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HYDRAULIC TRANSIENT IN A PIPELINE

Using Computer Model to Calculate and Simulate Transient

Master Thesis

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Picture on front page:

Pipeline failure due to the hydraulic transient in Lake Geneva

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ABSTRACT

A hydraulic transient, which is a flow condition where the velocity and pressure change rapidly with time, can collapse a water distribution system if that system is not equipped with adequate transient protection device(s). The occurrence of transients can introduce large pressure forces and rapid fluid accelerations into a water distribution system and if the system is not well protected, it can fail. A hydraulic transient normally occurs when a flow control component changes status (for example, a valve closing or pump stop), and this change flows through the system as a pressure wave. A valve can be closed in two ways; linear or stepwise (fast initial closure and slow subsequent closure). Pump stop could be due to planned stop, power failure or mechanical problem with the pump.

There are many transient analysis methods, but this thesis work employed only the graphical method and the method of characteristics to construct models (using the FORTRAN language) to calculate and simulate transients in a pipeline. Many scenarios and different valve closure operations were applied to the models (with a pump and without a pump in the pipeline) to study the transients. The model solutions were compared with the graphical solutions for the two transient flows.

It was found that the stepwise valve closure can reduce transients significantly than the linear valve closure operation. A pump with a high inertia can also reduce transients significantly, and this inertia can be achieved by using the stopstop program. The water velocity and the frictional coefficient were seen to be important factors which affect the hydraulic transient, even though both relate to each other. A high frictional coefficient reduces the velocity and as a results reduces the transients, and vice versa. Also low frictional coefficients lead to increase in water velocities and as a result increase in hydraulic transients, and vice versa. The graphical solution was found to agree well with the computer solution.

SUMMARY

Hydraulic transient is a flow condition where the flow velocity and pressure change rapidly (very fast) with time in pipelines filled with water. A hydraulic systems consists of many components, and when a flow control component changes status (for example, a valve closing or pump stop), it causes the change to move through the system as a pressure wave. When the system is not equipped with an adequate transient protection device(s) such as air vessel, surge tanks, etc to overcome the transient, it can cause the components of the system to fail. Hydraulic transient can collapse a water distribution system, and as such it is essential to analyse it in order to determine the values of the transient pressures that can result from flow control operations and to establish the design criteria for system equipment and devices so as to provide an acceptable level of protection against system failure due to pipe collapse or bursting.

A pump can stop due to power failure, pump failure or planned pump stop, and all of these have some effect on the transient. A valve can be closed in two ways; linear closure or stepwise closure (fast initial closure and then slow subsequent closure), and both have some effect on the transient. Stepwise valve closure operation can reduce transient than the linear valve closure operation.

The water velocity and the frictional coefficient also affect the hydraulic transient, even though both relate to each other. The higher the water velocity the higher would be the transient and vice versa. Also, a low frictional coefficient would lead to a high velocity and hence, high transient, and vice versa.

There are many transient analysis methods but the graphical method and the method of characteristics were used in this thesis work. Generally, the graphical method is considered to be one of the simplest and effective ways to calculate the hydraulic transient. The graphical method is also based on the method of characteristics.

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CHAPTER ONE

GENERAL INTRODUCTION

1.1 Background

A hydraulic transient is a flow condition where the velocity and pressure change rapidly with time. The occurrence of transients can introduce large pressure forces and rapid fluid accelerations into a water distribution system. When flow velocity changes rapidly because a flow control component changes status (for example, a valve closing or pump stop), it causes the change to move through the system as a pressure wave. Hydraulic transient can cause hydraulic equipments in a pipe network to fail, if adequate transient control measures are not in place to overcome the transient (if the pressure wave is strong enough).

Due to the devastating effects that a hydraulic transient can cause, its analysis is very important in determining the values of transient pressures that can result from flow control operations and to establish the design criteria for system equipment and devices so as to provide an acceptable level of protection against system failure due to pipe collapse or bursting.

Numerical models are used to analyse hydraulic transients due to the complexity of the equations needed to describe the transients. An effective numerical model allows the hydraulic engineer to analyse potential transient events and to identify and evaluate alternative solutions for controlling hydraulic transients, thereby protecting the integrity of the hydraulic system.

1.2 Objectives

The objectives of this work are to:

- develop a model to calculate hydraulic transients in a pipeline
- apply the model for different valve closure operations (linear and stepwise)

- apply the model with a pump in the pipeline
- apply many scenarios to study this phenomenon
- compare graphical solutions and computer solutions
- apply pump model to real cases

1.3 Scope of Work

In carrying out this work, the following research methodologies were applied

- Develop a model to calculate the hydraulic transient in a pipeline
- Compare the developed model with the graphical method
- Check the model for linear and step wise closure for different closure time, for instances $T < \mu$; $T \approx 2\mu,\ 5\mu,\ 10\mu,\ 20\mu,\ 50\mu,\ 100\mu,\ 200\mu,\ 400\mu$
- Check the model for a fixed closure time, $T \approx 5\mu$ but with different closure operations
- Check the model for the effect of different wave velocities, different frictional coefficient
- Modify the model in order to obtain maximum and minimum pressure envelopes
- Modify the model to allow for closure operation when the final valve status is not fully closed
- Calculate the hydraulic transients due to pump stop with controlled speed reduction of the motor (so called soft stop) and check valve closure.
- Modify the model to observe the effects of different rotational speed and time of pump stop
- Simulate real measurements (at Yndesäte and Kristinedal pump stations) by the model

1.4 Structure of Project Work

This project work is structured into six chapters.

Chapter one describes the background, objectives of the project work, research methodology, scope of work, and structure of the project work.

Chapter two reviews literature on FORTRAN programming and hydraulic transients. Chapter three looks at the research methodology. Results, and the discussions of the results are contained in chapters four and five respectively. Conclusions and recommendations are contained in chapter six.

CHAPTER TWO

LITERATURE REVIEW

2.1 Introduction to Transient Flow

In a water distribution system, system flow control is an integrated part of its operation, for instance, the opening and closing of valves, and starting and stopping of pumps. When these operations are performed very quickly, they can cause hydraulic transient phenomena to come into existence in the water distribution system, which can result in system damage or failure if the transients are not minimised.

When the steady state condition of a flow in a system is altered, the values of the initial flow conditions of the system, characterised by the measured velocity (V) and pressure (P) at positions along the pipeline (x), change with time (t) until the final flow conditions are established in a new steady-state condition.

From Figure 2.1, the physical phenomenon that occurs during the time interval T, between the initial and final steady-state conditions is known as the hydraulic transient. In general, surges are unsteady flows that results from relatively slow flow rate changes, and water hammer or hydraulic transients result from rapid flow rate changes.

Hydraulic transient evaluation is a complex task and it involves the determination of the values of the functions $V(x, t)$ and $P(x, t)$ during the time interval, T that resulted from a flow control operation performed in a time interval, Δt. In the case of hydraulic transients, density changes of the liquid (normally water) are essential to consider. In the case of surges (mass oscillations) the flow can be considered incompressible.

Figure 2.1 depicts how the transient evolution in a system looks like and it represents a view of the transient at a fixed point (x) just upstream of the valve that is being shut. In this graph, the pressure, P is represented as a function of time, t resulting from the operation of a control valve. In the figure, P_i is the initial pressure at the start of the transient, P_f is the final pressure at the end of the transient event, $P_{\min}$ is the minimum transient pressure, and $P_{\max}$ is the maximum transient pressure.

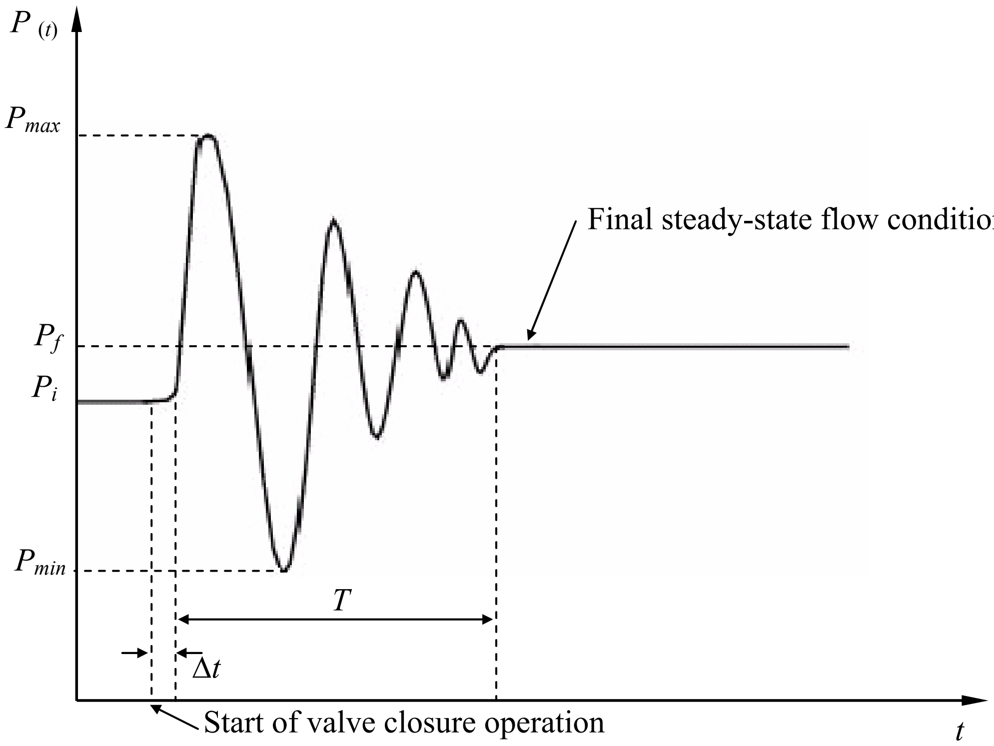

Figure 2.1 Hydraulic transient at position *x* in the system

2.1.1 Impacts of Transients

A wave is a disturbance that propagates energy and momentum from one point to another through a medium without significant displacement of the particles of that medium. A transient pressure wave, as in Figure 2.1, subjects system piping and other facilities to oscillating high and low pressures, and cyclic loads and these pressures can have a number of adverse effects on the hydraulic system.

Hydraulic transients can cause hydraulic equipments in a pipe network to fail if the transient pressures are excessively high. If the pressures are excessively higher then the pressure ratings of the pipeline, failure through pipe or joint rupture, or bend or elbow movement may occur. Conversely, excessive low pressures (negative pressures) can result in buckling, implosion and leakage at pipe joints during subatmospheric phases. Low pressure transients are normally experienced on the downstream side of a closing valve.

After pump stop in a pipeline with a check valve just downstream of the pump, the system flow reverses its direction towards the pump. Normally, in fairly long pipelines, the check valve will close when the flow just reverses (water velocity equals zero). However, in some situations the inertia of the valve causes it to delay its closure and the reverse flow causes it to close very rapidly through the remaining portion of its closure operation. This very fast, hard valve closure is known as valve slam. When the check valve is fully closed, it stops the reverse flow instantaneously, causing sometimes a loud noise (water hammer) in the pipeline. The noise associated with the slam is due to the impact of the disc into its seat. Surprisingly, a resilient-seated check valve can make the same metallic slam sound as a metal-seated valve (Val-Matic Valve and Manufacturing Crop., 2003). To prevent check valve slam, a check valve must either close very rapidly before appreciable reverse flow occurs or very slowly once reverse-flow has developed (Landon, 1993). In order to close rapidly, studies (Thorley, 1989) indicate that:

- The disc should have low inertia and friction,
- The travel of the disc should be short, or
- The motion should be assisted with springs.

To close slowly, a check valve needs to be equipped with external devices such as oil dashpots and the pump must be capable of withstanding some backspin. Dashpots devices have been proven to be effective (Val-Matic Valve and Manufacturing Crop., 2003).

Subatmospheric (low) pressure conditions increase the risk of some pipeline materials, diameters, and wall thickness to collapse (implode). Although the entire pipeline may not collapse, it can still damage the internal surface of some pipes by stripping the interior lining of the pipe wall. Even if the pipeline does not collapse, column separation (cavitation) could occur in a section of the pipeline, if the pipeline's pressure is reduced to the vapour pressure of the liquid. This column separation is caused by the differential flow in and out of a section of the pipeline. There are two distinct types of cavitation that can result and these are gaseous and vaporous cavitation.

Gaseous cavitation involves the formation of bubbles (dissolved air) and their collapse (coming out of the water). Gaseous cavitation causes small gas pockets formation in the pipe which could have the effect of dampening transients if these gas pockets are sufficiently large because they tend to dissolve back into the liquid slowly.

Vaporous cavitation is the boiling (vaporisation) of the water itself in the pipeline. Even liquid oxygen will boil and no one would ever call that hot. Fluids boil when the temperature of the fluid gets too hot or the pressure on the fluid gets too low. If you lower the pressure on the water it will boil at a much lower temperature and conversely if you raise the pressure the water will not boil until it gets to a higher temperature. In vaporous cavitation, a vapour pocket forms and then collapses in the pipe when the pipeline pressure increases due to more flow entering the region than leaving it. Collapse of the vapour pocket can cause a dramatic high pressure transient if the water column rejoins very rapidly, which can in turn cause the pipeline to rupture. Vaporous cavitation can also result in pipe flexure that damages pipe linings.

Cavitation can and should be avoided by installing appropriate protection equipment or devices in the system. To counteract vaporisation problems you must either decrease the suction head, lower the fluid temperature, or more generally slowdown velocity changes. The pipeline profile could also be important for the risk of cavitation.

Cyclic load occurs when the pressure fluctuations in the pipeline are very rapid, as in the case of water hammer. A large number of cyclic loads can results in pipe burst due to fatigue and pipeline fittings (bends and elbows) to dislodge, resulting in a leak or rupture. The occurrence of water hammer can release energy that sounds like someone pounding on the pipe with a hammer. This sound is due to the collapse of a large vapour cavity.

2.1.2 Causes of Hydraulic Transients

Hydraulic transients occur at flow changes (rapid) in pressurised conduits and this is due to

- Start and stop of pumps, especially stop due to power failure
- Load changes in hydropower plants
- Valves operations (shut-off valves)
- Check valve closure

- Air pockets in pipelines, especially at pump start
- Discharge of air through air vent, valves

2.1.3 Factors Affecting the Hydraulic Transients

The magnitude of the transient pressure peaks depends on many factors, and some of these factors are:

- Pipeline length, configuration. The longer the pipeline the stronger the hydraulic transients. Branched pipeline configuration is better in handling transients.
- Pipeline profile
- Rate of change of the flow. The more rapidly the flow changes, the higher are the generated hydraulic transients. Flow change depends on the valve operation, pump characteristics
- The elastic properties of the water and the pipes. Less elastic pipes are disadvantageous
- Possible contents of dissolved or gaseous gases in the water. Gas bubbles normally reduce transients
- Formation and appearance of vapour pockets (cavities) in the water
- Protective measures applied. These include surge chambers, air vessel, air valves, frequency-controlled pumps, etc.

2.2 History of Transient Analysis Methods

Various methods of analysis were developed for the problem of transient flow in pipes. They range from approximate analytical approaches whereby the nonlinear friction term in the momentum equation is either neglected or linearised, to numerical solutions of the nonlinear system. These methods can be classified as follows:

2.2.1 Arithmetic Method

This method neglects friction (Joukowsky, 1904; Allievi, 1925).

2.2.2 Graphical Method

This method neglects friction in its theoretical development but includes a means of accounting for it through a correction (Parmakian, 1963).

Generally, the graphical method is considered to be one of the simplest and effective ways to calculate the hydraulic transient. Basically there are some simplifications which must be done to make it possible to apply the graphical method. The transients are normally obtained only at the end points of the pipeline and the frictional losses are assumed to be concentrated to only one point (either the inlet or outlet of the pipeline depending on the problem), also the energy losses in the entire pipeline during the transient phase are thus approximated by means of the water velocity in one point (inlet or outlet) although the velocity in reality varies along the pipeline. The graphical method can be done by free hand or by the use of drawing software (for example AutoCAD), and that of course affects the results because it always depends on the person's accuracy. All of the graphical solutions in this thesis were performed as assignments during the unsteady water flow course.

2.2.3 Method of Characteristics

This method is the most popular approach for handling hydraulic transients. Its thrust lies in its ability to convert the two partial differential equations (PDEs) of continuity and momentum into four ordinary differential equations that are solved numerically using finite difference techniques (Gray, 1953; Streeter and Lai, 1962). The graphical method is also based on the method of characteristics.

2.2.4 Algebraic Method

The algebraic equations in this method are basically the two characteristic equations for waves in the positive and negative directions in a pipe reach, written such that time is an integer subscript (Wylie and Streeter, 1993).

2.2.5 Wave-Plan Analysis Method

This method uses a wave-plan analysis procedure that keeps track of reflections at the boundaries (Wood, Dorsch, and Lightner, 1966).

2.2.6 Implicit Method

This implicit method uses a finite difference scheme for the transient flow problem. The method is formulated such that the requirement to maintain a relationship between the length interval Δx and the time increment Δt is relaxed (Amein and Chu, 1975).

2.2.7 Linear Methods

By linearising the friction term, an analytical solution to the two PDEs of continuity and momentum may be found for sine wave oscillations. The linear methods of analysis may be placed in two categories: the impedance method, which is basically steady-oscillatory fluctuations set up by some forcing function, and the method of free vibrations of a piping system, which is a method that determines the natural frequencies of the system and provides the rate of dampening of oscillations when forcing is discontinued (Wylie and Streeter, 1993).

2.2.8 Perturbation Method

With this method, the nonlinear friction term is expanded in a perturbation series to allow the explicit, analytical determination of transient velocity in the pipeline. The solutions are obtained in functional forms suitable for engineering uses such as the determination of the critical values of velocity and pressure, their locations along the pipeline, and their times of occurrence (Walski, Chase, Savic, Grayman, Beckwith and Koelle, 2003).

2.3 Physics of Transient Flow

A hydraulic transient is generated when the flow momentum of the transported liquid changes due to the rapid operation of the flow control device in the hydraulic system. Mathematically hydraulic transient is analysed by solving the velocity $V(x, t)$ and pressure

P(x, t) equations for a well-defined elevation profile of the system, given certain initial and boundary conditions determined by the system flow control operations. In other words, the main goal is to solve a problem with two unknowns, velocity (V) and pressure (P), for the independent variables position (x) and time (t). Alternatively, the equations may be solved for flow (Q) and head (H).

The continuity equation and the momentum equation are needed to determine V and P in a one-dimensional flow system. Solving these two equations produces a theoretical result that usually reflects actual system measurements if the data and assumptions used to build the numerical model are valid.

2.4 Water Hammer Theory

Water hammer refers to the transient conditions that prevail following rapid system flow control operations. It can be used beneficially, as in the case of a hydraulic ram, which is a pump that uses a large amount of flowing water to temporarily store elastic energy for pumping a small amount of water to a higher elevation. More commonly, the destructive potential of water hammer is what attracts the attention of water engineers.

To generate equations describing the water hammer phenomenon, the unsteady momentum and mass conservation equations are applied to flow in a frictionless, horizontal, elastic pipeline. First, the momentum equation is applied to a control volume at the wave front following a disturbance caused by downstream valve action. The following equation may be developed, which is applicable for a wave propagating in the upstream direction:

$$\Delta p = -\rho a \Delta V \text{ or } \Delta H = -\frac{a}{g}\Delta V \tag{2.1}$$

where Δp = change in pressure, Pa

ρ = fluid density, kg/m^3

a = characteristic wave celerity of the fluid, m/s

ΔV = change in fluid velocity, m/s

ΔH = change in head, m

It can be seen from the equation that a valve action causing a positive velocity change will result in reduced pressure. Conversely, if the valve closes (producing a negative ΔV), the pressure change will be positive.

By repeating this step for a disturbance at the upstream end of the pipeline, a similar set of equations may be developed for a pulse propagating in the downstream direction:

$$\Delta p = \rho a \Delta V \quad \text{or} \quad \Delta H = \frac{a}{g}\Delta V \tag{2.2}$$

These equations are valid at a section in a pipeline in the absence of wave reflection. They relate a velocity pulse to a pressure pulse, both of which are propagating at the wave speed a. Assume that an instantaneous valve closure occurs at time t = 0. During the period L/a (the time it takes for the wave to travel from the valve to the pipe entrance), steady flow continues to enter the pipeline at the upstream end. The mass of fluid that enters during this period is accommodated through the expansion of the pipeline due to its elasticity and through slight changes in fluid density due to its compressibility.

The following equation for the numerical value of a is obtained by applying the equation for conservation of mass.

$$a = \sqrt{\frac{\dfrac{E_v}{\rho_v}}{1 + \dfrac{E_v}{E_p}.\dfrac{D}{e}.C}} \tag{2.3}$$

where E_v and E_p are the volumetric modulus of elasticity of the fluid and the pipe material (Pa) respectively, ρ_v is the density of the liquid (Kg/m^3), D and e are the internal diameter and the wall thickness (m) of the pipe respectively, and C = constant, which depends on the axial movement of the pipe. In practical calculations, $C \approx 1$.

2.5 Full Elastic Water Hammer Equations

The water hammer equations are one-dimensional unsteady pressure flow equations given by (Wylie and Streeter, 1993):

$$\frac{\partial H}{\partial t}+\frac{a^2\partial Q}{gA\partial x}=0 \qquad (2.4)$$

$$\frac{\partial Q}{\partial t}+gA\frac{\partial H}{\partial x}+\frac{fQ|Q|}{2DA}=0 \qquad (2.5)$$

In order to model the transient situation in a system, one has to solve these equations for a wide variety of boundary conditions of that system and its topologies. The full elastic water hammer equations cannot be solved analytically except by some approximate methods.

2.6 Transient Control

Ideally a hydraulic system design process will include an adequate investigation and specification of equipment and operational procedures to avoid undesirable transients. However, in reality, transients will still occur despite the design parameters; hence, remedial measures are required to keep transient conditions from seriously disturbing the proper functioning of an existing system. Unexpected sources of unsteady flow also appear in some newly constructed systems (Wylie and Streeter, 1993).

Two possible strategies for controlling transient pressures exist (Walski, Chase, Savic, Grayman, Beckwith and Koelle, 2003):

1. Focus on minimising the possibility of transient conditions during project design by specifying appropriate system flow control operations and avoiding the occurrence of emergency and unusual system operations.

2. To install transient protection devices to control potential transients that may occur due to uncontrollable events such as power failures and other equipment failure.

Systems that are protected by adequately designed surge tanks are generally not adversely impacted by emergency or other unusual flow control operations because operational failure of surge tank devices is unlikely. In systems protected by hydropneumatic tanks, however, an air outflow or air compressor failure can occur and lead to damage from transients. Consequently, potential emergency situations and failures should be evaluated and avoided to the extent possible through the use of alarms that detect device failures and control systems that act to prevent them.

Water usage and leaks in a distribution system can result in a dramatic decay in the magnitude of transient pressure effects.

2.7 Protection Devices

To the extent possible, the engineer would like to design flow control equipment such that serious transients are prevented. Using a transient model, the engineer can try different valve operating speeds, pipe sizes, and pump controls to see if the transient effects can be controlled to acceptable levels. If transients cannot be prevented, specific devices to control transients may be needed. A brief overview of various commonly used surge protection devices and their functions is provided in Table 2.1.

Some methods of transient prevention include (Walski, Chase, Savic, Grayman, Beckwith and Koelle, 2003):

- Slow opening and closing of valves: Generally, slower valve operating times are required for longer pipeline systems. Operations personnel should be trained in proper valve operation to avoid causing transients.

