washer was supported by a similar bar. The resulting trace was found to be in good agreement with that predicted using the one-dimensional bar equation, with only a slight oscillation being noticed. The rise time of the system was found to be of the order of 40 microseconds. An example trace from these tests is shown in figure (30). On the basis of these tests it was decided that the load washer/plate combination could be used as a quantitative measuring device for the forces exerted by the jets.
A strain gauge, (Showa type N11-FA-S120-11), was fixed to the outer wall of the water packet, 20 mm from the impacted end, so as to measure the circumferential strain. This was connected through a half Wheatstone bridge, a dummy gauge being used as the other arm. The amplifiers used, Fylde type 359TA, also provided the bridge balance unit.

An accelerometer, (Kistler type 8042), was mounted on a dummy pressure transducer placed in the walls of the water packet, via an extension piece, in order to measure the motion of the transducers. A diagram of the positioning of the strain gauge, together with the accelerometer mounting is shown in figure (31). Acceleration measurements were obtained at transducer positions one and two. It should be noted that the measurements of strain and acceleration were not, however, performed together, but on separate firings of the water cannon. The accuracy of the acceleration measurement is approximately 2 percent and the measurement of the strain is accurate to within 10 percent.
(5.1) Impact-chamber Pressures

(5.1.1) Description of Impact-chamber Pressure Records

A typical trace of pressure measured in the impact chamber using a vacuum in the barrel is shown in figure (32a), the pressure being measured at position one (figure 19). The slow rise and small pressures measured, compared to those predicted by the one-dimensional theory, suggested that some cushioning of the impact was taking place, probably because of gas in the barrel being compressed in front of the piston as it approached the water. This gas is assumed to have leaked past the piston from the driver chamber, especially during the early part of the piston motion when the pressure difference across the piston is greatest. When an air gap was included between the barrel and the impact-chamber, the character of the impact-chamber wall-pressure records altered fundamentally, as shown in a typical trace in figure (32b). It was found that the maximum pressure was raised to a value above that predicted by the theory and the rise time was reduced. An examination of the aluminium diaphragms after impact showed a firm imprint of the piston face. (Previous runs with an initially evacuated barrel had shown no such imprints, the diaphragm also being badly deformed). This indicated that sharp impacts were now occurring. Figure (33) shows the maximum pressure measured with various impact speeds of the 100 mm aluminium piston, illustrating the above-mentioned pressures, well in excess of that indicated by the one-dimensional theory. There is also a distinct difference between the maximum pressure recorded for the shortest aluminium piston at the higher piston speeds, compared to that
Nylon and mild-steel pistons also generate pressures above those predicted by one-dimensional theory, as may be seen in figures (35) and (36) respectively. The large-amplitude oscillation common to all of the sharp-impact pressure histories in the impact chamber is also a notable departure from the one-dimensional theoretical predictions. The magnitude of the pressure peaks recorded at the second and third transducer stations varies from that recorded at the first. The period of the pressure oscillations also changes. Typical records of the pressure measured at the three pressure stations are shown in figure (37). Figure (38) shows the maximum pressure recorded at the second and the third transducer positions for the 100 mm aluminium piston as a function of the piston impact velocity. It will be noticed that, particularly for the third transducer position, the pressure recorded is very variable and generally higher than predicted. It should be noted at this stage that these pressures must be taken into account when the design and safety of impact-cumulation devices are considered.

The high pressure phase is terminated by the reflection of the initial compressive wave from the front of the water. It is noticeable, however, that a second phase of high pressure is recorded some few milliseconds after the first. This is shown in a typical trace in figure (39).

This secondary phase of high pressure is believed to be derived from a second impact of the piston. The piston still has some residual gas pressure behind it and it pushes water into the nozzle. The effect is to produce a pressure-extrusion process in the nozzle. The time between the regions of high pressure varies from 1.5 to 3 milliseconds, depending on the piston material and its initial speed. It is likely that a
late-time jet velocity measurements, (see section 5.5).

(5.1.2) Discussion of Impact-chamber Pressures

The oscillation of pressure in the impact chamber is associated with the sharp-impact mode of piston/water energy transfer; not when the impact is cushioned. Changing the piston length does not alter the period of the oscillations. For the longest aluminium piston and for the Nylon pistons the periodic time of the oscillations is well within the time taken by the initial stress wave in the pistons to return to the piston-water interface. This is clearly shown in figure (40), the pressure history for the impact of the 200 mm aluminium piston at the first transducer position. The conclusion, then, is that the oscillations are not purely the result of the longitudinal wave motion within the pistons, (this being the only previous (24,124), explanation for the occurrence of pressure changes in the water before the relief wave arrives from the front of the water packet).

It is also noticeable from the pressure records that the period of oscillations of pressure generally vary with the impact velocity. The oscillation period reduces as the velocity increases. This is shown in figure (41) for the impact of the 50 mm aluminium piston, for pressures recorded at the first transducer position.
(5.2.1) Description of Nozzle Pressure Records

The pressures measured in the nozzle were similar to those measured and computed by Welsh (1980). He showed the existence of a second pressure peak a few hundred microseconds after the first. A typical trace of pressure with time in the short exponential nozzle is shown in figure (42). Multiple peaks of pressure are again present, and are larger and more numerous than the above predictions. It is also noticeable that these secondary peaks of nozzle pressure coincide with peaks of jet force. This will be further discussed in the section on jet force measurement, section (5.7). Figure (43) shows the nozzle pressure as a function of the piston velocity, and compares the experimental results with the numerical study of Rhyming (1972).

(5.2.2) Discussion of Nozzle Pressure Records

The secondary peaks in the nozzle pressure are thought to be caused by secondary flow processes within the water packet. These pressure peaks are associated with an increased flow velocity, as for the initial pressure rise. Measurements of the forces exerted by the jets, using the impact-plate system, show a clear increase in force, coinciding with the secondary nozzle pressure peaks. An example is shown in figure (44) and is further discussed in section 5.7. These secondary increases in jet force could have an effect on material destruction by a mechanism of pressurisation of cracks, the cracks either existing in the material already or having been originated by the initial impact of the jet. The secondary pressure peaks were argued (124) to be the result of a second
cnamoer pressure do not support this, the second phase of high pressure measured in the impact chamber occurring too late, being some milliseconds after the initial phase, and corresponding to the second phase of high pressure in the nozzle (see figure 39). The secondary pressures are therefore generated by the nozzle flow process.

(5.3) Collimator Pressures

(5.3.1) Description of Collimator Pressure Records

Collimator pressures were measured at the positions indicated in figure (21). A series of pressure records along the length of the fifth collimator, (see Table 3), is shown in figure (45). It will be noticed that the pressure drops rapidly with time at any given station; also the trend is toward a reduced high frequency content in the signals between stations 1 and 3. A plot of the maximum pressure along the collimators is shown in figure (46).

The collimator has the effect of reducing the high pressure generated in the water flowing through the nozzle. Numerical results using the leaky-lagrangian scheme described earlier, indicate that this pressure reduction is almost immediate once the water has entered the collimator. It has been experimentally shown, however, figure (46), that this is not the case. The pressure decays with distance along the collimator, the decay rate being dependent on the pressure of the water before entering the collimator.
The discrepancy between the experimental and the computed results lies in the choice of a zero-pressure boundary condition for the computations, whereas the air ahead of the water is really at a substantial pressure. On emerging from the converging portion of the nozzle, the front of the water is subject to the pressure of the air ahead of it. Any shortfall in pressure in this air relative to the water following causes expansion waves to travel into the water from the water/air interface. This leads to the reduction of pressure in the water. If the water jet is supersonic, relative to the water, when it emerges from the converging portion of the nozzle these expansion waves will be washed downstream with the jet. Thus the head of the jet, in this case, will be at a lower pressure on emergence than later portions of the jet, which were unaffected by the expansion waves.

The effect of the pressure reduction in the water is to produce a more stable jet on emergence from the nozzle. Thus breakup of the jet because of the effect of the surrounding air will be delayed (see chapter (8) for a discussion on jet breakup). This is shown to be the case by photographs taken of the jets from the collimators (section 5.6), with the jets becoming more coherent with increasing collimator length.

The rate of decay of the collimator pressure depends on the initial pressure in the water, and the jet speed. A decay equation for the collimator pressure, from figure (53), is given by:

\[ P(x) = P_{mx} e^{-k(x/l)} \]  

Where: \( P_{mx} \) is the maximum nozzle pressure,
x is the distance along the collimator,
\( K \) is a constant, and is found from figure (46) to be approximately 1.5.

(5.4) Piston Velocities

As indicated previously in section 4.2.2., the piston velocity measurement system using a magnetic sensor produced consistent results. Figure (47) shows a comparison between the theoretical piston velocity and the experimental results plotted to a base of initial driver pressure, for the 50 mm aluminium piston. The figure shows the theoretical piston velocity for the case of a steady expansion of the gas behind the piston. Such an expansion is assumed, instead of an unsteady expansion, because of the relatively long time for the piston to travel along the barrel, (approximately 50 milliseconds), and the multiple wave-reflections which can occur in the barrel during this time. The gas chamber was reduced in volume from 0.01308 m\(^3\) to 0.01248 m\(^3\) when the vacuum chamber was added to the water cannon, this resulting in a very small reduction in the theoretical piston velocity for the same bursting pressure.

(5.5) Jet-Head Velocities

The maximum jet velocity for a given piston velocity for the long exponential nozzle is compared in figure (48) with the theoretical jet velocity given by Rhyming, (1973), (see section 2.1). It will be noticed that the results obtained using air in the nozzle lie substantially below those using a vacuum in the nozzle, and that these latter results are in
the experimental measurement of the jet head velocity for jets produced in air from the conical nozzle. The results are again seen to lie somewhat below the theoretical results, as described above.

A typical output of the jet velocity measuring system, (section 4.2.3), is shown in figure (50), indicating the amount of light transmitted via the pin-holes with time. The trace shows the decay in velocity of the head of the jet with distance from the nozzle as the jet sequentially cuts off the light from the holes. This decay in velocity is plotted in figure (51) for jets from the conical nozzle.

