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How to cite this thesis
AN INVESTIGATION OF STEADY AND UNSTEADY FLOW IN PIPELINES FOR MINE HYDRO POWER SYSTEMS

deur
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THESIS

Voorgelê ter vervulling van der vereistes vir die graad

MAGISTER IN INGENIEURSWESE

in

UNIVERSITY OF JOHANNESBURG

INGENIEURSWESE

in die

FAKULTEIT INGENIEURSWESE

aan die

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NOVEMBER 1984
ABSTRACT

This thesis considers in detail the applicability to hydro power systems of the theories of steady and unsteady flow in pipelines. In doing so it highlights some of the shortcomings of these theories. An attempt is made by way of experimentation on a high pressure pipeline, to model some of the conditions which could occur in a full size future hydro power system. These experiments provide some quantitative data about the performance of some typical hydro power components such as pipes, orifices and valves, under steady and unsteady conditions.

A computer program is included which was used to provide theoretical data to compare with the experimental results. The program was found to be limited in its capacity to provide accurate simulation of the experimental pipeline, but this was thought to be due to the pipeline not correctly modelling a hydro power system.

Conclusions presented in this thesis will be of assistance to designers of future hydro power systems and to researchers continuing this work.
ACKNOWLEDGEMENT

The author gratefully acknowledges the assistance of the following people who have made this thesis possible.

DR. D.G. WYMER - whose supervision and guidance has been greatly appreciated.

PROF. H.J. LE ROUX - for his direction and helpful advice.

MR. K. HYDE - for his technical assistance in setting up the experimental equipment.

The work described in this thesis was performed as part of the research and development programme of the Research Organization, Chamber of Mines of South Africa.
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part I

theoretical approaches
to hydro power
1. **INTRODUCTION**

One of the most important aspects in the mechanization of stoping operations in gold mines is the provision of power in a suitable form. Hydraulic power is widely regarded as the ideal powering medium for stoping machinery, since high forces are needed in confined spaces.

Experience with conventional oil hydraulic systems under mining conditions has shown that it is impossible to prevent the leakage of oil. This leads to unnecessarily high fluid costs, pollution of the ore, and other disadvantages\(^1\).

As a result it was decided to pursue the most suitable alternative which was considered feasible at that time. This was based on the use of dilute oil-in-water emulsions which contained ninety five per cent water. These emulsions were relatively cheap and did not pollute the ore. However very little standard equipment was available which was suitable for use with these emulsions, and several years were spent subsequently on the development of suitable hydraulic equipment. Whilst some further development is needed, much of this equipment can now be used successfully in practical stoping systems.

This work led to attempts to run machinery on water alone as the hydraulic fluid. Little use could be made of most of the equipment developed for emulsions, so development of equipment to run on water alone started. Besides totally eliminating the problems of fluid costs and pollution of the ore, considerable simplification of the hydraulic system was possible.

Until this time it was assumed that the hydraulic power required would be generated by electrically driven pumps, which have certain basic disadvantages, the most
serious being that all the pumps and electrical equipment associated with electro-hydraulic power packs are generally unreliable in mining environments, expensive, and require excessive maintenance. This is aggravated substantially when considering the use of water alone. It has been found by Trew and Wymer (2) that high pressure water pumps are critically affected by the impurities in mine service water. Another disadvantage is that all of the electrical power introduced into the stopes is converted into heat.

Joughin (3) suggested that there was an alternative source of hydraulic power available for stoping machinery operating on water alone, pointing out that it would be possible to obtain the power directly from the cooling water which was piped to the stoping areas from surface.

It is likely that, in the near future, stoping machines can be developed that will operate successfully on water alone. If the chilled water for this purpose is carried from the surface in a continuous pipeline, all the disadvantages of electro-hydraulic power can be avoided completely. The hydrostatic head generated by the difference in elevation between the surface and the stope can be conveniently used to power machinery and at the same time provide the cooling necessary in the stopes.

If hydro power systems are to become a commonly used method of powering machinery in gold mines, it will be important to ensure complete reliability and safety without jeopardizing the economic viability of such systems. This can only be done if a complete understanding of all the mechanisms operating in the system is available. The scope of this investigation covers only the high pressure distribution pipes from the surface to the stoping machinery, and the components associated with this pipework, since the hydro power
systems envisaged would make use of the existing methods of returning the water to the surface. In this thesis, the general form of an envisaged hydro power system is first described. Then, existing steady-state and unsteady-state theories, relating to the performance of a hydro power system, are considered, and some of the shortcomings of these theories are highlighted.

A series of experiments is described, which were carried out to determine the steady-state characteristics of a pipe section, typical of that which might be used in a hydro power system. Further experiments were conducted on orifices and valves, to determine their steady state characteristics.

Measurement of acoustic wave velocities were carried out on a high pressure flexible hose to determine the wave velocity, since no theory is available for this.

Experiments are then described which were conducted to verify the unsteady-state theory, and to collect data on components for which no theory is available.
2. HYDRO POWER SYSTEMS

In many South African gold mines the working areas, or stopes, are situated at very great depths, typically between 1 500 metres and 3 000 metres below the surface. If, as suggested, chilled water is brought directly to these stopes from surface in a continuous pipeline the static pressure available for the powering of hydraulic machinery would be between 15 MPa and 29 MPa. The static pressure at any point in the system is defined by the following equation.

\[ P = \rho gh \]

where \( P \) = static pressure
\( \rho \) = density
\( g \) = acceleration due to gravity
\( h \) = elevation difference.

The unusually great elevation changes found in South African gold mines result in very high static pressures. The exact pressure available for the powering of machines will depend on the frictional losses in the system. If the case arises where this pressure is too low it could either be boosted on surface using centrifugal pumps or be increased at the point of usage by an intensifier. In cases where pressure is in excess of that required, the pressure could be controlled by using pressure regulating valves suitably positioned within the system.

In a hydro power system it is necessary to consider the power available for machinery along with the amount of cooling that can be provided. Wymer et al. (4) investigated this aspect and showed that even in relatively cool gold mines the volume of water required for cooling the stopes is greater than that required for powering machines regardless of whether conventional or mechanised mining is used. This means that the cooling
water used in mines is sufficient to provide power to machinery and cool the stopes without additional water being required.

An artists impression of a typical future hydro power system is shown in Figure 1. It consists of:

(i) a surface conditioning and refrigeration plant, which cleans the water returned from underground and cools it to approximately 5°C,

(ii) a high pressure pipeline which takes the chilled water from surface to the bottom of the shaft, the wall thickness of this pipe varying with depth below surface,

(iii) a number of high pressure distribution pipes which distribute the chilled water from the shaft bottom to the working areas, where they then branch to the individual machines.

Once used by the machines the used water drains into the existing underground collection dams. From here it is pumped in the medium pressure shaft pipes to higher levels and eventually to surface reservoirs, using a series of high lift centrifugal pumps. Since no extra volume of water is required the existing return lines and pumps would be adequate.

The working face in a South African gold mine is divided into panels which are typically between 20 and 50 metres in length. The total working face could be as much as 8 km in length. This means that a hydro power system would be required to provide power to up to 200 panels. A shaft system would consist of a main shaft and one or more sub-vertical shafts with mining operations extending up to 2 km from the shaft. A typical piping layout for a complete shaft system is shown in Figure 2.
1. Precipitation basins
2. Conditioning plant
3. Cooling towers
4. Refrigeration plant
5. High pressure pipe column carrying chilled water
6. Isolating and flow / pressure control valves
7. High pressure pipelines to stapes
8. Water powered rockdrills hand held or mounted
9. Water jet stope cleaning guns
10. Water powered gulley conveyor
11. Water powered face conveyor
12. Water powered impact ripping machines
13. Water driven ventilation fans in stapes
14. Water - hydraulic props
15. Distribution points with flow control valves
16. Low pressure water from stapes
17. Centrifugal dewatering pumps

Figure 1 AN ARTISTS IMPRESSION OF A TYPICAL FUTURE HYDRO POWER SYSTEM
Pipe diameters
- 300mm
- 200mm
- 150mm
- 100mm
- 50mm

Combined pressure regulating, flow control, isolating valves
Pressure relief valves
Isolating valves
Combined isolating/flow control valve

Figure 2 LAYOUT OF A TYPICAL HYDRO POWER SYSTEM FOR A SINGLE SHAFT
To provide hydraulic power to machinery, sufficient pressure must be available in any part of the system, regardless of the system demands. The pressure available at any point is dependent only on the losses in the system and the elevation at that particular point. The elevation at each point in the system is fixed, so the pressure available at each machine will depend on the flow rate in every part of the system. To account for the flow changes which will occur, and the elevation changes in the system, pressure control valves are required.

A pressure control valve can be a device as simple as an orifice or an adjustable throttle valve. Alternatively it could take the form of a regulating valve which automatically compensates for changing conditions and maintains the downstream pressure within prescribed limits. The choice and positioning of control valves in the system depend greatly on the design of the particular system and the pressure changes which can be tolerated at the machines. In certain cases such as a sub-vertical shaft where the elevation change is often so large as to raise the pressure above the useful level, a regulating valve at the top of the shaft rather than the bottom would allow lower pressure pipework to be used.

In any system of pipes it is obvious that individual pipes or sections need to be closed down for maintenance or in case of an emergency. Isolating valves are required so that this can be done with minimum disruption to other parts of the system and positioned accordingly.

In the event of a pipe rupture, the flow in the upstream pipes will increase until all of the available hydrostatic head is lost in friction. This condition is commonly known as runaway. Unless the flow is controlled by some means, the effects of this can be
hazardous to personnel and equipment. The normal method of controlling this flow is by use of a flow control valve, which can be a device such as an orifice sized to give an acceptable pressure loss under normal operating conditions, but to give a reasonable level of control under runaway conditions. Preliminary calculations for a typical hydro power system indicate that a properly sized orifice could limit the flow in an emergency to around five times the normal maximum flow rate. This is sufficient in many cases but, where more strict control is required and automatic isolation of the damaged line is preferable, a special type of valve is required. A flow control valve of this type can limit the flow to as little as around 120 per cent of the normal maximum and completely isolate the damaged pipe within seconds. The use of an orifice has the advantage of being completely fail safe, and in many cases a combination of both types of device could be a good choice.

Valve malfunction within the system can lead to greater pressures than designed for. To prevent the occurrence of a hazardous situation, pressure relief valves are required at appropriate points in the system. The most commonly used type of relief valve is a direct acting spring loaded relief valve. The disadvantage with this type of device is that it has limited speed of operation, and if a very sudden rise in pressure were to occur the valve would open only after the pressure had exceeded the relief valve setting. One relief valve which does not suffer from inertia effects is commonly known as a safety rupture diaphragm. It consists of a thin metal foil clamped in a special holder. When the pre-set pressure is reached the foil bursts venting the fluid to atmosphere. The advantages are that its speed of operation is many times faster than a spring loaded relief valve and it is intrinsically fail-safe. One disadvantage is that, unlike a spring loaded relief valve, it does not automatically reset once the pressure is returned to normal.
3. ANALYSIS OF STEADY-STATE CONDITIONS

For any point in the system at a given elevation, the steady-state pressure and flow is dependent only on the friction losses in the system. These losses can be calculated using a number of alternative methods. A choice has to be made of which methods are best suited to a particular application since each has its own advantages and disadvantages.

3.1 Friction in Pipes

The flow of water in pipes is usually turbulent and highly complex, being composed of many randomly fluctuating components which are superimposed on the main flow. As a result, no complete theory has been developed for the analysis of turbulent flow and even the most advanced theories are based partly on experimental information.

It is accepted that the relationship derived by Darcy (5) is the best equation to use when calculating the losses in pipes, and is given by:

\[ \Delta P = \frac{f l v^2 \rho}{2D} \]  

\[ \begin{align*}
\Delta P &= \text{head loss (Pa)} \\
L &= \text{pipe length (m)} \\
\rho &= \text{fluid density (kg/m}^3) \\
v &= \text{mean velocity in the pipe (m/s)} \\
D &= \text{pipe internal diameter (m)} \\
f &= \text{friction factor (non-dimensional)}
\end{align*} \]  

In this equation it is necessary to know how the friction factor, f, varies with Reynolds number and pipe surface roughness. Reynolds number, \( R_e \), is a non-dimensional coefficient defined by the equation:
\[ R_e = \frac{vD}{\mu} \]

where \( \mu \) = kinematic viscosity (m\(^2\)/s).

The surface roughness is a more complex phenomenon to define. There are a number of variables such as the size, shape and spacing of the irregularities to be taken into account. Many different methods have been proposed, the majority being highly complex, but none have become widely accepted. In a great number of cases the pipe is assumed to be smooth and the surface roughness is ignored altogether, which is considered to be unacceptable for the purpose of this work. Where roughness is considered, the most common way of accounting for it is by means of the simple expression

\[ e = \frac{\varepsilon}{D} \]

where \( e \) = ratio of average height of the irregularities to the normal inside diameter of the pipe.

\( \varepsilon \) = average surface roughness.

The most well known method for calculating the friction factor is the Colebrook and White equation(6). This equation is generally accepted as being suitable for the range of Reynolds numbers from 4 000 to 10\(^8\), and is given by the expression:

\[
\frac{1}{\sqrt{f}} = -2\log\left[\frac{\varepsilon/D}{3.7} + \frac{2.51}{Re^{1/4}}\right] \quad \cdots \cdots \cdots (2)
\]

However this equation is implicit, and \( f \) can only be calculated by trial and error. Olujic(7) considered eight alternative equations and compared them with experimental data for Reynolds' numbers in the range 4 000 to 10\(^8\) and concluded that the most practical equation to use, since it was explicit and accurate, was given by Shacham's equation(8).
\[
f = \left( 2 \log \left[ \frac{e}{3.7} - \frac{5.02}{R_e} \log \left( \frac{e}{3.7} + \frac{14.5}{R_e} \right) \right] \right)^{-2} \quad \text{...... (3)}
\]

This equation is thought to be the most suitable of those considered.

In the range of Reynolds numbers up to 4000 the flow is either laminar or in the critical zone between the laminar and turbulent zones. Friction losses in this range are unlikely to be of any interest because these losses would be considered negligible when compared with the average system pressure, and can be ignored.

3.2 Friction in Pipe Fittings

A generalized expression for friction losses in fittings is given by:

\[
\Delta P = \frac{K v^2 \rho}{2} \quad \text{........... (4)}
\]

It can be seen that equation (1) is a special form of this equation for pipes in which \( K = fL/D \). One method of allowing for pipe fittings is known as the "equivalent length" method. This method uses the Darcy equation to represent the loss in the fitting by use of an equivalent length of pipe \( L_e \), which is the length of straight pipe which would have the same friction losses associated with it. It has the advantage that the system then converts into a single pipeline for the purposes of calculating the losses. However, it has the limitation that the variation of the friction factor with Reynolds number is assumed to be the same as for pipes. This is not necessarily valid since the variation of the friction factor with Reynolds number depends to a great extent on the exact geometry of the fitting.
An alternative method known as the "velocity head" method uses equation (4) in which a value of $K$ is assigned to the fitting concerned. Values of $K$ for commonly used fittings are available from published experimental data (9). This method has the advantage that these values are available for a wide range of Reynolds numbers and fitting geometries.

A typical hydro power system such as that shown in Figure 2, involves relatively large lengths of pipes and relatively small numbers of fittings, such as tees, elbows and valves. Preliminary calculations have shown that the pressure losses associated with simple fittings are small in comparison with the losses in the pipes and can in most cases be neglected. The pressure losses associated with components such as isolating, pressure control and flow control valves will tend to be more significant and probably cannot be neglected. These components are best handled using the velocity head method, equation (4), rather than the equivalent length method. The variation of $K$ with Reynolds number and percentage valve opening is available in a few cases from published empirical data. In other cases it would be necessary to conduct experiments on the valve in question to derive the required values of $K$.

3.3 **Friction in Orifices**

As mentioned previously, orifices are the simplest form of pressure and flow control valves. To calculate the pressure losses associated with them they are treated in the same way as valves, namely by the velocity head method described in section 3.2, in that they are assigned a value of $K$ in equation (4) which is dependent on the Reynolds number and the geometry. As far as the geometry factor is concerned, the relevant parameters are the diameter and the length of the orifice. In a hydro power system it is envisaged that Reynolds numbers up to $10^8$ will have to be considered, the upper end of
the range being applicable to the control of a runaway situation following a pipe failure. Since the orifices would be used for controlling rather than measuring the flow as is the normal application, it would be necessary to consider orifice diameters down to 30 per cent of the pipe diameter. Strength considerations dictate that the length-to-diameter ratio must be relatively large, possibly up to a value of 5. The most commonly used expression for the flow characteristics of an orifice is given by the following equation (10).

\[ Q = C_d \frac{\pi d^2}{4} \frac{1}{\left[1 - \left(\frac{d}{D}\right)^4\right]} \sqrt{\frac{2(P_2 - P_1)}{\rho}} \]  

**Q** = flow rate \((m^3/s)\)  
**Cd** = non-dimensional orifice coefficient  
**d** = orifice diameter \((m)\)  
**D** = pipe diameter \((m)\)  
**P_1** = upstream pressure \((Pa)\)  
**P_2** = downstream pressure \((Pa)\)  
\(\rho\) = fluid density \((kg/m^3)\)

(Note: the non-dimensional coefficient, \(C_d\) is often referred to as the discharge coefficient but this is not strictly true because it takes into account other losses beside those at the point of discharge).