- Proper hydrant operation: Closing fire hydrants too quickly is the leading cause of transients in smaller distribution piping. Fire and water personnel need to be trained on proper hydrant operation.

- Proper pump controls: Except for power outages, pump flow can be slowly controlled using various techniques. Ramping pump speeds up and down with soft starts or variable-speed drives can minimize transients, although slow opening and closing of pump control valves downstream of the pumps can accomplish a similar effect, usually at lower cost. The control valve should be opened slowly after the pump is started and closed slowly prior to shutting down the pump.

- Lower pipeline velocity: Pipeline size and thus cost can be reduced by allowing higher velocities. However, the potential for serious transients increases with decreasing pipe size. It is usually not cost-effective to significantly increase pipe size to minimize transients, but the effect of transients on pipe sizing should not be ignored in the design process.

To control minimum pressures, the following can be adjusted or implemented; Pump inertia, Surge tanks, Air chambers, One-way tanks, Air inlet valves, and Pump bypass valves. To control maximum pressures, the following can be implemented; Relief valves, Anticipator relief valves, Surge tanks, Air chambers, and Pump bypass valves. These items can be used singly or in combination with other devices (Walski, Chase, Savic, Grayman, Beckwith and Koelle, 2003).

2.7.1 Pump Inertia

Pump inertia is the resistance of the pump to acceleration or deceleration. Pump inertia is constant for a particular pump and motor combination. The higher the inertia of a pump, the longer it takes for the pump to stop spinning following its shutoff and vice versa. Larger pumps have more inertia because they have more rotating mass. Pumps with higher inertias can help to control transients because they continue to move water through the pump for a longer time as they slowly decelerate. Pump inertia can be increased through the use of a flywheel. For long systems, the magnitude of pump inertia needed to effectively control transient pressures makes this control impractical due to the mechanical problems associated with starting high inertia pumps. Therefore, increasing pump inertia is not recommended as an effective option for controlling transient pressures for long piping systems (Walski, Chase, Savic, Grayman, Beckwith and Koelle, 2003).

2.7.2 Air Chambers and Surge Tanks

Air chambers and surge tanks work by allowing water out of the system during high-pressure transients and adding water during low pressure transients. They should be located close to a point where the initial flow change is initiated.

An air chamber is a pressure vessel that contains water and a volume of air that is maintained by an air compressor. During pump stop, the pressure and flow in the system decreases and as a result the air in the air chamber expands, forcing water from it into the system.

A surge tank is a relatively small open tank connected to the hydraulic system. It is located such that the normal water level elevation is equal to the hydraulic grade line elevation. During pump stop, the surge tank substitutes the pump and by gravity feeds the system with water. This controls the magnitude of the low pressure transient generated as a result of the pump stop.

2.7.3 One-Way Tank

This is a storage vessel under atmospheric pressure that is connected to the hydraulic system. It has a check valve (normally closed) connected to it which only allows water from the tank into the system. One-way tanks are primarily used in conjunction with pumping plants (Wylie and Streeter, 1993). The significant advantage of using a one-way tank rather than a surge tank is that the check valve allows the one-way tank to have a much lower height (Walski, Chase, Savic, Grayman, Beckwith and Koelle, 2003).

2.7.4. Pressure Relief and Other Regulating Valves

A pressure relief valve is a self-operating valve that is installed in a system to protect it from over pressurisation of the system. It is designed to open (let off steam) when safe pressures are exceeded, then closes again when pressure drops to a preset level. Relief valves are designed to continuously regulate fluid flow, and to keep pressure from exceeding a preset value.

An anticipator relief valve can be used instead of a pressure relief valve to control high pressure transient peaks. It is essential for protecting pumps, pumping equipment and all applicable pipelines from dangerous pressure surges caused by rapid changes of flow velocity within a pipeline, due to abrupt pump stop caused by power failure. Power failure to a pump will usually result in a down surge in pressure, followed by an up surge in pressure. The surge control valve opens on the initial low pressure wave, diverting the returning high pressure wave from the system. In effect, the valve has anticipated the returning high pressure wave and is open to dissipate the damage causing surge. The valve will then close slowly without generating any further pressure surges (M&M Control Service, INC).

Air inlet valves are installed at high points along the pipeline system to prevent vacuum conditions and potential column separation. Air enters the pipeline system during low pressure transient, and this air should be expelled slowly to avoid creating another transient condition. Before restarting the pumps, an adequate time should be allowed for the air that entered the pipeline to be expelled. There are varieties of valves that allow air to enter and leave a system, and their names depend on the manufacturer. These valves include air inlet valves, air release valves, vacuum relief valves, air vacuum valves, and vacuum breaker valves (Walski, Chase, Savic, Grayman, Beckwith and Koelle, 2003).

2.7.5 Booster Pump Bypass

Pump bypass with a valve is another protective device against pressure transients. Two pressure waves are generated as a result of reduction in flow due to booster pump stop; the wave travelling upstream is a positive transient, and the wave that travels downstream is a negative transient. A check valve in a bypass line allows free flow to the pipeline to prevent low pressures and column separation (Wylie and Streeter, 1993). The effectiveness of using a booster station bypass depends on the specific booster pumping system and the relative lengths of the upstream and downstream pipelines (Walski, Chase, Savic, Grayman, Beckwith and Koelle, 2003).

Table 2.1 Primary attributes and design variables of key surge-protection approaches

Protection Approach	Primary Attributes	Decision Variables
Check valve	Limits flow to one direction Permits selective connections Prevents/limits line draining	Size and location Specific valve configuration Antishock (dampening) characteristics
Pump bypass line	Permits direct connection and flow around a pump Can limit up-and-down surge	Size and location Exact points connected Check-valve properties
Open surge tank	Permits inflow/outflow to external storage May require water circulation Can limit up-and-down surge	Size and location Connection properties Tank configuration Overflow level
Closed surge tank (air chamber)	As pressure changes, water exchanged so volume of pressurised air expands or contracts Needs compressor	Location Volume (total/water/air) Configuration/geometry Orifice/connector losses
Feed tank (one-way tank)	Permits inflow into line from an external source Requires filling	Size and location Connection properties Tank configuration
Surge anticipation valve	Permits discharge to a drain Has both high- and low-pressure pilots to initiate action May accentuate down surge	Size and location High-and low- pressure set points Opening/closing times
Combination air-release and vacuum-breaking valve	When pressure falls, large orifice admits air Controlled release of pressurised air through an orifice	Location Small and large orifice size Specific valve configuration
Pressure-relief valve	Opens to discharge fluids at a preset pressure valve Generally opens quickly and closes slowly	Size and location High-pressure set point Opening/closing times

(Source: American Water Works Association, 2005)

2.8 Valve

A valve is a device used to control the flow of water. The control is achieved by closing, opening or partially obstructing various passageways. Valves have many applications and plumbing valves are the most commonly used valves in everyday life. Technically, valves are considered to be pipe fittings, and there are many different valve designs. Each of the many different valve designs has its own advantages and disadvantages. The gate valve slides up and down like a gate, the globe valve closes a hole placed in the valve, the angle valve is a globe valve with a 90° turn, and the check valve allows the fluid to flow only in one direction (Tuner and Çengel, 2005).

From a functional point of view, valves can be divided into the following groups (L. Jönsson and P. Larsen, 1975):

1. Valves which are either completely open or completely closed (on – off function)
2. Valves which can be used for a continuous control of the flow
3. Valves which only allow flow in one direction
4. Valves for special purposes such as pressure reducing valves, safety valves, air valves.

In a hydraulic system, valves are considered to be head loss, and this loss is described as:

$$h_{valve} = K_{valve} \frac{v^2}{2.g} \tag{2.6}$$

where K_{valve} = valve loss coefficient

v = cross sectional average velocity in the pipe close to the valve

The loss coefficient, K_{valve} as a function of the valve opening is normally experimentally determined for steady-state flows but is assumed to be applicable for unsteady conditions too. It has the value of $0 < K_{valve} < \infty$. The numerical value of K_{valve} might be obtained from valve manufacturers or could be estimated using simplifying assumptions. A check valve is normally described as having a low (or zero) loss coefficient when it is open, whereas a closed check valve is described by the condition that the flow is zero. $K_{valve} \approx 0$ when the valve is fully opened, and $K_{valve} = \infty$ when the valve is completely closed.

2.9 Pump

A pump is a device used to move liquids, or slurries from one place to another. A pump moves liquids or gases from a lower pressure region to a higher pressure region. It adds energy to the system (such as a water system) to overcome this difference in pressure and friction. In general, pumps fall into two major groups and these are rotodynamic pumps and positive displacement (reciprocating) pumps of which rotodynamic pumps are the most common ones.

2.9.1 Rotodynamic pumps

These are dynamic devices for increasing the pressure of a liquid. When the liquid passes through the pump, the kinetic (rotational) energy from the pump's motor is transferred through the impeller, propeller or rotor of the pump into fluid pressure (potential energy). Rotodynamic pumps generate the same head of liquid regardless of the density of the liquid being pumped. Rotodynamic pumps are divided into three main types: centrifugal pumps (radial flow pumps), mixed flow pumps, and axial (propeller) flow pumps.

Centrifugal pumps (radial flow pumps) utilise centrifugal force to move a fluid from one place to the other. They are the most common of the rotodynamic pumps. A centrifugal pump converts its electric motor's energy into kinetic energy and then into pressure of the fluid being pumped. They are used for a wide variety of purposes, such as pumping liquids for water supply, irrigation, and sewage disposal systems. Axial flow pump (the pressure is developed by the propelling or lifting action of the vanes of the impeller on the liquid), and Mixed flow pump (the pressure is developed partly by centrifugal force and partly by the lift of the vanes of the impeller on the liquid).

2.9.2 Positive displacement (displacement) pumps

In displacement pumps the pumping action is achieved by means of a part moving back and forth or with a rotating movement, whereby a fixed volume of water is trapped and pushed (displaced) out into the discharge pipe. Positive displacement pumps can be further classified as either rotary-type (for example the rotary vane pump) or reciprocating type (for example the diaphragm pump).

2.10 Pump Characteristics Curve

A pump characteristics curve is a diagram supplied by the pump's manufacturer to describe the relationship between the head and the capacity of the pump using various size impellers. Each pump has its own characteristics curve, and a typical pump curve is as shown in Figure 2.2. The curve also includes information about efficiency, horse power consumption, Net Positive Suction Head Required (N.P.S.H.$_{R.}$), etc. The Net Positive Suction Head Required (N.P.S.H.$_{R.}$) is the minimum absolute pressure at the suction nozzle at which the pump can operate, and each pump has a specific N.P.S.H.$_R$. To avoid pump cavitation, the N.P.S.H.$_A$ of the system must be greater than the N.P.S.H.$_R$ of the pump. In other words, the available N.P.S.H. must be higher than the required. Pump curves are generated by tests performed by the pump manufacturer and are based on a specific gravity of 1.0. Other specific gravities must be considered by the user (www.pumpworld.com).

The system curve represents the effect of flow rate, geometric head and hydraulic losses in a system in a graphic form. Hydraulic losses in piping systems are normally composed of pipe friction losses, valves, elbows and other fittings, entrance and exit losses, and losses from changes in pipe size by enlargement or reduction in diameter. The system curve must be developed by the user based upon the conditions of service, which include physical layout, process conditions, and fluid characteristics.

$$\text{System curve} = \Delta Z + \mathit{coefficient}.\ Q^2 \tag{2.7}$$

where ΔZ is the pressure difference (normally geometric height difference) between the reservoirs, *coefficient* is the hydraulic losses in the system, and Q is the flow rate.

The operating point is the point of intersection of the pump curve and system curve. The head that corresponds to the operating point is the steady state head, and the flow rate corresponding to the operating point is the steady state flow rate. The flow rate of a pump increases as the required head decreases.

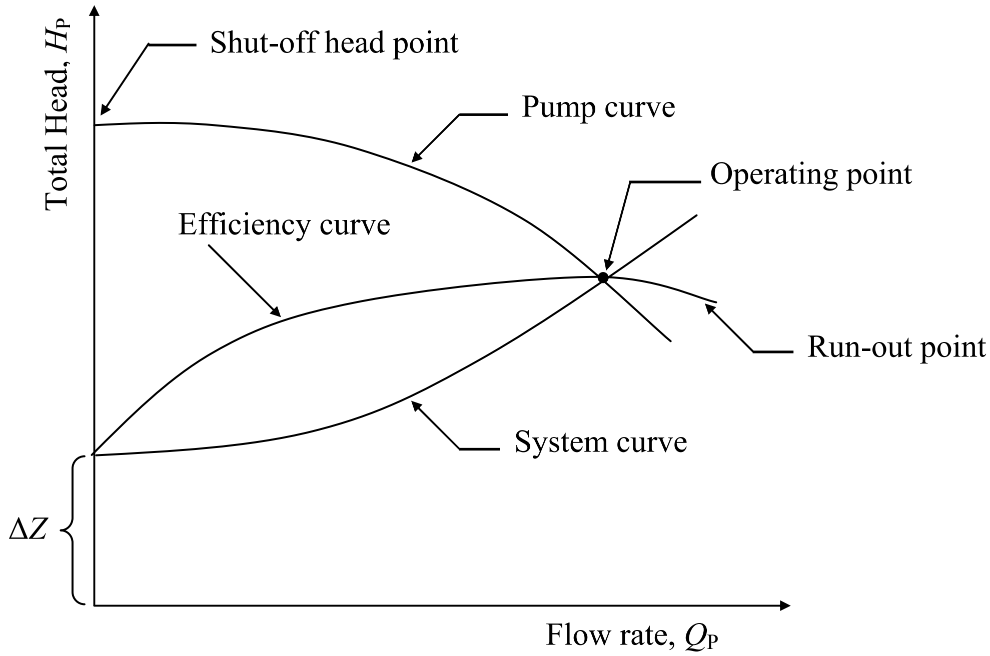

Figure 2.2 Typical system and pump characteristics curves

The shut-off head point is the head of the pump at no flow rate. The run-out point is the point where the flow rate of the pump is maximum and the head minimum, and beyond this point, the pump cannot operate. From the shut-off head point to the run-out point gives the pump's range of operation. Trying to run a pump off the right end of the curve will result in pump cavitation and eventually destroy the pump.

2.11 Suter Curve

This curve describes the general pump operation in a dimensionless form for three typical pumps, characterised by three different specific speeds. Figure 2.3 shows the complete characteristic curve (Suter curve). It describes the behaviour of the pump for all combinations of positive and negative discharge and speed. It is possible to control the operation of the pump by an initial discharge, speed and pressure head up- and downstream of the pump. There are several ways to control the drive of the pump in time: from a simple pump trip to the repeatedly stopping and starting of the pump, complete with motor characteristic and frequency-converter.

Turbine Zone	Dissipation Zone	Normal Zone	Reversed Speed Dissipation Zone
$v \leq 0$ $\alpha < 0$	$v < 0$ $\alpha \geq 0$	$v \geq 0$ $\alpha \geq 0$	$v > 0$ $\alpha < 0$

0 $\pi/2$ π $3\pi/2$ 2π

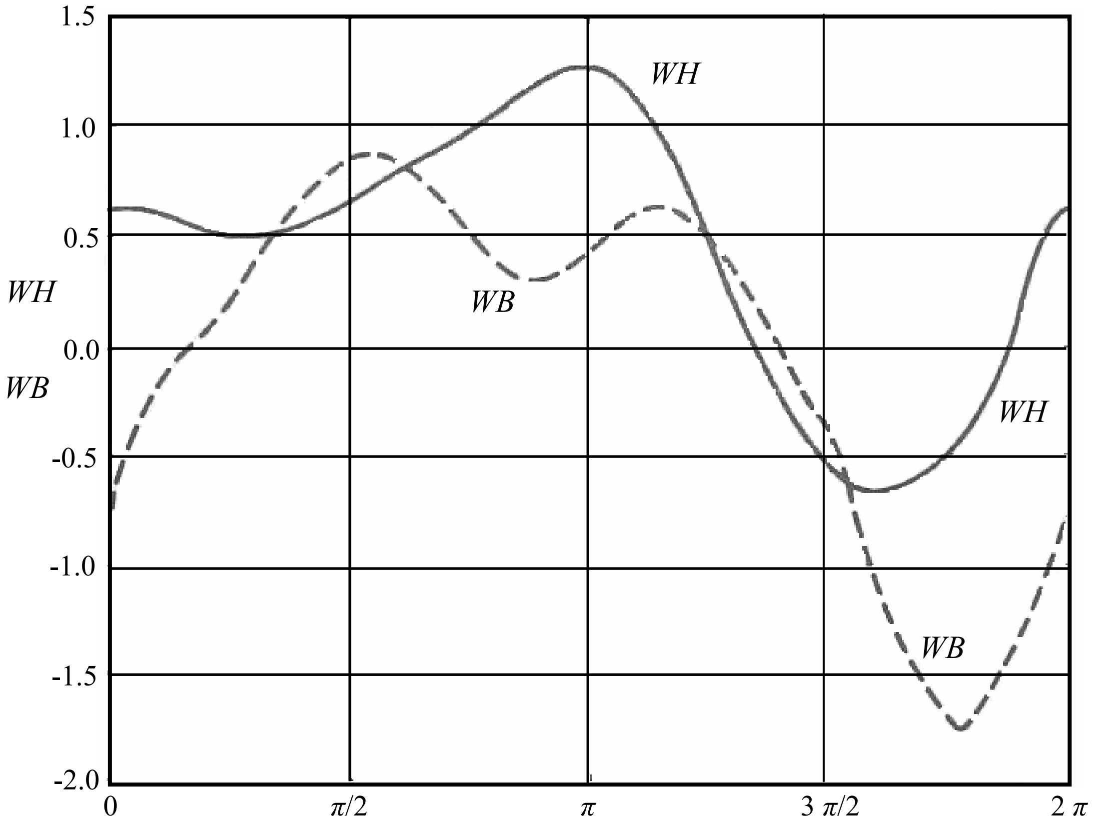

Figure 2.3 Complete pump characteristics curves (Suter Curve)
(Source: Wylie and Streeter, 1993)

2.12 Frequency Controlled Pump Systems (Soft -stop/Soft-start of Pumps)

Frequency controlled drive systems have become an important element in the field of drive systems enabling configurations with gradual speed control from zero rotations up to the rated speed or vice versa, i.e. soft start and soft stop respectively. The rotational speed of a pump can be controlled by controlling the frequency of the electric power supplied to its motor. When changing the frequency of the power to an AC motor, the ratio of the applied voltage to the applied frequency (V/Hz) is generally maintained at a constant value between the minimum and maximum operating frequencies. Operating at a constant voltage (reduced V/Hz) above a given frequency provides reduced torque capability and constant power capability above that frequency. The frequency or speed at which constant voltage operation begins is called the base frequency or speed.

Modern, highly developed pump systems call for the need to maintain the highest possible discharge or flow rate while keeping all system conditions such as the pressure values on the suction and discharge side of the pump within the permitted limits. Other systems demand a continuous supervision of the actual flow rate or the mechanical and thermal strain, i.e. vibrations, cavitation, structure born noise (SAM Electronics, 2004).

These requirements make it necessary to control the pump system. Rather than by throttling or by-passing of the pump, frequency controlled drive systems allow to adjust the speed of the pump in a wide range, thus allowing always a highly flexible and efficient operation.

Frequency controlled pump systems were originally introduced to:

- Adjust the capacity of the pump to the required flow rate, and
- Avoid the strong current peaks which occur at pump start (Soft start).

Later on soft stop was introduced as a way to control hydraulic transients at pump stop.

2.13 Piping Network

Piping systems normally consists of several pipes (some of different diameters) connected to each other in series or parallel which many involve several sources (supply of fluid into the system) and loads (discharges of fluid from the system). When the pipes are connected in series, the flow rate through the entire system remains constant regardless of the diameters of the individual pipes in the system. The total head loss in this case is equal to the sum of the head losses in individual pipes in the system, including the minor losses.

Head loss is combined of two major components; friction losses and minor losses. Friction losses are head losses due to the friction that the walls of the pipe impose on a liquid. Friction losses are dependent on the viscosity of the fluid and the turbulence of the flow. Head loss due to friction, h_f can be calculated using the Darcy-Weisbach equation:

$$h_f = f.\frac{L}{D}.\frac{u^2}{2g} \tag{2.8}$$

where f is the friction factor, L is the pipe length, u is the average velocity, D is the internal pipe diameter, and g the acceleration due to gravity

The friction factor, f, can be determined if you know the relative roughness of the pipe, k_s/D, and by solving for the Reynolds number, Re, and using the Moody Chart, which can be found in many fluid mechanics books. The Reynolds number can be found using the following equation:

$$\mathrm{Re} = \frac{\rho.u.D}{\mu} = \frac{u.D}{V} \tag{2.9}$$

where ρ = density of the fluid, μ = dynamic viscosity of the fluid, u = avearge fluid velocity, D = pipe diameter, and V = kinematic viscosity of the fluid.

For a pipe that branches out into two (or more) parallel pipes and then rejoins at a junction downstream (Figure 2.5), the total flow rate is the sum of the flow rates in the individual pipes.

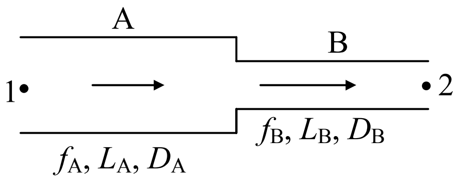

Figure 2.4 Pipes connected in series

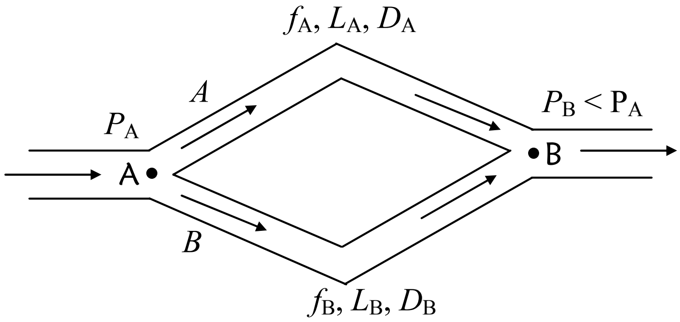

Figure 2.5 Pipes connected in parallel

Piping systems often involve sudden (gradual) expansion or contraction sections to accommodate change in flow rate or properties such as density and velocity. The loss coefficient, K_L for the case of sudden expansion (Figure 2.4) is determined to be

$$K_L = \left(1 - \frac{A_B}{A_A}\right)^2 \tag{2.10}$$

where A_B and A_A are the cross-sectional areas of the small and large pipes, respectively. $K_L = 0$ when there is no change ($A_B = A_A$) and $K_L = 1$ when a pipe discharges into a reservoir (A_A would be far larger than A_B). For sudden contraction, the loss coefficients are read from charts.

Pipeline system layouts should be designed in such a way as to avoid high points that are susceptible to air accumulation or exposure to low pressures (or both). In view of this, routes of layouts with undulating topographic profiles should be change to avoid these high points. Transient protection devices are normally unnecessary if the minimum transient head grade line is above the topographical profile of the piping system. For example, low-head systems with buried steel pipelines and diameter/thickness ratios (D/e) more than 200 should be avoided because of the risk of structural collapse during a transient vacuum condition, particularly if the trench fill is poorly compacted. Piping systems constructed above ground are more susceptible to collapse than are buried pipelines. With buried pipelines, the surrounding bedding material and soil provide additional resistance to pipeline deformations and help the pipeline resist structural collapse (Walski, Chase, Savic, Grayman, Beckwith and Koelle, 2003).