Velocity measurements of jets produced from the conical nozzle with various lengths of collimator attached are shown in figure (52) as a plot of maximum velocity with piston speed. The enhancement of velocity with collimator length is readily apparent. Plots of the decay of velocity of jets produced by the conical nozzle with the collimators attached are shown in figure (53). It will be noticed that the decay in velocity with distance from the nozzle for these nozzle configurations is much less marked than is the case with the shorter nozzles.

Velocity records from the longer collimators indicated an acceleration of the jet some distance from the nozzle exit, as figure (53) shows. This is probably associated with the secondary jetting noted in section 5.2.1, which is itself associated with the secondary pressure peaks in the nozzle.

In late-time velocity measurements, a secondary jet was noticed, some tens of milliseconds after the first jet. The second jet was completely separate from the first jet as shown by the photomultiplier record in figure (54). It will be noticed from this record that the second jet also has a high
Measurements of jet force were not, however, taken at this late time.

Runs were performed with the barrel of the water cannon completely sealed and initially full of atmospheric air. Under these circumstances, high jet velocities (i.e. > 500 m/s) were attainable at quite modest initial driver pressures (e.g. < 0.27 MN/m. (40 p.s.i.)). Moreover the pressures in the impact chamber, when operating under these conditions remained sufficiently low throughout the run to prevent triggering of the transient recorder, and were thought to be only a few atmospheres. The jet velocity remained high over a standoff of approximately five inches, (15 nozzle exit diameters), before any slowing was noted. Figure (55) shows the results of two tests under these conditions. The high velocities associated with this, sealed, water cannon, and the low pressures generated in the impact chamber are encouraging for the design of a practical water cannon. It is thought that compression of air in front of the piston causes a relatively slow acceleration of the water into the nozzle. In this way conditions are approached which more closely match that of the ideal moving water packet discussed in chapter (2).

Another point of note is that the noise generated by the water cannon using this sealed barrel system was much less than with that using an air vent.

(5.6) Jet Photographs

Plate (2) shows the development of the blast wave and air jet preceding the water jet from the nozzle. The shock-
nozzles. The pictures show that the jets break up rapidly after leaving the nozzle. Plate (4) shows, for the case of the short exponential nozzle, the improvement in jet coherence produced by evacuation of the nozzle prior to firing. The attached 'bow shocks' are clearly visible in this case, confirming a supersonic speed relative to atmospheric air.

Plate (3 c,d) shows the jets produced by the long exponential nozzle and by the conical nozzle with a collimator attached. The coherence of the jets is seen to be improved when compared to the jets from the shorter nozzles (Plate 3 a,b). The plates also show that the air jet ahead of the water is overtaken by the water. Plate (5) shows the development of the jet produced by the conical nozzle with one of the longest collimators attached, (collimator number 5, Table 3). It will be noticed that the jet undergoes a violent disruption a short time after it emerges from the nozzle. There is a secondary compression of the air around the jet as shown in plate (5 c), indicating that the disruption is rapid. The initial jet appears to be quite coherent and stable, however, as shown in plate (5 b). The secondary breakup is thought to be caused by the emergence of a high velocity portion of the jet, which strikes the slower moving water ahead of it, thus causing radial motion of the water. This secondary jetting occurs some 300-500 microseconds after discharge of the water, which agrees with the estimated time of disruption from the photographs. Semko (109) also noted that impulsive jets were subject to disruption associated with the varying speeds of discharge of various portions of the jet.

See chapter (8) for a discussion on the breakup of the jets.
Figure (56) shows typical plots of the force registered by the load-washer/plate arrangement described in section 4.2.6. It shows the change in the character of the force measured with standoff distance for the 36mm impact plate, with air in the nozzle.

A plot of jet impact force with standoff distance is shown in figure (57). The nozzle used was the long exponential nozzle. The drop in jet force with distance is readily apparent. By keeping the standoff distance fixed and by varying the impact plate diameter a plot of jet force against plate diameter was produced, figure (58). From both of these figures an increase in jet force occurs when the nozzle is evacuated prior to firing, and especially when the jet itself is discharged into a vacuum. Figure (59) shows the variation of jet impact force with plate diameter. It will be noticed that even at the smallest standoff distance of ten millimetres, the nine millimetre diameter plate recorded less than half of the force recorded by the largest plate. This indicates that the jet is broken up into drops at this distance, there being no evidence of a central core. The plots are made non-dimensional with respect to the maximum jet velocity as indicated by the results of separate, numerical, jet velocity studies (see section 2.2).

Fig (48) shows a plot of the maximum jet velocity with piston impact speed for the long exponential nozzle. This gives the maximum jet velocity to be approximately 10 times the piston impact speed. This compares with a factor of 17.6, given by Rhymings incompressible theory, appendix B.

It should also be noted that the theoretical velocity at the nozzle exit plane decays rapidly, as computed by Edwards and Welsh (1978), and by Glenn (1972) and as experimentally
apparent in the force records, figure (60). A typical velocity decay from the peak velocity to seventy percent of the latter occupies some 15 microseconds. Although the rise time of the transducers is very small (of the order of one microsecond), the addition of the force plates to the front face of the transducer increases this to some forty microseconds, as described earlier, thus it is not likely that the system will respond sufficiently rapidly to indicate the impact pressure, which only lasts for approximately 2.5 microseconds. It is also likely that the peak stagnation pressure force will be missed because of the very short duration of the maximum jet velocity. The force is proportional to the square of the velocity so that this implies a substantial loss in force output. Thus the results of jet force should be regarded as qualitative measurements of the effect of standoff, plate diameter and air or vacuum in the nozzle. The decay in jet velocity with distance from the nozzle exit for jets produced in air was quite marked (see section 5.5). This decay in the jet velocity is mostly responsible for the fall in the measured force with standoff, compared to jets produced in a vacuum.

The relationship between the secondary peaks of nozzle pressure and jet force is clearly shown in figure (44), as mentioned earlier. The pressure peaks coincide with an increase in jet force, and correspond with an increase in the local jet velocity.

This variation in jet exit velocity explains those force records which show an increase in force at long standoff. The faster water catches that which has been decelerated by the air, and this adds to the impulse at long distances, but, being separated in time, does not influence the force records.
The faster moving water exiting later in the discharge phase also causes jet disruption when it catches the slower moving water ahead of it. This may be seen in plate (5.c) for the jets produced by the conical nozzle with the 5th collimator attached.

(5.8) **Wall-Strains and accelerations in the Impact-Chamber**

The motion of the wall of the impact chamber was measured using a strain-gauge and an accelerometer as described in section (4.2.7) in order to determine the effect of this motion on the pressure transducer outputs. Typical traces of the output of the strain gauge, together with the pressure recorded at the equivalent transducer position at the same axial distance from the impacted end of the impact chamber are shown in figures (61) and (62). There is no correspondence between the strain and pressure histories and therefore no implied connection between these quantities. The strain recorded was compared to the strain theoretically given by thick-cylinder theory (99) for the impact pressure recorded. The corresponding graph is shown in figure (63). It will be noticed that the strain measured lies within 30 percent of that predicted, albeit for the limited number of cases and range of pressures tested.

Typical traces of the acceleration of the walls of the impact chamber, section (4.2.7), are shown in figure (64) for both the first and the second transducer mounting positions. Again, there is no correlation between these readings and the pressure oscillations recorded at these positions. The maximum accelerations are of the order of 10,000 g, giving a charge output from the transducers, by way of their
MN/m², or some one percent of the impact pressure. The velocity attained by the transducer may be estimated from the acceleration/time records by finding the area 'under' the curve. This was found to be of the order of 0.25 m/s. This will produce a water pressure of 0.4 MN/m², on the assumption of one-dimensional radial wave motion. Again, this is negligible when compared to the impact pressures being measured.

(5.9) **Blast Overpressures in the Atmosphere**

The signal from the miniature blast gauge is subject to spurious fluctuations because of mechanical oscillation, or 'ringing', of its diaphragm. A typical signal is shown in figure (65a). The solution is to filter the output from the gauges to subtract the high frequency noise associated with the vibrations as described previously in section (4.2.5). The resulting filtered signal is shown in figure (65b). The filtered signal has the characteristic shape of a blast wave, with an initial sharp pressure rise, followed by a more gradual reduction of pressure to a value below atmospheric. The peak overpressure given by the filtered output may be used to find the air pressure inside the nozzle by comparison with known shock tube data (see section 9.3).
It has been shown that the sharp impact of the piston on the water packet produces an oscillatory pressure pulse at the walls of the impact chamber. The maximum pressure generated may be substantially above the theoretical, one-dimensional pressure. If the piston motion is air-cushioned by having a completely closed barrel, then the pressure generated in the impact chamber can be small, less than one percent of the pressure caused by a sharp impact for a piston velocity of approximately fifty m/s. With a vacuum of 3400 N/m² absolute in the barrel, the pressure generated lies between the two cases outlined above, being approximately forty percent of the theoretical impact pressure, i.e. a few hundred atmospheres. Figure (32) shows this variation in the pressure for the cases of air or vacuum in the barrel. The conclusion, therefore, is that the variation of the vacuum in the barrel, and the provision of an air gap for the case of air in the barrel, exerts a strong influence on the pressure history in the impact chamber.

Comparison of the results of jet velocity obtained using a cushioned and non-cushioned impact shows that the former produces jets which are faster, for the same piston velocity. The moderate pressure developed for a relatively long time associated with a cushioned impact gives rise to a greater jet velocity than the higher, oscillating pressure that lasts for only a few hundred microseconds, arising from a sharp impact. Also, by introducing a substantial oscillation in the water packet pressure, and therefore a variation in the velocity of sections of the water packet, the sharp impact detracts from the ideal, uniform profiles of pressure and velocity, of a
pressures that are higher than the cushioned case the impact of the piston will detract from the kinetic energy transferred to the water by excessive compression and heating of the water. Spallation of the leading edge of the water may also occur for the case of the impacting piston as a result of reflection of the initial compressive wave from the front of the water packet. This was noted by Edwards and Farmer (32), dealing with the one-dimensional wave reflections. Interaction with annular relief waves may also produce spallation of the front of the water, as has been previously discussed. The non-ideal mechanisms described above are all detrimental to the production of a high-speed jet and so the improved results of maximum jet velocity obtained when a vacuum is used in the barrel, and the impact is thereby cushioned, may be explained.