This equation is essentially the same as equation (4) but in this case the constant \(K\) is now expressed as a function of the orifice diameter and the pipe diameter in the following way:

\[ K = \frac{1}{C_d^4} \left[\left(\frac{d}{D}\right)^4 - 1\right] \]  

\[ \]
The expression is normally used for sharp-edged orifices in which the length-to-diameter ratio is negligibly small. For this case the variation of the coefficient $C_d$ with Reynolds number has been extensively investigated and values have been published by the British Standards Institute\(^{11} \). However, investigations of long orifices have not adequately covered all possible combinations of length-to-diameter ratios and Reynolds numbers. For example, work done by Lichtarowicz et al\(^ {12} \) in which his own experimental data were compared with work done by others, covered length-to-diameter ratios up to 10 but was limited to a maximum Reynolds number of $10^5$.

Figure 3 shows the variation in the coefficient $C_d$ with Reynolds number for length-to-diameter ratios from 0.5 to 10. The coefficient shows a steady rise from almost zero, at low Reynolds numbers, to an almost constant value when the Reynolds number exceeds 10,000, the exception being the 0.5 length-to-diameter ratio which shows a marked decrease in coefficient from a Reynolds number of around 700 to 2,000.

Lichtarowicz concluded that for Reynolds numbers greater than $2 \times 10^4$ the orifice coefficient is no longer a function of Reynolds number and depends only on the length-to-diameter ratio. Figure 4 shows a graph of the coefficient $C_d$ against various length-to-diameter ratios up to 10, for the case when Reynolds number is greater than $2 \times 10^4$. Because of the rapid decrease in the coefficient $C_d$ below a length-to-diameter ratio of 1.5 Lichtarowicz recommended that orifices with length-to-diameter ratios of less than 1.5 should be avoided, since this could otherwise lead to uncertainties in the calculation of orifice flow characteristics.
Figure 3 VARIATION IN ORIFICE COEFFICIENT WITH REYNOLDS NUMBER, ACCORDING TO LICHTAROWICZ (12)
Figure 4 VARIATION OF THE ULTIMATE VALUE OF ORIFICE COEFFICIENT WITH LENGTH TO DIAMETER RATIO, ACCORDING TO LICHTAROWICZ (12)
For a length-to-diameter ratio of between 2 and 10, Lichtarowicz proposed the following empirical relationship:

\[ C_d = 0.827 - 0.0085 \frac{l}{d} \] ............ (7)

where \( l \) = orifice length
\( d \) = orifice diameter
4. ANALYSIS OF UNSTEADY CONDITIONS

Unsteady flow is that in which the velocity and other quantities at any particular point in the fluid change with time*. In the case of water the rate at which such changes occur determines the type of unsteady flow and the methods by which it can be analysed. When these changes occur slowly this condition is commonly referred to as surge. A typical example of this would be the flow of water between two reservoirs at slightly different levels. At the other extreme, velocity changes can occur so quickly that the forces associated with these changes are sufficient to cause the water to deform elastically. This condition causes compression waves to propagate throughout the system at very high speeds. These waves are called hydraulic transients and their effect is commonly known as "water hammer".

In general, water hammer becomes severe when the length of the pipes is great and the fluid velocity is high. It is envisaged that these conditions will occur in a hydro power system in a gold mine. One of the important consequences of water hammer is the occurrence of pressures higher than the normal system pressures, as calculated for steady state conditions. It is clear therefore that hydraulic transients must be taken into account if a mine hydro power system is to be designed properly.

* Strictly speaking turbulent flow is always unsteady due to local variations in velocity as described in Chapter 3. However, it is generally agreed that turbulent flow can be regarded as steady if the averaged values of velocity and other quantities at any particular point do not change.
4.1 Analysis Techniques

One of the simplest methods of calculating water hammer is by the use of Joukowski's equation(13) which is an approximate expression relating pressure increase to the change in velocity.

\[ \Delta P = C \nu \rho \] ...... (8)

where \( \Delta P \) = change in pressure
\( C \) = pressure wave propagation velocity
\( \nu \) = change in velocity
\( \rho \) = density of water

This expression is based on many simplifying assumptions. The more important of these are as follows:

(i) the friction losses are negligible,
(ii) the velocity changes instantaneously,
(iii) the pipe is rigid, and pressure changes do not alter the diameter,
(iv) the flow is one dimensional,
(v) the pipeline remains full of water at all times,
(vi) the pressure is uniform over any cross section, and is the same as at the pipe centre-line,
(vii) the stress waves propagating in the water do not interact with those propagating in the pipe wall.

These assumptions, particularly the first three, severely limit the applicability of Joukowski's equation.

Alleivi(14) produced an improved method of calculation in which the rate of velocity change was taken into account, using the equation:

\[ P = P_0 \left( \frac{N}{2} + \sqrt{\frac{N^2}{4} + N} \right) \] ...... (9)
\[
N = \left( \frac{\rho L v_0}{P_0 t} \right)^2
\]

\( \Delta P \) = change in pressure
\( P_0 \) = initial pressure in pipe
\( \rho \) = density
\( L \) = length of pipe
\( V_0 \) = initial velocity
\( t \) = time for valve closure

In order to take into account pipe friction and the pipe expansion due to increased pressure a much more sophisticated technique is necessary. This technique is based on the equations of dynamic equilibrium and continuity \((15)\).

\[
\frac{\partial V}{\partial x} + \frac{1}{g} \left( \frac{\partial H}{\partial x} + \frac{\partial v}{\partial t} \right) + f |v| v = 0
\]

\[
\frac{\partial H}{\partial x} + \frac{\partial v}{\partial t} + \frac{v}{2gD} = 0
\]

where
\( V \) = velocity
\( H \) = piezometric head
\( C \) = pressure wave propagation velocity
\( g \) = acceleration due to gravity
\( f \) = Darcy friction factor
\( D \) = pipe diameter
\( x \) = distance along pipe axis
\( t \) = time

These are quasi-linear hyperbolic partial differential equations and are impossible to solve as they stand. Therefore, it is necessary to convert the equations using a method which yields an approximate solution. The method most widely used is the method of characteristics \((16)\). This method converts the partial differential equations into ordinary differential equations which are then solved by an explicit finite
difference technique. The equations in their ordinary
differential form are:

\[
\frac{dQ}{dt} - \frac{gA}{c} \frac{dH}{dt} + \frac{fQ}{2DA} = 0 \quad \cdots (12) \text{ Valid only if } \frac{dx}{dt} = C
\]

and

\[
\frac{dQ}{dt} - \frac{gA}{c} \frac{dH}{dt} + \frac{fQ}{2DA} = 0 \quad \cdots (13) \text{ Valid only if } \frac{dx}{dt} = -C
\]

where \( Q \) = flow rate

\( A \) = cross sectional area of the pipe

The particular finite difference technique chosen was
the method of specified time intervals. This method has
the advantage that friction can be adequately accounted
for and that boundary conditions can be easily and
conveniently included. Before solution of the above
equations is possible in any particular pipe it is
necessary to calculate some of the terms.

The time interval \( dt \) is calculated using the expression:

\[
dt = \frac{dx}{C}
\]

where \( dx \) = smallest pipe length in the system

\( C \) = pressure wave propagation velocity for the
pipe.

The value \( dt \) used is the smallest value obtained after
evaluating the above equation for every pipe in the
system. Strictly speaking the relationship \( dx/dt = C \)
has then to be satisfied for every other pipe in the
system. Since \( C \) is constant for any pipe and \( dt \) has
been fixed, then \( dx \) could be calculated. However, for
any particular pipe its length will not normally be
divisible by this length \( dx \) an integer number of times.
To account for this it is necessary to select the
nearest integer number such that:

\[
\frac{L}{CN} \geq dt \quad \cdots (14)
\]
where \( L \) = length of pipe
\[ N = \text{number of integer divisions, ("reaches")}\]

This means that the length \( dx \) selected will in most cases be slightly smaller than it should be. The resulting error in \( dx \) is then accounted for by linear interpolation. Although this linear interpolation introduces a further approximation into the method, at least the error in \( dx \) is not ignored entirely.

Consider for example a system consisting of three pipes of different diameters, and assume the wave velocity is different in each pipe.

![Diagram of three pipes with different lengths and wave speeds](image)

Pipe No. 1 wavespeed = 1000 m/s
Pipe No. 2 wavespeed = 1200 m/s
Pipe No. 3 wavespeed = 1300 m/s

The smallest ratio of pipe length divided by wavespeed occurs in pipe number 3 and is equal to 0.00231 seconds.

Using the relationship in equation 14, pipe number 2 is divided into six reaches of 2,769 metres and pipe number 1 into ten reaches of 2,308 metres. This means that pipe 2 is now 16,615 metres instead of 17 metres and pipe 1 is 23,077 metres instead of 25 metres.

The pressure wave propagation velocity \( C \) is calculated using the following equation, which is derived in Appendix A.

\[
C = k_1 k_2 \sqrt{\frac{1}{\sqrt{\rho (nK + WE)}}} \quad (15)
\]
where \( k_1 \) = factor to account for pressure and temperature
\( k_2 \) = factor to account for dissolved gases
\( k \) = modulus of elasticity at 20°C and 0 MPa
\( \rho \) = density at 20°C and 0 MPa
\( E \) = modulus of elasticity of the pipe material
\( \psi \) = term used to account for pipe geometry and type of anchoring.

The friction loss factor \( f \) in equations 12 and 13 is the Darcy friction factor calculated using Shacham's equation.

Having calculated some of the terms the equations are then solved for a particular time \( t \) at discrete points along each pipe as defined above. The analysis is best accomplished by use of a digital computer, since the pipe system is usually quite complex in both its geometry and its operation. The use of the method of characteristics to analyse the transient behaviour of piping systems is well documented (17) and the accuracy of the results obtained has been verified experimentally in many cases. However, it is thought that this work may have limited applicability to hydro power systems as envisaged in gold mines since the previous work has been carried out in the relatively low pressure range. Hydro power systems are expected to be operated at much higher pressures, and introduce new boundary conditions.

It is well known that the accuracy of the method depends on the ability to specify correctly the boundary conditions and on the calculation interval used (18). Small time intervals obviously give the best accuracy but can require large amounts of computer time and this would until recently have necessitated the use of a main frame computer, making the analysis inconvenient and expensive. Micro computers are now readily available which are able to analyse even complex pipe systems.
4.2 Boundary conditions

For any uniform pipe length a characteristic grid can be constructed using length intervals $x$ and time intervals $t$ as follows:

The conditions at any one of the grid points can then be calculated by solving the differential equations 12 and 13 along the positive and negative characteristics. In order to do this it is necessary to know the steady state conditions at time $t = 0$, and the upstream and downstream boundary conditions. Common boundaries in a pipe system are:

(i) Reservoirs
(ii) Pumps
(iii) Series pipe junctions (consisting of step changes in pipe size)
(iv) Multiple pipe junctions (a number of pipes at one point of equal or varying diameters)
(v) Isolating valves
(vi) Orifices
(vii) Pressure control valves
(viii) Pressure relief valves
(ix) Rupture diaphragms
(x) Surge tanks
(xi) Air release valves
(xii) Dead ends
(xiii) Turbines
Each boundary, whether simple or complicated, has specific characteristics. An example of a valve characteristic is shown in the figure below:

Valve open \( \rightarrow 1.0 \)

Friction loss parameter

Valve closed \( \rightarrow 0 \)

4.3 The Computer Program

The program was written in Hewlett Packard Basic 2.0 for use on a Hewlett Packard 9826A micro-computer. The program was based on an existing program written by Sheer et al (19), for a CDC 6400 mainframe in FORTRAN and which was a global program encompassing many components and pipework configurations which were unnecessary for the analysis of hydro power systems.

Development of the program was undertaken on the basis that certain types of boundary conditions would occur in mine hydro power systems and that it would be necessary to consider these accurately. Several other modifications to the original program were made to improve the suitability for a hydro power system in a gold mine. A program listing and flow chart are presented in Appendix B. The particular boundaries used in the program are:

1. Series pipe junctions
2. Multiple pipe junctions
3. Dead ends
4. Nozzles
5. Isolating valves
6. Orifices
7. Rupture discs
8. Relief valves
9. Pressure control valves
Boundaries (i) to (v) are relatively simple to account for and are well documented. Boundaries (vi) to (ix) are documented but there are many conflicting opinions as to the method of accounting for their presence. These items are required in a hydro power system and the analysis can only be used confidently if there is sufficient information available for the treatment of these components. For the purpose of this study it was assumed that under transient conditions the steady state friction factor associated with these components remained valid even under transient conditions. Using this assumption the characteristic for each of the components is shown in Figure 5. In this illustration a transient in the form of a sine wave is chosen by way of example.

4.4 Illustration of the Use of the Computer Program to Analyse a Pipe System

The particular system chosen for analysis is shown schematically in Figure 6. This represents a simplified version of a pilot hydro power system currently being installed at No. 3 Shaft, Kloof Gold Mining Company Limited. It comprises a vertical section in the main shaft, a horizontal section in the haulage way on 23 level and a third section in the stope. The components included are a pressure control valve, a number of flow limiting orifices and two water jet stope cleaning guns. Each gun is represented by a valve, a pipe and a nozzle.

The minimum pipe length chosen was 25 metres. In reality the system contains pipe reaches down to a few centimetres in length, but if this minimum length were used the system would be divided into so many elements that unreasonable computer time would be used. In practice therefore a greater minimum length has to be used and this involves adjustment of component positions. In order to determine a reasonable interval
Figure 5 ASSUMED CHARACTERISTICS OF BOUNDARY CONDITIONS IN COMPUTER PROGRAM
Figure 6 SCHEMATIC OF SYSTEM ANALYSED BY THE COMPUTER PROGRAM
to be chosen, an analysis was carried out using minimum lengths of 3 metres and 25 metres respectively. For this purpose it was assumed that the valves in both cleaning guns were closed simultaneously, one in 0.3 seconds the other in 0.38 seconds, both with linear characteristics. Figure 7 shows the transient caused by this event at point A in the system (immediately upstream of the lower water jet gun).

It is clear from the Figure that a minimal increase in accuracy is obtained in going from 25 to 3 metres and yet the program took 21 times longer to run. It was considered therefore that using a 25 metre minimum length gives a result sufficiently accurate to be used for analysis.

The analysis of the system was carried out for two cases to illustrate the effect of certain components:

(i) the pressure regulating valve set to regulate downstream pressure to 17.5 MPa,

(ii) the pressure regulating valve and the orifices removed from the system.

Figure 8 shows the results for each case at points immediately upstream of the lower water jet gun and the regulating valve respectively (points A and B in Figure 6). It shows clearly the effect of the orifices and pressure control valves on the results. It shows that theoretically the pressure control valve and the orifices introduce significant attenuation of the hydraulic transients.

These results are then compared for position B case (ii) with the simplified results of Joukowski and Alleivi in Figure 9. The Figure shows clearly the large differences between the values obtained by using two
Figure 7 THE THEORETICAL RESULTS AT POSITION A IN THE SYSTEM FOR A MINIMUM PIPE LENGTH OF 3 METRES AND 50 METRES
Figure 8 THEORETICAL RESULTS FOR POSITIONS A AND B IN THE SYSTEM, WITH AND WITHOUT ORIFICES AND PRESSURE REGULATING VALVES.
simplified methods and the method of characteristics. It suggests that the methods of Joukowski and Alleivi are far too simplified to give meaningful results.

Figure 9 COMPARISON OF THE COMPUTER RESULTS WITH THE SIMPLIFIED RESULTS OF JOUKOWSKI AND ALLEIVI
part II

development of experimental facility and procedures
5. **INTRODUCTION**

It has been shown in Section 3 that there are many methods available for steady-state analysis of the system. It is not however known at this stage how good these theoretical methods are when applied to the type of components to be used in a hydro power system.

Section 4 has shown clearly that in developing the computer program for the unsteady state analysis a number of simplifying assumptions have been made. These appear not only in the mathematical treatment but also in the way in which the system components have been handled. It is not possible to judge how accurately this program will predict the performance of a mine hydro power system without experimental verification.

It was decided to construct an experimental pipeline in the Laboratory as a first step in the verification process. Ultimately it would be necessary to monitor a real hydro power system, but at the time of conducting the experiments, no system had been installed in a mine.

The experimental system in the laboratory consisted of a high pressure pipeline, 2" nominal bore, with a fast closing valve at one end and comprehensive instrumentation. The required flow rate of water through the pipeline was provided by a low pressure centrifugal pump for steady state tests, and a high pressure reciprocating pump for the unsteady-state tests. A schematic of the system is shown in Figure 9.
Figure 10  SCHEMATIC OF EXPERIMENTAL CIRCUIT SET UP FOR HIGH PRESSURE OPERATION
6. **EQUIPMENT**

6.1 **Power Units**

6.1.1 **High pressure**

The high pressure power unit consisted of an electrically driven triplex plunger pump, (Hammelmann HDP220 P70) capable of delivering 471 l/min at a pressure of up to 20 MPa. The pump was supplied from a 5000 litre tank using a small centrifugal boost pump. A photograph of the power unit is shown in Figure 11(a).