Corrosion weakens pipes and vapour separation causes pipes (steel, PVC, HDPE, and thin-wall ductile iron pipes) to experience fatigue failure, and any pipe exposed to these conditions may fail. Steel pipes are more economical than ductile iron pipes for large diameter pipes under high pressures. Wave celerity and the liquid velocity influence transient heads, hence, larger diameter pipes can be selected to obtain lower velocities so as to minimise transient heads for short pipelines delivering relatively low flows. Hydraulic systems with long pipelines almost always require transient protection devices and the diameter of the pipes should be selected to minimise construction and operating costs. The pipelines in a hydraulic system must be able to withstand the high transient peaks.

CHAPTER THREE

RESEARCH METHODOLOGY

The FORTRAN language (see appendix 1) was used to construct models to calculate pressure transient including friction. The principle is that the pipeline, in which transient should be studied, is divided into a number of equally long parts Δx. In Figure 3.1 the pipeline is represented by the x-axis between the coordinates *0* and *L*.

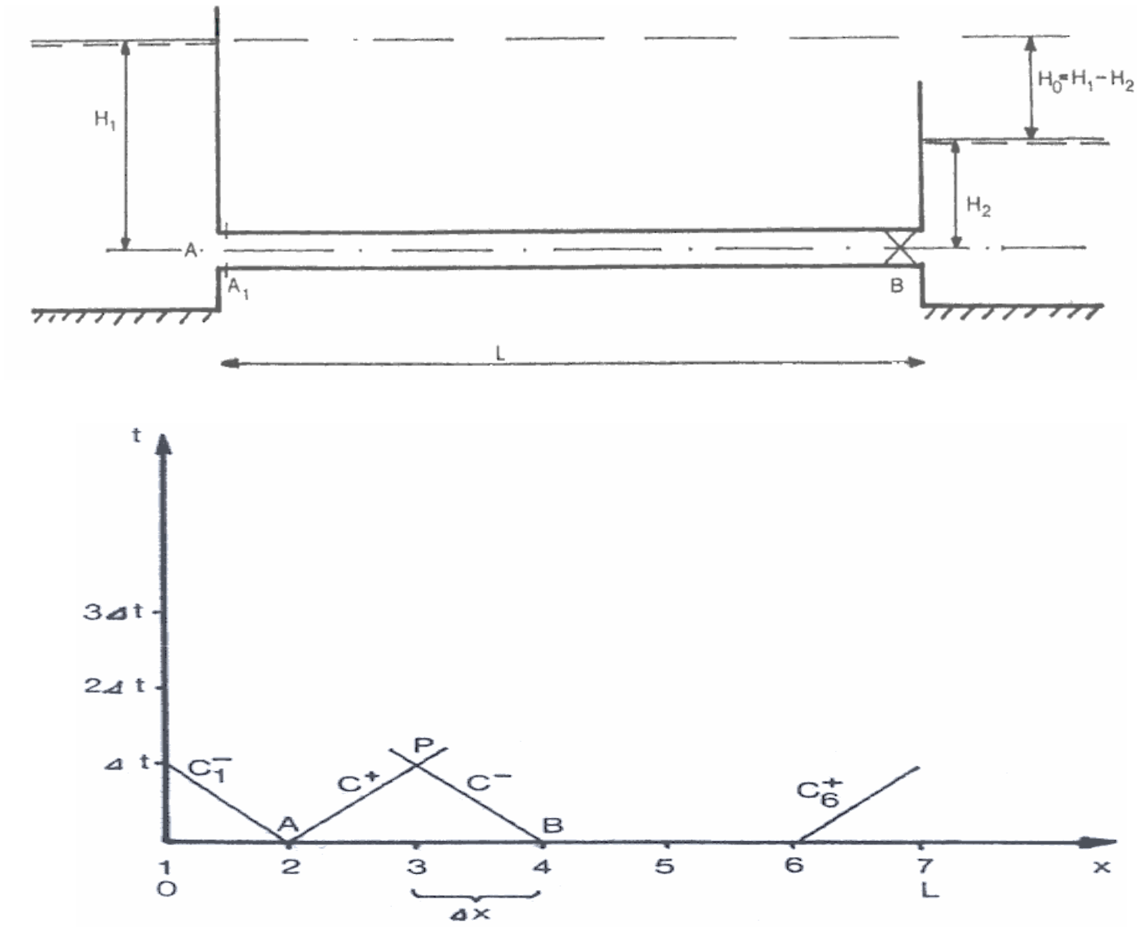

Figure 3.1 Numerical (computer) determination of pressure transient is based on division of the pipeline *0 - L* into equally long parts, defined by nodal points (L. Jönsson and P. Larsen, 1975)

In each end point some kind of boundary condition, for instance a valve at $x = L$ and reservoir at $x = 0$, must be known. The pipeline in Figure 3.1 is divided into six parts, Δx, defined by seven equidistant nodal points, numbered 1,….7. At time $t < 0$ steady-state conditions prevail in the pipeline and the water velocity and the pressure level in the different nodal points can be calculated with common frictional formula ($h_f = f.\ L/D\ .\ v^2/2g$ + possible local losses). At time $t = 0$ the flow rate is changed, for instance by initiating a valve closure operation. On the basis of the method of characteristics water velocity and pressure levels are computed in all nodal points at time $t = \Delta t$ using the corresponding values in the nodal points at $t = 0$ and the boundary conditions at $x = 0,\ L$. After that computational procedure is repeated for the nodal points for the time $t = 2.\Delta t$ using the previously computed values in the nodal points at time $t = \Delta t$ and the boundary conditions.

In this way the transient pressure evolution is computationally stepped forward in time with time step Δt .This time step Δt, should be chosen in such a way that:

$$\Delta x/\Delta t = a \tag{3.1}$$

where Δx = the distance between two consecutive nodal points.

Δt = the pressure wave velocity.

Equation 3.1 actually describes the C^+ and the C^- characteristics respectively depending on the sign of a. Consider a C^+ characteristic, which passes through nodal point 2 (i.e $x = \Delta x$) at time = 0 (point A). At time $t = \Delta t$ this characteristic passes through the point $x + a.\Delta t = x + \Delta x = 2.\Delta x$ = nodal point 3 according to equation 3.1. The points A and P are thus located on the same C^+ characteristic. In the same way one obtains, that the points B and P are located on the same C^- characteristic. The water velocity and the pressure level in point P can be obtained by:

$$C^+ :\ \frac{1}{g}(V_P - V_A) + \frac{1}{a}(H_P - H_A) + \frac{f_A.\Delta t}{2.g.D}.V_A.|V_A| = 0 \tag{3.2}$$

$$C^- :\ \frac{1}{g}(V_P - V_B) - \frac{1}{a}(H_P - H_B) + \frac{f_B.\Delta t}{2.g.D}.V_B.|V_B| = 0 \tag{3.3}$$

where:

V_A, V_B and V_P = water velocity at A, B and P respectively

H_A, H_B and H_P = pressure level at A, B and P respectively

f_A and f_B = The friction coefficient at A and B respectively

Friction is approximated by the starting velocity at A and B respectively.

This could be concisely written as:

$$C^{+}: \quad H_P = HCP - B.V_P \tag{3.4}$$

$$C^{-}: \quad H_P = HCM + B.V_P \tag{3.5}$$

where:

$$HCP = H_A + B.V_A - \frac{f_A.\Delta x}{2.g.D}.V_A.\left|V_A\right| \tag{3.6}$$

$$HCM = H_B - B.V_B + \frac{f_B.\Delta x}{2.g.D}.V_B.\left|V_B\right| \tag{3.7}$$

$$B = \frac{a}{g} \tag{3.8}$$

$$\Delta x = a.\Delta t \tag{3.9}$$

The variables HCP, HCM, B do only contain known data and can thus be calculated numerically. Equation 3.4 and (3.5) give:

$$H_P = \frac{HCP + HCM}{2} \tag{3.10}$$

$$V_P = \frac{H_P - HCM}{B} \tag{3.11}$$

After calculating the pressure level H_P and the water velocity V_P in the nodal point 3 at time, $t = \Delta t$, Figure 3.1 shows that exactly the same kind of calculation can be performed to give pressure levels and water velocities in the inner nodal point 2,….,6 at time $t = \Delta t$. Computation of the hydraulic condition in the nodal points 1 and 7 at time t =Δt utilizes a boundary condition and one characteristic. Assume that nodal point 1 is located in the inlet from the reservoir with boundary condition $H = H_0$ for all times $t > 0$ and that nodal point 7 is a valve that is closed instantaneously at time $t = 0$, i.e. $V = 0$ for $t > 0$. One then obtains for nodal point 1 at $t = \Delta t$:

$$H_1 = H_0 \quad \text{(boundary condition)}$$

$$V_1 = \frac{H_0 - HCM}{B} \quad (C_1^{-} \text{ characteristic})$$

Where *HCM* is computed for nodal point 2 at time $t = 0$.
In the same way one obtains for nodal point 7 at time $t = \Delta t$:

$$V_7 = 0 \qquad \text{(boundary condition)}$$

$$H_7 = HCP \qquad (C_6^+ \text{ characteristic})$$

Where *HCP* is computed for nodal point 6 at time $t = 0$.

Now the hydraulic conditions are known in al nodal points at time $t = \Delta t$ and with exactly the same technique as above the hydraulic conditions can be computed at time $t = 2.\Delta t$ in inner nodal points using the known hydraulic conditions at time $t = \Delta t$ as well as in the nodal boundary points 1 and 7 using boundary conditions and one characteristic (Jönsson L., Larsen, P., 1975).

3.1 Hydraulic transient due valve closure

To calculate the hydraulic transient due to valve closure by the FORTRAN model, the input data can be stated as follows:

1. Length of the pipeline (XL).
2. Diameter of the pipeline (DIA).
3. Inlet level (HIN) and outlet level (HOUT).
4. Frictional coefficient (f).
5. Wave velocity (WV).
6. Valve loss coefficient (XKVALVE)
7. Time end and also number of valve division (The pipe length should be divided to certain number of points)
8. Relation between valve opening (angles) and time closure (T).

3.1.1 Evaluation of the model for valve operation (TRANSIENT. FOR)

The model (see appendix 2) is applied to calculate the pressure in certain pipeline, and to evaluate the model, comparison has been done between the graphical solution for the problem and the model solution. The problem is stated as flows:

Two large reservoirs with water levels on H_1 and H_2 respectively are connected with a circular steel pipeline with the length L and inner diameter, *D*. the wall thickness is denoted by *e* (m). The frictional coefficient of the pipeline is $f = 0.009$ (constant). A butterfly valve is located in the pipeline immediately upstream of the lower reservoir for making it possible to shut off the flow, Figure 3.2 (Jönsson L., 2006)

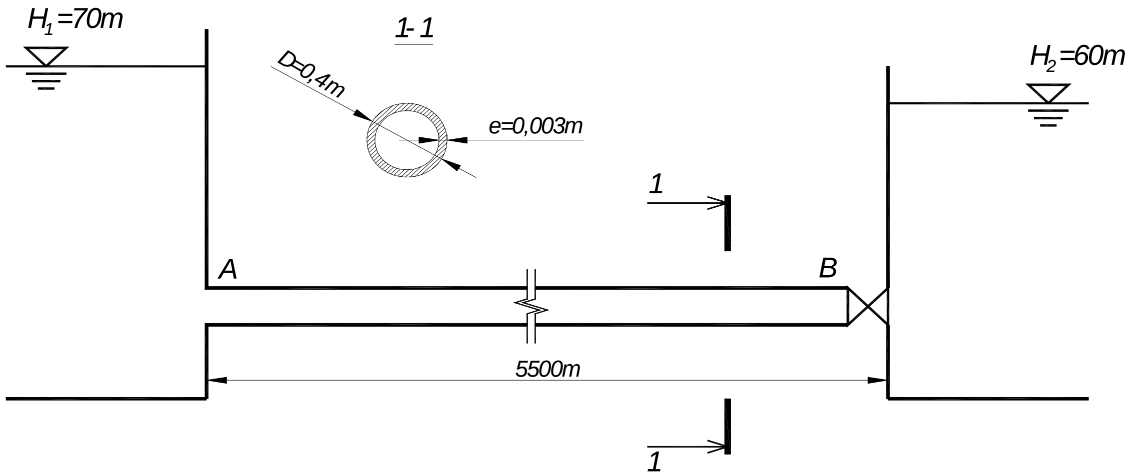

Figure 3.2 Flow situation with the downstream valve (Assignment 2, Unsteady water flow course)

The wave propagation velocity, *a*, could be determined by equation (3.12):

$$a = \sqrt{\frac{\frac{E_{H_2O}}{\rho_{H_2O}}}{1 + \frac{E_{H_2O}}{E_{pipe}} \cdot \frac{D}{e} \cdot C_1}} = \sqrt{\frac{\frac{2.1\times10^9}{10^3}}{1 + \frac{2.1\times10^9}{210\times10^9} \cdot \frac{0.4}{0.003} \cdot 1}} = 950(m/s) \qquad (3.12)$$

Where:

a = wave propagation velocity (m/s)

E_{H_2O} and E_{pipe} = modulus of elasticity of water and the pipe material respectively

ρ_{H_2O} = density of water (kg/m^3)

D and *e* = pipe diameter (m) and pipe wall thickness (m) respectively

C_1– Coefficient = 1

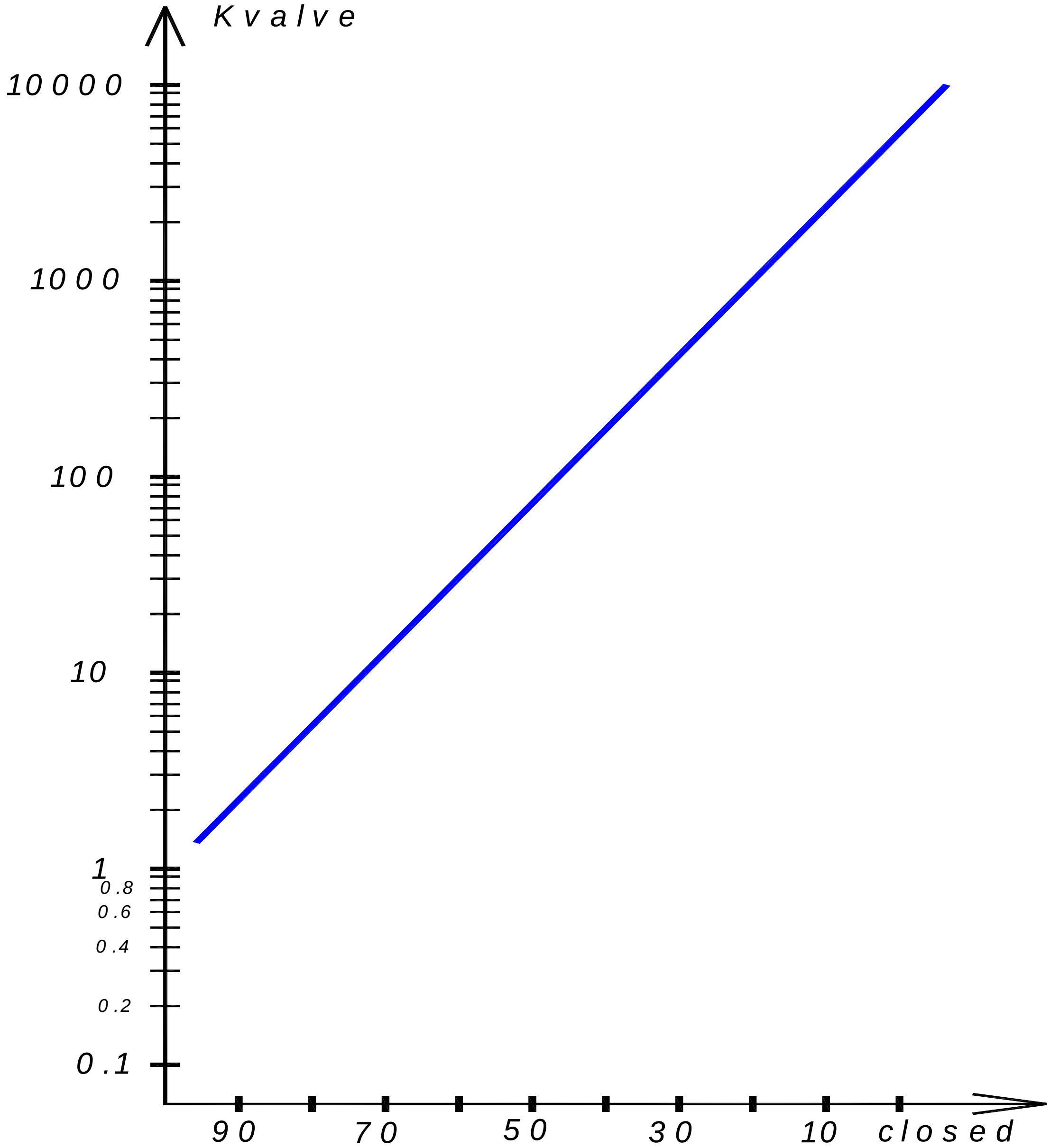

Figure 3.3 K_{valve} as a function of valve opening for a butterfly valve (Assignment 2, Unsteady water flow course)

The model is applied for both linear and stepwise valve closure. The valve operation can be seen in Figure 3.4 and Figure 3.5.

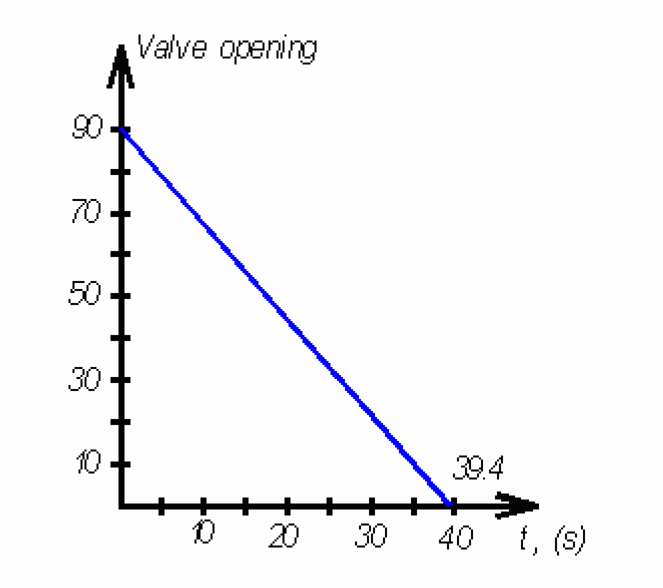

Figure 3.4 Linear valve closure (Assignment 2, Unsteady water flow course)

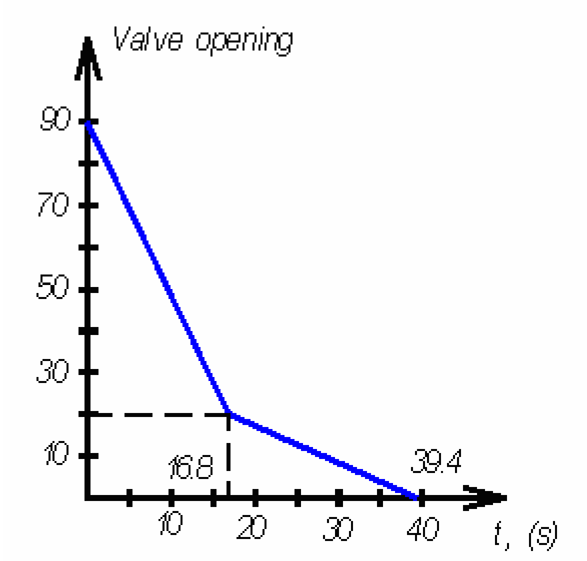

Figure 3.5 Stepwise valve closure (fast initial closure and a slow subsequent closure) (Assignment 2, Unsteady water flow course)

3.1.2 Checking the model for linear and stepwise closure for different time

To study the behaviour of the hydraulic transient for different times of valve closure, the model was applied for the following times, $T = 2\mu, 5\mu, 10\mu, 20\mu, 50\mu, 100\mu, 200\mu$

where $\mu = \dfrac{2L}{a}$. For step wise closure as can be seen in Figure 3.7, the point where t = 16.5 s and Angle = 35 is fixed after which the closure extends continuously and linearly to angle = 0 .

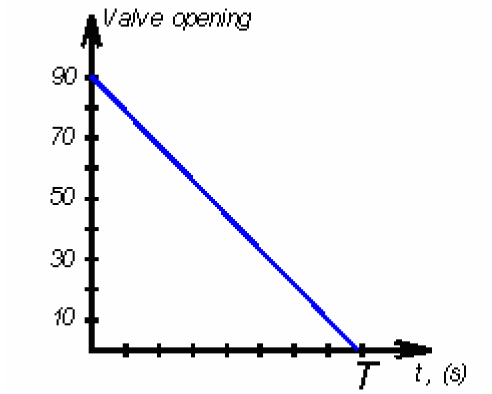

Figure 3.6 Linear valve closure for different time

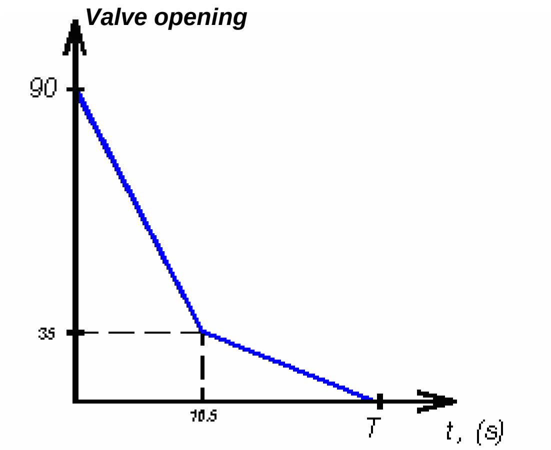

Figure 3.7 Stepwise valve closure for different time

3.1.3 Checking the model for a fixed closure time with different closure operation

The model has been applied for different closure operations for stepwise closure and with fixed time (T = 60 sec).

3.1.4 Effect of different wave velocities and different frictional coefficient

To study the effect of the wave velocity and the friction (pipe material) on the hydraulic transient different velocities and frictional coefficient are used (a = 1450, 950, 500 m/s and f = 0.05, 0.009, 0.005). Generally the valve operation in this case is the same as Figure 3.4. When changing the wave velocities the frictional coefficient was fixed to be 0.009 and when changing the frictional coefficients the wave velocity was fixed to be 950 m/s.

3.1.5 Modify the model to allow for closure operation where the final valve status is not fully closed

In this case the model was modified to calculate the hydraulic transient when the valve is not completely closed at the end (see appendix 3). The model was also applied to calculate the hydraulic transient for different time closures for both linear and step wise closure.

3.1.6 Modify the model in order to obtain maximum and minimum pressure

After making some modification the model could be used to calculate the maximum and the minimum pressure envelopes along the pipeline (see appendix 4). The model was applied to calculate the maximum and minimum pressure for different time closure, different wave velocities, different frictional coefficients and different ways of valve operation and that was for both linear and step wise valve closure. Generally, for the linear closure Figure 3.4 was used for valve operation and for different time closure Figures 3.6 and 3.7 were used.

3.2 Hydraulic transient due to pump stop and check valve closure

In this case the input data for the model are represented by pipeline length (XLAENGD), pipeline diameter (DIA), frictional coefficient (FRIK), inlet level (HIN), outlet level (HUT), wave velocity (VAGH), design head for the pump (HR), design discharge for the pump (QR), design rotation speed for the pump (XNR), and relation between rotation speed (XNR) and the time after initiation of the pump stop (T).