(5.10.2) Pressure-Extrusion Effect

After striking the water, the piston is slowed down, and may even rebound. Excess gas pressure behind the piston then pushes the piston and the remaining water ahead of it towards the nozzle. In this case, the flow mechanism becomes predominantly a pressure-extrusion process, and may form a secondary jet, which, as has been shown previously in section (6.4) may be completely separate from the first jet. The time of discharge of this second jet depends on the initial piston velocity, the piston material, the gas pressure and the shape of the nozzle. The second jet produced by a short exponential nozzle, for example, with an aluminium piston at a speed of 50 m/s, occurred some fifteen milliseconds after the first jet.

The pressure-extrusion process described above generates
through the nozzle. The pressure changes associated with the wave motions constitute the second phase of high pressure shown in figure (39).
(6.1) One-dimensional Wave-motion theory

As indicated previously in the introduction, the impact of the moving piston on the water in the impact chamber will generate water-hammer pressures. The pressure rise will cause stress waves to travel through the water, the piston and the walls of the impact chamber. The motion that is imparted to these by the passage of the waves is subsequently altered by wave reflection from boundaries, the continuing motion of the piston (if any) and the interaction between the water, impact chamber and the piston. It is also possible that the pressure in the water will be altered by the compression of the air in front of the piston before the impact. Indeed it is possible that the piston will not actually strike the water. The presence of air in front of the piston will be discussed further in section 8.4.1.

The one-dimensional models used by Welsh for the basis of his computations involved the interactions of waves travelling through the water and the piston. On impact, waves would travel through the piston and the water in a plane manner. The waves would reflect from the free end as tensile/rarefaction waves and, depending on their relative return times, either reduce the pressure at the interface or separate the water and the piston. The first one-dimensional impact model used is defined in appendix C. The models neglect both the radial oscillations caused by the passage of the longitudinal waves and the radial motion of the walls of the impact chamber.

Measurements showed that a major oscillation of pressure occurs in the impact chamber (section 5.1.1). This oscillation is at variance with the predictions of one-
(6.2) Departures from One-dimensional Behaviour

(6.2.1) Elastic Waves in the Piston

The equation for the longitudinal strain in elastic bars is well known (98, 16) and is given by:

\[
\frac{\partial^2 u}{\partial t^2} = C_0^2 \frac{\partial^2 u}{\partial x^2}
\]  

(6.1)

Where: \( u \) is the axial displacement parallel to the axis of the bar, at distance \( x \),

\( t \) is time, and

\( C_0 \) is the wave velocity.

This equation is one-dimensional and therefore cannot deal with radial motions of the bars. The oscillating wake of a pressure wave, as shown analytically by Skalak (110), and experimentally by Davies (29), is due to the radial inertia of the bar. The passage of the longitudinal wave sets the bar in radial motion, the radial strain being related to the axial strain by Poisson's ratio. The theory of Love (79) is applicable in this case. This theory accounted for the radial inertia of bars by assuming that the radial displacement, \( v \), at radius \( r \) is given by:

\[ v = -\nu r \frac{\partial u}{\partial x} \]  

(6.2)

where: \( \nu \) is Poisson's ratio.

This leads to the equation of motion for the longitudinal strain of the bar (Love page 428):
where: \( k \) is the polar radius of gyration of the crosssection.

Conway and Jakubowski (18) give an expression for the pressure at the interface of two identical impacting bars, based on this equation:

\[
\frac{\partial^2 u}{\partial t^2} - \nu k \frac{\partial^4 u}{\partial x^2 \partial t^2} = C_o^2 \frac{\partial^2 u}{\partial x^2} \quad (6.3)
\]

\[
P = \frac{8VC_o L \rho}{\pi} \sum_{i=1,3,5,...}^{\infty} \frac{1}{i(4L^2 + i\pi^2 V^2 k^2)} \sin \left( \frac{i\pi C_o t}{(4L^2 + i\pi^2 V^2 k^2)^{3/2}} \right) \quad (6.4)
\]

Where: \( t \) is time,
\( L \) is the length of the bars,
\( \rho \) is the density of the bars, and
\( C \) is the speed of sound of the bars.

Figure (66) shows this equation plotted as a function of time, for a steel bar 0.1 metres long, 0.0762 metres diameter, initially moving at 1.5 metres per second. (The steel piston, impacting on water at 50 metres per second changes its velocity by approximately 1.5 metres per second, see Appendix C). The effect of the radial inertia term may be seen as the oscillating wake of the theoretically square pressure pulse. The period of the oscillations is related to the time of travel of waves across the radius of the piston and in this case is approximately eight microseconds. The figure also indicates that the theoretical pressure fluctuations induced in the water by these radial oscillations could be a substantial fraction of the theoretical pressure. It should also be noted that the maximum pressure predicted by this
of those radial motions. Comparison with the experimental pressure records do not indicate any correlation between the above oscillations and those experimentally observed.

We can relate the pressure induced in the water by a perpendicular velocity change of the surface in contact with it, $\Delta u$, by the equation:

$$\Delta h = \frac{a}{g} \Delta u \quad (6.5)$$

where: $h$ is the pressure head,
$g$ is the gravitational constant and
$a$ is the speed of sound of the liquid.

This equation, derived from the one-dimensional wave equation, is known as Joukovsky's formula and is directly identified with the water-hammer equation (equation 1.1).

Figure (66) shows that the maximum pressure induced at the interface of two impacting bars by the radial motion of the steel piston is approximately one third of the one-dimensional impact pressure. In addition to providing information on the pressure at the impacted ends of the bars, the theory of Conway and Jakubowski predicts the velocity history at the free end of a bar to which a prescribed impulse is applied. This velocity variation has been used together with equation 6.5 to give an estimate of the pressure induced in the water packet. This is shown in figure (67). (See section 5.1.1 for the experimentally obtained pressures in the impact chamber).

Thus for the impact of a steel piston at 50 metres per second, the pressures generated by its radial motion are approximately $10 \text{ MN/m}^2$, or ten to fifteen percent of the impact pressure. There is again no correlation between these pressures, either in terms of magnitude or frequency, and those observed
(6.2.2) Radial Motion of the Impact-Chamber Walls

The first stress wave travelling through the walls of the impact chamber is produced by the initial pressure rise in the water, at a narrow section at the impacted end. The wave travels through the walls of the chamber and is reflected from the free outer diameter as a tensile wave. The compressive wave also travels along the length of the impact chamber, outstripping the wave travelling through the water. A schematic diagram of the waves at this stage indicating the relative speeds of the waves through the water and the steel is shown in figure (68). The resulting motion of the cylinder due to the wave movements within it is complex; however an estimate of the displacements may be found by applying a method given by Mehta (83). This study examined the motion of a thin annular disc subject to an internal step change of pressure. A 'discontinuous step' analysis was performed, involving splitting the disc into a number of elemental rings. The masses and surface areas of each ring were calculated and the motions of each found when subject to external forces. The motions of adjacent rings produced a strain and thereby a stress, which in turn was converted to a force for the next step of the calculation and so on. The time step used was related to the size of the rings by the wave velocity. The wave velocity used was the spherical wave velocity for mild steel of 6000 m/s instead of the slower, longitudinal or 'bar' velocity of 5200 m/s (11). An pressure of 73.2 MN/m² was applied at the bore of the disc for forty microseconds, this being the theoretical one-dimensional pressure imparted to the
The resulting displacement of the bore is shown in Fig (69). The change in pressure developed in the water due to the motion of the inner surface of the disc, using equation 6.5, is shown in Fig (70). It will be noted that the magnitudes of the oscillation of the pressure caused by the motion of the walls are negligible compared to that of the impact pressure.

The transducer itself is acceleration-sensitive, albeit only slightly, \((0.015 \text{ MN/m}^2/\text{g})\), so a measurement of the motion of the walls of the impact chamber was deemed necessary. Accordingly, a strain gauge was fixed to the outer diameter of the chamber and an accelerometer was attached to a dummy pressure transducer mounted inside the walls of the chamber as described previously. The results confirmed the above computations.

(6.2.3) Radial Waves in the Water

Three separate mechanisms for the generation of radial waves within the water may be proposed:

a) Leakage of water past the piston,

b) Radial waves generated as a result of radial stress relief waves traversing the piston, and

c) Curvature of the upstream diaphragm of the water packet leading to a non-uniform impact of the piston on the water.

Each of these possible mechanisms will be discussed in the following sections.

a) Leakage of water past the piston through the annular gap between the piston and the bore of the impact chamber may be
value by the impact of the piston.

The waves generated will be expansive and will tend to lower the pressure in the water. When these waves meet at the centre of the water they will create a deeper trough of low pressure. Depending on their initial amplitude a negative pressure may be developed which would cause the water to cavitate. The tap water used in the present tests cannot sustain negative pressures, which requires stringent laboratory conditions (12). This process of the generation of cavitation bubbles by the intersection of relief waves is analogous to that occurring in the impact of a drop on a surface, as discussed previously. The pressures predicted by numerical studies of the collapse of cavitation bubbles show a rapid decay of pressure with radial distance from the collapse centre, (55). The pressures generated by the rebound of the collapsing bubbles, also found in numerical studies (50,96) may be greater than the collapse pressures and are thought to be a major contribution to cavitation damage. These pressures are also localised and, although shocks have been photographed caused by the collapse and subsequent rebound of cavitation bubbles, (71), no experimental evidence is available which directly measures the pressures induced by this process. It is not thought likely, then, that the collapse and rebound of cavitation bubbles is responsible for the observed pressure oscillations.

b) Radial pressure waves may be generated in the water as a result of the radial stress-relief waves traversing the piston. These stress relief waves alter the motion of the front face of the piston which is in contact with the water. The change in strain may be substantial; however the period of the pressure oscillations generated in the water will be
c) As the piston approaches the water packet air is compressed in front of it. Using a gap to vent the air in the barrel gives a sharper impact, however air will still remain between the piston and the water, and will be compressed immediately before impact. This pressure rise in front of the water will lead to a deflection of the diaphragm, such that it becomes concave. The piston will therefore contact the outer rim of the diaphragm first and subsequently impact on the remainder, in a time-staggered fashion. Pressure waves, travelling from the outer, first-pressurised water will raise the pressure of the water in the centre. The impact of the piston on this already pressurised water will create pressures in excess of those predicted by one-dimensional theory. This process is akin to that which gives the pressures greater than predicted from the theory, during the impact of a spherical liquid drop on a target, which was mentioned previously.