6.1.2 **Low pressure**

The low pressure power unit consisted of an electrically driven multi-stage centrifugal pump capable of delivering 900 l/min at a pressure of up to 1.2 MPa. This pump was inserted into the circuit in place of the triplex plunger pump when required for steady state tests.

6.2 **Pipeline**

A photograph of the pipeline is shown in Figure 11(b). A detailed drawing, indicating the position of each of the pipe sections and each transducer, is shown in Figure 12. The majority of the pipeline consisted of pipe with an outside diameter of 60.3 mm and an inside diameter of 42.9 mm (conforming to Schedule 160 2" NB piping). This was connected by a series of 90° Bends to a 4" NB pipe leading to a tee with a dead branch*.

---

* This tee and branch served a dual purpose, the first being to trap air in the dead branch to act as an accumulator and damp out pump pulses, the second and most important being to provide a large ratio of pipe areas so that transients would be reflected from this point.
Figure 11  PHOTOGRAPHS OF THE HIGH PRESSURE EQUIPMENT SHOWING THE POWER UNIT AND THE PIPELINE WITH THE PNEUMATIC BALL VALVE INSTALLED.
Figure 12  DETAILED DRAWING OF TEST SECTION OF PIPE
The main section of the pipe was straight for a distance of 40,375 metres and was terminated by a fast closing valve. From here, 60 mm low pressure galvanised pipe was routed back along the high pressure pipe to the tank. With the exception of the 4,772 metre section of pipe containing the 90° bends, all the high pressure pipe was connected by flanges. These flanges were of a slip-on design which required welding internally and externally. Sealing of the flanges was effected by means of an O-ring. The straight 40 m section was loosely supported on a number of stakes at equal intervals. Care was taken to ensure that the supports did not restrict the longitudinal movement of the pipe.

6.3 Fast-closing Valve

Two types of fast closing valve were tried as described below.

(i) Pneumatically actuated ball valve

A photograph of the valve installed in the pipeline is shown in Figure 13, along with a schematic of the valve in Figure 14. The valve consisted of a 50 mm high pressure ball valve, a pneumatic rotary actuator, a solenoid operated 2-way pilot valve and a pneumatic pressure regulator. Application of a pilot voltage to the solenoids caused air pressure at approximately 0.7 MPa to be directed to either side of the rotary actuator which in turn operated the ball valve.

(ii) Manually operated poppet valve

A photograph of the valve installed in the pipeline is shown in Figure 15, along with a schematic of the valve in Figure 16. The flow
Figure 13 PHOTOGRAPH OF PNEUMATICALLY ACTUATED BALL VALVE

Figure 14 SCHEMATIC OF PNEUMATIC ACTUATOR
Figure 15 PHOTOGRAPH OF THE QUICK CLOSING POPPET VALVE

Figure 16 SCHEMATIC OF THE QUICK CLOSING POPPET VALVE
through the valve passed between the valve cylinder and the valve poppet. This restriction caused a net force on the poppet in the direction of the valve seat. The poppet was restrained by a rod, controlled by a catch mechanism. When the catch was released the poppet accelerated towards the valve seat until it reached its closed position. The flow through the valve was only restricted in the final stages of the valve movement. The theoretical values of closure time, and distance before valve poppet has an effect on flow can be found in Appendix C part(a).

6.4 Instrumentation

It was decided to use a data acquisition and processing system that had been constructed for future use with a full scale hydro power system, namely the pilot system currently being installed at Kloof gold mine. This was more than adequate for the purpose, and it gave the additional advantage in that it afforded an opportunity to commission and test the monitoring system prior to its use underground. A block diagram of the system is shown in Figure 17.

The data acquisition system consisted of a Hewlett Packard 6942A multiprogrammer comprising a main frame memory and a series of plug-in control cards. Control of the multiprogrammer was achieved by use of a Hewlett Packard 9826A desk top computer, and the results could be produced in printed and graphic form.

A current to voltage converter was installed in the system to convert the 4 to 20 mA signal from the transducers into a 0 to 12V analogue signal required for the multiprogrammer. The strain gauges gave a voltage signal which was amplified to give a 0 to 10V signal to the multiprogrammer. The data system was located some
Figure 17  MONITORING SYSTEM BLOCK DIAGRAM
distance from the pipeline and was linked to the individual transducers using a 150 metre eight-twinned-pairs screened cable. Details of the most important transducers are as follows:

(i) Pressure transducers

Five Hottinger Baldwin Messtechnik P7-500 two wire absolute pressure transducers were used with an operating range of 0 to 50 MPa, and a maximum overload of 100 MPa. The manufacturer quoted an accuracy of ± 0.5% of the full range. An automatic check of the transducer accuracy could be carried out using a pre-set check signal equivalent to 25 MPa. All components were hermetically sealed in a stainless steel body.

(ii) Temperature transducers

A Gould model T3000 temperature transducer was used consisting of a platinum resistance temperature sensor, and hermetically sealed electronics in a stainless steel case. The temperature range was set at 0°C to 150°C. The manufacturer quoted an accuracy of ± 0.2% over the full range. The probe was provided with a stainless steel high pressure thermowell to protect from the effects of high pressures.

(iii) Flowmeters

Three Bestobell bi-directional turbine flow meters type M2/1250 size B30, B90, B120 each different size being used over the optimum flow range specified. They were used in conjunction with the recommended flow straighteners and a frequency to current convertor. The flowmeter gave a repeatability of 0.5% and a linearity of
(iv) Strain gauges

Two Micro-measurements CEA stacked rosette strain gauges were used, each having a resistance of 350 ohms and a nominal gauge factor of 2.12. Each of the rosettes was combined with a temperature compensation rosette, to form one half of a full bridge.

Various other pieces of equipment were used from time to time and are described as they occur in the text.
7. EXPERIMENTAL PRELIMINARIES

7.1 Calibrations

The system was first pressure tested to 40 MPa for 1 hour. Air was then purged from the system by bleeding the pipeline at its high points and then recirculating the fluid for 3 hours. All the pressure transducers were calibrated using a dead weight gauge tester.

Figure 18 shows the calibration carried out for the B30 and B120 flowmeters against the originally installed B90. All three were factory calibrated but as an additional check they were calibrated against each other. It can be seen from the figure that they all agree to within acceptable limits.

Because the pipe strain was required under dynamic conditions it was necessary to calibrate the strain gauges in position on the pipe. The pipeline was pressurized from 0 MPa through to 27 MPa using a small hydraulic hand pump, and at a number of different values of pressure, the longitudinal and circumferential strain readings were taken. Figure 19, shows the output from the multiprogrammer against the water pressure. These results were used to calibrate the strain, since under static conditions the theory can be used to accurately calculate the true strain(20).
Figure 18  CALIBRATION CHART FOR FLOWMETERS

- B30 Serial No. Z009/78
- B120 Serial No. Z021

Ideal curve

Flow rate (litres/min.)

B90
Figure 19 CIRCUMFERENTIAL AND LONGITUDINAL STRAIN CALIBRATION GRAPH
7.2 Pipe Roughness Measurements

Ten samples of 2 inch nominal bore pipes were selected from the assembly off-cuts. Each was measured at eight points to determine its wall thickness and inside diameter. The results are shown in Table 1 and from this the following average values were calculated:

- mean inside diameter: 40,998 mm
- mean wall thickness: 9,911 mm

The internal surface roughness of each pipe sample was then measured in an axial direction. The traces are shown in Figure 20, together with the average surface roughness calculated on each trace. From these the average surface roughness was calculated as 12,79 pm.

7.3 Valve Closure Characteristics

The accuracy of transient analysis depends on the ability to correctly describe the closure characteristic of the valve causing the transient. When describing a valve it is usual to describe it with its inherent flow characteristic, which is the characteristic measured at constant pressure drop. This is because the manufacturer of the valve has no way of knowing the pressure at which it will be used in practice. Few valves operate at a constant pressure drop so it is necessary to know the installed flow characteristics, which is unique for each valve installation.

To measure the characteristics of any valve it is necessary to know how K (from equation 4 Section 3.2) varies with valve stroke, and how the stroke varies with time. So in each case two sets of independent results are required.
### INSIDE DIAMETERS (mm)

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### WALL THICKNESS (mm)

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Table 1

RESULTS OF INSIDE DIAMETER AND WALL THICKNESS MEASUREMENTS OF TEN SAMPLES OF PIPE RANDOMLY SELECTED FROM CONSTRUCTION OFF-CUTS
Figure 20 TRACES OF ROUGHNESS TAKEN FOR TEN DIFFERENT PIPE SAMPLES
(i) Pneumatically actuated ball valve.

For ball valves the variation of $K$ with stroke can be found in the literature. The fully open valve of $K$ is however different in every installation and needs to be measured. Confirmation of the published results was also desirable and so a complete set of measurements was taken.

To measure the variation in $K$ with valve stroke a steel plate was attached to the valve body which was marked in 1 degree intervals from 0 degrees through to 90 degrees. A pointer was then attached to the valve handle to enable the rotation of the valve to be measured in any position. The valve was rotated manually and the upstream pressure and flow across the valve measured for a number of valve angles twice in succession. In calculating $K$ due allowance was made for the pipe pressure loss between the measurement points.

The results are shown in Table 2 along with the calculated valve function as defined in Appendix C, part(b). Figure 21 shows the variation in $K$ against angle of rotation, and Figure 22 shows the calculated valve function.

Figure 21, shows a similar characteristic to that of Miller (9), the only difference being that in this case the inside diameter of the valve was 50 mm and the pipe was only 41 mm, so the valve moved 22.92° before there was any effect on the pressure drop. This is clearly seen in Figure 22.

To measure the variation in the valve stroke with time a linear rotary potentiometer was
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<td>669 607</td>
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<td>4,48</td>
<td>2 483 127</td>
<td>1 535 676</td>
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<td>2 370 031</td>
<td>1 448 796</td>
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<td>70</td>
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<td>5 061 246</td>
<td>4 113 795</td>
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<td>74</td>
<td>1</td>
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<td>8,93</td>
<td>10 095 787</td>
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<td>16,86</td>
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<td>169 667 752</td>
<td>0,0766</td>
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<td>348 506 992</td>
<td>0,0535</td>
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<td></td>
<td>2</td>
<td>-</td>
<td>-</td>
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<td></td>
</tr>
</tbody>
</table>
Figure 21  MEASURED CHARACTERISTIC OF BALL VALVE, WHERE 
\[ K = 2 \frac{\Delta P}{V^2} \]

NOTE: SUPPRESSED ZERO ON HORIZONTAL AXIS

Figure 22  DERIVED BALL VALVE CHARACTERISTIC
fitted to the valve. The valve was then closed pneumatically, firstly with no pressure in the pipeline then with the relief valves set at 10 MPa. The results are shown in Figure 23.

Under zero pressure conditions the valve closes in an almost linear manner in 0.061 secs. When the system is operating normally the pressure rises upstream of the valve as it closes. The valve starts to close rapidly while the pressure rises. The last 14 per cent of closure then becomes extremely slow, taking 0.12 secs to close. This is due to flow forces through the valve increasing as the valve is just about to close. This shows the differences in characteristic that can be expected even under steady state conditions. If transients occurred during this closure period this curve would be further modified.

(ii) Manually operated poppet valve.

To measure the variation in K with valve stroke a bracket was attached to the valve so that the stroke could be varied manually, and accurately measured using a vernier. The variation in K was then obtained through the full movement of the valve, which was approximately 41 mm, by measuring the upstream pressure and flow. It was found that only the final 2.3 mm of the valve stroke had any significant effect on K. The results for this final portion of the stroke are shown in Table 3 which also shows the valve function calculated as described in Appendix C part(b). The valve function against valve closure for the final 2.3 mm is shown in Figure 24.
Figure 23 CLOSING CURVES OF BALL VALVE UNDER TWO DIFFERENT CONDITIONS

Upstream pressure rising to a maximum of 10MPa

Fully open

Fully closed

Zero pipe pressure
<table>
<thead>
<tr>
<th>Valve Movement Stroke</th>
<th>Upstream Pressure (kPa)</th>
<th>Flow rate (l/min)</th>
<th>$K$ (between measurement points)</th>
<th>$K$ (across valve)</th>
<th>Valve function $V_f$</th>
</tr>
</thead>
<tbody>
<tr>
<td>2.30</td>
<td>1.00</td>
<td>439</td>
<td>198.6</td>
<td>139.6</td>
<td>0</td>
</tr>
<tr>
<td>2.19</td>
<td>0.95</td>
<td>459</td>
<td>190.9</td>
<td>158.0</td>
<td>18.4</td>
</tr>
<tr>
<td>2.07</td>
<td>0.90</td>
<td>517</td>
<td>189.3</td>
<td>181.0</td>
<td>41.4</td>
</tr>
<tr>
<td>1.96</td>
<td>0.85</td>
<td>540</td>
<td>182.7</td>
<td>203.0</td>
<td>63.4</td>
</tr>
<tr>
<td>1.84</td>
<td>0.80</td>
<td>565</td>
<td>172.0</td>
<td>239.6</td>
<td>100.0</td>
</tr>
<tr>
<td>1.73</td>
<td>0.75</td>
<td>620</td>
<td>157.0</td>
<td>315.6</td>
<td>176.0</td>
</tr>
<tr>
<td>1.61</td>
<td>0.70</td>
<td>640</td>
<td>134.0</td>
<td>447.3</td>
<td>307.7</td>
</tr>
<tr>
<td>1.50</td>
<td>0.65</td>
<td>667</td>
<td>130.6</td>
<td>490.7</td>
<td>351.1</td>
</tr>
<tr>
<td>1.38</td>
<td>0.60</td>
<td>679</td>
<td>120.0</td>
<td>591.7</td>
<td>452.1</td>
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<tr>
<td>1.27</td>
<td>0.55</td>
<td>688</td>
<td>107.3</td>
<td>749.9</td>
<td>610.3</td>
</tr>
<tr>
<td>1.15</td>
<td>0.50</td>
<td>695</td>
<td>96.0</td>
<td>946.4</td>
<td>806.8</td>
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<tr>
<td>1.04</td>
<td>0.45</td>
<td>710</td>
<td>85.3</td>
<td>1224.6</td>
<td>1085.0</td>
</tr>
<tr>
<td>0.92</td>
<td>0.40</td>
<td>720</td>
<td>69.0</td>
<td>1897.9</td>
<td>1758.3</td>
</tr>
<tr>
<td>0.81</td>
<td>0.35</td>
<td>730</td>
<td>58.7</td>
<td>2658.9</td>
<td>2519.2</td>
</tr>
<tr>
<td>0.69</td>
<td>0.30</td>
<td>741</td>
<td>48.7</td>
<td>3921.1</td>
<td>3781.5</td>
</tr>
<tr>
<td>0.58</td>
<td>0.25</td>
<td>750</td>
<td>38.7</td>
<td>6284.7</td>
<td>6145.1</td>
</tr>
<tr>
<td>0.46</td>
<td>0.20</td>
<td>751</td>
<td>26.4</td>
<td>13523.1</td>
<td>13383.5</td>
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<tr>
<td>0.35</td>
<td>0.15</td>
<td>768</td>
<td>19.0</td>
<td>26699.2</td>
<td>26559.6</td>
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<tr>
<td>0.23</td>
<td>0.10</td>
<td>780</td>
<td>17.6</td>
<td>31601.9</td>
<td>31462.3</td>
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<tr>
<td>0.12</td>
<td>0.05</td>
<td>800</td>
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<td>35728.3</td>
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<tr>
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<td>0.0</td>
<td>0</td>
<td>0</td>
<td>0</td>
<td>0</td>
</tr>
</tbody>
</table>

Table 3 RESULTS TO MEASURE CHARACTERISTIC OF POPPET VALVE

Figure 24 DERIVED CHARACTERISTIC OF THE QUICK CLOSING POPPET VALVE
The variation of stroke with time required the fitting of a linear displacement transducer but with such a long stroke the transducer tried was not responsive enough. Instead a black and white flag was attached to the valve stem and a Zimmer optical displacement transducer was used. Figure 25 shows the output traces obtained for each of the four tests carried out. The measured valve characteristics are compared in Figure 26.

These graphs show the four closing times as being:

12,45 ms.
17,83 ms.
19,10 ms.
19,67 ms.

The closing time was also determined theoretically (Appendix C, part(a), and was found to be 21,3 ms. Which was in reasonable agreement with the experimental results.
Figure 25 RESULTS OBTAINED FOR POPPET VALVE MOVEMENT USING ZIMMER OPTICAL DISPLACEMENT TRANSDUCER
Figure 26 POPPET VALVE MOVEMENT FOR EACH OF THE FOUR TESTS CARRIED OUT
8. PROCEDURES

8.1 Steady-state Friction Losses

The multi-stage centrifugal pump capable of giving 900 litres per minute at 1.2 MPa was installed at the upstream end of the pipeline. Three pressure gauges, one 600 kpa, one 250 kpa and the other 100 kpa were calibrated using a dead weight gauge tester. The 100 kpa and 650 kpa gauges were installed at positions H and A respectively in Figure 12. Whenever the pressure at position A fell below 250 kpa the 650 kpa gauge was replaced with a 250 kpa gauge.

The flow through the pipeline was varied and the pressure drop readings taken, from which the friction factor, $f$, was calculated using equation (2). The pressure gauges were then replaced with two KIowa 0 to 2 MPa pressure transducers, an amplifier and an X-Y plotter. Readings were taken at various flow rates for both transducers, and the friction factor was calculated using equation (2). The original transducers were then refitted and a number of readings taken using the instrumentation described in Section 6.4.