3.2.1 Evaluation of the model for pump stop (SOFTST.FOR)

This model is based on the pipeline configuration according to Figure 3.8, a pump delivering water from one reservoir to another one. The pump is equipped with a check valve and is located at the upstream reservoir. Computer program, SOFTSTOP.FOR, computes the hydraulic transient at pump stop with preset slowdown of the pumping speed (defined in the input data file) which could simulate a soft stop procedure. By letting the slowdown time be sufficiently rapid, it is also possible to simulate a case with no soft stop for the case with negligible inertia. The FORTRAN source code is shown in appendix 5. The pump data for different speeds, flow rates and heads are described in non-dimensional from using the SUTER curve (Figure 2.3) for a typical centrifugal pump.

In this case the model was constructed to calculate the hydraulic transient due to pump stop either by normal closure or due to power failure. In the same way as for the first case with valve closure the model was applied to a certain pipeline in order to make comparison with graphical solution to evaluate the model. The problem is stated as follows:

Water is pumped from a low-lying reservoir to an upper reservoir via L = 2000 m. Long pipeline see Figure 3.8. The pipeline is made of steel with an inner diameter of D = 0.25m and wall thickness of e = 0.008m. The friction coefficient, f = 0.01. The water level difference between the reservoirs is ΔZ = 70m (Jönsson L.; 2006).

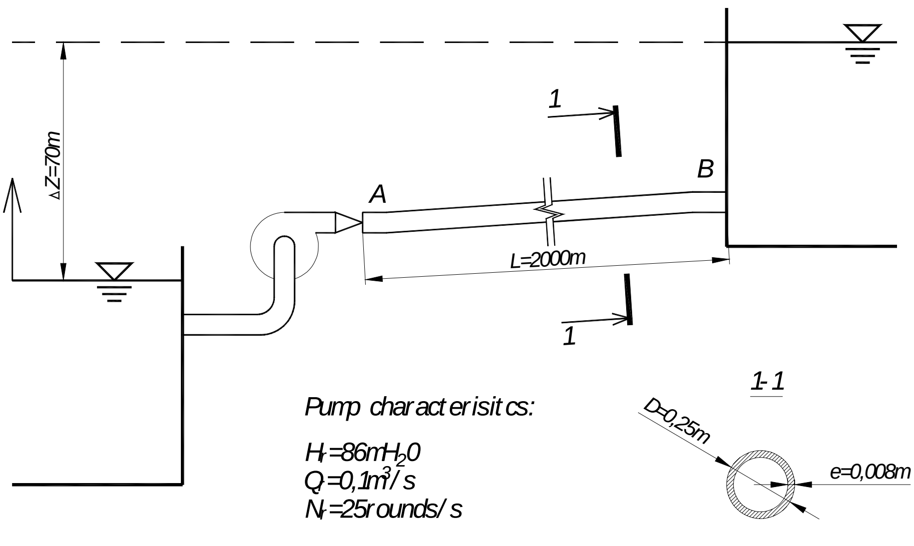

Figure 3.8 Pump and pipeline connecting two reservoirs (Assignment 3, Unsteady water flow course)

Hr, Qr and Nr represent the pressure, discharge and rotation speed during the steady state respectively. The wave propagation velocity, a, could be determined using equation 3.12:

$$a = \sqrt{\frac{\dfrac{E_{H_2O}}{\rho_{H_2O}}}{1+\dfrac{E_{H_2O}}{E_{pipe}}\cdot\dfrac{D}{e}\cdot C_1}} = \sqrt{\frac{\dfrac{2.1\times10^9}{10^3}}{1+\dfrac{2.1\times10^9}{210\times10^9}\cdot\dfrac{0.25}{0.008}\cdot 1}} = 1265(m/s)$$

The graphical method dealt with a pump stop for a pump with high inertia which caused the pump to slow down in a certain way which was obtained by means of the graphical method. In order to make a comparison with the result from computer method SOFTSTOP.FOR, the same slowdown procedure was defined in the input data file for this program. This data was thus taken from the graphical solution.

3.2.2 Applying the model for different times and rotation speeds

To study the effect of different ways of controlled pump stop, different times versus different rotational speeds were applied. The reason for this is that, nowadays pump can be equipped with a special kind of electronic control circuit which makes it possible to slowdown the pump speed at the pump stop in controlled way. This will of course have a significant impact on the hydraulic transient.

3.2.3 Applying the model to simulate real measurement

In this step the model was used to simulate real measurement for hydraulic transient from different pump stations.

3.2.3.1 Yndesäte pumping station

Yndesäte sewage pumping station is located in the Helsingborg municipality and pumps sewage water to the next pumping station, Påarp Östra. The pressurized pipeline profile is described by the following horizontal and vertical coordinates:

Table 3.1 Pipeline profile at Yndesäte pumping station

Length from pumping station (m)	Vertical coordinate (m)
0	45.80
184	48.00
323	48.60
494	48.95
681	54.40
763	55.30

The profile is thus rising continuously without any local peaks. Relevant data for the pumping station and the pipeline:

Pumps: CP3200-610, Flygt (two in parallel).
Valve: Swing check valve.
Pump sump water level: 41.8m
Geometric head: 13.5m
Pipeline length: 763m
Pipe material: Cast iron, diameter 250 mm, wall thickness 8 - 10 mm

The pressure transducer was attached to the pressure side of the pipeline and immediately after the check valve. The vertical location of the transducer was about 3.8 m above the pump sump water level. One pump running and stop of the pump, measured transient pressure can be seen in Figure 4.33. Sample rate 50 Hz, measurement time 50 s. Steady state pressure in the measurement point about 26 m (absolute pressure) implying a total pump head of 19.6 m (geometrical head + friction)implying flow of 50 l/s. The measured maximum pressure amounts to 34.4 m absolute pressure and the measured minimum pressure to 6.2 m absolute pressure. The pressure drops immediately to a pressure below the atmospheric pressure (10.2) due to the location of the measurement point and head loss in the passively rotating pump after pump stop (no inertia). When the flow at the check valve tends to reverse the check valve closes almost instantaneously and the pressure rises. After that one obtains the typical oscillating pressure due to pressure wave propagation back and forth through the entire pipeline between the close check valve and the downstream end of the pipeline. The average value of the oscillatory pressure time period was found to be about T = 3.27 s which gives a pressure wave velocity a = 932 m/s according to the equation:

$$T = \frac{4L}{a} \tag{3.13}$$

Where L = length of the pipeline (m), a = pressure wave velocity and T = time period of pressure oscillations.

The measured pressure wave propagation value should be compared with theoretical one according to the expression:

$$a_{theor} = \sqrt{\frac{\frac{E_{H_2O}}{\rho_{H_2O}}}{1+\frac{E_{H_2O}}{E_{pipe}}.\frac{D}{e}.constant}} \tag{3.14}$$

where:

E_{H_2O} and E_{pipe} = modulus of elasticity (N/m^2) for water and pipe material respectively ρ_{H_2O} = density of water (kg/m^3), D and e = inner pipe diameter (m) and pipe wall thickness (m) respectively.

Constant relates to the axial behaviour of the pipeline, set to $1-\mu^2$ assuming no axial movement (μ = Poisson's constant). Using E_{pipe} = 100×10^9 N/m^2 for cast iron, $E_{H_2O} = 2.1\times10^9$ N/m^2 for water, D = 0.25m, e = 0.009m, ρ_{H_2O} = 1000 kg /m^3 one obtains:

a = 1150 m/s

which is considerably higher than the measured one. One possible reason for the deviation might be given by the fact that sewage water is pumped. In such case there is a possibility for the existence of tiny gas bubbles in the pipeline, reducing the wave speed (Jönsson, 2003).

3.2.3.2 Kristinedal pumping station

The Kristinedal pumping station is located in the Sjöbo municipality and pumps municipal sewage water in a transfer scheme. The profile of the pipeline is described as follows. It rises 15 m during the first km of the pipeline length. After that the pipeline is undulating with three low points, with a rise of 4.5 m, 2.5 m, 7 m respectively after each low point. Relevant data for the pumping station and the pipeline:

Pumps: Flygt CP3127-50, 7.4 KW, 2900 rpm. The station is equipped with a facility for soft start and soft stop.
Valve: Ball check valve.
Pump stump water level: Approximately 2.3 m below the pressure transducer.
Geometrical head: 19 m.
Location of the pressure transducer: 2.3 m above pump sump water level.
Pipeline length: 3240 m
Pipeline material: PVC, diameter 225 mm, NT 16, wall thickness, e = 14 mm (according to average standard values for this kind of pipe material).

At steady state with one pump running the measured pressure was 31.4 m absolute pressure, corresponding to a pump head of 31.4 + 2.3 - 10.2 = 23.5 m and a flow rate Q = 18 l/s (v = 0.45 m/s).In this case two measurements were used for the study and measured transient pressures can be seen in Figure 4.35 and Figure 4.36:

1. KRID01: One pump running and a soft stop (24s) equipped to this pump. Sample rate 25 Hz, measurement time 180 s. The measured maximum pressure was 33.7 m absolute pressure, which was approximately the same as steady state operating pressure. The measured minimum pressure amounted to 15.3 m absolute pressure. The effect of the soft stop is seen as a gradually decreasing pressure down to the minimum pressure 15.3 m absolute pressure at which point in time the valve closes. After that the oscillating pressure occurs, also comprising a disturbance. The wave propagation velocity determined, on the basis of average time period, T = 40.5 s of the oscillating pressure was found to be a = 320 m/s, equation 3.13. The theoretical wave propagation speed, equation 3.14, was found to be a_{theor} = 455 m/s, which is significantly higher than the measured value.

2. KRID02: One pump running and stop of this pump without any soft stop, measured transient pressure can be seen in Figure 4.36. Sample rate 25 Hz, measurement time 200 s. The measured maximum pressure was found to be 36.2 m absolute pressure and the measured minimum pressure 14.5 m absolute pressure. At pump stop the pressure drops instantaneously 17 m absolute pressure at which point in time the valve closes. After that the pressure drops more slowly to the minimum pressure 14.5 absolute pressure and subsequently the oscillation pressure with an average time period $T = 41.12$ s occurs. The pressure wave velocity, based on the oscillation pressure time period, was found to be $a = 315$ m/s which is in very good agreement with the previous estimates of the wave velocity (Jönsson, 2003).

CHAPTER FOUR

RESULTS

In this part all the results of the model simulation are shown as a relation between the hydraulic pressure (m H_2O) and the time (s). One has to know that the term of pressure level in all the graphs represents the real pressure ($\frac{P}{\gamma}$)+ geometrical head (Z) and the atmospheric pressure considered to be as reference level (zero level). In the case with valve closure at the downstream end of the pipeline the graphs represent the transient pressure immediately upstream of the valve (except for envelope cases). In the case with pump stop the graphs represent the transient pressure immediately downstream of the check valve. In the two real cases (Yndesäte and Kristinedal) the graphs represent the pressures immediately downstream of the check valve.

4.1 Transient flow due to valve closure operations

4.1.1 Evaluating the model by comparing it with the graphical solution

In evaluating the model, its solution for both linear and stepwise valve closure operations was compared with that of the graphical solution and the results are shown in Figures 4.1 and 4.2. The valve closure operation can be seen in Figures 3.4 and 3.5

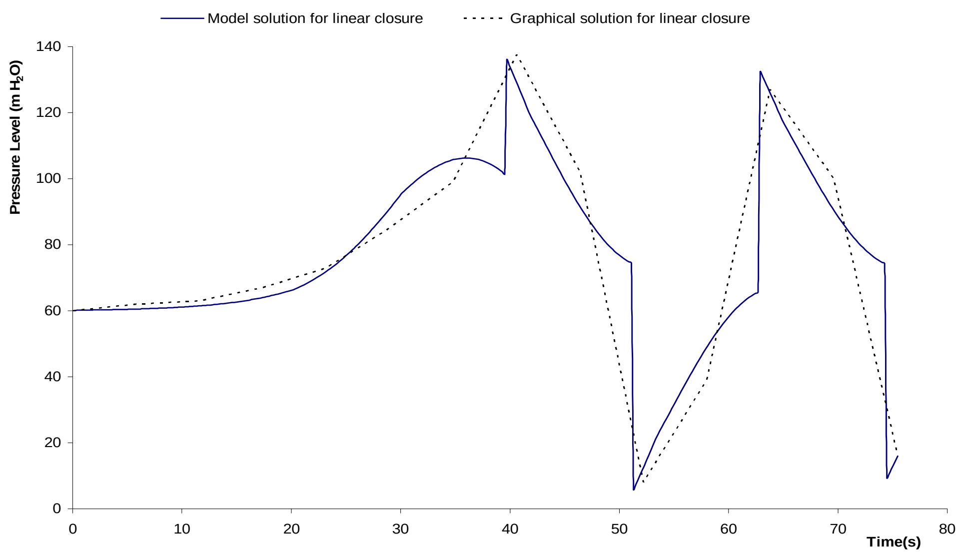

Figure 4.1 Linear valve closures (model and graphical solutions)

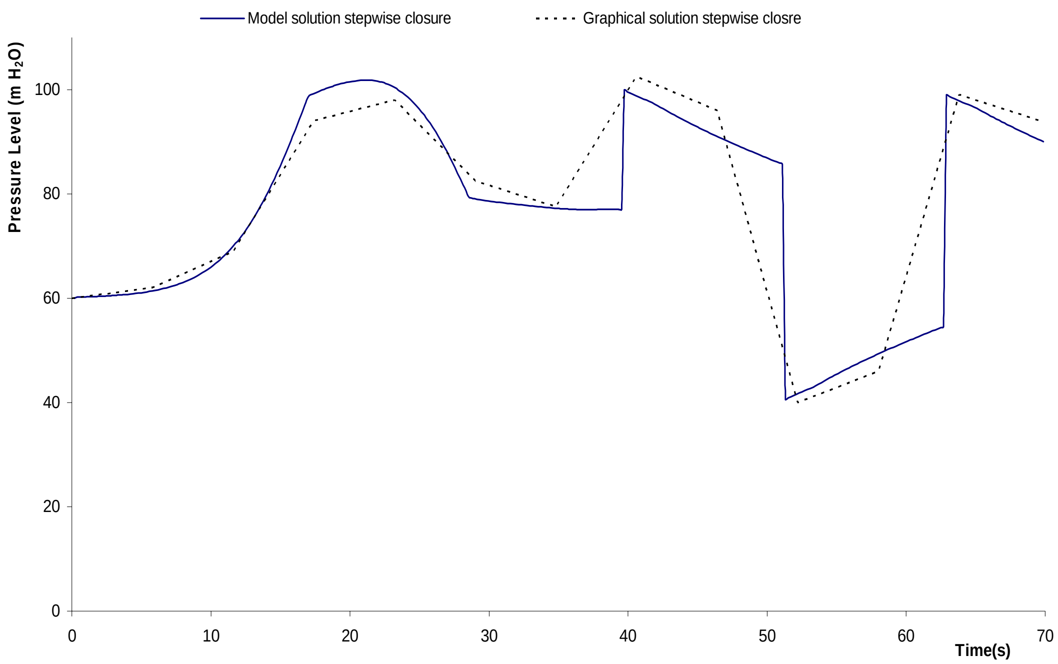

Figure 4.2 Step wise valve closures (model and graphical solution)

Figure 4.1 shows that the maximum pressure for the model solution for the linear valve closure is 136 m H_2O and the minimum pressure is 7 m H_2O, while the maximum pressure for the graphical solution is 137.6 m H_2O and the minimum pressure is 8 m H_2O. Figure 4.2 shows the maximum pressure for the model solution for stepwise valve closure is 99.3 m H_2O and the minimum pressure is 41 m H_2O, whiles the maximum pressure for the graphical solution is 102.5m H_2O and the minimum is 40 m H_2O.

4.1.2 Linear valve closure operations for different time

The results of closing the valve linearly in different times; $T = \mu$, 2 μ, 5 μ, 10 μ, 20 μ, 50 μ, 100 μ, 200 μ (where $\mu = 2L/a$, L = pipe length, a = wave velocity) can be seen in Figures 4.3, 4.4, 4.5 and 4.6. The maximum pressure varies from H = 186.5 m H_2O when $T = 2L/a$ to H = 87 m H_2O when $T = 400L/a$ whiles the minimum pressure varies between -38 m H_2O and 53 m H_2O. The details of the linear valve closure operations are tabulated in Table 4.1.

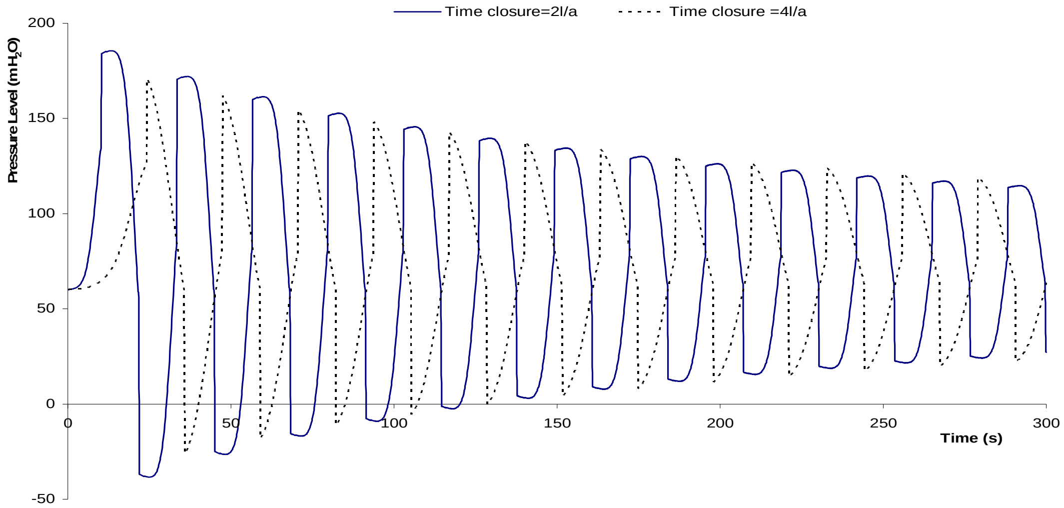

Figure 4.3 Linear valve closure operations for *T* = 2*L*/*a* and *T* = 4*L*/*a*

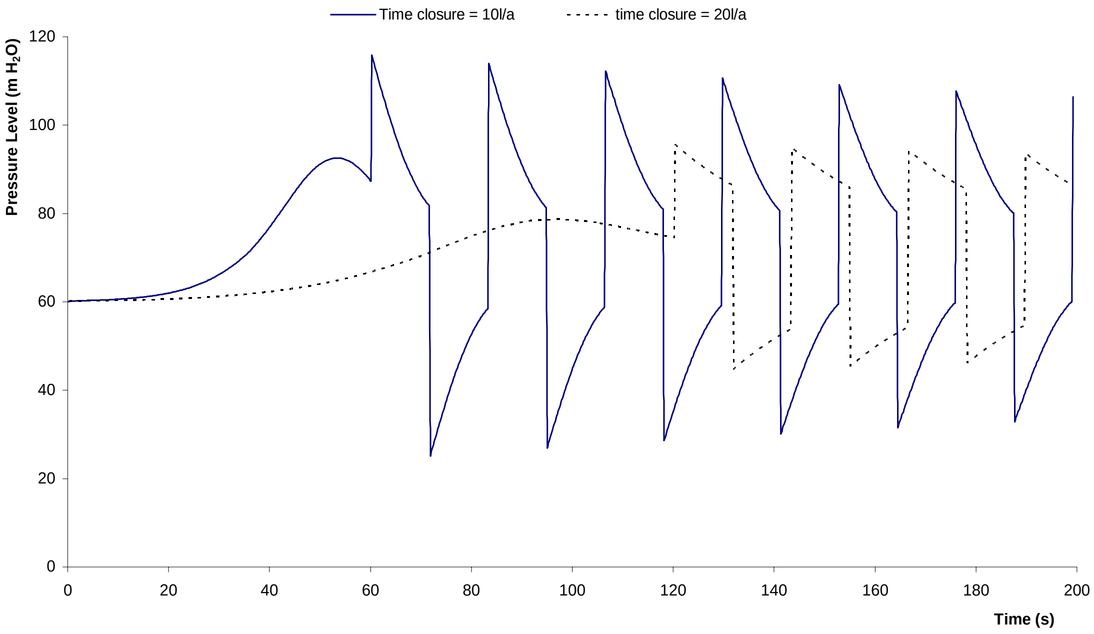

Figure 4.4 Linear valve closure operations for *T* = 10*L*/*a* and *T* = 20*L*/*a*

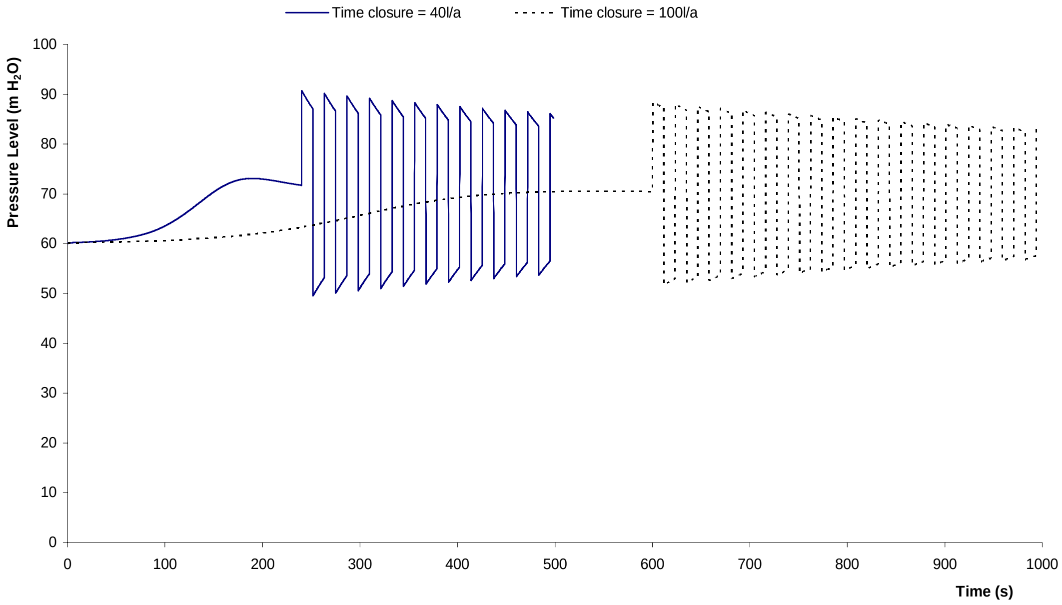

Figure 4.5 Linear valve closure operations for *T* = 40*L*/*a* and *T* = 100*L*/*a*

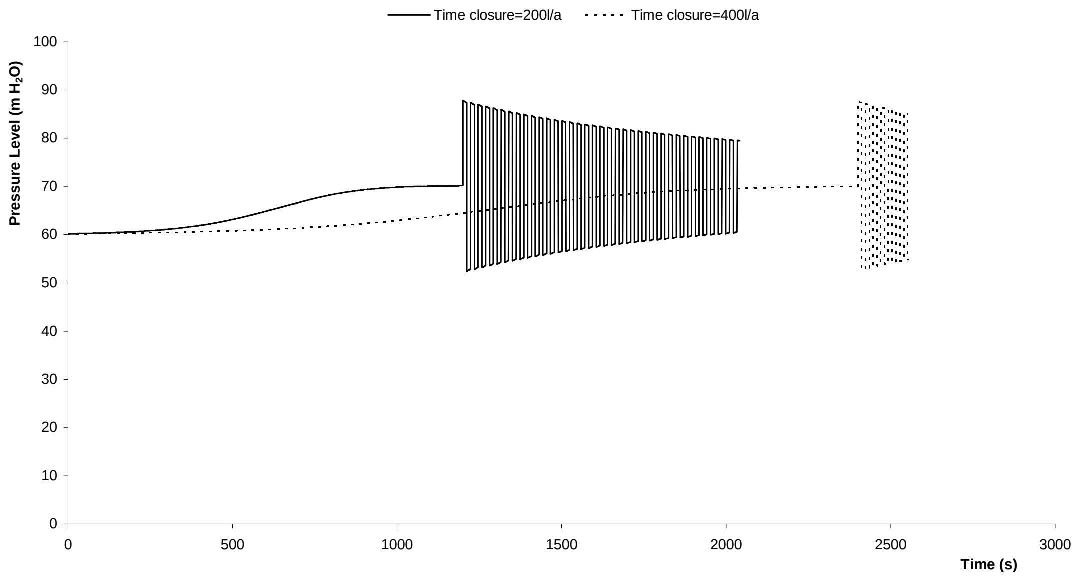

Figure 4.6 Linear valve closure operations for *T* = 200*L*/*a* and *T* = 400*L*/*a*

Table 4.1 Linear valve closure as a function of time (s) and valve opening (°)

	Time (s)							
Valve opening	μ	2 μ	5 μ	10 μ	20 μ	50 μ	100 μ	200 μ
90°	0	0	0	0	0	0	0	0
60°	3.4	7.8	21	40	84	200	420	860
40°	5.6	13.4	36	68	136	330	700	1400
0°	10	24	60	120	240	600	1200	2400

4.1.3 Stepwise valve closure operations for different time

The model was applied to the stepwise valve closure operation for different times (T = 5μ, 10 μ, 20 μ, 50 μ, 100 μ, 200 μ) and the results are shown in Figures 4.7, 4.8 and 4.9. The maximum pressure varied from H = 96 m H_2O when T = 10L/a to H = 87.3 m H_2O when T = 400L/a whiles the minimum pressure varied between 44.8 m H_2O and 52.8 m H_2O. The general diagram that shows the valve operation can be seen in Figure 3.7 and the details of the valve closure operations are tabulated in Tables 4.2, 4.3, 4.4, 4.5, 4.6, and 4.7.