(6.2.4) Evaluation of the cause of the Experimental Pressure Oscillations

As mentioned previously the experimental and theoretical analysis of the motion of the walls of the impact chamber showed that the pressure oscillations associated with this motion could not account for the observed pressure oscillations. Similarly comparison of the experimental pressure records with the results of the computed pressures due to vibration of the piston eliminates piston vibration as a cause of the pressure fluctuations.

The remaining explanations of the non-ideal behaviour of
as proposed in section 6.2.3. The first of these, that of leakage of water past the piston cannot alone be the explanation for the pressure anomalies as this process must cause a pressure reduction in the water, whereas the pressures recorded have been above the theoretical pressures. The second method of generation of radial waves in the water was by the passage of radial waves in the piston. It has already been noted that a change of piston material does not alter the period of oscillation of the pressure, thus this mechanism may be discounted. The remaining suggestion is that of a non-flat impact between the piston and the water, described in section 6.2.3, caused by the deflection of the thin upstream diaphragms. This mechanism was evaluated using a computer model of the impact between a piston and a curved liquid surface and is described in the following sections.

6.2.5 Modelling of the Impact of a Piston on a Curved Water Surface

A computer model of the piston-water impact was formulated on the basis of an axisymmetric version of the Flux-Corrected-Transport (F.C.T) code due to Boris, Book et. al. (1973-1976) (5,6,7). In common with other finite difference codes F.C.T operates in conjunction with a flow field subdivided into a mesh of cells for which the equations of conservation of mass, momentum and energy are solved repeatedly over a series of small time steps. The main advantage of F.C.T. over other, similar, finite difference schemes, is its ability to handle shocks without undue smearing of the shock front. This feature renders it well suited to a transient flow of the type
The computational mesh is shown in figure (71). The model concentrates on representing a time-staggered impact of the piston on the water and, accordingly, incorporates an idealised piston of infinite length. On impact with a given interface cell this piston produces velocity and density changes appropriate to one-dimensional impact and maintains these values for the duration of the calculations. Contact occurs first at the outermost annulus and proceeds across the interface. The true rate of the radial spread of contact was not known and in practice would depend on the piston velocity and on the degree of concavity in the water surface just before impact. Two rates of spread of the contact were examined in the computer runs, assuming a maximum deflection of the diaphragm of 0.5 and 2.5 mm.

Computed pressure histories at each transducer position are shown in figure (72), and are compared with experimental pressure records. Several of the features of the experimental pressure records were found to occur in the computed results. Notably, a major non-uniformity in the piston-water interface pressure distribution was found to occur, with a centre pressure of more than double the one-dimensional pressure. This resulted in the generation of maximum wall pressures at the first transducer position which were well in excess of the one-dimensional pressure and which came within three percent of the measured values. Furthermore a definite oscillation in the wall pressure is predicted, with peaks quite well synchronised with the experimental equivalents. The predicted troughs, however, though correctly timed, are by no means as deep as those on the measured records. It is believed that these troughs are caused by relief of the high pressures by the leakage of water past the impacting piston as described
The arrival time of the pressure relief wave from the front of the water packet was well predicted by the computer calculations.

The rise time of pressure at the first transducer position is illustrated for both the 0.5 and the 2.5 mm of initial concavity. The less concave case gives a shorter rise time but a slightly lower peak pressure than the second case.

The time-staggered impact model has thus reproduced several of the most important features of the measured wall pressure traces and represents a strong argument in favour of the underlying hypothesis.

(6.3) Experimental One-dimensional Impact

In order to confirm this hypothesis tests were carried out with the upstream diaphragm replaced by a short Nylon piston, as shown in figure (73a). The impact chamber was turned around such that transducer 3 became the upstream transducer. This allowed the Nylon piston to be completely inserted into the impact chamber. The Nylon piston left no radial clearance between itself and the walls of the chamber, being a tight fit in the bore. This piston was then struck by the 100mm aluminium piston fired as usual. Typical pressure records produced by this arrangement are shown in figure (73b). It will be noticed that the character of the pressure histories has been substantially altered from those produced using a diaphragm in front of the water. The pressure histories are more similar to those expected from the one-dimensional theory, which is shown by the dashed line in figure (73b), than the previous experimental pressures. No excursions of
there is no oscillatory behaviour. These results thus lend considerable weight to the proposed model of a non-flat impact of the piston on the water and suggest a means of producing one-dimensional waves in the water packet.
(7.1) Tests with Hollow, Water-Filled Pistons

As indicated in section 2.2.2, computed results have shown that, for a water packet to nozzle length ratio of less than one, a moving water packet entering the nozzle produces a faster jet than a water packet set into motion by the impact of a piston. The fluid piston does not produce such high pressures in the impact chamber and is therefore inherently safer. In view of these advantages, attempts were made to achieve a practical realisation of the theoretical ideal of a fluid piston. Two different approaches were used: hollow pistons filled with water and pistons which pushed water into the nozzle. The design of the hollow pistons is shown in figure (74), three types being used. The first was constructed of aluminium, with a rigid base. The water was held in place by a brass end cap screwed to the aluminium walls and holding a thin (0.025 mm) Mylar diaphragm. The short exponential nozzle was machined to provide a buffer, designed to retard the hollow piston while allowing the water to continue through the nozzle. It was found that at only moderate impact speeds the brass ring became distorted when the piston was stopped. Various buffers were used in an attempt to nullify this problem including a copper 'spring', plasticene and Apeizon 'Q' compound. The jet velocities recorded varied considerably and were generally low. It was thought that a contributor to this could be the effect of the fixed base of the piston, creating negative pressures when the water moved away from it. Thus the second type of piston was developed. This took the form of a tubular Nylon water-capsule sealed at each end by a Mylar diaphragm. Early runs caused damage to the end caps used to secure the Mylar
This is because, owing to the similarity in densities, the inertia forces due to the Nylon and the water portions produced practically equal base reactions. This in turn ensured that the diaphragms were not called upon to transfer load between the water and the Nylon when both were accelerated by the applied pressure.

After several runs with this arrangement, however, the Nylon walls broke and so the third piston was designed. This consisted of a tubular aluminium water-capsule with a mild-steel base which was an internal sliding fit in the tube. On impact with the nozzle the aluminium tube would be retarded while the base, acting as a secondary piston, and the water would continue moving. The length of the moving base was fixed in such a way that under axial acceleration the combined inertia forces of the steel/water column produced a base pressure identical with that created by the aluminium tube. Again this ensured that no diaphragm force was needed when the combined assembly was accelerated axially.

The result of each of these tests is shown in figure (75) as a plot of piston velocity against jet velocity. The figure shows that the results of jet velocities are generally low in comparison with those obtained theoretically. As a result of damage to the hollow pistons only a limited number of runs were carried out. The variation in velocity shown on figure (75) could have arisen from blockage of the nozzle by the upstream diaphragm. The main problem was found to be prevention of solid/solid impact between the walls of the pistons and the nozzle. Even with a buffer material the energy of the rapidly moving piston was mostly expended in distorting itself. Further work is necessary to develop a system capable of delivering a uniform packet of water, at
Fluid Piston Driven by a Rigid Piston

As an alternative to the water being held in the pistons a 'free' packet of water was then used. This was accelerated from rest and driven along the barrel by a piston acting on the rear face of the water, the piston itself being driven by gas pressure as usual.

The barrel of the water cannon was modified to include a section at the breech end which could hold water. This is shown in fig (76). A piston was placed between the water and the gas chamber as shown in the figure, such that when the front diaphragm burst as a result of increase in gas pressure the piston, itself accelerating along the barrel, pushes the water in front of it. A moving packet of water thus enters the nozzle.

This system overcomes the problem of piston damage as a result of collision with the nozzle, however in trials involving various types of driving piston the jet velocities measured were very low. This was assumed to be due to the braking effect of water leaking past each of the variety of piston seals used, and creating a large, retarding, shear force. The absence of a continuous large acceleration would allow gravity to cause the initially flat water front to sag and the resulting contact with the nozzle would be asymmetric.

The various novel systems described above were temporarily abandoned in favour of conventional impact-cumulation, pending the acquisition of improved apparatus, e.g superior sliding or rolling piston seals.
Photographic and other evidence for jet breakup and dispersion

Plate (3 a,b) shows schlieren pictures of the jets produced by the conical nozzle and the short exponential nozzle respectively. It shows that the jets break up rapidly after leaving the nozzles. The jets from the longer nozzles, however, break up less rapidly as may be seen by comparing with plate (3 c,d), which shows the jets produced from the long exponential nozzle. Pictures by Edney (1976) of jets produced by the cumulation device at the C.E.R.A.C. institute show jet dispersion similar to this. Such a rapid breakup implies that the useful damage that the jet could inflict on a target is greatly reduced, as the greatest damage is produced by a coherent jet at high speed. (The impact pressure is maintained for a time related to the cross-sectional area of the impacting jet, and the "quasi-static" stagnation pressure relies on a continuous jet striking the target). The photographs imply that the jet is broken into small droplets, the damage effect of which is small, for both of the above reasons, compared to the unbroken, high-speed jet. Droplet impact on materials at high speed, is well documented in studies of rain on aircraft and turbine blades (65).