Then four orifices were manufactured in carbon steel, each with a length of 45 mm and with diameters of 8, 10, 12, and 14 mm. Each orifice was concentric about the centre line and care was taken to ensure as sharp an entry and exit as possible. Each of the orifices was placed in turn in the line, the pressure 1,204 metres upstream and 0,098 metres downstream was measured, while the flow rate was varied by by-passing part of the flow back to tank. The positions of the transducers 1,204 metres upstream and 0,098 metres downstream were chosen as the nearest transducer positions available on either side of the orifice in the circuit. The results were inserted in equation 5 to calculate the orifice coefficient, $C_d$. 
8.2 Wavespeed in a Hydraulic Hose

There is no theory available for calculating the wave speed in a hydraulic hose. Since hoses will tend to be used with hydro power systems it was decided to determine the wavespeed experimentally. A 4-spiral 1 inch hydraulic hose was chosen for the test.

The test was carried out using two KYOWA 0-20 MPa pressure transducers and a WATANABE Mark IV linear recorder. The hose was raised to a specific pressure then the pulse generated by a hydraulic unloading valve was sent down the hose, the wavespeed being measured by the difference in position of the pulses on the recorder.

8.3 Pipeline Transients

Five tests were carried out in which the manually operated poppet valve was closed and the pressures were recorded inside the pipe at a position 35,486 metres upstream of the valve and at the valve positions A and H in Figure 12. To be able to capture the values required at the maximum sampling rate, a micro-switch was attached to the valve and the movement of the valve was used to set the monitoring system in operation.

The black and white interface required for the Zimmer optical displacement transducer was then fitted to the valve end of the pipeline. The valve was then closed as before and the movement of the valve end of the pipe was measured. This test was conducted three times.

The black and white interface was removed and a 14 mm diameter orifice was installed in the pipeline 4,287 metres upstream of the valve as indicated in Figure 12. The valve was then closed as in the previous tests. The 14 mm diameter orifice was then replaced with a 10 mm diameter orifice and the test repeated.
A two inch nominal bore pipe tee was then fitted to the system upstream of the quick acting valve as shown in the insert in Figure 12. The spring loaded pressure relief valve shown in Figure 27 was fitted to this tee. The valve was set to allow slight seepage at 8 MPa and the quick closing valve operated, as in previous tests. The test was repeated with the relief valve set at 12 MPa and at 16 MPa.

A safety rupture diaphragm holder, of the type to be used on the Kloof hydro power pilot system, was fitted with a magnetic switch to detect disc failure by means of a 9 volt signal as shown in Figure 28. The holder with the appropriate disc was then inserted into the pipeline at position B in Figure 12. The valve was then closed as in the previous tests and the pressure at position A recorded, along with the voltage of the magnetic switch which indicated the disc failure. When failure of the disc occurred the voltage fell from 9V to 0V.
Figure 27  PHOTOGRAPH OF RELIEF VALVE INSTALLED IN THE PIPELINE

Figure 28  PHOTOGRAPH OF RUPTURE DISC ASSEMBLY AND MAGNETIC SWITCH
part III

experimental results
and discussion
9. STEADY FLOW CONDITIONS

9.1 Pipe Friction Losses

In the design of hydro power systems, it is essential to know the friction losses in the pipes. Under maximum flow conditions, the Reynolds number will be greater than 4,000, and so the flow will be turbulent. In turbulent flow there is an irregular secondary motion of fluid particles at right angles to the flow which is superimposed on the principal motion in the flow direction. This turbulent mixing introduces shear stresses, known as Reynolds stresses, and causes the dissipation of energy. As Reynolds number increases the secondary motions which constitute the turbulence become more intense and approach the pipe wall where the flow is laminar. This causes the laminar sub-layer to decrease in thickness such that the surface irregularities start to protrude through. The rougher the pipe the lower the value of Reynolds number at which this happens\(^\text{(21)}\). In this rough zone of flow the thickness of the laminar film is small compared with the height of the surface irregularities and the turbulent flow round each bump generates a wake of eddies which give rise to a resistance force known as 'form drag'.

The results of the tests given in Figure 29 show the friction factor against Reynolds number in the range 4,000 to 300,000. In the Figure the theoretical line calculated using equation (3) Section 3.2 is shown and an example of the theoretical calculation is given in Appendix C part (a). The theoretical results agree with the measured values in the range of Reynolds number from 30,000 to 100,000, but not either side of this. In the lower range the differences are due to the pipe containing many joints, these joints being internally welded. This welded restriction would act as a series of orifices and at the low Reynolds numbers below 20,000 it was shown by Lichtarowicz that the orifice
Figure 29 RESULTS FOR FRICTION LOSS MEASUREMENTS IN THE PIPELINE, COMPARED WITH THE THEORETICAL VALUES
coefficient varies between almost zero and 0.8. This means that any orifice effect of the restrictions would be expected to raise the measured friction factor considerably as the Reynolds number falls. At the higher values of Reynolds number the protrusions would start to have a greater significance and would cause the form drag effect to become important, since while the average roughness was only 12.79 \( \mu \text{m} \) the protrusions were as great as 170.5 \( \mu \text{m} \).

9.2 Orifice Friction Losses

Figure 30 shows the flow through a sharp-edged orifice compared with that in a long orifice and also shows four conditions that can occur in a long orifice by changing the length or Reynolds number. The parameters which affect the performance of long orifices are:

- (i) length to diameter ratio,
- (ii) upstream pipe diameter to orifice diameter ratio,
- (iii) downstream pipe diameter to orifice diameter ratio,
- (iv) Reynolds number,
- (v) Surface conditions, such as sharpness of edges and orifice surface finish.

None of the investigations carried out previously have a similar set of conditions as experienced here. Also in all cases investigated, including those by Lichtarowicz\(^{12}\), pressure was taken by corner tappings, and no account was taken of pressure recovery in the downstream section of pipe. For hydro power type installations pressure recovery will occur and has to be taken into account.
Figure 30 COMPARISON OF TURBULENT FLOW THROUGH ORIFICES UNDER VARYING CONDITIONS
The results in Figure 31 show the pressure drop across each of the orifices, against the flow rate. These results show very good correlation with regard to fitting a straight line. The correlation coefficient, $r^2$ which is used as a guide to the accuracy of the least squares curve fit varied between 0.936 and 0.998. Figure 32 shows the variation in orifice coefficient against Reynolds number, based on flow rate in the orifice. The results could not be represented adequately by one curve, and a good least squares fit could only be obtained by grouping together the 8 mm and 10 mm diameter orifices, and the 12 mm and 14 mm orifices. From the figure it can be seen that there is some factor causing the change in orifice coefficient. The sudden change in behaviour associated with an increase in diameter from 10 to 12 mm is thought to be caused by instability, which one author claimed is due to the change-over from non-cavitating to cavitating flow (22). For an $\eta/d$ ratio of 1.5 and a Reynolds number of greater than $6 \times 10^4$, a sharp rise in orifice coefficient was found by Koennecke (23). Spikes and Pennington (22) saw a similar rise but this was followed by a rapid fall. In all of these cases Lichtarowicz (12) attributes this behaviour to cavitation.

Cavitation is produced because the sharp corner causes a local region of low static pressure. If this static pressure is sufficiently low, dissolved air will be released from the liquid or vapour bubbles will form. In this case a cavitation parameter, $K$, is defined as:

$$K = \frac{P_1 - P_2}{P_2 - P_v}$$

where $P_1$ = upstream pressure
$P_2$ = downstream pressure
$P_v$ = vapour pressure of water
Figure 31: PRESSURE DROP AGAINST REYNOLDS NUMBER FOR FOUR LONG ORIFICES

Equations derived from least squares curve fit, in the form:

\[ Y = mx^n \]

<table>
<thead>
<tr>
<th>Orifice dia (mm)</th>
<th>m</th>
<th>n</th>
<th>( r^2 )</th>
</tr>
</thead>
<tbody>
<tr>
<td>8</td>
<td>0.329</td>
<td>2.27</td>
<td>0.936</td>
</tr>
<tr>
<td>10</td>
<td>2.709</td>
<td>2.47</td>
<td>0.959</td>
</tr>
<tr>
<td>12</td>
<td>3.394</td>
<td>2.24</td>
<td>0.995</td>
</tr>
<tr>
<td>14</td>
<td>4.45</td>
<td>1.95</td>
<td>0.998</td>
</tr>
</tbody>
</table>
Figure 32 ORIFICE COEFFICIENT DERIVED FROM EXPERIMENTAL RESULTS AGAINST REYNOLDS NUMBER FOR FOUR DIFFERENT LONG ORIFICES
This parameter is the inverse of that normally used but was considered by Bragg (24) to be more useful. The term $P_v$ is neglected in this case since it is very small in comparison with the other terms. Figure 33 shows the cavitation parameter against Reynolds number for each of the orifices. Bragg (24) gave the onset of cavitation as occurring at approximately $K = 1.5$, which means that both the 12 mm and 14 mm orifices were not cavitating but both the 8 mm and 10 mm diameter orifices were. This provides a very satisfactory explanation for the large change in orifice coefficient occurring between the 10 mm and the 12 mm orifice shown in Figure 32. Conditions under which orifices cavitate should in practice be avoided since damage can be caused to the pipes downstream of the orifice as well as the orifice itself.
Figure 33  CAVITATION PARAMETER AGAINST REYNOLDS NUMBER FOR LONG ORIFICES
10. UNSTEADY FLOW CONDITIONS

10.1 Wavespeed in a Hydraulic Hose

The wavespeed in a steel pipe has been verified experimentally many times and can be calculated by using equation (5) in Appendix A. In the case of a hydraulic hose the terms $c_1$ and $E$ in the equation are unknown, and wavespeed was measured experimentally, therefore.

An example of the results is given in Figure 34. Table 4 lists all of the 149 results taken and Figure 35 shows the variation of wavespeed against pressure for this particular hose. The solid line is the best straight line given by a least squares curve fit. The dotted line in the graph represents the rate of change of wavespeed with pressure that would be expected according to Pearsall's results (25) for steel pipes, Appendix A.

The results show the wavespeed rising with pressure, more rapidly than in the case of steel pipes and, since expansion of the internal diameter with pressure would lead to an effective stiffening of the hose walls, this seems justified.

In cases where no theoretical analysis is available such as non-cylindrical concrete pipes it is normal to convert the pipe to an equivalent steel pipe and include extra factors in the function $\psi_{\text{eff}}$ to convert the function to that found experimentally. In this case the results could be converted to that of an equivalent steel pipe by taking the function $\psi_{\text{eff}}$ as being:

$$\psi_{\text{eff}} = 2.63 \left[ 1_d - 0.9P \right]$$

This function can then be used in equation:

$$C = \frac{1}{\sqrt{\rho \left( \psi_{\text{E}} + \psi_{\text{E}} \right)}}$$

to calculate the wavespeed, according to the best straight line through the result.
Figure 34  TYPICAL RESULTS TAKEN TO MEASURE THE WAVESPEED IN A HYDRAULIC HOSE
<table>
<thead>
<tr>
<th>Pressure (MPa)</th>
<th>Measured values of Wavespeed (m/sec)</th>
<th>Number of Values</th>
<th>Average (m/sec) ± Range (m/sec)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1,8</td>
<td>1103.9, 1102.4, 1111.7, 1059.4, 1073.8</td>
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<tr>
<td>3,0</td>
<td>1072.4, 1084.2, 1097.2, 1079.7, 1093.2</td>
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<td>1088.6 ± 10.9</td>
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<tr>
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<td>1087.8 ± 4.9</td>
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<td>1076.8 ± 5.6</td>
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<tr>
<td>6,0</td>
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<td>1087.6 ± 6.3</td>
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<td>7,0</td>
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<tr>
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<td>1099.6 ± 8.1</td>
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<tr>
<td>10,0</td>
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<td>1112.0 ± 11.2</td>
</tr>
<tr>
<td>11,0</td>
<td>1096.2, 1102.4, 1091.8, 1105.5, 1110.1</td>
<td>11</td>
<td>1096.2 ± 10.8</td>
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<td>12,0</td>
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<td>1136.2 ± 9.3</td>
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</table>

Table 4 WAVESPEED IN A 4-SPIRAL ONE-INCH HYDRAULIC HOSE AT VARIOUS PRESSURES
<table>
<thead>
<tr>
<th>Pressure (MPa)</th>
<th>Measured values of Wavespeed (m/sec)</th>
<th>Number of Values</th>
<th>Average (m/sec)</th>
<th>Range (m/sec)</th>
</tr>
</thead>
<tbody>
<tr>
<td>14.0</td>
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<td>16.0</td>
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<td>8</td>
<td>1158.4</td>
<td>17.2</td>
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<td>18.0</td>
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<td>25.1</td>
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<tr>
<td>20.0</td>
<td>1185.1 1179.9 1161.6 1165.2 1174.5</td>
<td>7</td>
<td>1172.3</td>
<td>10.8</td>
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</table>

Table 4 (CONTINUED) WAVESPEED IN A 4-SPiral ONE-INCH HYDRAULIC HOSE AT VARIOUS PRESSURES
Figure 35 ACOUSTIC WAVE VELOCITY OF HYDRAULIC HOSE
These results are valid only for a 4-spiral one inch hose. The behaviour of other hoses would have to be established through additional experiments.

10.2 Pipeline Transient Behaviour

10.2.1 Pressure measurements

Figure 36 shows the initial transient occurring at position H, at the valve, and its arrival at position A, at the far end of the pipeline (see Figure 12), for one of the five cases measured.

From these results it was possible to measure the acoustic wave velocity to within ± 1% by comparing the digital images stored in the computer. The values obtained are shown along with the theoretical values according to Pearsall (25) in the table below:

<table>
<thead>
<tr>
<th>Temperature °C</th>
<th>22.0</th>
<th>29.4</th>
<th>29.7</th>
<th>30.4</th>
<th>30.5</th>
</tr>
</thead>
<tbody>
<tr>
<td>Wavespeed m/sec</td>
<td>1449</td>
<td>1481</td>
<td>1472</td>
<td>1475</td>
<td>1477</td>
</tr>
<tr>
<td>Predicted using Pearsall's results (See Appendix C part (d))</td>
<td>1442</td>
<td>1461</td>
<td>1462</td>
<td>1463</td>
<td>1463</td>
</tr>
<tr>
<td>Per cent difference</td>
<td>-0.5</td>
<td>-1.4</td>
<td>-0.7</td>
<td>-0.8</td>
<td>-1.0</td>
</tr>
</tbody>
</table>

All of the values measured are in close agreement with the theoretical values, in every case slightly lower. This would suggest the presence of very small amounts of dissolved gasses.

The maximum pressures recorded at positions A and H along with the rise time of the pressure at H were also measured from the traces, and are given in the Table below:
Figure 36 TRANSIENT OCCURRING AT POSITIONS A AND H IN THE PIPELINE FOLLOWING THE CLOSURE OF THE QUICK CLOSING VALVE, AT 22°C
<table>
<thead>
<tr>
<th>Test Number</th>
<th>1</th>
<th>2</th>
<th>3</th>
<th>4</th>
<th>5</th>
</tr>
</thead>
<tbody>
<tr>
<td>Maximum pressure at H (MPa)</td>
<td>55.0</td>
<td>55.0</td>
<td>55.4</td>
<td>55.4</td>
<td>54.4</td>
</tr>
<tr>
<td>Maximum pressure at A (MPa)</td>
<td>27.0</td>
<td>29.0</td>
<td>29.0</td>
<td>28.7</td>
<td>27.9</td>
</tr>
<tr>
<td>Rise time at position H (ms)</td>
<td>0.83</td>
<td>0.94</td>
<td>0.94</td>
<td>0.94</td>
<td>0.88</td>
</tr>
</tbody>
</table>

It is significant to note that repeatability is such that in each case the pressure recorded at position H, just upstream of the valve, varies by only 2%. Figure 37 shows the theoretical pressure rise predicted at positions A and H according to the computer program.

The differences between the practical and theoretical results are considered to be due to the inaccuracies in defining the valve closure characteristics adequately, and the limited extent to which the experimental pipeline could model a real hydro power system.

Figure 37 THE THEORETICAL PRESSURES IN THE PIPELINE ACCORDING TO COMPUTER ANALYSIS
The pressure recorded at position H, immediately upstream of the valve, is then shown for one case in Figure 38. It is shown over a much larger time period so that the reflected waves at times 2L/c and 4L/c are visible. From this, the distance of the major reflection point in the system can be calculated to be between 45.1 and 46.62 metres from position H. This coincides with the position of the 4" nominal bore tee which was positioned for that purpose. In this case the reflection coefficient, which is the ratio of pipe areas, is 10.73, and so a good reflection would be expected. Figure 38 also shows the attenuation of the pipeline to be very low since the initial pressure wave has only reduced by 0.9 MPa after travelling along 180 metres of pipeline. Many authors such as Ansari (26) have reported damping of pressure waves to be influenced by rate of change of pressure in such a way that the greater the rate of change of pressure the less damping occurs. In this case the rise time varies between 830 and 940 microseconds. This is very quick indeed and would explain the very low damping. The steady state friction effects according to Darcy were calculated using the instantaneous flow rates predicted by the computer. They were found to be very small and were not considered to contribute significantly to the 0.9 Mpa pressure loss.