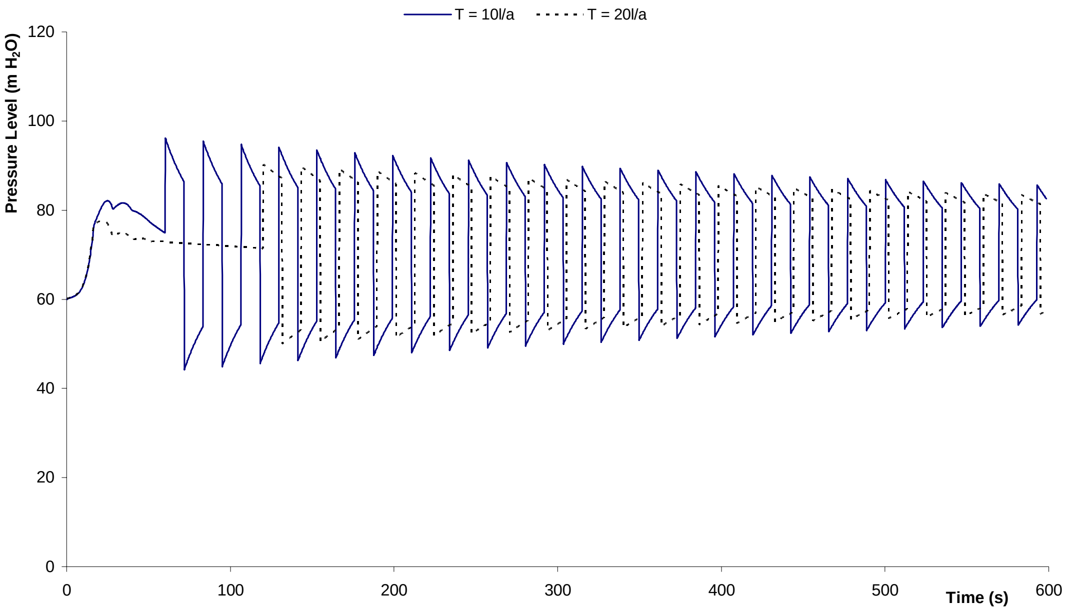

Figure 4.7 Stepwise valve closure operations for *T* = 10*L*/*a* and *T* = 20*L*/*a*

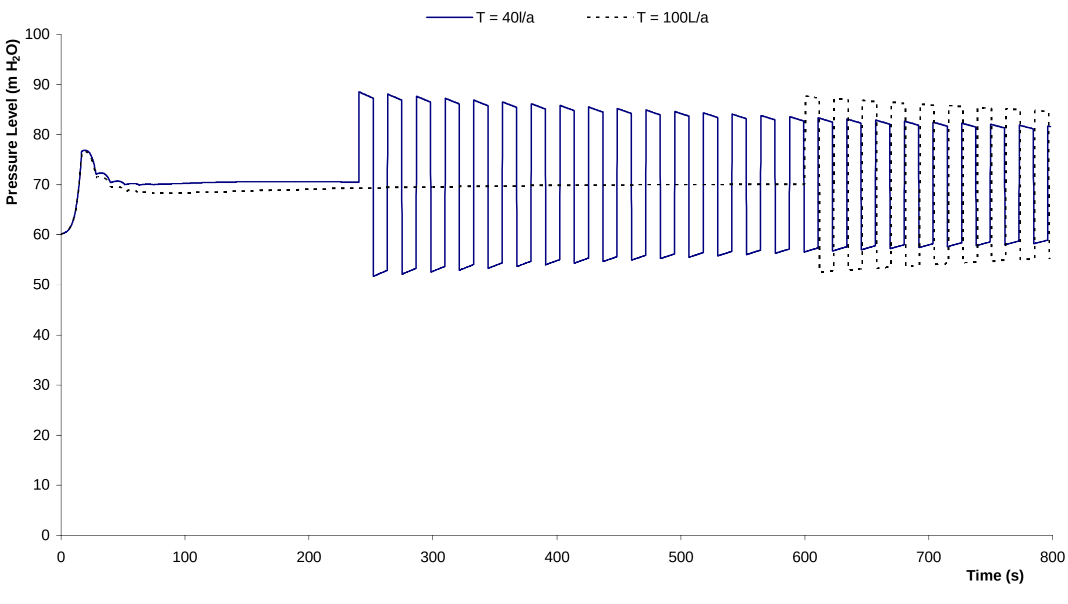

Figure 4.8 Stepwise valve closure operations for *T* = 40*L*/*a* and *T* = 100*L*/*a*

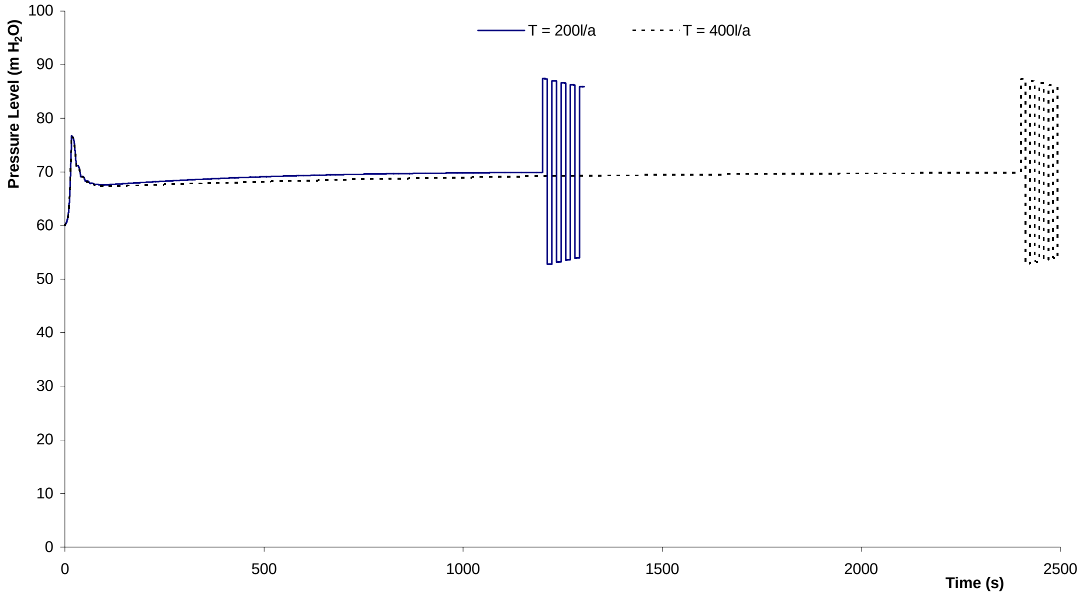

Figure 4.9 Stepwise valve closure operations for *T* = 200*L*/*a* and *T* = 400*L*/*a*

Table 4.2 Stepwise valve closure operation when *T* = 5*μ*

Valve Opening (angle in degrees)	Time (s)
90	0
35	16.5
32	20
0	60

Table 4.3 Stepwise valve closure operation when *T* =10*μ*

Valve Opening (angle in degrees)	Time (s)
90	0
35	16.5
34	20
0	120

Table 4.4 Stepwise valve closure operation when *T* = 20*μ*

Valve Opening (angle in degrees)	Time (s)
90	0
35	16.5
32	40
0	240

Table 4.5 Stepwise valve closure operation when *T* = 50*μ*

Valve Opening (angle in degrees)	Time (s)
90	0
35	16.5
32.4	60
0	600

Table 4.6 Stepwise valve closure operation when *T* = 100*μ*

Valve Opening (angle in degrees)	Time (s)
90	0
35	16.5
33	100
0	1200

Table 4.7 Stepwise valve closure operation when *T* = 200*μ*

Valve Opening (angle in degrees)	Time (s)
90	0
35	16.5
21	1000
0	2400

4.1.4 Fixed valve closure time and different valve closure operations

The model was applied to a scenario where the valve closure time was fixed (T = 5μ) but the valve closure operations varied, and the results are as shown in Figures 4.10 and 4.11. The valve operation profiles are described in Tables 4.8, 4.9, 4.10, and 4.11.

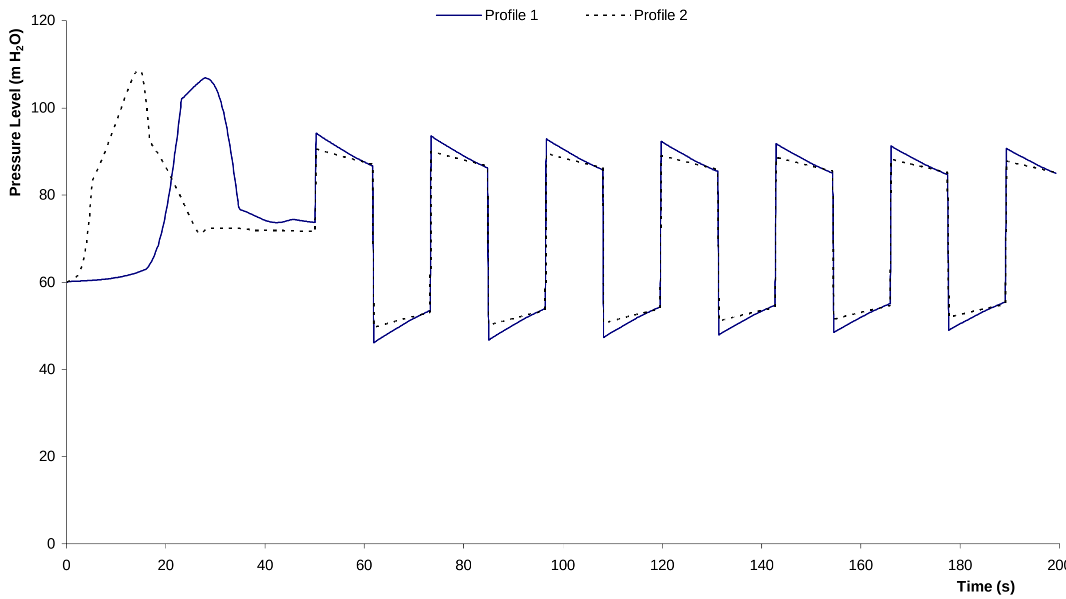

Figure 4.10 Effect of different profile of valve closure operations

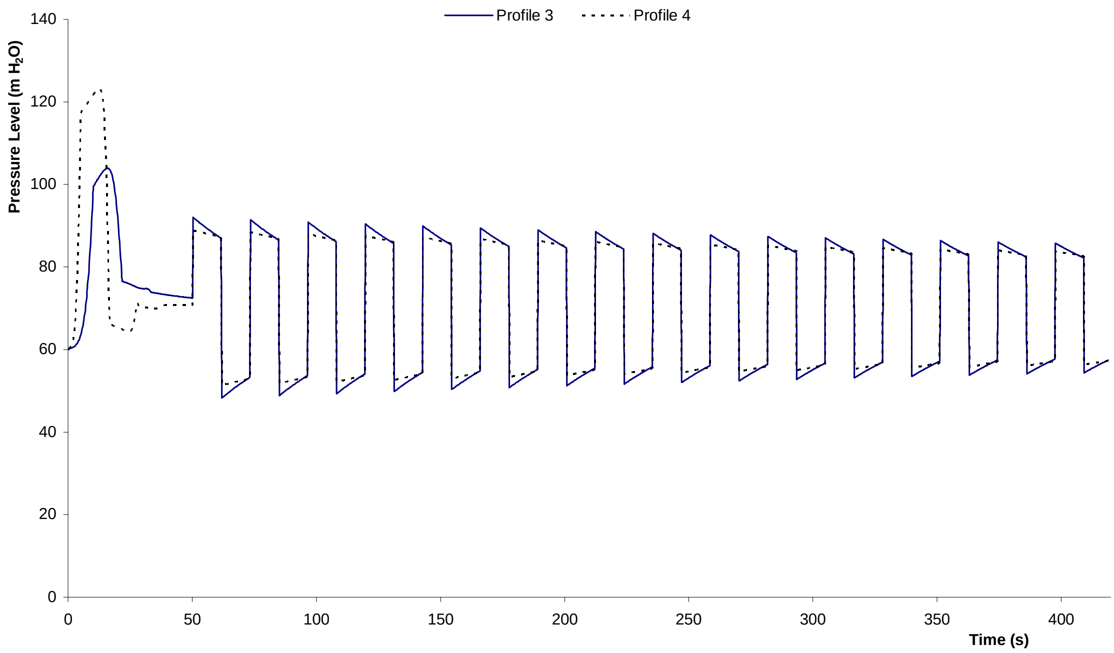

Figure 4.11 Effect of different profile of valve closure

Table 4.8 Valve closure operation (profile 1)

Valve Opening (angle in degrees)	Time (s)
90	0
57	16.5
18	23
0	50

Table 4.9 Valve closure operation (profile 2)

Valve Opening (angle in degrees)	Time (s)
90	0
30	5
13	15
0	50

Table 4.10 Valve closure operation (profile 3)

Valve Opening (angle in degrees)	Time (s)
90	0
20	10
10	30
0	50

Table 4.11 Valve closure operation (profile 4)

Valve Opening (angle in degrees)	Time (s)
90	0
10	5
5	25
0	50

4.1.5 Effect of different pipeline frictional coefficient

The model was applied to many scenarios where different pipeline frictional coefficients ($f = 0.009, 0.025, 0.05, 0.10$) were used but the wave velocity fixed to 950 m/s, and the effects are as shown in Figures 4.12 and 4.13. The maximum pressure varied between 95 m H_2O when f = 0.1 and 136 m H_2O when f = 0.009, whiles the minimum varied between 48 m H_2O and 7 m H_2O. For closure operation see Figure 3.4.

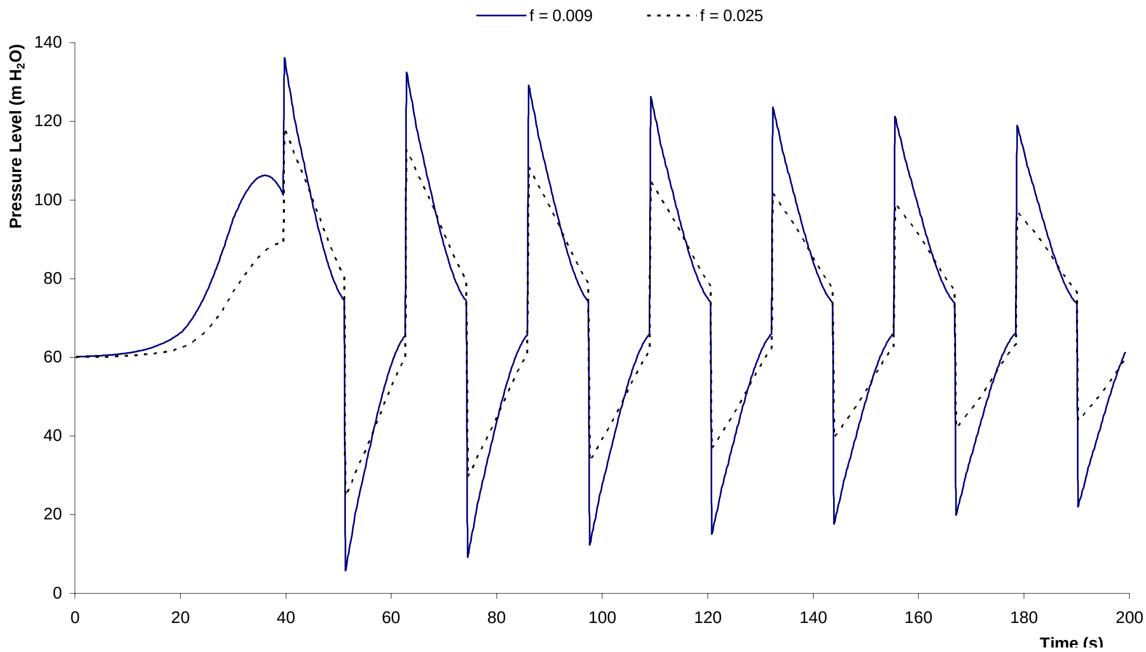

Figure 4.12 The effect of different frictional coefficient ($f = 0.009$ and $f = 0.025$)

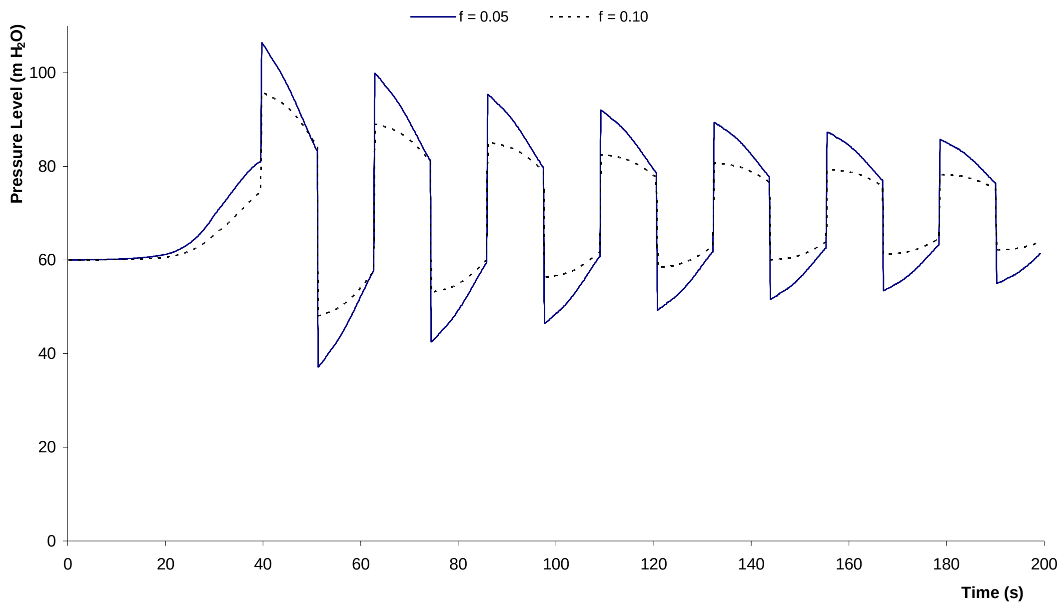

Figure 4.13 The effect of different frictional coefficient ($f = 0.05$ and $f = 0.10$)

4.1.6 Effects of different wave velocities

With the frictional coefficient fixed to 0.009, the model was applied to many scenarios where different wave velocities (*a* = 300, 500, 950 and 1450 m/s) were used and the effects are as shown in Figures 4.14 and 4.15. The maximum pressure varied between 143.5 m H_2O when *a* =1450 m/s and 102.5 m H_2O when *a* = 300 m/s, while the minimum pressure varied between -1 m H_2O and 42 m H_2O

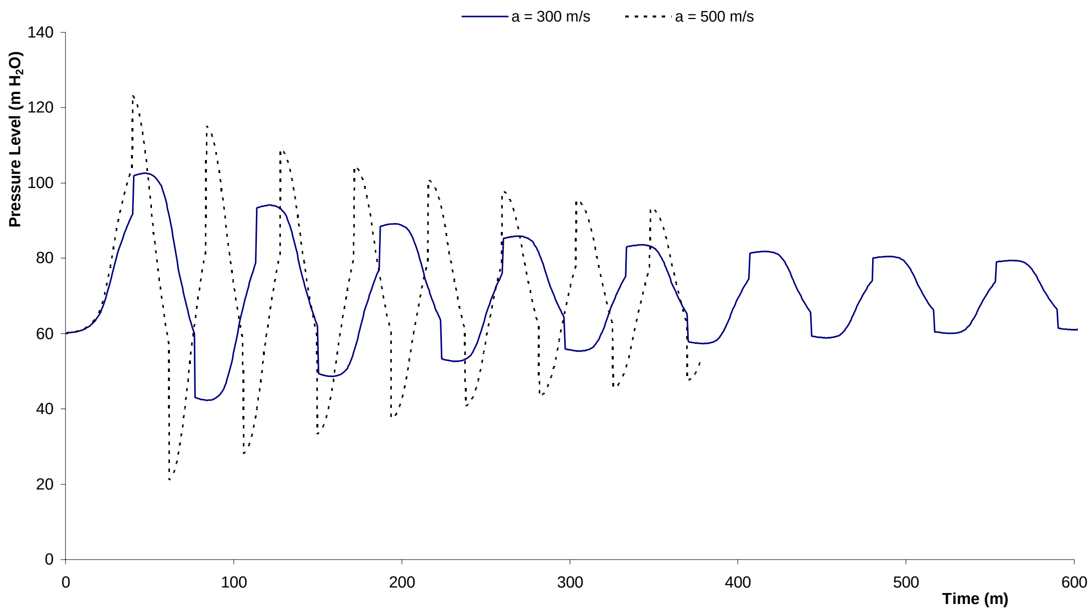

Figure 4.14 Effects of different wave velocities (*a* = 300 and 500m/s)