Edney suggested that such photographs could be misleading in that there may be a central core of coherent water surrounded, and obscured, by a shroud of drops. This core could be highly effective for cutting purposes. In support of this proposal Edney noted an initial fall in jet velocity, followed by an acceleration. This he took to be the stripping off of a cloud of droplets by the action of the surrounding air, the apparent acceleration being caused by the remaining
droplets was said, (31), to have been caused by 'instabilities within the jet' and the action of aerodynamic forces on the jet. The instabilities, however, were not defined and they are further discussed in section 8.3. Leach and Walker (74) showed by X-ray photography that a continuous jet had this shroud of drops obscuring a central core. Force measurements of the impact of the jets produced in the present study, however, as discussed in section 5.7, show no indication of a central core and so the present jets are assumed to have broken up completely. The breakup of continuous jets is well documented: a useful review is to be found in a paper by McCarthy and Molloy (81). The mechanisms put forward for the breakup of a continuous jet, however, such as the growth of instabilities within the jet, (Rayleigh (100)), take a relatively long time to disturb the jet and therefore cannot explain the very rapid breakup of the unsteady jets from the water cannon.

(8.2) **The effect of air in the nozzle on jet velocity**

The effect of air in the nozzle has been discussed in section 2.2.2. Edney (1976) confirmed the numerical computations of Locher (80), showing no difference in jet velocity using air or vacuum in the nozzle. In contrast, Cooley (24) found a distinct difference, some 20 percent, in the jet velocity depending on whether air or vacuum was used in the nozzle. Cooley's velocity measurement system involved the sequential breaking of two pencil leads placed in the path of the jet.

The present study has confirmed that jets produced with a vacuum in the nozzle can be significantly quicker than jets
is thought to be a result of the reduced dispersion of the jet using a vacuum, the air producing less of a deceleration on the more rod-like jet.

(8.3) **Jet Breakup Mechanisms**

(8.3.1) **Previously Suggested Mechanisms of Jet Breakup**

Field and Lesser (41) gave an explanation for the breakup of supersonic jets, based on wave interaction within the body of the jet. Characteristically, a small region of the front of the jet will have a high velocity and a high pressure, as shown in fig (11), a plot produced using the one-dimensional code of Edwards and Welsh (1978). When the liquid leaves the nozzle the constraining influence of the nozzle walls is removed. If the water jet emerges at supersonic speed then the relief waves travelling through the water will be of the form shown in figure (77). Region 4 is a region of interaction between oppositely moving expansions and is probably cavitated. Field and Lesser indicated that, within one nozzle diameter, an explosive decompression will occur, with the jet breaking into slugs of approximately the same size as the nozzle exit diameter. Jet breakup was also shown to occur some thirty nozzle exit diameters downstream of the nozzle exit as a result of the growth of Taylor instability. This process would occur with all water jets fired into air and would seem to set a limit on their maximum effective range.

Edney produced pictures of supersonic jets showing that the jets produced in a vacuum disperse much less rapidly than
droplets as explained previously, the main body of the jet remaining coherent. Photographs taken by Field and Lesser show also that the jet produced in a vacuum does not show any immediate, gross disturbance. It is apparent, then, that the presence of air is required for the very rapid dispersion of such jets. We can, therefore, distinguish between the breakup of a jet and its dispersion. For jets to break up completely a disturbance of the jet and a mechanism for dispersion are required. Air provides the dispersion, possibly by the Magnus effect (34), or by a non-uniform pressure distribution on the leading surface of the jet (4), which latter mechanism has been found to give the characteristic 'mushroom' shape to the leading edge of the undisturbed jet. The Magnus effect could account for the rapid radial dispersion of the jets. The hypothesis (34) is that the outer portions of the jet acquire an intense vorticity during their rapid motion adjacent to the nozzle walls. Given a mechanism for jet breakage, the resulting particles at the edges of the jet would already possess a high angular velocity which, coupled to their large velocity relative to the air would generate a Magnus-effect lift directed radially outwards.

The jets produced by Edney (1976) were said to be subject to an initial disturbance, which was not defined and which produced the droplet shroud, the remaining part of the jet staying relatively coherent and at a high velocity.

(8.3.2) Present Experimental Evidence for Jet Breakup

Velocity measurements indicated that the water jets being produced by the present work were subsonic, with breakup of the jet occurring very rapidly on exit from the nozzle. Force
The disparity between the present findings and that of previous authors (31,41), in that subsonic jets were found to be dispersing rapidly, led to the explanation of jet dispersion due to the Magnus effect (34), discussed above. Further work showed that the removal of the air in the nozzle greatly improved the coherence of the jet, with the maximum velocity being increased, as mentioned previously. This improvement in coherence may be seen in plate (4), and the increase in velocity in figure (48). Moreover the addition of collimators to the end of the conical nozzle improved the coherence of the jets. The Magnus effect theory could not account for the improvement in coherence caused by the vacuum in the nozzle and by increase in collimator length and so a different explanation was sought.

(8.3.3) Effect of Air in the Nozzle on Initial Jet Breakup

The major difference between the jets produced in the present work and those produced by Field and Lesser is the presence, in the present work, of an air jet ahead of the water jet. Field and Lesser note the presence of an air shock but treat the air pressure as negligible, compared to the much higher pressures in the water, for the purposes of calculating the distance to breakup. Photographs presented by Field and Lesser, comparing the breakup of subsonic and supersonic jets,
said, (41), to be at a pressure of some 10 MN/m$^2$, compared to
the 100 MN/m$^2$ range for the water. Plate (2) shows the
emergence of the preceding air jet and blast ahead of the
water from the short exponential nozzle. The characteristic
diamond pattern of shocks and expansions of a supersonic air
jet may be clearly seen. Thus the wave diagram for the air
jet may be drawn as shown in figure (78). This is the
pressure field into which the water jet emerges. The
expansion wave travelling into the (subsonic) water from the
front of the jet thus reduces the pressure in the water, not
to atmospheric, but to the pressure of the air ahead of the
jet. The water jet is therefore at a substantial overpressure
when it emerges from the nozzle. It is clear that relief
waves from the sides of the jet will meet on the centre-line
to produce regions of cavitation in a similar manner to that
described previously. The jet will therefore be subject to a
substantial disturbance. The corresponding wave diagram is
shown in figure (79). The rapid breakup of the present
subsonic jets may thus be accounted for, with subsequent
radial dispersion of the jet occurring as a result of the
forces imposed on the ablated particles by the surrounding
air. A secondary event is that the front of the water jet
will pass through regions of alternating high and low
pressure, caused by the Prandtl-Meyer expansion plume of the
supersonic air jet. It is likely, however, that the jet will
be broken by this time so that the effect of this is thought
to be only minor.

The rapid fall in jet-head velocity can lead to a false
estimate of the maximum jet velocity. The system devised by
Cooley (24), for example, measured the time of flight between
Measurement will, in this case, give an average velocity, being a function of the initial velocity and the jet velocity decay. This may partly explain why he found a distinct difference in the jets produced with and without air in the nozzle. Edney, on the other hand, found the jet velocity from successive photographs of the jet as it emerged from the nozzle, a more accurate method, which showed little difference in the maximum velocity of jets produced with air or vacuum in the nozzle. Present results have shown that the presence of air in the nozzle is detrimental to the maximum velocity attainable, with a notable improvement in coherence and maximum velocity being observed using vacuum in the nozzle.

Thus there is conflicting evidence concerning the effect of having either air or a vacuum in the nozzle. The first is that of Edney and Locher, who found little change in jet velocity between either method, supported by the measurements of Welsh who found good agreement between his experimental results and the computed results. The second is that of Cooley and the present investigator who both found a significant change in the maximum velocity. The present investigator has identified a possible mechanism for the rapid slowing of the jets with distance from the nozzle; however a unified theory of impulsive jet breakup is given in the following section.
Cushioned and Non-cushioned Impact

The preceding section has introduced a mechanism, based on that for supersonic jets of Field and Lesser (41), for the rapid breakup of subsonic jets; however this itself poses a dilemma: previous work (124) had produced jets that were subsonic and yet had measured velocities close to the computed results with a system using two measuring stations ten centimetres apart. The discrepancy lies in the different flow starting conditions of the various experimental water cannons. A closed, evacuated barrel has been seen, (section 5.1.1), to have given a cushioning of the impact between the piston and the water, the air in front of the piston coming from leakage of the driver gas past the piston. A closed barrel with no vacuum has also been shown to give a large cushioning effect, pressures in the impact chamber being very small. The use of an air gap between the barrel and the water, gives a more impulsive impact, (section 5.1.1). Photomultiplier results have indicated (section 5.5) that the decay in velocity of the jets resulting from a sharp impact is greater than those of a cushioned impact, which show little velocity decay close to the nozzle exit. Moreover the previous results, (34,124) using a cushioned impact lay close to the theoretical results, indicating that the decay in jet velocity was small for this case. Thus it can be concluded that, all other parameters remaining constant, that a cushioned impact produces a jet whose velocity decays less rapidly than that produced by a sudden impact.
Figures (80) and (81) show the piston and water wave trajectories and the state of each region of the water respectively, based on one-dimensional wave motions. The figures show that the rarefactions caused by the reflection of the compressive wave from the leading edge of the water packet will combine with rarefactions transmitted from the rear of the piston to produce regions of fluid at a certain pressure and velocity, with a cavitated zone in between the regions, (32). Similarly annular relief of the high pressures generated in the water packet as described previously may also cause a region of cavitation within the water and combination with the rarefaction from the front of the water will cause spallation of the leading edge of the water. The possibility of spallation of this type has been described previously (23,78). Thus the velocity distribution of the water packet on entering the nozzle will be far from ideal. The non-ideal behaviour is accentuated by the bursting of the diaphragm ahead of the water: it is highly improbable that the diaphragm will tear and fold to the walls so as to expel all the air from its vicinity. A much more likely outcome is that water will flow through the rupturing aperture, leaving pockets of air trapped behind the diaphragm which will be continuously folding and which will eventually expel the trapped air as bubbles into the flow. This air will then be carried along with the water. Negative pressure caused by interaction of expansion waves would promote the formation of bubbles of gas taken out of solution as well as forming cavitation bubbles. The result of these mechanisms is that there is a region of water in which air has been entrained.
nozzle this water will be compressed, pushing air ahead of it out of the nozzle. Any vapour is likely to have been recompressed by the time the water approaches the nozzle exit. Entrained gas, however, will still be under high pressure on emergence even if it emerges at subsonic speed as described in section 8.3.3. It will thus decompress rapidly, causing a disturbance of the high speed jet which, acted upon by the atmosphere will break up and give the lateral spreading seen in plate (3). Longer nozzles, with the same area ratio, will produce jets which are more coherent as a result of the reduced disturbance of the jet, the entrained air bubbles expanding through a smaller pressure difference. This improvement in jet coherence has been noted in section 5.6.