Figure 39 shows the pressure recorded at one of the intermediate positions, E, in the pipeline, 1715 metres from position H. The pressure has in this short distance reduced significantly in value but this is not due to the attenuation of the pipeline but to the 'line packing' effect. If Figures 38 and 39 are compared it is seen that the waves occur almost simultaneously at both positions due to their close proximity, 1715 metres, but the shape of the wave has changed. The wave still contains the same energy but has reduced in height but increased in width.
Figure 38 PRESSURE TRANSIENT AT POSITION H, CLOSE TO THE VALVE

Figure 39 PRESSURE TRANSIENT RECORDED AT AN INTERMEDIATE POSITION IN THE PIPELINE
This effect which is an important phenomena in transient analysis is often a predominant factor in these long high pressure pipelines. Line packing is caused by the changes in fluid density which occur as the compression wave moves away from the boundaries causing the wave. A complete explanation of this effect can be found in Chaudhry (17) pp. 192-193.

The transients at positions A and H are shown in Figure 40 over a period of 0.3 seconds. The spikes at positions 1, 2 and 3 represent the initial transient at position H and its two subsequent positive reflections. These waves can easily be identified along with the same wave passing position A. However after 2 seconds other waves appear which cannot be so easily identified. It is felt that these occurrences are similar to those studied by many other authors in recent years (27), (28) which they suggested was due to interaction between the fluid and the pipe. Although the identification is difficult it is possible to measure propagation speeds of 4 800 metres per second between some of the occurrences, which is similar to the speed of compression waves in steel. However the complexity of the result makes it impossible to draw positive conclusions.

10.2.2 Pipe Movement

An example of the results taken to measure movement of the valve end of the pipeline is shown in Figure 41. The measurement equipment gave movement only in one direction so the negative movement is not shown. Unfortunately it was not possible to synchronize the starting of the displacement transducer and the computer. It seems reasonable to assume that the initial pipe movement occurs as the column of fluid is brought to rest. The results for each of the tests are given in Table 5. The first movement of approximately 15 mm has a considerable rise time of 62 ms, which
Figure 40  PRESSURE TRANSIENT RECORDED AT POSITION H, CLOSE TO THE
VALVE AND POSITION A 34.81 METRES AWAY.

System temp.
304 °C
Table 5 RESULTS OF TESTS CARRIED OUT TO ASCERTAIN PIPE MOVEMENT AT VALVE END OF EXPERIMENTAL PIPELINE

<table>
<thead>
<tr>
<th>Test No.</th>
<th>1st Movement (mm)</th>
<th>2nd Movement (mm)</th>
<th>3rd Movement (mm)</th>
<th>Time of occurrence of 2nd movement after 1st (secs)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>15.19</td>
<td>2.24</td>
<td>1.41</td>
<td>0.2704</td>
</tr>
<tr>
<td>2</td>
<td>14.87</td>
<td>2.21</td>
<td>1.29</td>
<td>0.2716</td>
</tr>
<tr>
<td>3</td>
<td>14.80</td>
<td>2.36</td>
<td>1.37</td>
<td>0.2718</td>
</tr>
</tbody>
</table>

Figure 41 EXAMPLE OF RESULTS FOR THE PIPE MOVEMENT TESTS CARRIED OUT USING THE ZIMMER OPTICAL DISPLACEMENT TRANSDUCER
compares with a period for the negative pressure wave in the fluid to return of 62.4 ms. This is expected, since, until the negative pressure wave returns the pipeline would have a force imbalance, due to the pressure imbalance towards the valve end of the pipeline. It would be expected that any oscillation would occur at the frequency of the transients with sinusoidal motion, heavily damped. The pipeline does have this sinusoidal damped motion but with a decreasing frequency, which is not the same as that of the transients. This suggests that the pressure imbalance alone does not determine the movement of the pipe. This has been seen previously by Hatfield et al. (27), who conclude that this effect results from an interaction of transients in both the liquid and the pipe.

### 10.2.3 Pipe Strain

The strain gauges located at position A in Figure 12 were used to measure circumferential and longitudinal strain in the pipe when a transient passed through the fluid. The results are shown in Figures 42 and 43. The following information was derived from these results:

<table>
<thead>
<tr>
<th>Position</th>
<th>Maximum pipe dimension change (mm)</th>
<th>Final steady dimension change (mm)</th>
<th>Rise time to steady state value (secs)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Diametrically at outside surface</td>
<td>0.0311</td>
<td>0.0294</td>
<td>0.340</td>
</tr>
<tr>
<td>Longitudinally</td>
<td>3.204</td>
<td>3.099</td>
<td>0.340</td>
</tr>
</tbody>
</table>
Figure 42  CIRCUMFERENTIAL STRAIN IN THE PIPELINE FOLLOWING CLOSURE OF THE QUICK CLOSING VALVE.

Figure 43  LONGITUDINAL STRAIN IN THE PIPELINE FOLLOWING CLOSURE.
It should be noted that the results were derived directly from the instruments and the microstrain scale is inverted for this reason.

In both cases the initial rise is slow, the strain overshoots its final value and in the case of the circumferential strain comes to rest. In the case of the longitudinal strain it undershoots once more before rising to its steady position. The rise times and the occurrence of overshoot and undershoot do not coincide with either the pipe movement or the pressure transients. Figure 44 showing both circumferential and longitudinal strain together with the transient occurring close to the strain gauges illustrates that the strain is unaffected by the transients. It merely changes to accommodate the general trend of pressure rise without being subject to the sharp fluctuations.

10.3 Transients through Orifices

No theory is available for the effect of orifices on unsteady flows. In a hydro power system there are likely to be many orifices and their effect is very important. The theoretical analysis used in the computer program is based on the assumption that the pressure wave after passing through the orifice was modified only by the steady state head loss. Figure 45 shows the effect of a 14 mm orifice on the pressure wave at positions A and H. It is clearly seen when compared with Figure 36 that there is no significant difference in the initial pressure wave at H, compared with the case than when no orifice was installed. The small discontinuities during the rise time are due to reflections returning from the nearby valve. However, the pressure at the far end (position A) rises to a maximum of 22.1 MPa which is significantly smaller than in the case without orifices, Figure 36.

When a 10 mm diameter orifice is introduced into the system the situation changes dramatically. Figure 46
Figure 44  CIRCUMFERENTIAL AND LONGITUDINAL STRAIN TOGETHER WITH THE TRANSIENT CAUSING THE STRAIN CHANGES
Figure 45  INITIAL PRESSURE TRANSIENT AT POSITIONS A AND H IN THE PIPELINE WITH A 14 MM DIAMETER ORIFICE INSTALLED IN THE PIPELINE

Figure 46  PRESSURE TRANSIENT POSITIONS A AND H IN THE PIPELINE
shows the pressure wave at positions A and H. In this case the wave has been broken by reflections into three pressure waves, the first having a rise time of only 0.5 ms and rising to a pressure of 57.48 MPa. The next two pressure waves are attenuated fairly rapidly. The three reflections with the smaller orifice are due to the reflection coefficient being much greater than for the 14 mm orifice and so the transient cannot pass through the orifice so easily. At the far end of the system, position A, the maximum pressure recorded is 25.0 MPa, similar to the case for the 14 mm orifice. However, the static pressure is now 12 MPa so the pressure rise is only 13 MPa, compared with 17 MPa for the 14 mm orifice.

Figure 47 shows the pressure at position H over a much longer period of time for both the 10 mm and the 14 mm diameter orifices. Here the initial waves and the very slightly damped reflections can be clearly seen. It is significant to note that the pressure wave occurring at 0.25 seconds in the case of the 10 mm orifice is again identified as not being directly due to the internal transients in the fluid, since it occurs at an irregular time.

From these results it can be seen that the size of orifice makes very little difference to the magnitude of the transients. However it can be clearly seen that, in the case of small orifices, additional pressure spikes can be produced by reflection. This illustrates the need for careful consideration in hydro power systems before positioning of orifices.

When the result is compared with the computer model in Figure 48 very little agreement is found. This is again considered to be due to the inaccuracies in defining the value closure characteristics and the inability of the experimental pipeline to accurately model a real hydro power system.
Figure 47 A COMPARISON BETWEEN THE INITIAL PRESSURE TRANSIENT
10.4 Behaviour of Relief Valves

Pressure relief valves are sometimes used to control transients by opening to allow rapid outflow of liquid from the pipeline.

Figures 49, (i), (ii) and (iii) show the pressure at the fast-closing valve when closed rapidly, with the relief valve set to open at 8, 10 and 16 MPa respectively.

The following information was taken from the figures:

<table>
<thead>
<tr>
<th>Valve pressure (MPa)</th>
<th>Pressure of first transient (MPa)</th>
<th>Pressure of return of first transient (MPa)</th>
<th>Pressure of first transient above preset limit (MPa)</th>
<th>Pressure of second transient above preset limit (MPa)</th>
</tr>
</thead>
<tbody>
<tr>
<td>8</td>
<td>16,52</td>
<td>15,36</td>
<td>8,52</td>
<td>7,36</td>
</tr>
<tr>
<td>10</td>
<td>11,87</td>
<td>18,19</td>
<td>-0,13</td>
<td>6,19</td>
</tr>
<tr>
<td>16</td>
<td>15,48</td>
<td>19,61</td>
<td>-0,52</td>
<td>3,61</td>
</tr>
</tbody>
</table>

These results show clearly that, as expected, the relief valve does not respond adequately to the transients.
Figure 49  PRESSURE TRANSIENT OCCURRING WITH A SPRING LOADED RELIEF VALVE INSTALLED IN THE PIPELINE
In every instance valve 'chatter' is present since the valve is unable to respond to the pressure changes occurring in the pipeline.

The valve is unstable over the range of operating conditions tested and independent of the spring load exerted by the adjustment mechanism. It is possible to calculate when this instability is likely to occur (29). This is however, a very complicated procedure. It is important in hydro power systems to ensure that valve 'chatter' does not occur at any frequency.

Figure 49(iv) shows the transient recorded at position A in the pipeline for each of the relief valve settings. It is seen that the effects of valve chatter are not apparent at this distance. However, the pressure at this position has not been affected in any substantial way by the presence of the relief valve. In all three cases the pressure measured exceeds the relief valve settings frequently. The circumferential and longitudinal strain were measured at the same position and the results are shown in Figure 50. The circumferential and longitudinal strains in each case show signs of responding to the valve chatter. The strain variations occur at a frequency of approximately 30 Hz. This could cause fatigue damage to a piping system if it were allowed to continue.

10.5 Behaviour of Safety Rupture Diaphragms

Under quasi-static conditions the stress in a safety rupture diaphragm immediately prior to bursting is defined by the equation:

\[ \sigma = \frac{Pr}{2t} \]

where 
- \( r \) = radius of the dome.
- \( t \) = thickness of material at the top of the dome.
- \( P \) = burst pressure.
Figure 50  CIRCUMFERENTIAL AND LONGITUDINAL STRAIN FOR EACH OF THE THREE RELIEF VALVE SETTINGS
Since the disc has little inertia it should fail whenever the burst pressure is exceeded. In any material, however, a finite rate of change of stress occurs and a very rapid change can occur without failure occurring. Figure 44 shows that the pipe strain follows the average pipe pressure at a relatively slow rate. The very thin disc material of a safety rupture diaphragm would tend to respond much more quickly to changes in pressure, but the rate at which it responds is unknown.

In Figure 51, the initial transients rise and fall above the failure pressure very frequently, with a high rate of change. Immediately that the transients stop falling below the failure pressure, failure occurs. When a disc with a lower burst pressure pressure is used (Figure 52), failure occurs almost immediately and although this transient also has a fast rise time it does not fall in the first instance below the failure pressure. The criterion for diaphragm rupture therefore appears to be whether or not the transients fall below the nominal burst pressure.
Figure 51  PRESSURE TRANSIENT RECORDED AT THE VALVE WHEN A RUPTURE DISC WITH A BURST PRESSURE OF 12.1 MPa IS INSTALLED IN THE PIPELINE.

Figure 52  PRESSURE TRANSIENT RECORDED AT THE VALVE WHEN A RUPTURE DISC WITH A BURST PRESSURE OF 6.7 MPa IS
11. CONCLUSIONS

(i) The steady state friction losses in pipes for hydro power systems can be much greater under certain conditions than those predicted theoretically using the Darcy friction factor.

(ii) The behaviour of long orifices cannot be predicted accurately unless the relationship between the orifice coefficient and Reynolds number is known for that orifice.

(iii) When calculating the flows through orifices under severely restricted flow conditions, such as during a runaway situation, account should be taken of a possible large change in orifice coefficient due to cavitation.

(iv) It is possible to calculate the wavespeed in a one-inch hydraulic hose of the type tested to within six per cent by using the expression:

\[ \psi_m = 2.63 [1d - 0.9P] \]

in the equation

\[ c = \frac{1}{\sqrt{\rho(1K + \psi_E)}} \]

Similar expressions could be derived empirically for other types of hose.

(v) The theoretical formulae for the wavespeeds in pipes are valid over the range of pressures used in hydro power systems and can be used confidently to give accurate values of wavespeed.

(vi) For accurate analysis of water hammer it is important that the installed flow characteristics of any valves used are known.
(vii) In real piping systems, insufficient information about the behaviour of one or more boundary conditions can lead to serious inaccuracies in the predictions of maximum pressures.

(viii) The strains in the pipe walls do not follow the fluid transients, and so transient pressures in excess of failure pressures will not necessarily cause failure of a pipe.

(ix) Steel hammer will occur in the pipe in conjunction with water hammer. Account may be taken of its effects, by using the theory developed by Hatfield et al (27).

(x) Pipe movement due to hydraulic transients cannot be predicted easily but generally the more flexible the piping system the greater the likelihood of interaction between the structure and the fluid. Allowance should be made in hydro power systems for the large pipe movement which will occur.

(xi) The steady state friction losses through orifices are not valid under transient conditions. Further, orifices do not necessarily reduce transient pressures and can if badly positioned increase the number of transients. At this stage, insufficient information is available to predict accurately their behaviour. Further research is needed in this area.

(xii) Relief valves tend to chatter under transient conditions, and may cause damaging fluctuating strains in hydro power systems.
(xiii) Safety rupture diaphragms may or may not remain intact when exposed to pressure spikes exceeding the nominal burst pressure, depending on the characteristic of the transient waveform.
REFERENCES

1. WYMER, D.G. The use of water hydraulics in machines for mechanized gold mining in South Africa. Int. Conf. Mining and Machinery, 1979, Institute of Engineers, Brisbane, Australia.


APPENDIX A

THE VELOCITY OF WATER HAMMER WAVES IN WATER

It can be shown (25) that the velocity of water hammer waves in water at atmospheric pressure and 20°C is given by the following expression.

\[ C = \sqrt{\frac{1}{\rho \left( \frac{1}{K} + \frac{d}{tE} \right)}} \]  

\[ \ldots \ldots (1) \]

where,

\( C \) = velocity of water hammer waves in water
\( \rho \) = density of water, at 20°C
\( K \) = bulk modulus of water, at 20°C
\( t \) = wall thickness of pipe
\( d \) = diameter of pipe
\( E \) = modulus of elasticity of pipe at 20°C

This expression assumes that the pipe is unrestrained in the longitudinal direction, is perfectly elastic, and thick walled.

Halliwell (30), introduced a non-dimensional parameter into the equation, giving instead:

\[ C = \sqrt{\frac{1}{\rho \left( \frac{1}{K} + \frac{d}{tE} \right)}} \]  

\[ \ldots \ldots (2) \]

to account for the differences in pipes and the anchoring of the pipe under various conditions. If the pipe is thick walled and free to move then;

\[ \psi = \frac{d}{t} \]

and equation 2 returns to the simpler case of equation 1. The other conditions he introduced were:
<table>
<thead>
<tr>
<th>Condition</th>
<th>Thick walled elastic pipe</th>
<th>Thin walled elastic pipe</th>
</tr>
</thead>
<tbody>
<tr>
<td>Anchored against longitudinal movement for entire length</td>
<td>[ \psi = 2(1 + \nu) \frac{R_o^2 + R_i^2 - 2\psi R_i^2}{R_o^2 - R_i^2} ] [ \frac{R_o^2 - R_i^2}{R_o^2 - R_i^2} ] [ \frac{R_o^2 - R_i^2}{R_o^2 - R_i^2} ]</td>
<td>[ \psi = \frac{d(1 - \nu^2)}{t} ]</td>
</tr>
<tr>
<td>Anchored against longitudinal movement at upper end</td>
<td>[ \psi = 2 \frac{R_o^2 + 1.5R_i^2 + \psi(R_o - 3R_i^2)}{R_o^2 - R_i^2} ] [ \frac{R_o^2 - R_i^2}{R_o^2 - R_i^2} ]</td>
<td>[ \psi = \frac{d(1.25 - \nu^2)}{t} ]</td>
</tr>
<tr>
<td>Frequent expansion joints</td>
<td>[ \psi = 2 \left( \frac{R_o^2 + R_i^2 + \psi}{(R_o^2 - R_i^2)^2} \right) ]</td>
<td>[ \psi = \frac{d}{t} ]</td>
</tr>
</tbody>
</table>

where \( \nu \) = Poisson's ratio

\( R_o \) = external radius of pipe

\( R_i \) = internal radius of pipe

\( d \) = pipe diameter

\( t \) = wall thickness.