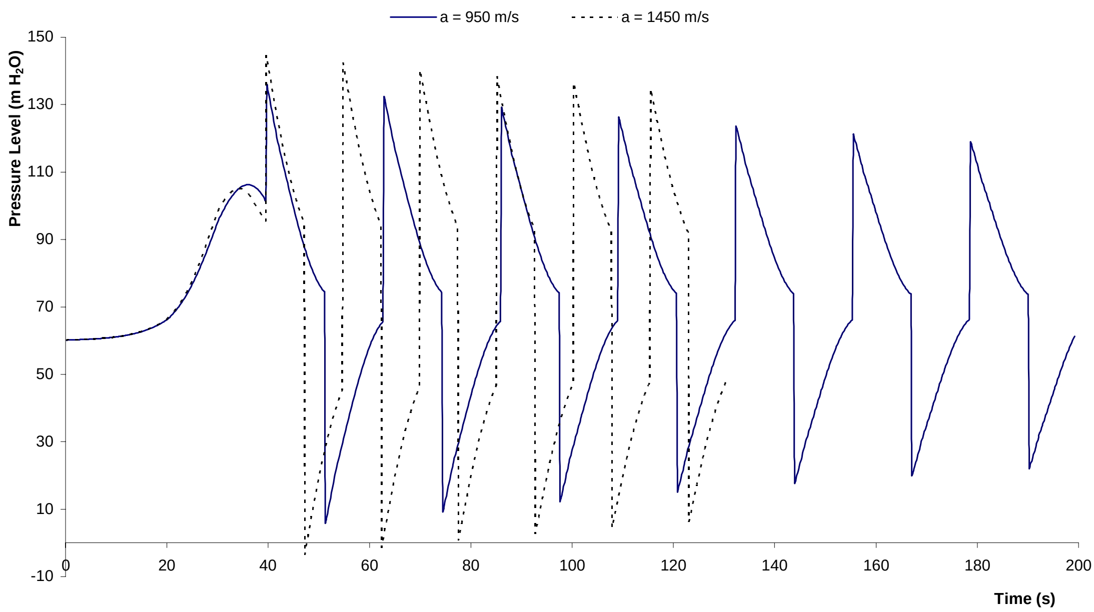

Figure 4.15 Effects of different wave velocities (*a* = 950 and 1450m/s)

4.1.7 Maximum and minimum pressure envelopes at every grid point in the pipeline during the linear valve closure

The model was modified to determine the maximum and minimum pressure envelopes at every grid point in the pipeline for the linear valve closure operation and the result is shown in Figure 4.16

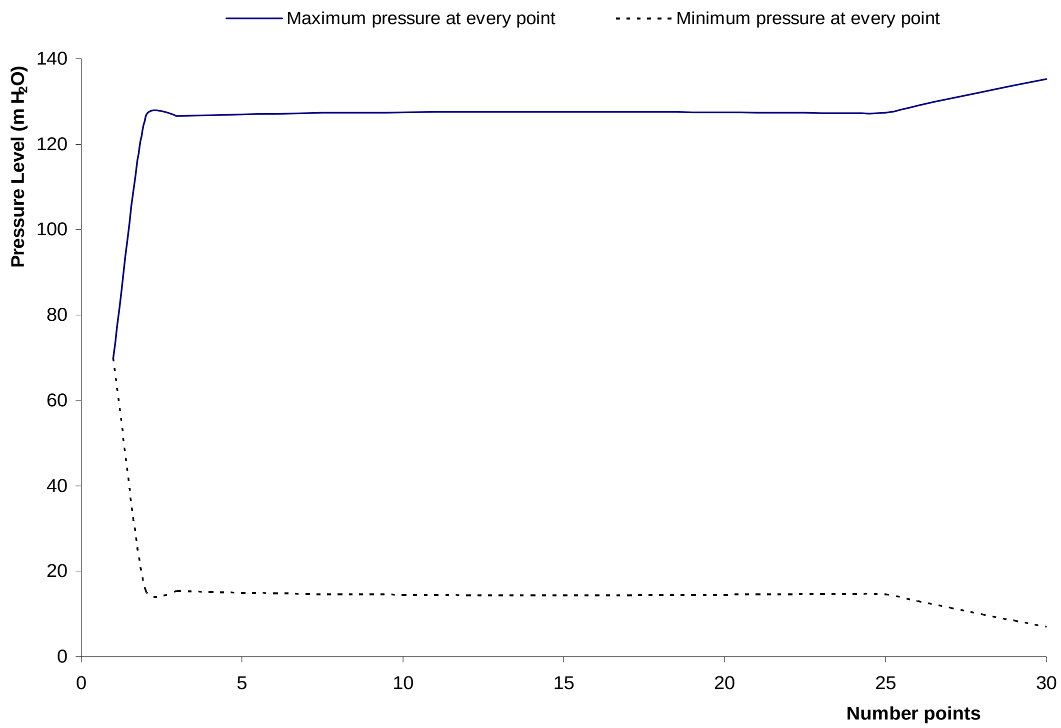

Figure 4.16 Maximum and minimum pressure at every point for linear valve closure

4.1.8 Maximum and minimum pressures at all points in linear valve closure for different times.

The model was used to determine the maximum and minimum pressures for times, $T = 10L/a$ and $T = 100L/a$ in the case of the linear valve closure, and the results are as shown in Figures 4.17 and 4.18

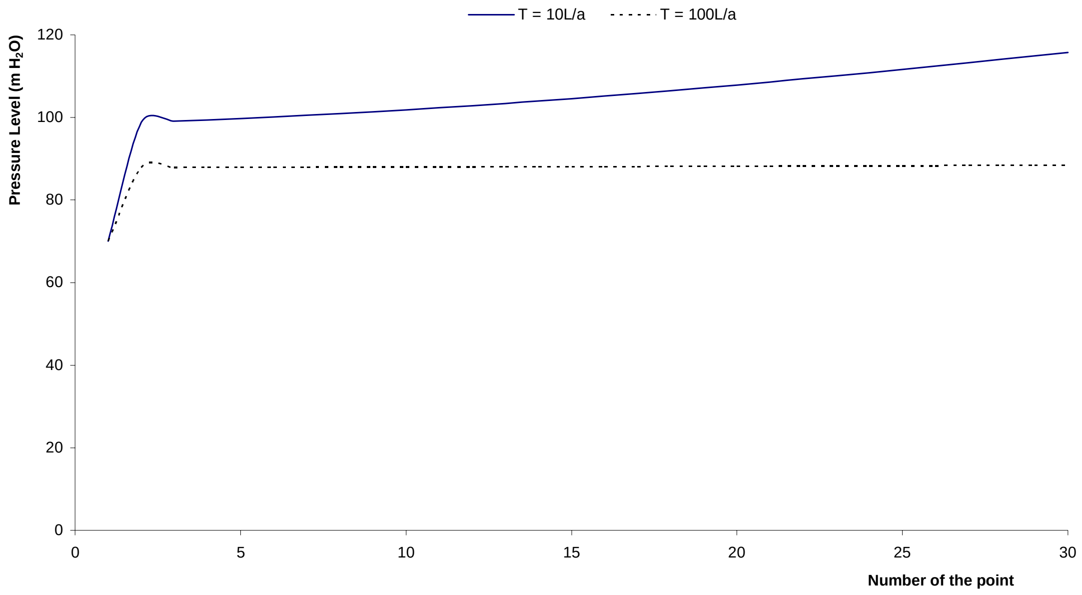

Figure 4.17 Maximum pressures for *T* = 10*L*/*a* and *T* = 100*L*/*a* during the linear valve closure

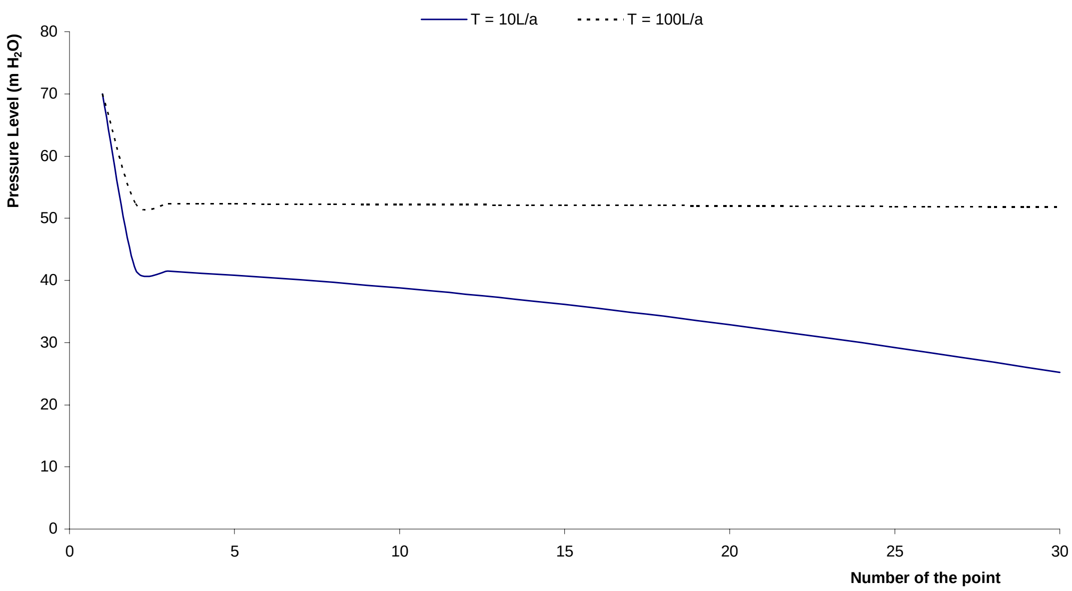

Figure 4.18 Minimum pressures for *T* = 10*L*/*a* and *T* = 100*L*/*a* during the linear valve closure

4.1.9 Maximum and minimum pressures at all points for linear valve closure for different wave velocities.

For wave velocities, *a* = 300m/s and 950m/s, the maximum and minimum pressures at all the points were determined using the model and the results are as shown in Figures 4.19 and 4.20 respectively.

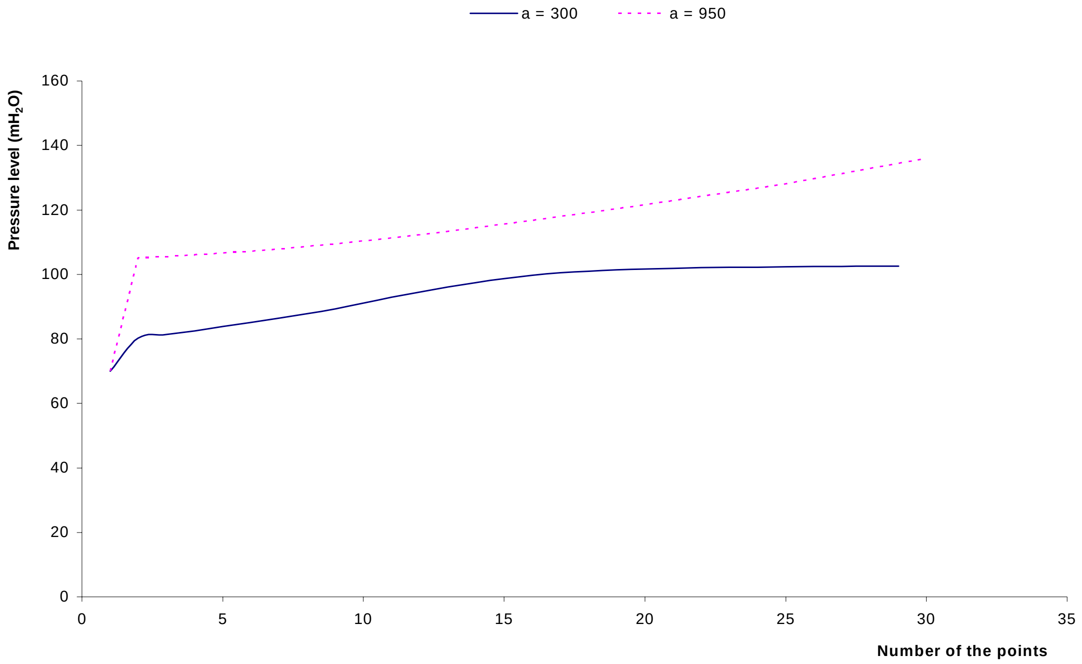

Figure 4.19 Maximum pressures at all points for linear valve closure for *a* = 300m/s and 950m/s

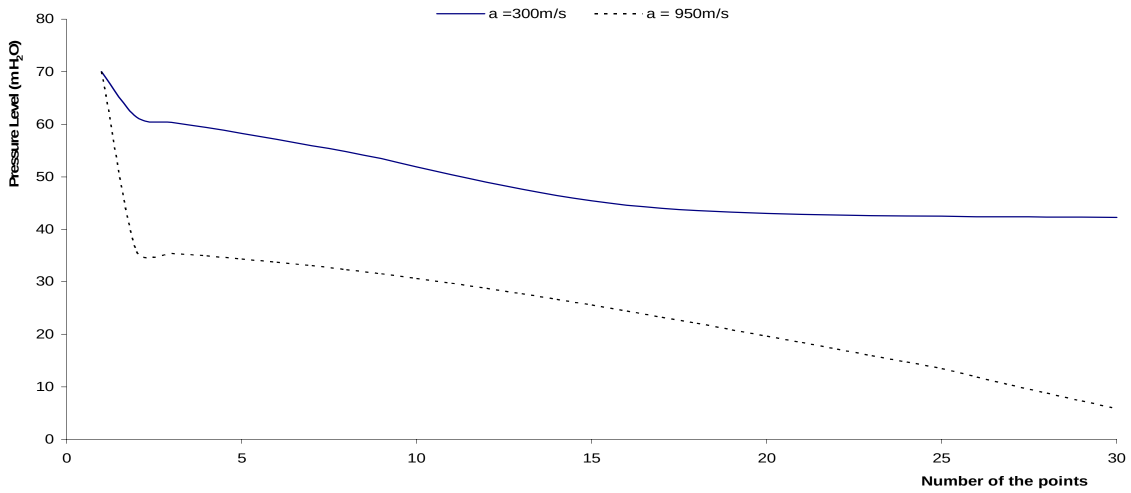

Figure 4.20 Minimum pressures at all points for linear valve closure for *a* = 300m/s and 950m/s

4.1.10 Maximum and minimum pressures at all points for linear valve closure for different frictional coefficient

Using different frictional coefficients (f = 0.1, 0.05), the model was used to determine the maximum and minimum pressures at all the points for linear valve closure and the results are shown in Figure 4.21 and 4.22 respectively.

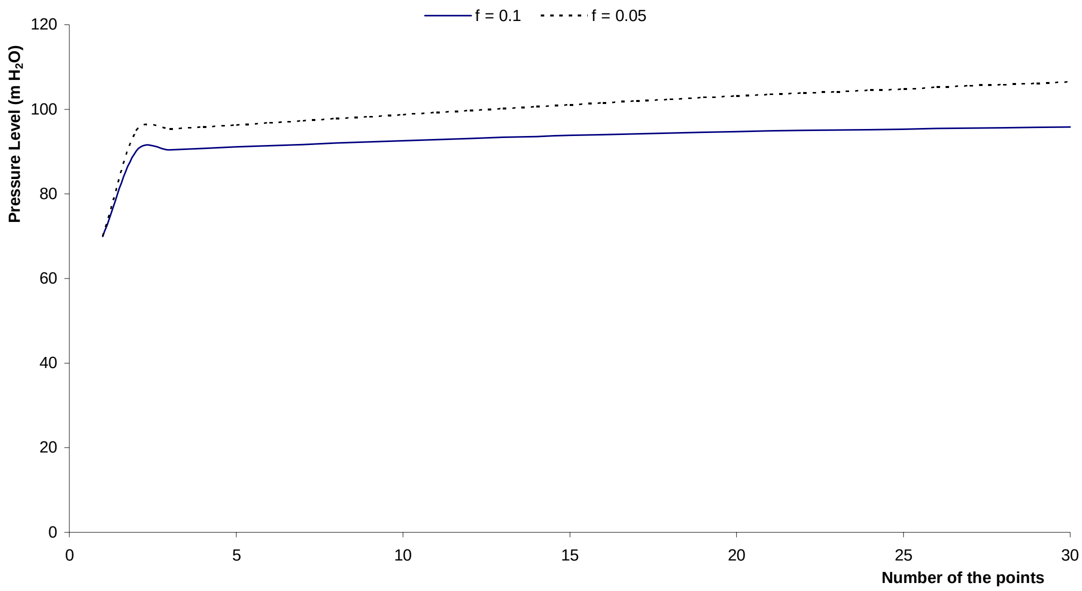

Figure 4.21 Maximum pressures at all points for linear valve closure for *f* = 0.1 and 0.05

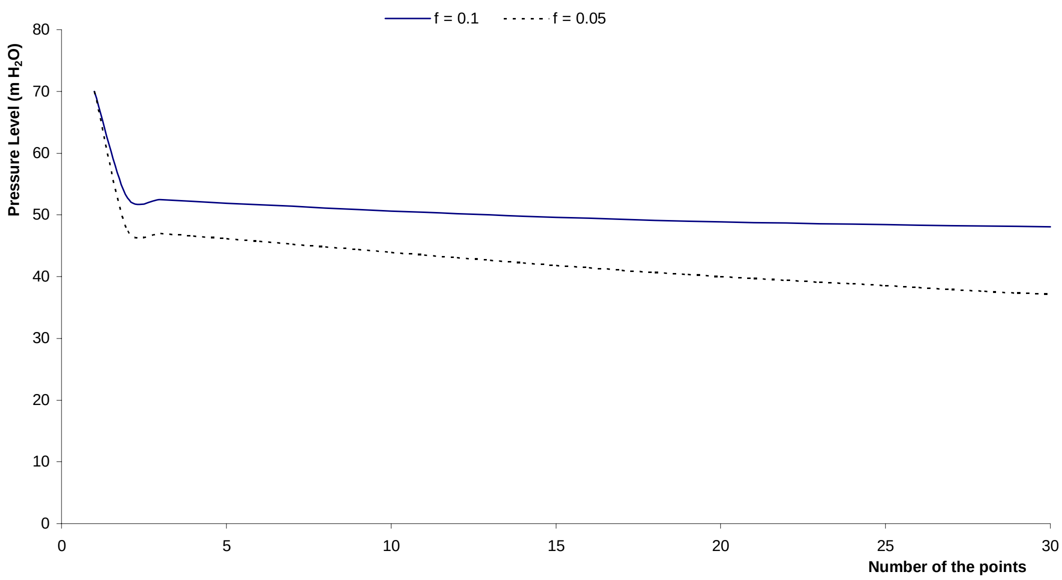

Figure 4.22 Minimum pressures at all points for linear valve closure for *f* = 0.1 and 0.05

4.1.11 Maximum and minimum pressures at all the points for stepwise valve closure for different times.

For the stepwise valve closure operation, the model was used to determine the maximum and minimum points of all the points for times, $T = 10L/a$ and $T = 100L/a$ and the same values in table 4.2 and 4.5 were used. The results are shown in Figures 4.23 and 4.24 respectively.

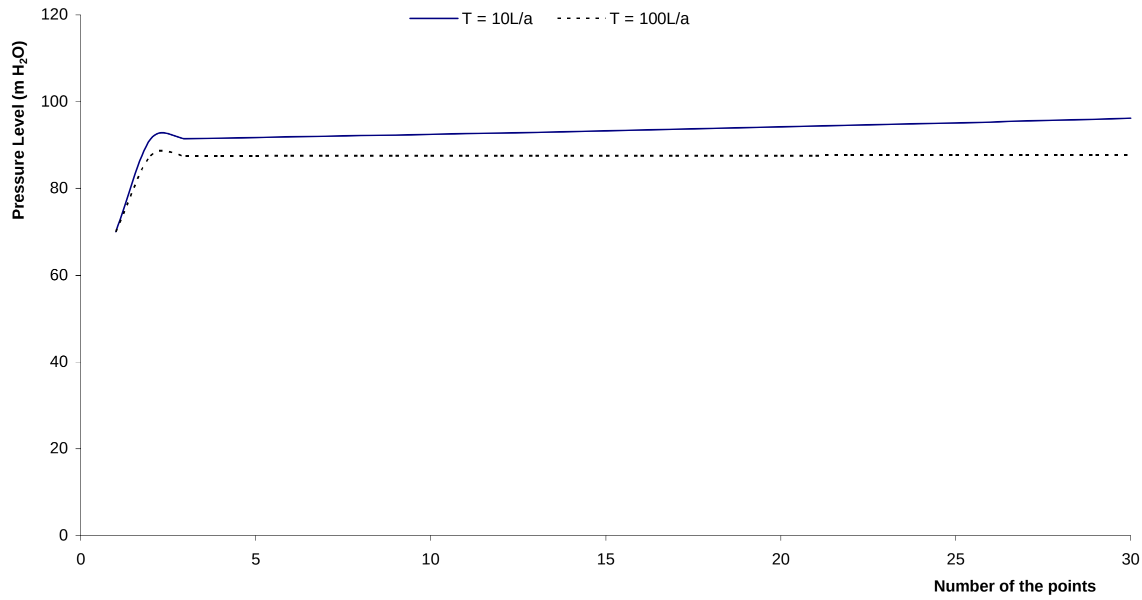

Figure 4.23 Maximum pressures at all points for stepwise valve closure for *T* = 10*L*/*a* and *T* = 100*L*/*a*

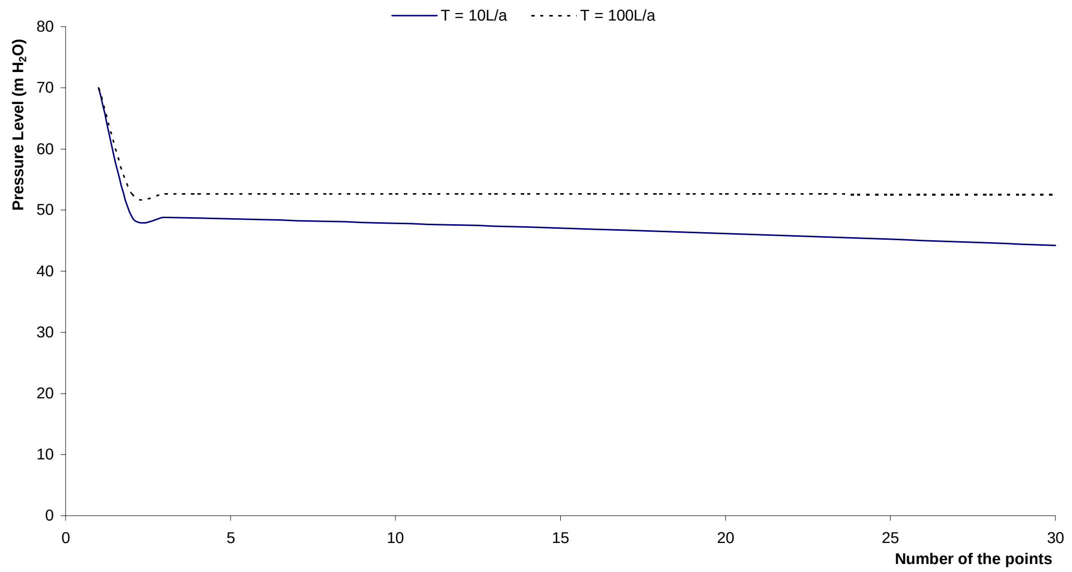

Figure 4.24 Minimum pressures at all points for stepwise valve closure for *T* = 10*L*/*a* and *T* = 100*L*/*a*

4.1.12 Closure operation when the final valve status is not fully closed

The model was modified to allow for the valve closure operations when the final valve status is not fully closed. The valve operation is stopped when the valve opening is 20 ^{o} (after which it becomes constant) for both the linear and stepwise valve closure operations. The valve operations for the linear and stepwise valve closure are described in Table 4.12

and Table 4.13 respectively. For times, T = 10L/a and T = 100L/a, the pressure levels for the linear valve and stepwise valve closure operations are shown in Figures 4.25 and 4.26 respectively.