(8.4.3) **Effect of Collimators on Jet Breakup**

Collimators reduce the high pressure of the water which was generated in the nozzle, relief waves travelling upstream from the (less) pressurised air ahead of the water. Compressive waves are still being generated in the nozzle, however, while this pressure relief occurs, so that the pressure history will not remain 'steady' along the collimator. This effect changes the shape of the pressure histories along the length of the collimator, see figure (45). The initial portion of the jet is at a low pressure and is therefore coherent on emergence (see plate 5,b). Figure (82) shows a typical pressure history taken from one of the longest collimators. As mentioned previously the variation in the jet exit velocity may cause enough disturbance to the jet to cause its disruption. Figure (82) shows that the secondary nozzle and collimator pressure
the likelihood is that disruption of the jet will occur, where the velocities are dissimilar, leading to jet breakup. This effect may be seen on plate (5,c) as a disruption of the later portions of the jet.

(8.4.4) Prevention of Jet Breakup

The cushioned piston-water impact produces a slower acceleration of the water, due to the compression of gas in front of the piston; no rapid excursions to high pressure are produced, as indicated previously in section (5.5), and thereby no spallation of the water or internal cavitation. It is thought for this cushioned impact that less air will be entrained as a result of the breaking of the diaphragm, which initially moves more slowly than with the sharp impact, allowing the air surrounding it to escape. The behaviour of the jet on emergence is then precisely as described previously (31), with a cloud of particles being formed around the initial part of the jet. The initial disruption necessary to form this cloud comes from the mechanism proposed earlier, i.e. the interaction of relief waves from the sides of the jet disrupting subsonic jets on emergence from the nozzle.

Air in the nozzle is, then, detrimental to the production of coherent subsonic jets, aiding disruption in two separate ways: the sudden expansion of entrained gas, and the production of interacting relief waves within the body of (subsonic) jets. The breakup will be minimised by giving the water a gradual acceleration instead of an impulsive blow, which minimises the first cause of breakup. The jets will then travel some fifteen nozzle exit diameters before any
to subject to an initial disturbance.
(9.1) Generation of the Air Shock and Blast

The passage of the water through the nozzle generates high pressures not only in the water but also in the air ahead of it. Field and Lesser (41), state, for example, that the air is compressed to a typical maximum pressure of 10 MN/m², as compared to a maximum pressure in the water of 100 MN/m². These high pressures in the air result in a blast wave being formed before the water has emerged from the nozzle, as shown in plate (2). Locher, (1974), allowed for the effects of air pressure ahead of the water packet in an analysis of the flow of water through the nozzle, as mentioned in section 2.2.1. Further to this work, Lemcke and Locher (75) produced a method of generating strong shock waves by attaching a shock tube to the end of the nozzle of an impact-cumulation water cannon. The high speed water acts as a driver for the shock wave. Shock velocities of several thousand metres per second were achieved using this method with a prototype device. Pressures in the 10 MN/m² range were measured in the shock tube, confirming the estimate of Field and Lesser. The jet velocities generated in these trials were significantly higher than in the present work owing to the use of higher piston speeds. The experiments showed that shock speeds closely agreed with shock tube theory for the given jet (interface) velocity.

(9.2) Shock-Wave Theory

This is a well established theory, e.g. ref (80), relating the pressure difference across a shock to the shock velocity. The well known Rankine-Hugoniot relations are applicable to shock conditions inside a shock tube, where a shock may be
how a shock is formed by the acceleration of a driving piston in a gas filled tube and presents the standard equations relating the velocity of the piston with the speed of sound of the driven gas. Figure (83) shows schematically the wave system inside a tube under these conditions. Lemcke and Locher showed that the strength of the shock generated by the water cannon could be found, knowing the jet velocity, the jet acting as the piston as in the above example. As the shock so formed emerges from the nozzle it forms a spherical blast wave.

(9.3) Blast Waves and their decay

The theory of blast waves was first developed by G.I. Taylor (112). A blast wave results from the sudden expansion of compressed gas into the atmosphere. Such expansions follow the detonation of high explosives or the arrival of a shock-wave at the open end of a shock tube. The classic shape of the pressure history of a blast wave is shown in figure (84). Typically the blast wave will exhibit a fast rise to a pressure substantially above atmospheric pressure, followed by a slower fall to a pressure which is below atmospheric, and will continue to be so for a short length of time. Behind the blast the air is set into motion, as is the case for the passage of a shock-wave. The magnitude of these effects, the strength of the blast, depends on the quantity and rapidity of energy released by the gas expansion. Glass (45) researched the decay laws in various geometrical frames and indicated that the peak overpressure of a strong blast decayed according to the geometries of the situation. Thus, for example, spherical blast wave decayed as $R^3$, and a cylindrical
decay of weak blast waves found the following, however:
Spherical decay as:

\[ \frac{1}{R(\log_{e}R)^2} \]  \hspace{1cm} (9.1)

Cylindrical as:

\[ \frac{1}{R} \]  \hspace{1cm} (9.2)

Thus the decay of a blast depends not only on distance but also on the shock strength. Recently, Phan (94) examined the decay of blast overpressure empirically by measuring the blast overpressure at a large number of measuring stations along several azimuthal lines radiating from the exit of a simple shock tube. He found that the rate of peak overpressure decay with distance is a function of the initial driver chamber pressure, the decay being more rapid when the driver chamber pressure was higher. To account for this, the primary shock, \( P_2 \), itself a function of driver pressure, was introduced as a non-dimensional parameter.

It was found that the decay of the blast overpressure could be expressed as:

\[ \frac{P - P_1}{P_2 - P_1} = k \left( \frac{x}{c} \right)^n \]  \hspace{1cm} (9.3)

where:

- \( k \) is a constant,
- \( P_1 \) is atmospheric pressure,
- \( P - P_1 \) is the overpressure,
- \( P_2 - P_1 \) is the primary shock strength
- \( x \) is the distance from the exit of the shock tube, measured along the axis,
- \( c \) is the calibre of the tube,
\( P \) is the pressure downstream of the shock.

From experimental measurements the values of \( n \) and \( k \) were found, giving:

\[
\frac{P - P_1}{P_2 - P_1} = 0.4 \left( \frac{X}{c} \right)^{-1.2} \quad (9.4)
\]

Phan found that the decay of overpressure with distance from the nozzle exit varied with the azimuth angle measured from the shock tube axis. Both \( n \) and \( k \) in the above equation are reduced with increasing angle as the shock is being weakened due to the effect of expansion waves produced by the sharp corner at the exit. Fig (85) shows the resulting shock profile for various corner angles for an incident shock Mach number of 1.5, from reference (112). Phan showed the variation of \( n \) with angle to be approximately linear:

\[
n = -1.2 + 0.134 \theta \quad (9.5)
\]

\( k \) was found to be:

\[
k = 0.4 \times 10^{-0.502 \theta} \quad (9.6)
\]

where: \( \theta \) is the azimuth angle measured in radians.

The general expression for the decay of blast overpressure with distance and angle from a shock tube is, then, given by:

\[
P - P_1 = 0.4 \left( 10^{-0.502 \theta} \right) \left( P_2 - P_1 \right) \left( \frac{X}{c} \right)^{-1.2 + 0.134 \theta} \quad (9.7)
\]

As mentioned in section (9.2) Lemcke and Locher (75) showed that the shock generated by the nozzle flow of a water cannon
Using this knowledge and the above equation it is possible to estimate the blast overpressure at any point around the water cannon.

(9.4) **Blast noise levels**

Blast noise levels may be characterised by a number of different criteria (17):

i) Peak overpressure,

ii) Rise time to peak overpressure,

iii) Pressure wave duration,

iv) Pressure envelope duration.

These criteria are defined diagrammatically in figure (84).

The peak overpressure is the most commonly used definition of noise and may be defined in decibels (dB) by the expression:

\[
\text{SPL (dB)} = 20 \log_{10} \left( \frac{\Delta P}{P_{\text{ref}}} \right)
\]

where: SPL is the Sound Pressure Level,

Pref is the reference acoustic pressure, \(2 \times 10^{-5} \text{ N/m}^2\), taken to be the threshold of hearing, and

\(\Delta P\) is the overpressure.

Damage to the ear occurs at overpressures of about 2 p.s.i. (176 dB), (94), and, the ear being the most sensitive organ to blast noise, this provides a 'ready reckoned' maximum blast. Coles, (17), however, correlating data on recommended noise limits, gives the maximum permissible impulsive noise to be 140 dB. This is quoted as the level at which a 'temporary threshold shift' will occur for approximately ninety percent of people i.e. a temporary loss of hearing.
An experimental impact-cumulation water cannon designed and tested by Cooley (21) was operated in field conditions at a quarry and at a mine. Measurements of the noise were made, recording some 135 to 140 dB at a range of 30 metres (100 ft). The jet velocities employed were of the order of 2500 metres per second.