The equations above are valid at atmospheric pressures at a temperature of 20°C and with no undissolved gasses present in the water.

Pearsall (25) studied the effect of temperature and pressure on the velocity of the water hammer waves and concluded that it is possible to account for changes in pressure and temperature by introducing a factor \( k_1 \), where:

\[
k_1 = \frac{a_0 + a_1T + a_2T^2 + a_3T^3 + a_4T^4}{a_0} \quad \text{...... (3)}
\]

and

\[
a_0 = 1402.86 + 1.532P + 0.1518P^2 - 1.1858 \times 10^{-5}P^3
\]

\[
a_1 = 5.0239 + 0.90 \times 10^{-3}P - 2.2711 \times 10^{-4}P^2 + 7.5536 \times 10^{-7}P^3
\]

\[
a_2 = -0.05691 - 1.5532 \times 10^{-4}P + 4.6587 \times 10^{-6}P^2 - 1.5514 \times 10^{-8}P^3
\]

\[
a_3 = 2.88493 \times 10^{-4} + 2.29447 \times 10^{-6}P - 5.09 \times 10^{-8}P^2 + 1.55059 \times 10^{-10}P^3
\]

\[
a_4 = 6.31 \times 10^{-7} - 9.9176 \times 10^{-9}P + 2.0419 \times 10^{-10}P^2 - 5.6253 \times 10^{-13}P^3
\]
and \( T \) = temperature (°C)  
\( P \) = pressure (MPa)  
\( c \) = acoustic velocity as calculated using equation (2) (m/sec)

He also showed that if the water contains even a small quantity of undissolved gas bubbles the acoustic velocity is greatly changed. For concentrations of air of greater than 1 in 1,000 by volume he proposed an additional factor \( k_2 \), where:

\[
k_2 = \frac{1}{\sqrt{\frac{P_a}{yK + P_a}} \cdot \frac{1}{C_0}} \quad .... (4)
\]

and \( C_0 \) = the acoustic velocity calculated using equation (2)  
\( P_a \) = the absolute gas pressure  
\( K \) = bulk modulus of water  
\( y \) = proportion of gas by volume

To calculate the velocity of water hammer waves in water for any given temperature, pressure and gas content, therefore, it is necessary to use the full expression:

\[
c = k_1 k_2 \sqrt{\frac{1}{\sqrt{\frac{P_a}{yK + P_a}}}} \quad .... (5)
\]

Because of the complexity of this equation, tables have been produced to allow \( k_1 \) and \( k_2 \) to be calculated easily. Portions of Pearsall's tables have been interpolated to give some values of \( k_1 \) and \( k_2 \) in the ranges that would be required in a mine hydro power system.
APPENDIX B

Flow chart and program listing

UNIVERSITY OF JOHANNESBURG
CALCULATE VELOCITY AND PRESSURE AT LEFT END OF CURRENT PIPE SECTION AND JOKOWSKYS CONSTANTS

TYPE OF CONNECTION OF RIGHT HAND SIDE OF CURRENT PIPE SECTION

IS T GREATER THAN THE NUMBER OF PIPE SECTIONS

IS T GREATER THAN TIME FOR PIPE SURGES

PRINT AND PLOT RESULTS FOR THE END OF EACH PIPE

OUTPUT TO DISC STORAGE

END
OPTION BASE 1
DIM Title$*100,Idno$*205,Wt3(400)
PRINTER IS 1
DISP "DO YOU WANT PRINTOUT?";
ON KEY 0 LABEL "YES" GOTO 80
ON KEY 1 LABEL "NO" GOTO 110
GOTO 50
PRINTER IS 701
PRINT USING 100
IMAGE /
PRINT "*************************************************************************************"
PRINT "*
PRINT "COMTAP
PRINT "*
PRINT "A PROGRAM TO EVALUATE TRANSIENTS IN HYDRAULIC SYSTEMS"
PRINT "*
PRINT "*************************************************************************************"
BULK
IDENTIFICATION NUMBER

OPTION BASE 1
DIM Title$*100,Idno$*205,Wt3(400)
PRINTER IS 1
DISP "DO YOU WANT PRINTOUT?";
ON KEY 0 LABEL "YES" GOTO 80
ON KEY 1 LABEL "NO" GOTO 110
GOTO 50
PRINTER IS 701
PRINT USING 100
IMAGE /
PRINT "*************************************************************************************"
PRINT "*
PRINT "COMTAP
PRINT "*
PRINT "A PROGRAM TO EVALUATE TRANSIENTS IN HYDRAULIC SYSTEMS"
PRINT "*
PRINT "*************************************************************************************"
BULK
IDENTIFICATION NUMBER

11L
10 OPTION BASE 1
20 DIM Title$*100,Idno$*205,Wt3(400)
30 PRINTER IS 1
40 DISP "DO YOU WANT PRINTOUT?";
50 ON KEY 0 LABEL "YES" GOTO 80
60 ON KEY 1 LABEL "NO" GOTO 110
70 GOTO 50
80 PRINTER IS 701
90 PRINT USING 100
100 IMAGE /
110 PRINT "*************************************************************************************"
120 PRINT "*
130 PRINT "*
140 PRINT "*
150 PRINT "*
160 PRINT "*
170 PRINT "*
180 PRINT "*************************************************************************************"
190 PRINT USING 200
200 IMAGE /
210 OFF KEY
220 DISP "INPUT YOUR OWN TITLE IF REQUIRED?"
230 INPUT Title$;
240 DISP "INPUT IDENTIFICATION NUMBER";
250 INPUT Idno$;
260 PRINT USING 270
270 IMAGE /
280 PRINT "Title$; IDENTIFICATION NUMBER ";Idno$
290 PRINT USING 270
300 INTEGER I,J,K,Kk,S1,N5,N1,N4,L,M,L1,L2,L3,L4,Kkk
310 DIM P1(35),Pd(35),Cc(35),R5(35),E3(50),E4(2,5),E7(2,5),C7(2,5),Tx(500),
320 Vy(500)
330 DIM Pt(35),Ps(35),T6(35),T5(35),S2(35),S8(35),J1(35),J2(35),J3(35),J4(35),
340 S5(500)
350 DIM Pav(35),T9(35),F1r(35),CB(35),S4(35),H(500),V(500),R6(500),H11(500),
360 T2(500),Vp(500)
370 DIM Ord(500),T2(500),E1(500),Pf1(35)
380 DIM Rn(35),Kt(35),Wt1(400,20),Wt2(400,20),Tn(13),R0(13),Kv(13),Tv(13),
390 A1(25)
400 DIM A2(25),B1(25),dc(25),Prv(500),XB(500),Rf(35)
410 INTEGER I1, Jj
420 DISP ""
430 DISP "NAME OF DATA FILE ?";
440 INPUT Df$;
450 ASSIGN aPath3 TO Df$;
460 ENTER aPath3;Tempsys
gosub Density
470 GOTO 770
480 Density:
490 FOR I=1 TO 13
500 ENTER aPath3;Tn(I),Ro(I)
510 NEXT I
520 FOR I=1 TO 13
530 IF Tn(I)>Tempsys THEN 550
540 NEXT I
550
550  \text{Rou}=\frac{(Tn(I)-\text{Tempsys})\times(Ro(I-1)-Ro(I))}{(Tn(I)-Tn(I-1))}+Ro(I)
560  \text{RETURN}
570  \text{Kin}_{\text{vis}}: \!
580  \text{FOR } I=1 \text{ TO } 10
590    \text{ENTER } \text{aPath3};Tv(I),Kv(I)
600     Kv(I)=Kv(I)\times1.E-6
610  \text{NEXT } I
620  \text{FOR } I=1 \text{ TO } 10
630     \text{IF } Tv(I)\geq \text{Tempsys} \text{ THEN } 650
640  \text{NEXT } I
650     \text{Kin}=\frac{(Tv(I)-\text{Tempsys})\times(Kv(I-1)-Kv(I))}{(Tv(I)-Tv(I-1))}+Kv(I)
660  \text{RETURN}
670  \text{Bul}_{\text{mod}}: \!
680  \text{FOR } I=1 \text{ TO } 4
690    \text{ENTER } \text{aPath3};Tb(I),Blk(I)
700     Blk(I)=Blk(I)\times1.E+9
710  \text{NEXT } I
720  \text{FOR } I=1 \text{ TO } 4
730     \text{IF } Tb(I)\geq \text{Tempsys} \text{ THEN } 750
740  \text{NEXT } I
750     \text{Bul}=\frac{(Tb(I)-\text{Tempsys})\times(Blk(I-1)-Blk(I))}{(Tb(I)-Tb(I-1))}+Blk(I)
760  \text{RETURN}
770  \text{ENTER } \text{aPath3};Ts,X7,T4,59,53,51,No0,S2rv
780  \text{Nd}=X7
790  Xy=1
800  Lx=0
810  Ip=1
820  De=1
830  S7=2.0
840  \text{FOR } I=1 \text{ TO } S1
850    \text{ENTER } \text{aPath3};S2(I),S8(I),J1(I),J2(I)
860    \text{ENTER } \text{aPath3};J3(I),J4(I)
870    \text{READ } P1(I),Pd(I),Rf(I),Pt(I),T6(I),T5(I)
880  \text{NEXT } I
890  \text{FOR } I=1 \text{ TO } S1
900     Pf(I)=.017
910  \text{NEXT } I
920  \text{E}=2.06E+11
930  \text{FOR } I=1 \text{ TO } S1
940     \text{IF } (Pd(I)/Pt(I))<25 \text{ THEN } 970
950     T3=(Pd(I)/Pt(I))\times(1-.27^2))
960  \text{GOTO } 1000
970     Z1=((Pd(I)+2*Pt(I))^2+(Pd(I))^2)/4
980     Z2=((Pd(I)+2*Pt(I))^2-(Pd(I))^2)/4
990     T3=(2\times1.27*Z1/Z2)-(2\times.27\times(Pd(I)/2)^2)/Z2
1000     Pav(I)=\text{SQRT}(Bu/(\text{Rou}*(1+(Bu/E)))\times T3))
1010  \text{NEXT } I
1020  X3=P1(I)/Pav(I)
1030  TB=1
1040  \text{FOR } I=2 \text{ TO } S1
1050     \text{IF } (P1(I)/Pav(I)-X3)>0 \text{ THEN } 1080
1060  \text{TB}=I
1070  X3=P1(I)/Pav(I)
1080  \text{NEXT } I
1090  X3=P1(TB)/Pav(TB)
1100  \text{IF } S3=0 \text{ THEN } 1790
1110  \text{FOR } J=1 \text{ TO } 48
1120    \text{ENTER } \text{aPath3};E3(J)
1130  \text{NEXT } J
1140 FOR N=1 TO 5
1150   ENTER aPath3;E4(1,N),E4(2,N)
1160 NEXT N
1170 IF S5=1 THEN 1210
1180 FOR N=1 TO 5
1190   ENTER aPath3;E7(1,N),E7(2,N)
1200 NEXT N
1210 PRINT "^^^^^^^^ VALVE CHARACTERISTICS ^^^^^^^"}
1220 PRINT
1230 FOR J=1 TO 48
1240   PRINT USING 1250;E3(J)
1250   IMAGE 6X,D.3D,
1260 NEXT J
1270 PRINT USING 200
1280 PRINT "$$$$$$$ VALVE MOVEMENT $$$$$$$$$$$$$$$$
1290 PRINT USING 200
1300 PRINT " VALVE NUMBER 1"
1310 PRINT " TIME(Secs) ANGLE (Deg) "
1320 FOR N=1 TO 5
1330   PRINT USING 1340;E4(1,N),E4(2,N)
1340   IMAGE 2X,2D.2D,11X,3D.1D
1350 NEXT N
1360 IF S5=1 THEN 1790
1370 PRINT
1380 PRINT " VALVE NUMBER 2"
1390 PRINT " TIME(Secs) ANGLE (Deg) "
1400 FOR N=1 TO 5
1410   PRINT USING 1340;E7(1,N),E7(2,N)
1420 NEXT N
1430 DISP " DO YOU WANT TO PLOT VALVE CHARACTERISTICS? ";
1440 ON KEY 0 LABEL "YES" GOTO 1470
1450 ON KEY 1 LABEL "NO" GOTO 1790
1460 GOTO 1440
1470 PLOTTER IS 705,"HPGL"
1480 CSIZE 4
1490 VIEWPORT 15,97*RATIO,10,97
1500 WINDOW 0,90,0,100
1510 AXES 10,10,0,0
1520 VIEWPORT 0,100*RATIO,0,100
1530 LORG 6
1540 FOR I=0 TO 90 STEP 10
1550   MOVE I,0
1560   LABEL I
1570 NEXT I
1580 LORG 8
1590 FOR I=0 TO 100 STEP 10
1600   MOVE 0,I
1610   LABEL I
1620 NEXT I
1630 LINE TYPE 7
1640 I=0
1650 FOR J=3 TO 48
1660   DRAW I,E3(J)*100
1670   I=I+2
1680 NEXT J
1690 LINE TYPE 1
1700 LORG 1
1710 MOVE 40,-8
1720 LABEL "ANGLE Deg"
1730 MOVE -5,50
1740 LDIR 64.4
1750 LABEL "RESISTANCE"
1760 LDIR 0
1770 MOVE 45,90
1780 LABEL "VALVE CHARACTERISTIC"
1790 PRINT USING 1800
1800 IMAGE 2/
1810 OFF KEY
1820 PRINT "**********~**********************************************
1830 PRINT "* STEADY FLOW THROUGH SYSTEM=";T4;"L/min *
1840 PRINT "**********
1850 PRINT USING 1860
1860 IMAGE 4/
1870 PRINT " ++++++++++++--------------PIECE DETAILS--------------+++++++++
1880 PRINT "+
1890 PRINT "PIPE NO. LENGTH IN. DIA ROUGHNESS ELV-LEFT ELV-RIGHT"
1900 PRINT " m m m m m m"
1910 Ggg=0
1920 Dt=X3
1930 T=0.
1940 FOR I=1 TO Ndt
1950 T=T+Dt
1960 NEXT I
1970 IF (T-Ts)<0 THEN 1940
1980 Ts=T-.5*Dt
1990 T=0.
2000 K=1
2010 FOR I=1 TO S1
2020 PRINT USING 2030:S2(I),P1(I),Pd(I),Rf(I),T6(I),T5(I)
2030 IMAGE 1X,3D,5X,4D,D,2X,D.3D,2X,D.6D,2(2X,5D),3X,#
2040 Kkk=S8(I)
2050 IF Ggg>S1 THEN 2220
2060 ON Kkk GOTO 2110,2130,2150,2180,2200,2090,2070
2070 PRINT "PRESS REG VALVE";J1(I)
2080 GOTO 2220
2090 PRINT "ORIFICE AT END OF PIPE NO.";S2(I)
2100 GOTO 2220
2110 PRINT "ONE PIPE=";J1(I)
2120 GOTO 2220
2130 PRINT "VALVE,PIPE NO.";J1(I)
2140 GOTO 2220
2150 PRINT "TWO PIPES NOS";J1(I);"AND";J2(I)
2160 J=ABS(J2(I))
2170 GOTO 2220
2180 PRINT "BLANK END"
2190 GOTO 2220
2200 PRINT "NOZZLES"
2210 GOTO 2220
2220 T9(I)=INT(P1(I)/(Dt*Pav(I)))
2230 IF T9(I)=0 THEN 2250
2240 GOTO 2260
2250 T9(I)=1
2260 P1r(I)=P1(I)/T9(I)
2270 CB(I)=Dt/P1r(I)
2280 S4(I)=K
2290 K=K+T9(I)+1
2300 Gg=6g+1
2310 NEXT I
2320 PRINT USING 200
2330 PRINT "ADDITIONAL PIPE INFORMATION"
2340 PRINT "PIPE NO. REACHES REACH LENGTH WALL THICKNESS ACcouSTIC"
2350 PRINT " m m m/s"
2360 NEXT I
2370 FOR I=1 TO S1
2380 PRINT USING 2390: S2(I), T9(I), Pr(I), Rf(I), Pav(I)
2390 IMAGE 3D, 7X, 4D, 10X, 5D, 10X, D, 6D, 14X, 5D
2400 NEXT I
2410 PRINT USING 200
2420 PRINT "ORIFICE INFORMATION"
2430 PRINT USING 200
2440 PRINT "NODE NUMBER UPSTREAM DIAMETER ORIFICE DIAMETER ORIFICE TH"
2450 PRINT " mm mm mm"
2460 FOR I=1 TO N2
2470 ENTER &Path3; S2p
2480 S2pp=S4(S2p)+T9(S2p)
2490 ENTER &Path3; T2(S2pp), Ord(S2pp), Oth(S2pp)
2500 PRINT USING 2510; S2pp, T2(S2pp), Ord(S2pp), Oth(52pp)
2510 IMAGE 2X, 4D, 13X, D, 16X, 4D, 16X, 3D, D
2520 NEXT I
2530 PRINT ""
2540 FOR I=1 TO S2rv
2550 ENTER &Path3; Frv
2560 Ff=S4(Frv)+T9(Frv)
2570 NEXT I
2580 ASSIGN &Path3 TO *
2590 H(I)=S9+T6(I)
2600 L1=1000
2610 L2=1000
2620 L3=1000
2630 L4=1000
2640 N2=1000
2650 N3=1000
2660 M2p1=0.
2670 M2p2=0.
2680 FOR I=1 TO S1
2690 IF S8(I)=4 THEN 2810
2700 IF S2(I)=M2p1 THEN 2730
2710 GOTO 2780
2720 GOTO 2780
2730 K=S4(I)
2740 H(K)=S9+T6(I)
2750 T4=T4/2.0
2760 PRINT "TWO BRANCHES FEED SYSTEM"
2770 GOTO 3020
2780 IF S2(I)-N2<0 THEN 2840
2790 IF S2(I)-N2=0 THEN 2810
2800 IF S2(I)-N3<=0 THEN 2810
2810 X4=0.
2820 PRINT "DEAD END FOUND PIPE NO."; S2(I)
2830 GOTO 2970
2840 IF S2(I)-L1<0 THEN 2900
IF S2(I)-L2<0 THEN 2880
GOTO 2900
X4=X1(I)
GOTO 2970
IF S2(I)-L3<0 THEN 2960
IF S2(I)-L3=0 THEN 2940
IF S2(I)-L4<=0 THEN 2940
GOTO 2960
X4=X1(2)
GOTO 2970
X4=1.0
k=S4(I)
Head=H(K)
T1=X4*T4/(15000*3.142*Pd(I)^2)
R2=Pf(I)*P1(I)*T1^2/(2*9.8142*Pd(I))
L=T9(I)+1
Xx=1.0/T9(I)
FOR J=1 TO L
   V(K)=T1
   H(K)=Head-(J-1)*R2*Xx
   R6(K)=H(K)-T6(I)+(J-1)*(T6(I)-T5(I))*Xx
   I<=K+1
NEXT J
Kkk=S8(I)
L=J1(I)
IF L=0 THEN 3180
FOR Nn=1 TO 51
   IF L-S2(Nn)=0 THEN 3160
NEXT Nn
PF~INT II PIPE NO. ";S2(I);"REFERS TO NON EXIST PIPE ";J1(I)
L=Nn
Ll=S4(L)
K=S4(I)+T9(I)
ON Kkk GOTO 3390,3410,3730,4360,4310,3230,3200
H(L1)=Prv(k)+T5(I)
X8(K)=((ABS(H(L1)-H(K)))*9.81)/(V(K)^2*.5))
GOTO 4360
A1(Ip)=(3.142*T2(K)^2)/4
A2(Ip)=(3.142*Ord(K)^2)/4
Bi(Ip)=SQR(1-(A2(Ip)/A1(Ip))^2)
IF Oth(K)/Ord(K)==2 THEN 3280
GOTO 3300
Dc(Ip)=.827-(.0085*Oth(K)/Ord(K))^2
IF Oth(K)/Ord(K)<1.5 THEN 3330
Dc(Ip)=(.136*(Oth(K)/Ord(K))+.606)
R2=.05096B*(((T4/60000)*Bi(Ip))/(A2(Ip)*1.*Dc(Ip)))^2
S5(K)=(19.62*R2)/V(K)^2
H(L1)=H(K)+R2
Ip=Ip+1
GOTO 4360
H(L1)=H(K)
GOTO 4360
IF Ibv=! THEN 3500
J=0
3430 J=J+1  
3440 FOR N=1 TO 5  
3450 C7(J,N)=E7(J,N)  
3470 NEXT N  
3480 IF J=2 THEN 3570  
3490 GOTO 3440  
3500 J=0  
3510 J=J+1  
3520 FOR N=1 TO 5  
3530 C7(J,N)=E4(J,N)  
3540 NEXT N  
3550 IF J=2 THEN 3570  
3560 GOTO 3510  
3570 A8=C7(2,1)  
3580 FOR J=1 TO 48  
3590 A9=(J-1)*2  
3600 IF (A9-A8)<0 THEN 3670  
3610 IF (A9-A8)>0 THEN 3640  
3620 X9=E3(J)  
3630 GOTO 3680  
3640 A9=A9-2  
3650 X9=E3(J-1)+(E3(J)-E3(J-1))*5*(A8-A9)  
3660 GOTO 3680  
3670 NEXT J  
3680 X9=1.0/X9  
3690 R2=((X9^2-1)^V(K)^2*.5)/9.81  
3700 H(L1)=H(K)-R2  
3710 Ibv=2  
3720 GOTO 4360  
3730 M=ABS(J2(I))  
3740 IF M=0 THEN 3910  
3750 FOR Nn=1 TO 51  
3760 IF M-52(Nn)=0 THEN 3790  
3770 NEXT Nn  
3780 PRINT "PIPE NO.";S2(I);"REFFERS TO NON EXISTANT PIPE NO.";M  
3790 M=Mn  
3800 Mm=S4(N)  
3810 H(L1)=H(K)  
3820 H(Mn)=H(K)  
3830 IF J4(I)=0 THEN 4360  
3840 IF J3(I)=0 THEN 3860  
3850 GOTO 3900  
3860 IF J2(I)-J4(I)=0 THEN 4360  
3870 N2=J2(I)  
3880 N3=J4(I)  
3890 GOTO 4360  
3900 N1=J3(I)  
3910 IF N1=0 THEN 3970  
3920 FOR Nn=1 TO 51  
3930 IF N1-52(Nn)=0 THEN 3960  
3940 NEXT Nn  
3950 PRINT "PIPE NO.";S2(I);"REFFERS TO NON EXISTANT PIPE NO.";N1  
3960 N1=Nn  
3970 N5=N1  
3980 FOR N4=1 TO 2  
3990 IF S8(N1)=7 THEN 4300  
4000 PRINT "BRANCH TERMINATED BY NOZZLES"  
4010 H11(N4)=H(K)-T5(N1)  
4020 IF H11(N4)>0 THEN 4040
PRINT "HLL(","N4;"> IS NEGATIVE"
N1=J4(I)
IF N1=0 THEN 4110
FOR Nn=1 TO S1
   IF N1-S2(Nn)=0 THEN 4100
NEXT Nn
PRINT "PIPE NO.",S2(I);"REFER TO NON EXIST PIPE NO.";N1
N1=Nn
NEXT N4
P11=0.
FOR Kps=L TO N5
   P11=P11+F1(Kps)
NEXT Kps
P12=0.
FOR Kps=M TO N1
   P12=P12+F1(Kps)
NEXT Kps
Cons=(Pd(L)/Pd(M))^5*Pf(M)*P12/(Pf(L)*P11)
Rax=SNR(ABS(Cons*Hll(1)/Hll(2)))
X1(2)=1.0/(1.0+Rax)
X1(1)=1.0-X1(2)
L1=J1(I)
L2=J3(I)
L3=(J2(I))
L4=(J4(I))
COTO 4360
Hll(N4)=H(K)-T5(N1)-Hst
COTO 4040
RS(I)=H(K)-T5(I)
PRINT "NOZZLE AT PIPE NO.",S2(I);"HEAD=";RS(I)
Ce(I)=T1
GOTO 4360
R1=H(K)-T5(I)
NEXT I
FOR I=1 TO S1
   Pf1(I)=Pf(I)
NEXT I
FOR I=1 TO S1
   Pf1(I)=Pf(I)
NEXT I
K=S4(I)
IF V(K)=0 THEN Pf(I)=0
IF Pf(I)=0 THEN 4470
Rn(I)=V(K)*Pd(I)/Kin
Bct=LGT(Rf(I)/Pd(I)/3.7+14.5/Rn(I))
Pf(I)=1/((-2*LGT(Rf(I)/Pd(I)/3.7-5.02*Bct/Rn(I)))^2)
NEXT I
Hh=0.
FOR I=1 TO S1
   IF Pf(I)>Pf1(I)+.0005 THEN 4520
   IF Pf(I)<Pf1(I)-.0005 THEN 4520
   Hh=Hh+1
   IF Hh=S1 THEN 4570
NEXT I
PRINT ""
GOTO 2600
PRINT "PIPE NUMBER MOD FRICTION FACTORS ROUGHNESS"
FOR I=1 TO S1
   PRINT USING 4610;S2(I),Pf(I),Rf(I)
NEXT I
IMAGE 5X,3D,12X,D.3D,12X,D.6D
**INITIAL SYSTEM CONDITIONS**