Table 4.12 Incomplete linear valve closure operations

	Time (s)	
Valve Opening (angle in degrees)	$T = 5\mu$	$T = 50\mu$
90	0	0
30	26	500
20	60	600
20	1000	1000

Table 4.13 Incomplete stepwise valve closure operations

	Time (s)	
Valve Opening (angle in degrees)	$T = 5\mu$	$T = 50\mu$
90	0	0
35	16.5	16.5
20	60	600
20	1000	1000

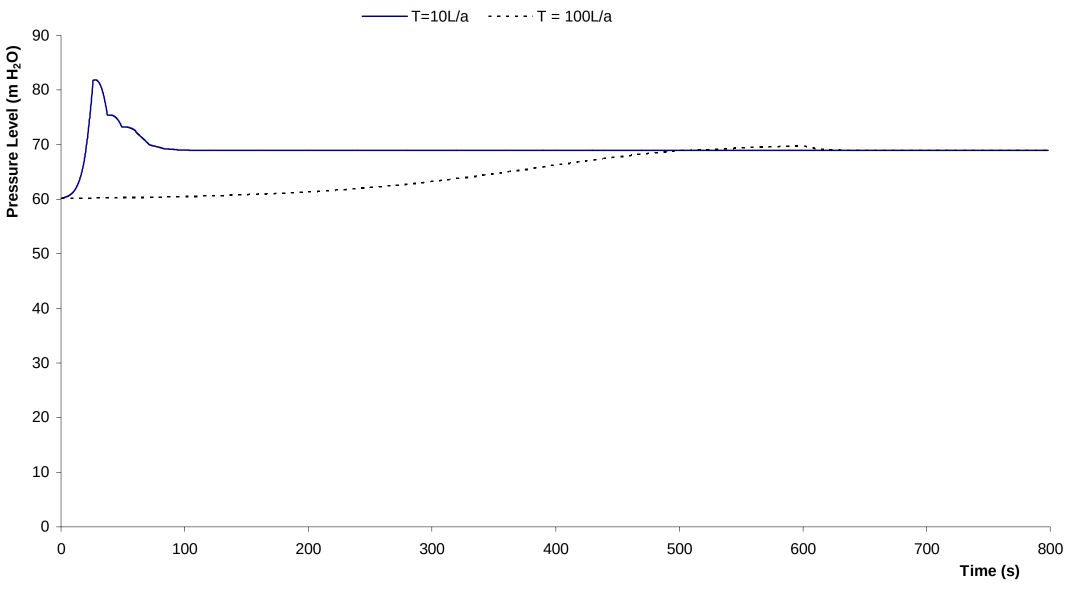

Figure 4.25 Pressure levels for incomplete linear valve closure for T = 10L/a and T = 100L/a

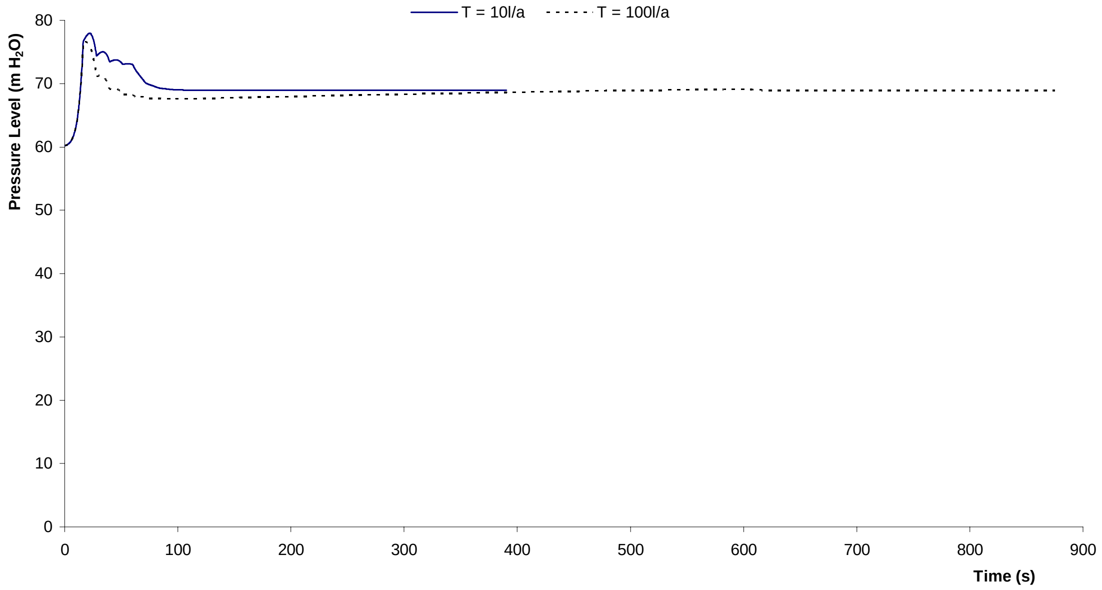

Figure 4.26 Pressure levels for incomplete stepwise valve closure for *T* = 10*L*/*a* and *T* = 100*L*/*a*

4.1.13 Maximum and minimum pressures at all points for incomplete linear valve closure at time *T* = 10*L*/*a* and *T* = 100*L*/*a*

The model was modified to allow for closure operation when the final valve status in the two valve closure operations is not fully closed. For the linear valve closure, the maximum and minimum pressures at all the grid points for time, *T* = 10*L*/*a* and *T* = 100*L*/*a* are as shown in Figure 4.27 and Figure 4.28 respectively.

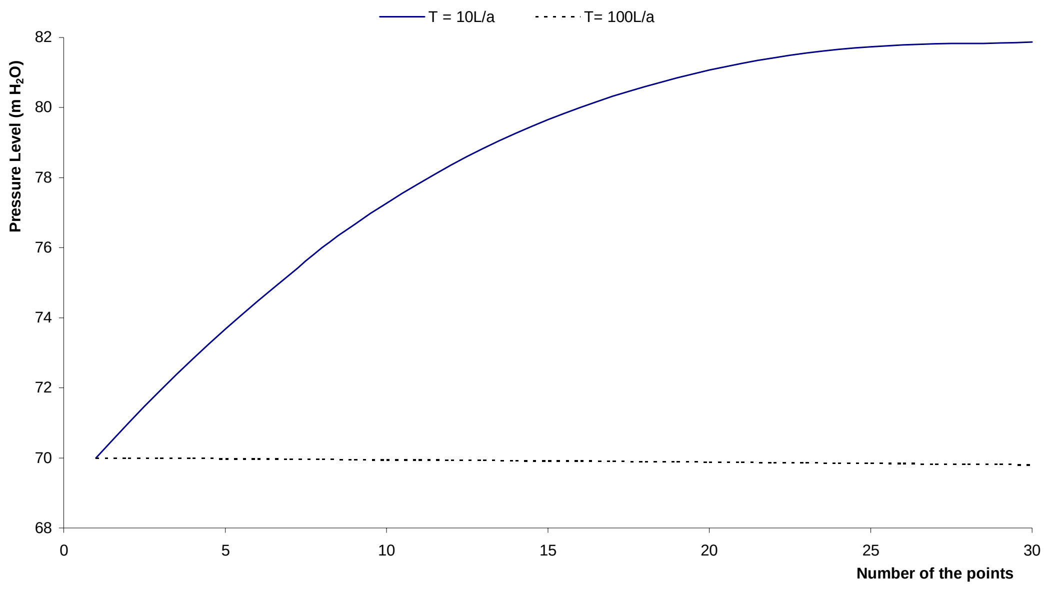

Figure 4.27 Maximum pressures at all the points for incomplete linear valve closure for *T* = 10*L*/*a* and *T* = 100*L*/*a*

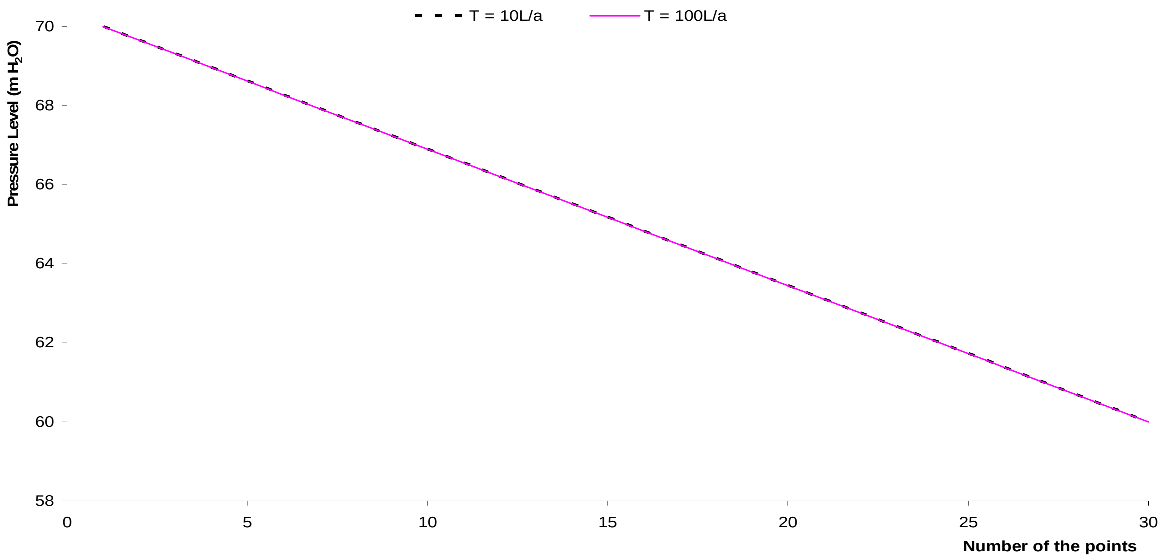

Figure 4.28 Minimum pressures at all the points for incomplete linear valve closure for *T* = 10*L*/*a* and *T* = 100*L*/*a*

4.2 Transient flow due to pump stop and check valve closure

The transient pressures produced in the pipeline in this case are mainly due to the pump stop with controlled speed reduction of the motor (so called soft stop) and the check valve closure.

4.2.1 Evaluating the model by comparing its solution with that of the graphical solution

The main input data for the model are length and diameter of the pipeline, geometrical head and the pump characteristics (Figure 3.8) curves, also the frictional coefficient, $f = 0.01$ and the wave velocity, $a = 1265$ m/s. Figure 4.29 shows the comparison between the model and graphical solution for the transient pressures at the pump for each time step. It could be seen from the Figure that the maximum and minimum pressures for the model solution are 131.7 m H_2O and 6 m H_2O respectively, whiles the maximum and minimum pressures for the graphical solution are 124.5 m H_2O and 5.2 m H_2O respectively.

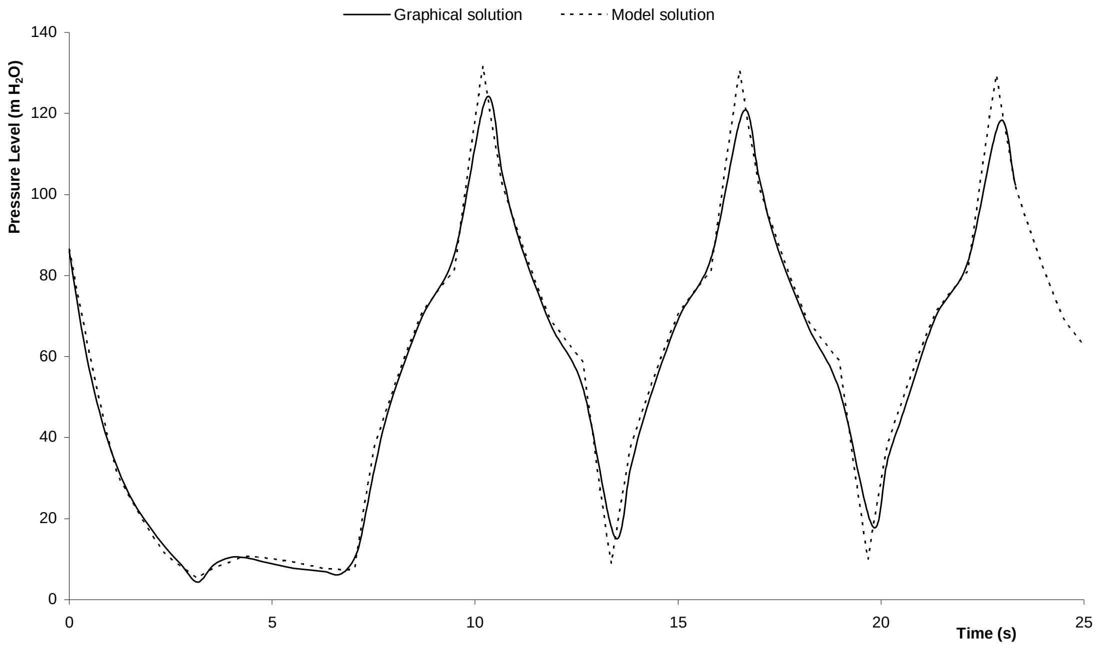

Figure 4.29 Comparison between the model and graphical solutions for the transient pressure levels at the pump for each time step

One of the most important input data for the model is to describe how the pump slows down with time, in other words, to describe the relationship between the rotational speed and time when the pump is switched off. In this case the relationship between the rotational speeds and times are described in Table 4.14. This relation was obtained from the graphical solution.

Table 4.14 Relationship between the pump's rotational speed and time

Rotational Speed, *N* (r.p.m)	**Time (s)**
25	0
16.5	1.18
12.25	2.37
9.25	3.95
6.5	6.72

4.2.2 The effect of rotational speed and time

The model was applied to six different cases with the same pipeline and the same pump as in section 4.2.1 with different ways of slowing down the pump. The effect of the different ways of slowing down the pump after the pump stop on the hydraulic transient can be seen in Figure 4.30, 4.31, and 4.32. The relationship between the time (s) and the rotation speed, *N* (r.p.m) for all the cases are shown in Tables 4.15, 4.16, 4.17, 4.18, 4.19 and 4.20.

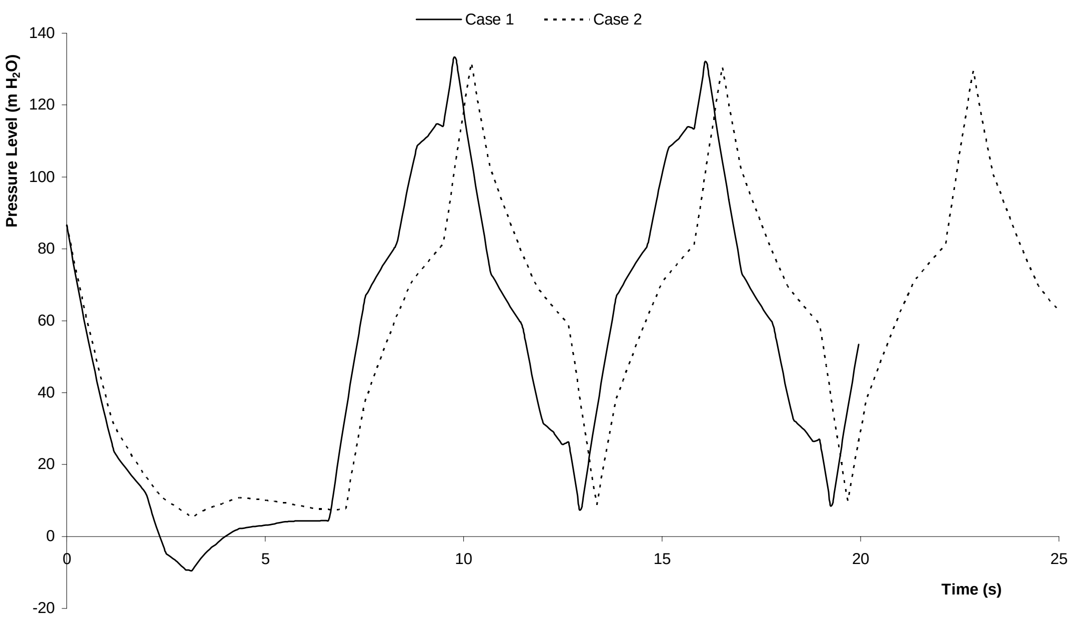

Figure 4.30 Hydraulic transient at different times of pump stop (check valve close at *T* = 6.5 s in case 1 and *T* = 7 s in case 2)

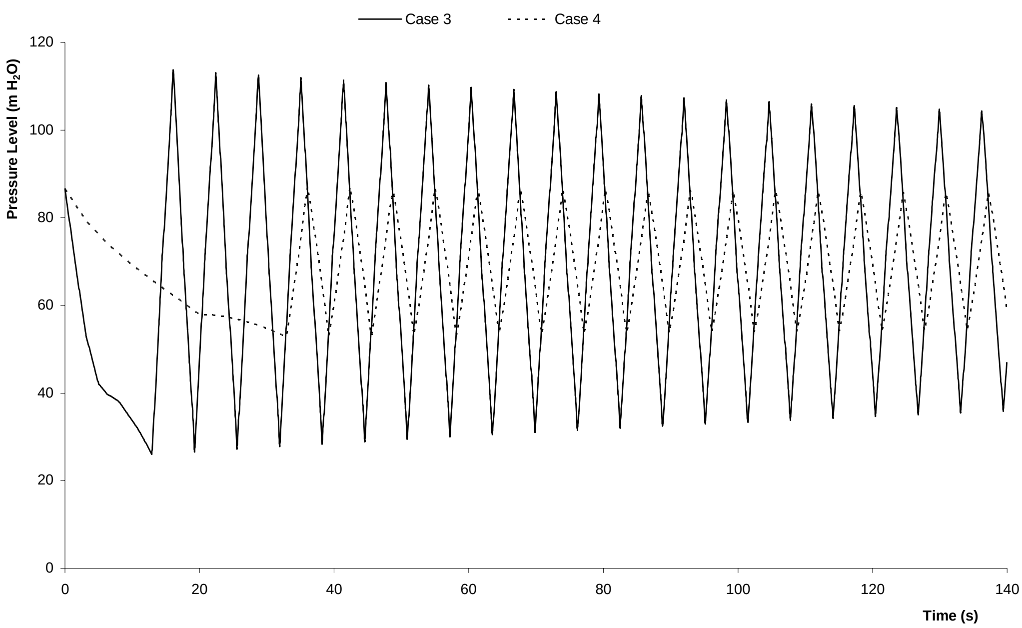

Figure 4.31 Hydraulic transient at different times of pump stop (check valve close at *T* = 13 s in case 3 and *T* = 33 s in case 4)

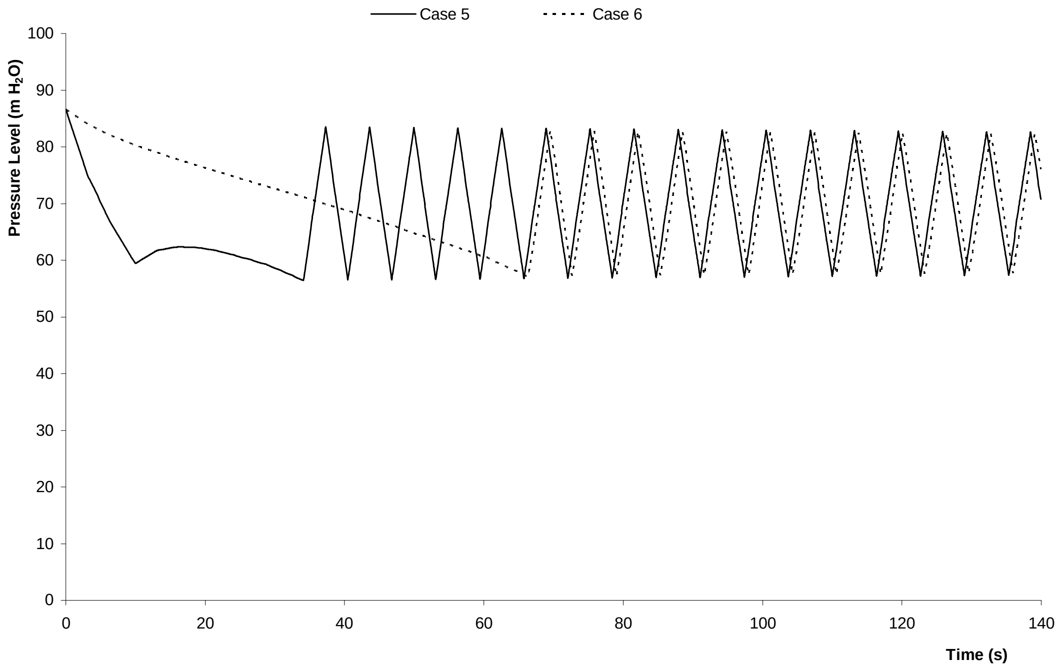

Figure 4.32 Hydraulic transient at different times of pump stop (check valve close at *T* = 34 s in case 5 and *T* = 66 s in case 6)

Table 4.15 Rotational speed and time in case 1

Rotational speed, *N* (r.p.m)	**Time (s)**
25	0
15	1.2
12.5	2.0
7.5	2.5
5	3

Table 4.16 Rotational speed and time in case 2

Rotational speed, *N* (r.p.m)	**Time (s)**
25	0
16.5	1.18
12.25	2.37
9.25	3.95
6.5	6.72

Table 4.17 Rotational speed and time in case 3

Rotational speed, *N* (r.p.m)	**Time (s)**
25	0
17.5	5
10	16
6.25	23
3.5	30

Table 4.18 Rotational speed and time in case 4

Rotational speed, *N* (r.p.m)	**Time (s)**
25	0
18.75	20
13	70
7.5	100
3.75	158

Table 4.19 Rotational speed and time in case 5

Rotational speed, *N* (r.p.m)	**Time (s)**
25	0
20	10
13.75	80
5.25	200
2.5	316

Table 4.20 Rotational speed and time in case 6

Rotational speed, *N* (r.p.m)	**Time (s)**
25	0
19.5	50
10	150
5.75	350
1.25	520

4.3 Simulating real measurements by the model

Figure 4.33 illustrates the measured hydraulic transient at Yndesäte pump station and the model simulation. Point $t = 0$ represents the time when the pump stopped and the measurements before this point represent the steady state conditions. The maximum and minimum measured pressures are 28 m H_2O and 0 m H_2O respectively, while the maximum and minimum model pressures are 21.5 m H_2O and -1 m H_2O respectively. When the model was applied to make the simulation, the time for pump stop was considered being less than 2L/a second (in this case the time considered to be less than 1 second) due to the fact that the pump was considered to have a low inertia which means that an instantaneous pressure drop will be obtained when the power is switched off. The relations between the rotational speed and the time after pump stop (Yndesäte) are described in Table 4.21.