The present investigator found blast overpressures of approximately 188 dB at a standoff of 0.2 metres, for the jets produced by the conical nozzle, using a piston velocity of 50 m/s. A typical record of the blast measured is shown in figure (65). The experimental overpressure is somewhat higher than that predicted from equation 10.7, which gives, at this standoff, an overpressure of some 178 dB for a nozzle air pressure of 3 MN/m², (30 atm). The discrepancy may be due to equation 10.7 being invalid at this relatively large distance, in terms of calibre, from the nozzle. Alternatively the air pressure in the nozzle may be higher than that expected. Measurement of the air pressure ahead of the water in the nozzle has, however, not been attempted.

It should be noted that the blast overpressure measured is greater than the safety limits described above and it is recommended that precautions are taken to ensure that personnel are protected from this noise.
1) The jet head velocity decay of jets produced by the 
impact cumulation water-cannon can be substantial, leading to 
a greatly diminished cutting potential. This has been found 
to be associated with the bursting of the jet on emergence 
from the nozzle, due to the overpressure of the water. The 
subsequent action of the air on the broken jet provides the 
mechanism for the deceleration of the jet head.

2) An impulsive blow of the piston on the water, as 
envisaged in the classic impact-cumulation water-cannon, is 
detrimental to the production of a coherent jet. This is 
thought to be the result of excessive entrainment of air in 
the water as a result of the effect of the downstream 
diaphragm on the starting conditions of the flow. 
Compression of this air in the nozzle, results in the 
disruption of the jet on emergence from the nozzle due to the 
rapid expansion of the air.

3) The coherence of the jets produced by the water-cannon 
may be improved by the use of a vacuum in the nozzle. This 
means that there is no entrainment of air in the water thus no 
large scale breakage of the jet on emergence from the nozzle.

4) For the case of a sharp piston-water impact the use of a 
vacuum in the nozzle of the water cannon increases the jet 
velocity compared to that with air in the nozzle.

5) The addition of a collimator to the end of the nozzle 
reduces the overpressure of the water, so that the initial 
portion of the jet can emerge as a coherent rod. The pressure
character and the length of the collimator has to be many nozzle exit diameters before any significant improvement of the jet is evident.

6) For the case of a sharp piston-water impact the addition of a collimator substantially increases the velocity of the emerging jet for the same initial piston velocity.

7) A mechanism has been formulated for the rapid initial breakup of subsonic jets, being an extension of the mechanism for breakup of supersonic jets, and which allows for the effect of the air in front of the jet. The pressure of the jet can remain high on emergence as a result of this mechanism and can lead to subsequent disruption of the jet by expansion of entrained air and interaction of radial relief waves.

8) Secondary peaks of jet nozzle pressure coincide with an increase in the local jet velocity. This could lead to an enhanced cutting potential due to pressurisation of cracks in the material. The cracks could have been created by the initial impact of the jet.

Secondary jetting can occur a short time after the initial jet, and separate from it, with the possibility of enhanced material damage as described above. This jetting comes about as a result of a pressure-extrusion process in the nozzle.

9) The pressure in the impact chamber varies greatly depending on the amount of cushioning of the piston impact. For an impulsive blow, in which the air is allowed to escape from in front of the piston, the pressures experienced in the impact chamber can be substantially above the one-dimensional impact pressure. This is a result of the non-flat impact of
10) The noise from the conventional impact cumulation water-cannon presents a major safety hazard. Measurements indicated a large overpressure close to the nozzle, and although this will decay with increased distance from the nozzle it is recommended that operators should be protected from this noise. The use of such a water cannon should be restricted such that it is not used in confined spaces without suitable precautions. The conversion of the piston impact to a cushioned blow substantially reduces the noise from the cannon. With the barrel of the cannon sealed and with air in the barrel the noise is negligible.
1) **Cutting tests**

The present work has identified means of producing fast, coherent jets of water from an impact-cumulation water cannon. The next step is to perform cutting trials on samples of various materials to verify the cutting potential of the system. To this end higher piston velocities than are presently used are required in order to increase the jet velocity.

2) **Long standoff force measurements**

In order to quantify the cutting effectiveness of the jets at long standoff distances, (>0.2 metres to 1.0 metre), a series of force measurements should be performed. Using these measurements and the data from the cutting tests will enable design criteria to be formulated with regard to standoff distances for cutting effectiveness and for safety.

3) **Design of a prototype**

The design and development of a prototype, industrial impact-cumulation water cannon to include a multi-shot capability.

4) **Computing**

Although the one-dimensional code performs well in estimating the nozzle pressures it should be enhanced to include the effect of air in the nozzle. A two-dimensional code would be useful in this regard, and this may also be adapted to model the flow outside the nozzle exit.
The effect of the upstream diaphragm on the starting conditions of the flow through the nozzle should be investigated, with regard to the entrainment of air in the water.

6) **Air Pressure in the Nozzle**

   A direct measurement of the pressure of the air in the nozzle ahead of the water should be made, in order to confirm the numerical computations and to correlate with the measurements of blast noise.
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### Piston Properties

<table>
<thead>
<tr>
<th>Material</th>
<th>Density Kg/m</th>
<th>Speed of sound m/s</th>
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<tr>
<td>Mild steel</td>
<td>7923</td>
<td>5172</td>
</tr>
<tr>
<td>Aluminium</td>
<td>2763</td>
<td>5143</td>
</tr>
<tr>
<td>Nylon</td>
<td>1133</td>
<td>1029-1600</td>
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### TABLE 2

**Nozzle dimensions**

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<tr>
<th>Nozzle</th>
<th>Length (mm)</th>
<th>Inlet diameter (mm)</th>
<th>Outlet diameter (mm)</th>
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</thead>
<tbody>
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<td>76.2</td>
<td>7.5</td>
</tr>
<tr>
<td>Long Exponential</td>
<td>225</td>
<td>76.2</td>
<td>7.5</td>
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<tr>
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<td>152.4</td>
<td>76.2</td>
<td>9.0</td>
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Collimator Lengths:
(See figure 26, All dimensions in mm.)

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<th>B</th>
<th>C</th>
<th>D</th>
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<td>155.0</td>
<td>13.0</td>
<td>38.0</td>
<td>63.0</td>
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</tbody>
</table>
Plate 2: The development of the blast wave and the air jet ahead of the water. Short exponential nozzle.

a) 50 b) 150 c) 250 d) 340 Microseconds
Plate 3: a) Short exponential nozzle
b) Conical nozzle
c) Long exponential nozzle
d) Conical nozzle with collimator
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a) Air
b) Vacuum
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Fig. 1 Comparison between the pressure generated by water impact and by water stagnation pressure with water velocity.
Fig. 2 Impact pressure for an aluminium piston
Fig. 3 Impact pressure for a Nylon piston
Fig.4 The impact of a cylindrical packet of water on a rigid body, showing wave motions and radial jetting.
Fig. 5 The impact of a spherical drop on a rigid target, showing the early-time wave behaviour and the 'contact points'.
Fig. 6 The impact of a spherical drop on a rigid target, showing the wave motion and radial jetting.
Fig. 7 Impact pressure distribution for liquid jet (after Johnson and Vickers)
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(a) Pressure-extrusion
(b) Impact-extrusion
(c) Impact-cumulation
Fig. 9 Schematic diagram of the cumulation process in the nozzle
a) Water packet approaching nozzle
b) Generation of compressive waves
   c) Acceleration of water front, a
Fig. 10 Comparison of the results of experiments by Edney with the computed results of Glenn, for vacuum and air in the nozzle. Velocity timings taken for the jet on emergence and for late-time.
Exponential nozzle, Area ratio 100, Length ratio 4, Initial water packet velocity 140 m/s.

A = 1200 microseconds
B = 1350 "
C = 1456 " (discharge)
D = 1500 "
E = 1650 "

Fig.11 Nozzle Pressure History (after Welsh)
Fig. 12 Artificial Viscosity, effect on a shock, (after Pidsley)
Pressure history, Conical nozzle

Initial water packet velocity 50 m/s
Fig. 14 The maximum pressure with time, short exponential nozzle (after Welsh), showing secondary nozzle pressure peaks. Initial water packet velocity = 28 m/s.
Fig. 15 The Water Cannon

- Piston
- Pressure measurement points
- Chamber
- Barrel
- Gas chamber
- Nozzle holder
- Nozzle
- Piston velocity module
- Impact Chamber
Fig. 16 The end of the barrel and the impact chamber, showing the air vent.
Fig. 17 New Water-Packet Arrangement

DIAPHRAGM

BARREL

SPACER

AIR GAP SECTION

IMPACT CHAMBER

NOZZLE SECTION

water out

water in
Fig. 18 Mounting of the pressure transducers, showing the grease column
Fig. 19 Impact chamber pressure measurement points
Fig. 20 Short exponential nozzle, pressure measurement point
Fig. 21 Collimator pressure measurement points (see Table 3), Dimensions in mm
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a) Diode system   b) Magnetic pickup
Fig. 23 Photodiode measurement of jet velocity
a) actual record b) extended plot of (a)
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Fig. 25 The Schlieren system
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Fig. 27 Blast gauge mounting
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Fig. 30: Calibration traces for the force measurement apparatus, using the impulse delivered by a 1.8 m steel bar dropped from 20 mm. a) 80 mm b) 9 mm impact plates.
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b) present system, air gap
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Fig. 34: Maximum water packet pressure with piston velocity, first transducer position, 0.05 metre aluminium piston.
Fig. 35: Maximum water packet pressure with piston impact, first transducer position, 0.1 metre Nylon piston
Fig. 36 Maximum water packet pressure with piston impact, first transducer position, 0.1 metre mild-steel piston
Fig. 37 Typical plot of the pressure recorded at the three impact chamber transducer positions.
Fig. 38 Maximum pressure at water packet transducer positions 1 and 2, with piston velocity, 0.1 m Aluminum piston.
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a) The impact-chamber at the first transducer position

b) The short exponential nozzle
Fig. 40: Pressure history for the 0.2 m aluminium piston, showing the return time of the longitudinal wave through the piston.
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Fig. 4.2 Typical nozzle pressure history, Short exponential nozzle
Fig. 43 Nozzle pressure with piston velocity, Short exponential nozzle
Fig. 4.4 Correlation between the secondary peaks of nozzle pressure with jet impact force.