**TRANSIENT ANALYSIS**

```
J=0
J=J+1
FOR N=1 TO 5
  C7(J,N)=E4(J,N)
NEXT
IF J=2 THEN 5000
GOTO 4940
Z=Z+1
FOR L=1 TO 51
  Kk=S4(L)
  K=Kk
  A7=C8(L)*Pav(L)
  IF (T9(L)-1)>.0 THEN 5070
  GOTO 5150
  L=T9(L)
  Vr=V(K)+A7*(V(K)-V(K+1))
  Vs=V(K)+A7*(V(K+2)-V(K+1))
  Hr=H(K)+A7*(H(K)-H(K+1))
  Hs=H(K)+A7*(H(K+2)-H(K+1))
  K=K+1
NEXT J
IF I=1 THEN S1000
I=I+1
FOR I=1 TO S1
  K=S4(I)
  Kk=K+T9(I)
  IF E3(I)=0 THEN 5000
  IF S3=2 THEN 5000
  J=0
  J=J+1
  FOR N=1 TO 5
    C7(J,N)=E4(J,N)
  NEXT
  IF J=2 THEN S1000
  GOTO 4940
  Z=Z+1
  FOR L=1 TO 51
    K=S4(L)
    Kk=K+T9(L)
    Vr=V(K)+A7*(V(K)-V(K+1))
    Vs=V(K)+A7*(V(K+2)-V(K+1))
    Hr=H(K)+A7*(H(K)-H(K+1))
    Hs=H(K)+A7*(H(K+2)-H(K+1))
    K=K+1
  NEXT J
  IF I=1 THEN S1000
  I=I+1
  FOR I=1 TO S1
    K=S4(I)
    Kk=K+T9(I)
    PRINT USING 4760;S2(I),V(K),R6(K),V(Kk),R6(Kk)
  NEXT I
  PRINT CHR$(12)
  GOTO 4940
  S6=S4(S1)+T9(S1)
  T=T+Dt
  Z=0
  Ip=1
  IF E3(I)=0 THEN 5000
  IF S3=2 THEN 5000
  J=0
  J=J+1
  FOR N=1 TO 5
    C7(J,N)=E4(J,N)
  NEXT
  IF J=2 THEN 5000
  GOTO 4940
  Z=Z+1
  FOR I=1 TO S1
    K=S4(I)
    Kk=K+T9(I)
    PRINT USING 4760;S2(I),V(K),R6(K),V(Kk),R6(Kk)
  NEXT I
  PRINT CHR$(12)
  GOTO 4940
```
5219 \texttt{R6(K)=-C3/C4}
5220 \texttt{Vp(K)=0}
5221 \texttt{GOTO 5610}
5222 \texttt{FOR J=1 TO L}
5223 \texttt{R6(K)=H(K)-T6(I)+(J-1)*(T6(I)-T5(I))*Xx}
5224 \texttt{K=K+1}
5225 \texttt{NEXT J}
5226 \texttt{NEXT L}
5227 \texttt{FOR I=1 TO S1}
5228 \texttt{L=T9(I)+1}
5229 \texttt{Xx=1.0/T9(I)}
5230 \texttt{K=S4(I)}
5231 \texttt{FOR J=1 TO L}
5232 \texttt{R6(K)=H(K)-T6(I)+(J-1)*(T6(I)-T5(I))*Xx}
5233 \texttt{K=K+1}
5234 \texttt{NEXT J}
5235 \texttt{NEXT L}
5236 \texttt{FOR L=1 TO S6}
5237 \texttt{V(L)=Vp(L)}
5238 \texttt{H(L)=R6(L)}
5239 \texttt{NEXT L}
5240 \texttt{FOR I=1 TO S1}
5241 \texttt{L=T9(I)+1}
5242 \texttt{Xx=1.0/T9(I)}
5243 \texttt{K=S4(I)}
5244 \texttt{FOR J=1 TO L}
5245 \texttt{R6(K)=H(K)-T6(I)+(J-1)*(T6(I)-T5(I))*Xx}
5246 \texttt{K=K+1}
5247 \texttt{NEXT J}
5248 \texttt{NEXT L}
5249 \texttt{FOR L=1 TO S6}
5250 \texttt{V(L)=Vp(L)}
5251 \texttt{H(L)=R6(L)}
5252 \texttt{NEXT L}
5253 \texttt{FOR I=1 TO S1}
5254 \texttt{L=T9(I)+1}
5255 \texttt{Xx=1.0/T9(I)}
5256 \texttt{K=S4(I)}
5257 \texttt{FOR J=1 TO L}
5258 \texttt{R6(K)=H(K)-T6(I)+(J-1)*(T6(I)-T5(I))*Xx}
5259 \texttt{K=K+1}
5260 \texttt{NEXT J}
5261 \texttt{NEXT L}
5262 \texttt{FOR L=1 TO S6}
5263 \texttt{V(L)=Vp(L)}
5264 \texttt{H(L)=R6(L)}
5265 \texttt{NEXT L}
5266 \texttt{FOR I=1 TO S1}
5267 \texttt{L=T9(I)+1}
5268 \texttt{Xx=1.0/T9(I)}
5269 \texttt{K=S4(I)}
5270 \texttt{FOR J=1 TO L}
5271 \texttt{R6(K)=H(K)-T6(I)+(J-1)*(T6(I)-T5(I))*Xx}
5272 \texttt{K=K+1}
5273 \texttt{NEXT J}
5274 \texttt{NEXT L}
5275 \texttt{FOR L=1 TO S6}
5276 \texttt{V(L)=Vp(L)}
5277 \texttt{H(L)=R6(L)}
5278 \texttt{NEXT L}
5279 \texttt{FOR I=1 TO S1}
5280 \texttt{L=T9(I)+1}
5281 \texttt{Xx=1.0/T9(I)}
5282 \texttt{K=S4(I)}
5283 \texttt{FOR J=1 TO L}
5284 \texttt{R6(K)=H(K)-T6(I)+(J-1)*(T6(I)-T5(I))*Xx}
5285 \texttt{K=K+1}
5286 \texttt{NEXT J}
5287 \texttt{NEXT L}
5288 \texttt{FOR L=1 TO S6}
5289 \texttt{V(L)=Vp(L)}
5290 \texttt{H(L)=R6(L)}
5291 \texttt{NEXT L}
5292 \texttt{FOR I=1 TO S1}
5293 \texttt{L=T9(I)+1}
5294 \texttt{Xx=1.0/T9(I)}
5295 \texttt{K=S4(I)}
5296 \texttt{FOR J=1 TO L}
5297 \texttt{R6(K)=H(K)-T6(I)+(J-1)*(T6(I)-T5(I))*Xx}
5298 \texttt{K=K+1}
5299 \texttt{NEXT J}
5300 \texttt{NEXT L}
5301 \texttt{GOTO 5610}
SinglePipe:

R6(K) = (Rat * C3 - C1) / (C2 - Rat * C4)

R6(L) = R6(K)

Vp(K) = C3 + C4 * R6(K)

Vp(L) = C1 + C2 * R6(L)

RETURN

Nozzle:

IF Vp(K) >= 0 THEN 5900
GOTO 5950

C9 = Cc(I) / SQR(R5(I))

Aaa = ABS(C9^2 - (4 * C4 * (C3 + (C4 * T5(I)))))

D = 0.5 * (C9 / C4) * (C9 - SQR(Aaa))

Vp(K) = D

IF Vp(K) >= 0 THEN 5980
R6(K) = T5(I)
Vp(K) = C3 + C4 * T5(I)
GOTO 5990
R6(K) = (D - C3) / C4
RETURN

Branch:

J = ABS(J2(I))
FOR Jj = 1 TO 51
IF J - S2(Jj) = 0 THEN 6050
NEXT Jj
Const = C8(Jj) * Pav(Jj)
J = S4(Jj)
Vs2 = V(J) + Const * (V(J + 1) - V(J))
Hs2 = H(J) + Const * (H(J + 1) - H(J))
IF J2(I) = 0 THEN 6110
GOTO 6150

RETURN

Valve:

IF S3 = 1 THEN 6430
Ibv = Ibv + 1
IF Ibv = 1 THEN 6360
Ibv = 0
J = 0
FOR N = 1 TO 5
C7(J, N) = E7(J, N)
NEXT N
IF J = 2 THEN 6430
GOTO 6300
J = 0
6370 J=J+1
6380 FOR N=1 TO 5
6390 C7(J,N)=E4(J,N)
6400 NEXT N
6410 IF J=2 THEN 6430
6420 GOTO 6370
6430 IF T-C7(1,2)>0 THEN 6520
6440 IF T-C7(1,2)=0 THEN 6500
6450 IF T-C7(1,1)>0 THEN 6480
6460 A8=C7(2,1)
6470 GOTO 6730
6480 J=2
6490 GOTO 6710
6500 A8=C7(2,2)
6510 GOTO 6730
6520 IF C7(1,3)<=0 THEN 6500
6530 IF (T-C7(1,3))<0 THEN 6570
6540 IF (T-C7(1,3))>0 THEN 6590
6550 A8=C7(2,3)
6560 GOTO 6730
6570 J=3
6580 GOTO 6710
6590 IF C7(1,4)<=0 THEN 6530
6600 IF (T-C7(1,4))<0 THEN 6660
6610 IF (T-C7(1,4))=0 THEN 6640
6620 J=4
6630 GOTO 6710
6640 A8=C7(2,4)
6650 GOTO 6730
6660 IF (C7(1,5))<0 THEN 6640
6670 IF (T-C7(1,5))<0 THEN 6700
6680 A8=C7(2,5)
6690 GOTO 6730
6700 J=5
6710 Rati=(C7(2,J)-C7(2,J-1))/(C7(1,J)-C7(1,J-1))
6720 A8=C7(2,J-1)+Rati*(T-C7(1,J-1))
6730 IF A8=90 THEN 7050
6740 FOR J=1 TO 48
6750 A9=(J-3)*2
6760 IF (A9-A8)>0 THEN 6800
6770 IF (A9-A8)<0 THEN 6830
6780 X9=E3(J)
6790 GOTO 6840
6800 A9=A9-2.0
6810 X9=E3(J-1)+(E3(J)-E3(J-1))*0.5*(A8-A9)
6820 GOTO 6840
6830 NEXT J
6840 Xxx=1.0-X9^2
6850 A=C2*C4^2*Xxx
6860 B=2.0*(C2*C3*C4*Xxx-9.81*(C2-C4)*X9^2)
6870 C=C2*C3^2*Xxx+2.0*9.81*X9^2*(C3-C1)
6880 IF (B^2-4*A*C)>0 THEN 6910
6890 D=-B/(2*A)
6900 GOTO 6920
6910 Vp(K)=C3+C4*D
6920 IF Vp(K)>0 THEN 7010
6930 IF Vp(K)=0 THEN 7010
6940 B=2.0*(C2*C3*C4*Xxx+9.81*(C2-C4)*X9^2)
6950 C=C2*C3^2*Xxx-2*9.81*X9^2*(C3-C1)
6960 IF \( B^2-4AC > 0 \) THEN 6990
6970 D = -B/(2*A)
6980 GOTO 7000
6990 D = (-B + SQRT(B^2-4*A*C))/(2*A)
7000 \( V_p(K) = C_3 + C_4*D \)
7010 R6(K) = D
7020 R6(L) = (C_3 + C_4*D-C_1)/C_2
7030 \( V_p(L) = V_p(K) \)
7040 GOTO 7090
7050 \( V_p(K) = 0 \).
7060 \( R_6(K) = -C_3/C_4 \)
7070 \( R_6(L) = -C_1/C_2 \)
7080 \( V_p(L) = 0 \)
7090 RETURN
7100 Orifice:
7110 \( X_9 = \text{SQRT}(1/(C_5(K)+1)) \)
7120 X_9 = 1 - X_9^2
7130 A = C_2*C_4^2 * X_9
7140 B = 2.0 * (C_2*C_3*C_4*C_9*X_9 - 9.81*(C_2-C_4)*X_9^2)
7150 C = C_2*C_3^2*X_9 + 2*9.81*X_9^2*(C_3-C_1)
7160 IF \( B^2-4A*C > 0 \) THEN 7190
7170 D = -B/(2*A)
7180 GOTO 7200
7190 D = (-B - SQRT(B^2-4*A*C))/(2*A)
7200 \( V_p(K) = C_3 + C_4*D \)
7210 IF \( V_p(K) >= 0 \) THEN 7290
7220 B = 2.0 * (C_2*C_3*C_4*X_9 + 9.81*(C_2-C_4)*X_9^2)
7230 C = C_2*C_3^2*X_9 - 2*9.81*X_9^2*(C_3-C_1)
7240 IF \( B^2-4A*C > 0 \) THEN 7270
7250 D = -B/(2*A)
7260 GOTO 7280
7270 D = (-B + SQRT(B^2-4*A*C))/(2*A)
7280 \( V_p(K) = C_3 + C_4*D \)
7290 \( R_6(K) = D \)
7300 \( R_6(L) = C_3 + C_4*D-C_1)/C_2 \)
7310 \( V_p(L) = V_p(K) \)
7320 RETURN
7330 Press_reg:
7340 \( X_9 = \text{SQRT}(1/(X_8(K)+1)) \)
7350 X_9 = 1 - X_9^2
7360 A = C_2*C_4^2 * X_9
7370 B = 2.0 * (C_2*C_3*C_4*X_9 - 9.81*(C_2-C_4)*X_9^2)
7380 C = C_2*C_3^2*X_9 - 2*9.81*X_9^2*(C_3-C_1)
7390 IF \( B^2-4A*C > 0 \) THEN 7420
7400 D = -B/(2*A)
7410 GOTO 7430
7420 D = (-B - SQRT(B^2-4*A*C))/(2*A)
7430 \( V_p(K) = C_3 + C_4*D \)
7440 IF \( V_p(K) >= 0 \) THEN 7520
7450 B = 2.0 * (C_2*C_3*C_4*X_9 + 9.81*(C_2-C_4)*X_9^2)
7460 C = C_2*C_3^2*X_9 - 2*9.81*X_9^2*(C_3-C_1)
7470 IF \( B^2-4A*C > 0 \) THEN 7500
7480 D = -B/(2*A)
7490 GOTO 7510
7500 D = (-B + SQRT(B^2-4*A*C))/(2*A)
7510 \( V_p(K) = C_3 + C_4*D \)
7520 \( R_6(K) = D \)
7530 \( R_6(L) = C_3 + C_4*D-C_1)/C_2 \)
7540 \( V_p(L) = V_p(K) \)
RETURN

IF T-Ts<0 THEN 7580
GOTO 7840

IF X7-Z=0 THEN 7600
GOTO 7700

Wt3(Xy)=T
A=A+1
FOR S=1 TO S1
Kr(S)=S4(S)+T9(S)
Wt1(Xy,A)=R6(Kr(S))
Wt2(Xy,A)=Vp(Kr(S))
A=A+1
NEXT S

RETURN

FOR S=1 TO S1
K=S4(I)
Kk=K+T9(I)
PRINT USING 7800;S2(I),T,Vp(K),R6(K),Vp(Kk),R6(Kk)
IMAGE 2X,3D,4X,2D,2D,4(7X,4D,1D)
NEXT I

RETURN

FOR I=1 TO 51
Mxde=Wt1(I,I)
TB=1
FOR Xy=2 TO Ts/Dt/X7-1
IF Wt1(Xy,I)-Mxde<0 THEN 8010
TB=Xy
Mxde=Wt1(Xy,I)
NEXT Xy
Mxde=Wt1(TB,I)
PRINT USING 8040;S2(I);Mxde
IMAGE 5X,3D,6X,5D,D,#
Mide=Wt1(I,I)
TB=1
FOR Xy=2 TO Ts/Dt/X7
IF Wt1(Xy,I)-Mide>0 THEN 8110
TB=Xy
Mide=Wt1(Xy,I)
NEXT Xy
Mide=Wt1(TB,I)
PRINT USING 8140;Mide;(Mxde-Mide)
DISP "DO YOU WISH TO PLOT, STORE DATA OR BOTH?";
ON KEY 0 LABEL "PLOT" GOTO 8210
ON KEY 1 LABEL "STORE" GOTO 8230
ON KEY 4 LABEL "BOTH" GOTO 8250
GOTO 8170
GOSUB Plott
GOTO 9560
GOSUB Data_store
GOTO 9560
GOSUB Plott
GOSUB Data_store
GOTO 9560
Plott: 
PLOTTER IS 705, "HPGL"
OFF KEY
ALPHA OFF
OUTPUT 705; "VS"
LDIR 0
DISP "PIPE NO FOR WHICH PLOT REQUIRED?";
INPUT Ppn
FOR I=1 TO S1
IF S2(I)-Ppn=0 THEN 8390
Ppm=I
NEXT I
Nodn=S4(Ppm)+T9(Ppm)
DISP "NODE NO. YOU ARE PLOTTING IS"; Nodn;
A=Ppm
PEN 1
LINE TYPE 1
CSIZE 4
DISP "INPUT XMIN, XMAX, YMIN, YMAX";
INPUT X1, X2, Y1, Y2
DISP "INPUT XINTERVAL, YINTERVAL";
INPUT X5, Y5
VIEWPORT 18, 99*RATIO, 10, 97
WINDOW X1, X2, Y1, Y2
LDIR 0
OFF KEY
AXES X5, Y5, X1, Y1
LINE TYPE 1
VIEWPORT 0, 100*RATIO, 0, 100
LORG 6
PEN 1
FOR I=X1 TO X2 STEP X5
MOVE I, Y1
LABEL I
IF I=X2 THEN 8660
NEXT I
LORG 8
FOR I=Y1 TO Y2 STEP Y5
MOVE X1, I
LABEL I
NEXT I
LINE TYPE 8
MOVE X1, Y1
MOVE Wt3(1),Wt1(1,A)
8740 PEN 1
8750 FOR Xy=2 TO Ts/Dt/X7
8760 DRAW Wt3(Xy),Wt1(Xy,A)
8770 NEXT Xy
8780 VIEWPORT 18,100*RATIO,3,97
8790 IF D2=7 THEN 8830
8800 IF D2=8 THEN 9400
8810 DISP "NEW WINDOW NEEDED , INSERT YMIN,YMAX,YINT?";
8820 INPUT Y3,Y4,Y6
8830 LINE TYPE 1
8840 WINDOW X1,X2,Y3,Y4
8850 AXES XS,Y6,X2,Y1
8860 LORG B
8870 VIEWPORT 0,100*RATIO,0,100
8880 FOR I=Y3 TO Y4 STEP Y6
8890 MOVE X2,I
8900 LABEL I
8910 NEXT I
8920 MOVE Wt3(1),Wt2(1,A)
8930 LINE TYPE 5
8940 PEN 2
8950 FOR Xy=2 TO Ts/Dt/X7
8960 DRAW Wt3(Xy),Wt2(Xy,A)
8970 NEXT Xy
8980 MOVE 0,0
8990 LINE TYPE 4
9000 DRAW X2,0
9010 VIEWPORT 0,100*RATIO,0,100
9020 CSIZE 4
9030 WINDOW 0,1,0,1
9040 LINE TYPE 1
9050 PEN 1
9060 LORG 6
9070 MOVE .5,.05
9080 LABEL "TIME Secs"
9090 LDIR 64.4
9100 MOVE 0,.5
9110 LABEL "HEAD metres"
9120 LDIR 0
9130 CSIZE 3
9140 MOVE .6,1
9150 LABEL "END OF PIPE NO.";Ppn
9160 MOVE .6,.93
9170 LABEL "NODE NO.";Nodn
9180 LORG 3
9190 CSIZE 2
9200 MOVE 0,1
9210 LABEL Idno$
9220 MOVE .97,.50
9230 LDIR 64.4
9240 LORG 4
9250 CSIZE 4
9260 LABEL "VELOCITY m/sec"
9270 DISP "DO YOU NEED A TRANSPARANCY";
9280 ON KEY 8 LABEL "YES" GOTO 9320
9290 ON KEY 9 LABEL "NO" GOTO 9400
9300 ON KEY 7 LABEL "Y_NOVEL" GOTO 9370
9310 GOTO 9280
DISP "WAITING FOR A CONTINUE KEY!";
PAUSE
GOTO 8520
OUTPUT 705; "VS,7"
D2=8
GOTO 8520
OFF KEY
DISP "DO YOU NEED ANOTHER PLOT?";
ON KEY 0 LABEL "YES" GOTO 8280
ON KEY 1 LABEL "NO" GOTO 9450
GOTO 9420
RETURN

Data_store:
DISP "INPUT FILE NAME?, BETWEEN ""S"";
INPUT Nf$
Fl=INT((Ts/Dt/X7)*16*(S1+1))/1000)
F1=F1+5
CREATE BDAT Nf$, Fl
ASSIGN ~Pathl TO Nf$
FOR Xy=1 TO Ts/Dt/X7
FOR A=1 TO S1+1
OUTPUT ~Pathl; Wt1(Xy,A), Wt2(Xy,A), Wt3(Xy)
NEXT A
NEXT Xy
ASSIGN ~Path1 TO *
NEXT Xy
RETURN
END
(a) **Calculation of Closing Time of Poppet Type Valve**

Let the linear distance between seat and valve be \( s \) mm, and seat angle = \( \theta \)°.

Then the surface area of the frustrum of the cone is \( \pi s (R+r) \).

In this case, referring to the diagram:

\[
\text{Surface Area} = \frac{\pi l}{\sin \theta} \left\{ 2r + l \tan \theta \right\}
\]

Effective orifice area between poppet and cylinder = \( \frac{\pi}{4} (D_{\text{cyl}}^2 - D^2) \)

Effective orifice diameter, \( d_o = \sqrt{(D_{\text{cyl}}^2 - D^2)} \)

Pressure drop across valve = \( \frac{8Q^2 \rho}{C_d^2 \pi^2 d_o^4} \)

Acceleration of valve = \( \frac{8Q^2 \rho}{4} \times \frac{\pi D^2 \times 1}{\sqrt{\frac{8Q^2 \rho}{4}}} \)
\[ \frac{2 \Omega^2 \rho D^2}{C_D \pi D_0^4} = \frac{2 \Omega^2 \rho D^2}{C_D \pi m (D_{cyl}^2 - D^2)^{1/2}} \quad \ldots(1) \]

where

- \( \rho \) = density (kg/m\(^3\))
- \( \Omega \) = flow rate (m\(^3\)/sec)
- \( D \) = poppet diameter (m)
- \( D_{cyl} \) = cylinder diameter (m)
- \( C_D \) = orifice coefficient
- \( m \) = poppet mass (kg)

The valve closure affects the flow only when the effective orifice area between poppet and seat is equal to the area between poppet and cylinder.

Therefore

\[ \frac{n}{\sin \theta} \left( 2r + \tan \theta \right) = \frac{\pi}{4} (D_{cyl}^2 - D^2) \]

In this case \( = 6.86 \text{ mm} \)

Valve moves \((40 - 6.86) \text{ mm}\) before reaching this position with an acceleration of \(176.8 \text{ m/s}^2\) calculated using equation 1, Section 3.1. At this point the velocity is \(3.423 \text{ m/sec}\). If acceleration is assumed to be constant for the final \(6.86 \text{ mm}\), then closure of final stage takes \(1.91 \text{ ms}\).

Total time for closure over \(40 \text{ mm}\) stroke is \(21.3 \text{ ms}\).
(b) VALVE FUNCTION REQUIRED FOR THEORETICAL ANALYSIS

To allow all valves used experimentally to be compared on a similar basis, it is necessary to define a common function for all valves. The valve function, $V_f$, used was defined as:

$$V_f = \sqrt{\frac{1}{k\alpha + 1}}$$

where $k$ is a coefficient from the equation:

$$p = \frac{kV_f^2}{2g}$$

$\alpha$ is an area function to allow for difference in areas of the value and the pipe immediately upstream:

$$\alpha = \frac{(\text{Area of pipe immediately upstream})^2}{(\text{Valve fully open area})}$$

So if the values of $k$ against angle of opening is known $V_f$ can be calculated.

$k$ varies between zero and infinity which means $V_f$ varies between 1 when the valve is fully open and zero when it is closed.
(c) **Pressure drop in high pressure pipe**

The ten surface roughness samples shown in Figure 18 gave the average roughness as 12.79 ± 3.96 \( \mu \)m. The results of inside diameter measurements Table 1 gave an inside diameter of 40.998 ± 0.676 mm.

Roughness, \( e = \frac{\varepsilon}{d} = 0.000312 \)

Velocity, \( V = \frac{475/60000 \times 4}{0.040998^2} = 5.9969 \text{ m/sec} \)

At a temperature of 22°C, the kinematic viscosity of water is:

\( \mu = 0.97 \times 10^{-6} \text{ m}^2/\text{sec} \)

Reynolds number \( = \frac{Vd}{\mu} = \frac{5.9969 \times 0.040998}{0.97 \times 10^{-6}} = 253465 \)

Using equation 3, Section 3.1,

friction factor, \( f = \left\{-2\log\left[\frac{0.000312 - 5.02}{3.7}\right] \log 253465 \right\}^{-2} = 0.01736 \)

and using equation 1,

Pressure drop \( = \frac{0.01736 \times 5.9969^2 \times 998}{2 \times 0.040998} = 7.599 \text{ kPa/m} \)
CALCULATED WAVE VELOCITY IN PIPE

The pipe is considered to be a thick walled pipe since the ratio of thickness to diameter is greater than 0.04, (in this case 0.24). Appendix A equation 5 gives the wave velocity as

\[ C = k k_2 \frac{1}{\sqrt{1 + \psi}} \]

and for thick walled pipes not anchored,

\[ \psi = 2 \left( \frac{R_o^2 + R_i^2 + \psi}{R_o^2 - R_i^2} \right) \]

In this case, \[ \psi = 2 \left( \frac{60.8^2 + 40.998^2 + 0.27}{60.8^2 - 40.998^2} \right) = 5.876 \]

and \[ C = 1436.54 \text{ metres/sec at } 20^\circ\text{C and zero pressure.} \]

Using Table 1 in Appendix A which gives velocity ratios related to atmospheric pressure and 20°C the values of the wave velocity in the range measured experimentally can be calculated. In this example the water was considered to contain no undissolved gases and \( k_2 = 1 \).

<table>
<thead>
<tr>
<th>Temp °C</th>
<th>Pressure (MPa)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>0</td>
</tr>
<tr>
<td>20</td>
<td>1436.54</td>
</tr>
<tr>
<td>30</td>
<td>1462.40</td>
</tr>
</tbody>
</table>