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b) Jet impact force
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Fig. 46 Collimator pressure drop
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Fig. 48 Jet velocity with piston velocity, long exponential nozzle, 100 mm aluminium piston
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Conical nozzle
100 mm aluminium piston
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Fig. 52: Velocity of jets from the conical nozzle with collimators added. Numbers refer to the length of the collimators, see table 3.
Fig. 53 Velocity decay with distance of jets from collimators
Fig. 54: Photomultiplier record showing secondary jetting
Fig. 55 Examples of photomultiplier records of jets produced with a closed barrel, indicating only small velocity decay.

Jet Speed

= 450 m/sec.

Jet Speed

= 325 m/sec.
Fig. 56  Typical force records
36 mm plate at 10, 35, 60 and 85 mm standoff
Fig. 57 Force with standoff for air and vacuum in the nozzle
Fig. 58 Force with impact plate diameter, for air and vacuum in the nozzle
Fig. 59 Force with impact-plate diameter at various standoffs
Fig. 60 Force record showing the decay in jet force with time
Fig. 61 Strain of the impact chamber with pressure. 

a) Strain History

b) Pressure History
Fig. 62 strain of the impact chamber with pressure,  
a) Strain history  
b) Pressure history
Fig. 63  Circumferential strain of the outer wall of the impact chamber
Fig. 64 Typical accelerometer records for a) 1st and b) 2nd impact chamber transducer positions.
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Fig. 67 Pressure induced in the water due to axial motion of bar
Fig. 68 Schematic diagram of the initial wave motion in the impact chamber, showing the time-scale of wave travel.
Fig. 69 Displacement of the bore of the impact chamber, after Mehta
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F  Free boundary
R  Rigid—reflective boundary
S  Staggered impact boundary
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Fig. 74 Design of the water-filled pistons

a) Hollow aluminium

b) Hollow Nylon

c) Hollow aluminium with moving steel base
Fig. 7.5 Jet velocities, water-filled pistons

- x Hollow aluminium tube
- Δ "" "", moving base
Fig. 76 Breech end modification to hold water
1 Undisturbed jet
2 Side wave region
3 Corner wave region
4 Underexpanded, cavitated region

Fig. 77 Wave diagram for an emerging, supersonic jet, after Field.
Fig. 78 Air jet expansion
Pa - Air pressure
Pw - Water pressure
Po - Atmospheric pressure

Fig. 79 Pressure wave diagram for subsonic jet on emerging from the nozzle
Fig. 80 One-dimensional constructional diagram of the impact of a 100 mm aluminium piston on the water packet, showing the various flow regimes
Fig. 81 Constructional diagram showing the pressure and velocity of various regimes of the piston and water, defined in Fig. 80.
Fig. 82 Pressure record, 5th collimator, First Transducer Position
Fig. 8.3 Wave system created by an accelerating piston in a gas filled tube.
A – B Blast overpressure
A – B1 Rise time
A – C Pressure duration
A – D Pressure envelope duration

Fig. 8: Classic blast wave and definitions of blast noise (see text)
Fig. 85 Diffracted shock profile around a 90 degree corner, initial Mach no. 1.5, (after Skews)
The derivation of the water-hammer pressure due to liquid impact

Cook's Analysis

Let $V$ be the velocity of a column of water. The kinetic energy of a layer of thickness $h$ and of unit area is:

$$\frac{1}{2} \rho \Delta h V^2$$  \hspace{1cm} (1)

Where: $\rho$ is the density of the fluid.

After impact with a solid surface the velocity energy of this layer is destroyed and is replaced the potential energy which is given by:

$$\frac{1}{2} \beta \rho^2 \Delta h$$  \hspace{1cm} (2)

where: $\beta$ is the compressibility of the liquid,

$P$ is the pressure.

Equating (1) and (2) gives:

$$P = V \sqrt{\rho / \beta}$$  \hspace{1cm} (3)

Now the compressibility, $\beta$, is related to the bulk modulus, $k$, by the expression:

$$\beta = \frac{1}{k}$$  \hspace{1cm} (4)

and also:

$$c^2 = \frac{k}{\rho}$$  \hspace{1cm} (5)

where: $c$ is the speed of sound in the water.

Equating (4) and (5) and substituting into (3) gives:

$$P = \rho. c. V$$
Saint-Venant presented this analysis in studies of the impact of elastic bars. He equated the momentum that a bar had acquired with the force that produced it using Newton's law of motion:

A force, $F$, acting on a bar of area $W$ will produce a compression, $j$, per unit length.

If $Kt$ is the length of the bar which has been compressed in a time, $t$, then the total compression is equal to $Kjt$.

We presume that the speed of the end of the bar is uniform and equals $V$, say. Equating the distance that the end of the bar has moved in the time, $t$, we have:

$$V \cdot t = K \cdot j \cdot t$$

or:

$$V = K \cdot j \quad (1)$$

The force on the end of the bar may be given as:

$$F = E \cdot w \cdot j \quad (2)$$

So that if we equate momentum with force we have:

$$F \cdot t = K \cdot t \cdot w \cdot V \cdot \rho$$

or:

$$E \cdot w \cdot j \cdot t = K \cdot t \cdot w \cdot K \cdot j \cdot \rho \quad (3)$$

i.e.:

$$K = \sqrt{\frac{E}{\rho}} \quad (4)$$

So that $K$ is the speed of sound of the bar, $C$, and equation (3) becomes:

$$\rho \cdot w \cdot C \cdot V = F = P \cdot w$$

or:

$$P = \rho \cdot C \cdot V \quad (5)$$
Rhyming's analytical formulae

The following formulae were given by Rhyming for the incompressible flow of a packet of water through an exponential nozzle.

The maximum speed of the jet, $U_{mx}$, is given by:

$$U_{mx} = U_0 \left\{ \frac{(R \log R)}{Lr} \right\}^{\frac{1}{2}} \quad R \gg 1 \quad (3.1)$$

Where:

- $U_0$ is the initial water packet velocity,
- $R$ is the area ratio of the nozzle and
- $Lr$ is the length ratio of the nozzle, defined as the ratio of the lengths of the nozzle and the water packet, $L/l$.

The maximum pressure, $P_{mx}$, generated in the nozzle is related to the maximum velocity by the expression:

$$P_{mx} = \frac{1}{4} \left\{ \frac{1}{2} \int U_{mx}^2 \right\} \quad (3.2)$$

The velocity decay at the nozzle exit is a function of time and the nozzle shape and is given by:

$$U(t) = U_{mx} \left\{ 1 + \frac{1}{2} \int^t \right\}^{-1} \quad (3.3)$$

Where:

$$\beta = \left\{ \frac{21Lr}{R U_0}(\log R) \right\}^{\frac{1}{2}} \quad (3.4)$$
The one-dimensional impact model of Welsh

The stress generated in the piston by the impact on the water is given by:

\[ \sigma = \rho_p C_p (V_o - V_i) \]  \hspace{1cm} (1)

Where:  
- \( V_o \) is the piston velocity before impact,
- \( V_i \) is the piston velocity after impact,
- \( C_p \) is the velocity of stress waves in the piston,
- \( \rho_p \) is the density of the piston.

The pressure, \( P \), in the water is given by:

\[ P = \rho_w C_o V_o \left( 1 + \frac{2V_o}{C_o} \right) \] \hspace{1cm} (2)

With subscript 'w' referring to the water.

The velocity at the interface, \( V_i \), is common to both the water and the piston. The pressures are also equal so that we can equate equations (1) and (2), to obtain:

\[ V_i = \frac{\left[ -\left( \rho_w C_w + \rho_p C_p \right) + \left( \rho_w C_w - \rho_p C_p \right)^2 \right]^2}{8 \rho_w \rho_p C_p V_o^2} \] \hspace{1cm} (3)

This is, then, the impact velocity.

Thus, for example, a steel piston, initially travelling at 50 metres per second will have an impact velocity of 48.39 metres per second, i.e. it will have slowed down by only 1.61 metres per second.
The Formation of a Shock by a moving piston

Consider an infinite tube, filled with a perfect gas and fitted at one end with a piston, as shown in the diagram below. By moving the piston towards the gas a compression is produced in the gas. The gas will acquire the velocity of the piston.

Consider now the piston moving with velocity, \( u \), and having its velocity increased in small steps, \( \Delta u \). The elementary compression waves produced by this motion will be propagated with the velocity of sound relative to the gas, i.e. with velocity:

\[
\frac{dx}{dt} = u + a
\]

relative to the tube.

Considering the changes of pressure and momentum of a small element of gas of mass,

\[
\frac{dm}{dx} = A \rho \frac{A}{dx},
\]

a relationship is obtained between the pressure and velocity change due to an elementary wave:

\[
A \Delta P = dm \Delta u \quad (1)
\]

In the limit:
Since, for an ideal gas:
\[ C_p \, dt = \frac{dp}{\rho} \]
then (2) may be written:
\[ C_p \, dt = a \, du \quad (3) \]
and, as
\[ a^2 = \gamma \, R \, t \quad , \text{and,} \quad C_p = R(\gamma/y - 1) \]
then:
\[ du - 2 \, da/(\gamma - 1) = 0 \quad (4) \]

Initial conditions are \( U = 0, \ a = a \), so (4) becomes:
\[ u/a - 2a/(\gamma - 1)(a) = -2/(\gamma - 1) \quad (5) \]
i.e. by knowing the local velocity, \( u \), we can determine the speed of sound, \( a \).

The equation \( dx/dt = u + a \) is called the Characteristic Equation, it shows that, relative to the tube, each wave is propagated in a linear manner.

For an accelerating piston the \( x/t \) diagram is, then, shown as:

\[ X \]
\[ shock \]
\[ \frac{dx}{dt} = a + U_p \]
\[ \frac{dx}{dt} = a_o \]
\[ U_p \]
\[ t \]
of sound causes a shock-wave to form. This, then, is the
cause of the formation of the blast from the water cannon, the
blast being simply the shock-wave expanding in the atmosphere.