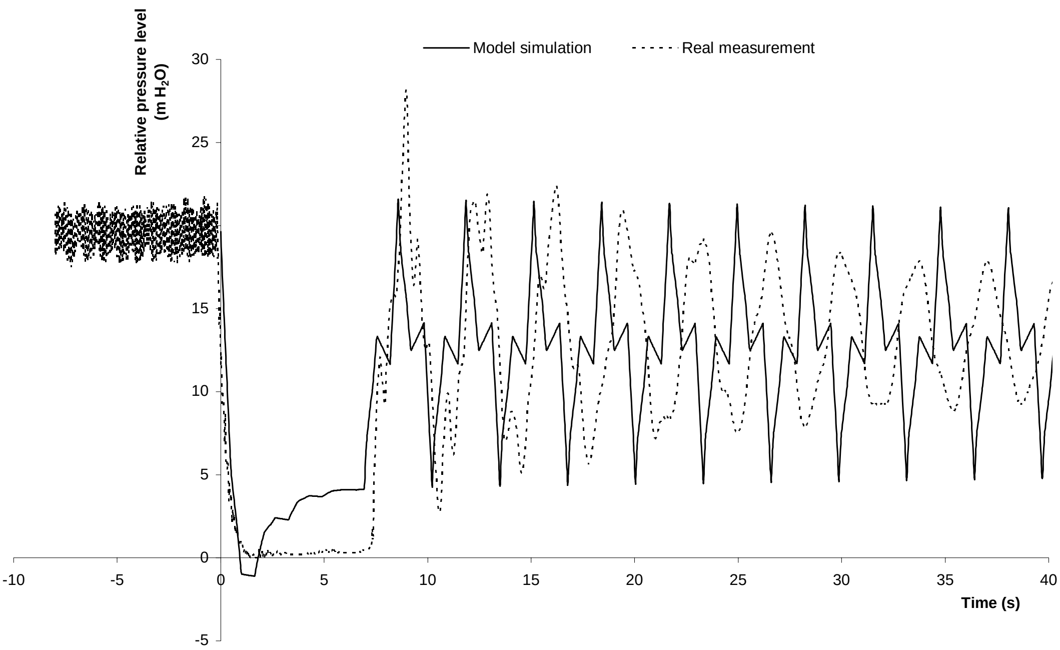

Figure 4.33 Model simulation and real measurement of hydraulic transient at Yndesäte pumping station

Table 4.21 Rotational speed and time (Yndesäte pump station)

Rotational speed, *N* (r.p.m)	**Time (s)**
25	0
20	0.3
16	0.5
13	0.8
10	1

Figure 4.34 shows the computed hydraulic transient due to the theoretical wave velocity and the actual wave velocity at Yndesäte pump station. Figure 4.35 shows the measurement and the simulation for Kristinedal pump station. The slowing down of the pump at the Kristinestad pump station was controlled by the soft stop software. The maximum and minimum measured pressures are 26 m H_2O and 7.6 m H_2O respectively, while the maximum and minimum model pressures are 28.5 m H_2O and 8 m H_2O respectively. Since there was no data to describe how the pump slows down, the simulation was done by using trial and error. The most appropriate scenario for the slowing down of the pump as a relation between the rotational speed and time is shown in Table 4.22.

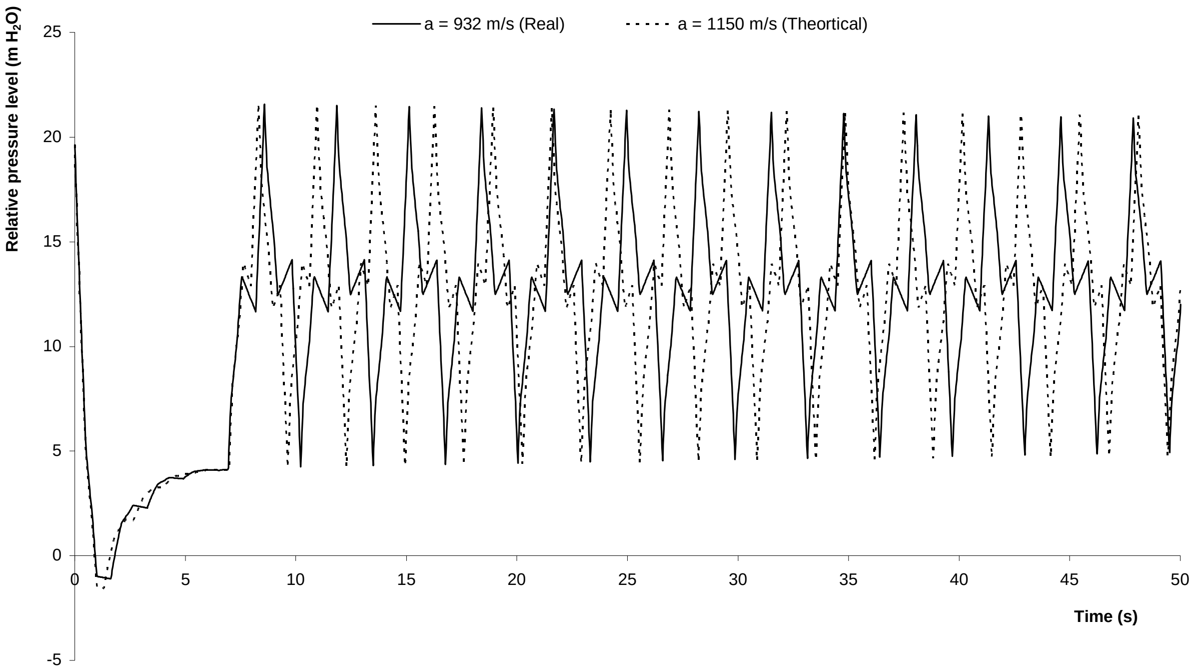

Figure 4.34 Hydraulic transient due the theoretical wave velocity and measured wave velocity at Yndesäte

Table 4.22 Rotational speed and time for Kristinestad pump station (equipped with Softstop)

Rotational speed, *N* (r.p.m)	**Time (s)**
25	0
20	5
15	15
12	24
10	30

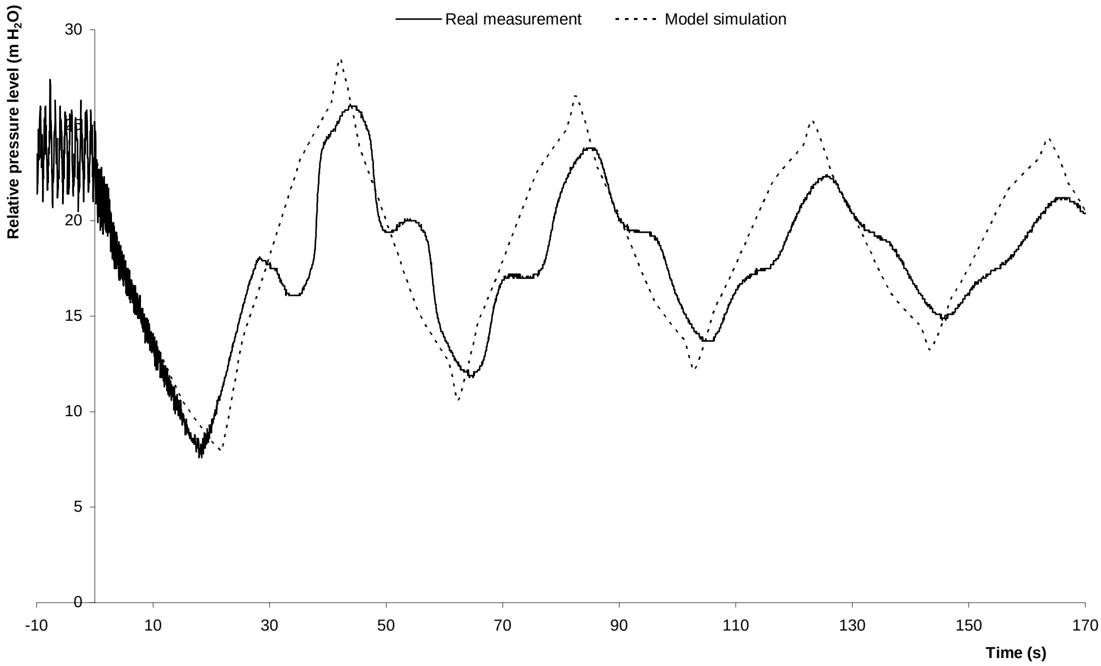

Figure 4.35 Model simulation and real measurement of hydraulic transient at Kristinedal pumping station (pump equipped with facility for Softstop)

Figure 4.36 shows the measurement and the model simulation for Kristinesdal but in this case there was no soft stop. From the measurement it is clear that the pressure starts to drop immediately after pump stop and that indicates that the pump has low inertia, so the same scenario for the relationship between the rotational speed and the time at the Yndesäte pump station was also applied in this case. The maximum and minimum measured pressures are 28.3 m H_2O and 6.8 m H_2O respectively, while the maximum and minimum model pressures are 30.7 m H_2O and 4.4 m H_2O respectively.

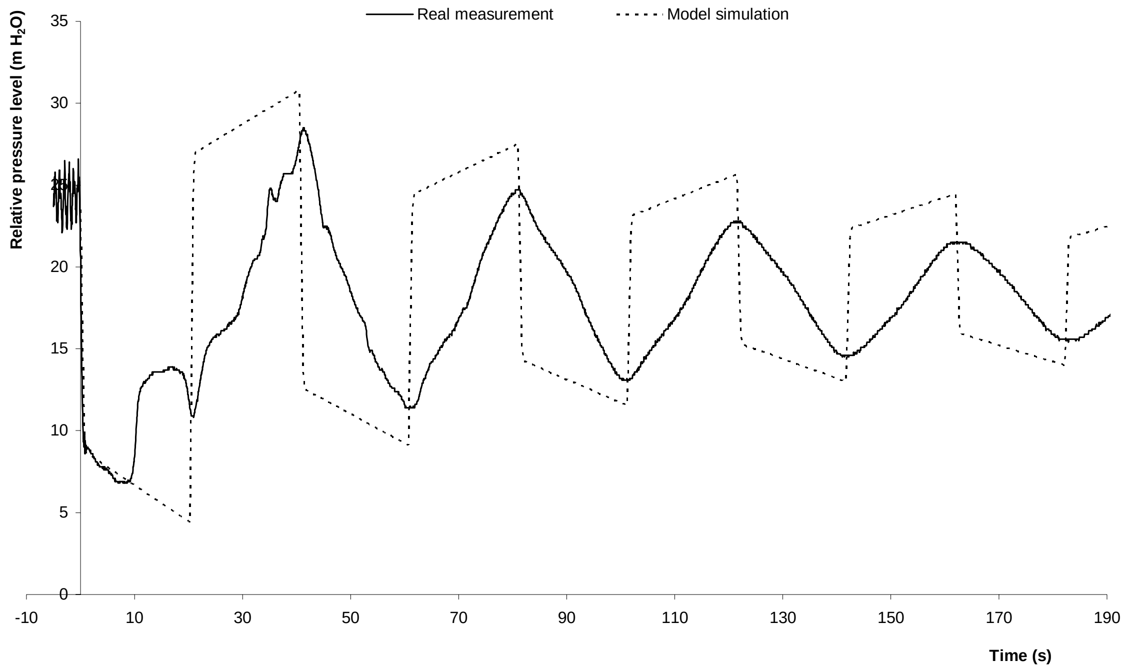

Figure 4.36 Model simulation and real measurement of hydraulic transient at Kristinedal pumping station (without soft stop)

CHAPTER FIVE

DISCUSSION

The hydraulic transient is considered to be an important reason which causes pipeline failure in the hydropower plant, waste water system, and drinking water system. This work concerned to study the phenomenon and to apply many scenarios to see its behaviour in different conditions. Mainly there are two reasons that can cause the hydraulic transient, one is closing valve in pipeline system, and the other is pump stop either due to ordinary stop or due to power failure. There are many ways to calculate the hydraulic transient for design purposes, in this study computer modelling was used as main method to perform calculations for hydraulic transient. The computer method is really more practical and effective at least in comparison to the graphical method.

5.1 Hydraulic transient due to valve closure

The evaluation for the model solution for both cases (transient due to valve closure and due to pump stop) shows that the difference between the model solution and the graphical solution for linear and stepwise closure varied between 2 to 1 m H_2O respectively for both maximum and minimum pressure. This difference considered to be acceptable especially with assumption that the graphical solution is not very accurate due to the manual way of determining the hydraulic transient. Also the difference between the linear valve closure and the step wise closure varied between 36 m H_2O to 33 m H_2O respectively for both maximum and minimum pressure and that shows how the step wise closure can significantly reduce the hydraulic transient.

The valve closing time has a significant effect on the hydraulic transient. The maximum hydraulic transient dropped about 100 m H_2O when the closure time varied from μ to 200μ while the minimum dropped about 90 m H_2O in linear valve closure. For the time closure μ (more or less equivalent to instantaneous closure) the pressure rise according to Kutta - Joukowski theory is described as:

$$\Delta H_{tr} = \frac{\Delta V \cdot a}{g} \tag{5.1}$$

where ΔV = total velocity change from initial value to zero, m/s

a = wave propagation velocity, m/s

g = acceleration of gravity, m/s^2

In the case of linear valve closure and instantaneous closure the discharge (Q) was found to be Q = 0.156 m^3/s and the section area of the pipe = 0.125 m^2 corresponding to an initial velocity calculated to be V = 1.25 m/s. Now since velocity change (ΔV) = 1.25 m/s, wave velocity, a = 950 m/s and the gravity (g) = 9.81 m/s^2 it is possible to compute the pressure rise (ΔH_{tr}).

$$\Rightarrow \Delta H_{tr} = 121 \text{ m}$$

Since the initial pressure at the valve = 60 m H_2O, the maximum pressure will be 121 + 60 = 181 m H_2O and the comparison between this value and the calculated maximum pressure by the model (186 m H_2O) shows that there is only a slight difference between them and this indicates the reliability of the model's performance.

In stepwise closure model, the maximum pressure dropped about 10 m H_2O while the minimum dropped about 8 m H_2O so it is very clear that closing the valve slowly can reduce the hydraulic transient. Anyway it is important to recognise that the pressures lower than -10 H_2O are not realistic in our case (z = 0).

The valve operation also has impact on the hydraulic transient. The ideal way to close the valve to reduce the transient is by closing it rapidly in the beginning (till valve opening becomes between 30° - 25°) and then close it slowly.

The water velocity and the frictional coefficient are considered to be important factors which affect the hydraulic transient. Of course both of them relate to each other because for example the high frictional coefficient will definitely lead to reducing the velocity. Anyway the increase of the velocity will lead to an increase in hydraulic transient and vice versa. Also low frictional coefficient leads to increase in hydraulic transient and vice versa.

The envelope graphs show that maximum pressures as well as minimum pressures occur immediately upstream of the valve (provided the pipeline is horizontal as has been assumed in the computations. A non horizontal pipeline might give different results).

The hydraulic transient, when the valve is not completely closed, is less compared to the case when the valve is completely closed and that is obviously because not all the discharge reflect in the pipeline (some of it goes through the valve), this could explain why the pressure is less in incomplete valve closure.

5.2 Hydraulic transient due to the pump stop and check valve closure

The evaluation of the graphical solution as compared to the more exact computer model solution shows that the difference between the model and the graphical solution was about 7 m H_2O both for maximum and minimum pressures which might be considered to be acceptable, at least in this case. Thus, the simple graphical solution provides acceptable results. However, a pipeline might be especially sensitive to low pressures and in some case it might be necessary to use the computer model in order to obtain a necessary accuracy.

One of main problems in dealing with model is how to determine the relation between the rotation speed and the time after pump stop because it is an important input data for the model. The model which was used here assumed that the decrease in pump speed was prescribed. The graphical solution was used to calculate the pump run down (change of speed). In order to make a comparison between the graphical solution and the software results it was necessary to use the pump run down from graphical solution and input it into the software case. Generally, the pump run down can be determined by assuming reasonable profile for the change of speed after determining the closing time for the check valve.

The pump stop can be controlled by equipping the pump with a kind of control circuit called softstop, which can help in making pump stop slow down. The softstop could be considered as a virtual more or less high inertia of the pump. Slowing down the pump has significant impact in reducing the hydraulic transient. For instance, the initial pressure decrease becomes more gradual the lower the pump decrease rate is. At the same time the pressure peaks (max/min) decrease. Also one has to note that the check valve will close before the pump will come to complete stop, the water column in the pipeline tends to reverse long before the pump comes to its prescribed zero rotational speed. After the check valve closure, the pump is isolated from the pipeline and the pump will not affect the transient any more.

The model's application on some measurements from different pump stations showed good results. One could notice that the initial low pressure phase is very similar (especially the time duration) for the model and for the measurement. This is an indication that the model simulation is good. In Yndesäte pump station the difference between the maximum measured pressure and maximum simulated pressure = 6.5 m H_2O while the difference between the minimum pressure = 1 m H_2O. The pump was considered to have low inertia so the preset pump down had to be fast enough in order to simulate the initial pressure decrease.

In Kristinedal pump station for the case in which the pump was equipped with soft stop the difference between the maximum measured pressure and the maximum simulated pressure = 2.5 m H_2O and the difference between the minimum = 1.6 m H2O. In this case, because the pump was equipped with softstop, the pressure reaches its minimum value after about 30 seconds (pretty long time), indicating that the check valve is closed. In the case with no softstop the difference between the measured and the simulated pressure is about 2.5 m H_2O for both minimum and maximum pressures. These results show that, the model is pretty dependable for design purposes.

CHAPTER SIX

CONCLUSION AND RECOMMENDATION

6.1 Conclusion

One purpose of this study was to compare the graphical solution of two different hydraulic transient flows with its corresponding solutions from more detailed computer models. This required the development of the two computer models. It was found that the graphical solution agreed well with the computer solution.

A second purpose was to apply the valve computer model to determine the effect of different parameters, such as valve closure operation, wave velocities, etc on the transient.

A third purpose of the study was to apply the pump computer model to some real cases where hydraulic transient measurements had been performed, both with and without soft stop arrangement. A good agreement was obtained.

6.2 Recommendations

To decrease and control the hydraulic transient in pipelines the following recommendations should be considered:

- Use stepwise closure with proper operation (rapid closure in the beginning and then slower).

- Close the valve as slowly as possible.

- The pipeline material and the expectable flow should be considered in the pipeline design.

- The point before the valve directly should be considered as a critical position in the pipeline design (horizontal pipeline case).

- The pump stations should be equipped with softstop program to increase the inertia of the pump.

- Some kind of safety factor should be applied to determine the maximum and minimum designing pressures.

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Web Link

Pump World: Pump characteristics curve

www.pumpworld.com (11-12-2006)

APPENDICES

Appendix 1 Information about the FORTRAN programming language

1.1 Brief History about FORTRAN

A team of programmers at the International Business Machines Corporation (IBM) led by John Bacus developed the FORTRAN programming language, which is one of the oldest programming languages, and it was first published in 1957. The name FORTRAN is an acronym for FORmula TRANslation, because it was designed to allow easy translation of mathematical formulas into code. Often referred to as a scientific language, FORTRAN was the first high-level language, using the first compiler ever developed.

1.2 Areas of Application of FORTRAN

FORTRAN is useful for a wide variety of applications; some of the more outstanding ones are as follows:

- Number crunching - due to the more natural (like its true algebraic form) way of expressing complex mathematical functions and its quick execution time, FORTRAN is easy and efficient at processing mathematical equations.

- Scientific, mathematical, statistical, and engineering type procedures - due to its rapid number-crunching ability FORTRAN is a good choice for these types of applications.

1.3 Programming Languages

A program is a sequence of instructions written in a programming language. A programming language is a language used to communicate with the computer. For a program to be executable, the program must be written in the binary machine code recognised by the processor. Machine code programming is difficult and prone to error. Furthermore, since each type of processor has its own machine code, a program written for one type of processor is not executable by any other. There are two types of programming languages, and these are High level and Low level languages.

1.3.1 High Level Languages

These languages are only understood by machines (machine codes). High level language instructions are not executable. Instead, a high level language source program is read as input by a program called a compiler, which checks its syntax and, if it is free from errors, compiles an equivalent machine code object program. (If the source program contains syntax errors, the compiler outputs a number of messages indicating the nature of the errors and where they occur).

Although it is in machine code, the object program is incomplete because it includes references to subprograms which it requires for such common tasks as reading input, producing output and computing mathematical functions. These subprograms are grouped together in libraries which are available for use by all object programs. To create an executable program, the object program must be linked to the subprogram libraries it requires. The executable program may then be loaded into memory and run. The steps required to compile, link and run a FORTRAN program are illustrated by Figure A1.

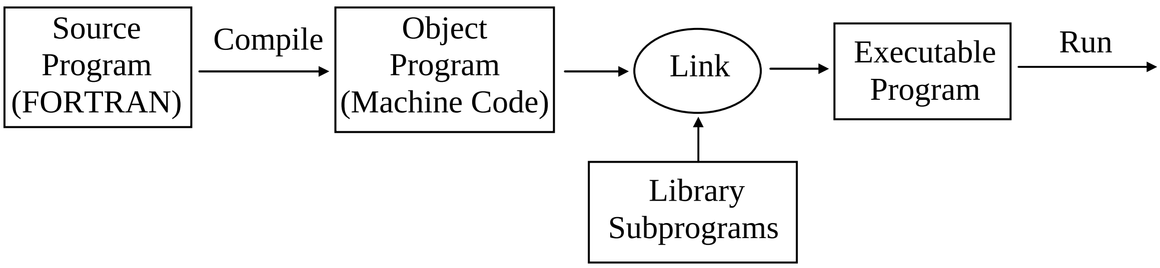

Figure A1 Compiling, linking and Running a FORTRAN Program
(Source: Osei-Bonsu Charles, 1998)

1.3.2 Low Level Languages

Low level languages are machine specific. These languages provide little or no abstraction from a computer's microprocessor. The word "low" does not imply that the language is inferior to high-level programming languages but rather refers to the small or nonexistent amount of abstraction between the language and machine language; because of this, low-level languages are sometimes described as being "close to the hardware."

1.4 Errors in Programming

FORTRAN programming, including all other programming languages, is prone to programming errors, and these are syntax, logic and runtime error.

1.4.1 Syntax Error

This is the error detected during the compilation of the source program.

1.4.2 Logic Error

This error can never be detected during compilation or execution of the program.

1.4.3 Runtime Error

This error is only detected during the running or execution of the program.

1.5 Coding Sheet/Coding Format

The FORTRAN statement has to follow certain values of layout it must follow, and these are indicated below.

1) The first rule is that the last 8 Columns (73 – 80) are not used at all, or to be more accurate, they are not treated as part of the program but can be used for identification purposes.

2) The second rule is that the first 6 columns are used for special purposes, and that as a result the FORTRAN statement always begin at (or after) Column 7.

Special purposes for which the first 6 Columns are use are:

a) Column 1 is used for comments. If the first column contains a ‘c’ or an asterisk (*), then the whole line is treated as comment and ignored when the program is being compiled.

b) Column 6 is a continuation line. If Column 6 of a line contains any character other than space and zero, then the line is treated as continuation of the previous line. There can be up to 19 continuation lines following any initial line.

c) Columns 2 – 5 are the statement number fields. These are used solely for statement numbers, and will be blank if the statement is not labelled. Only statements that are referred to, by others need to be numbered.

d) Columns 7 – 72 are for the statement body. The main body of the statement will be contained between Columns 7 and 72 inclusive.

Appendix 2 Program to calculate the hydraulic transient in a pipeline (TRANSIENT.FOR)

```
C   THIS PROGRAM WILL CALCULATE THE HYDRAULIC TRANSIENT IN PIPE
LINE
      DIMENSION H(1:1000),H1(1:1000),Q(1:1000),HCP(1:1000),Q1(1:1000),
     &HCM(1:1000)
      OPEN(UNIT=7,FILE='MOSABIN.DAT',STATUS='OLD')
      OPEN(UNIT=8,FILE='MOSABOUT.DAT',STATUS='OLD')
      READ(7,*)XL,DIA,HIN,HOUT
      READ(7,*)N,FRIC,WV,XKVALVE0
      READ(7,*)TIMEEND,M
      READ(7,*)ANGLE1,ANGLE2,ANGLE3,ANGLE4,T1,T2,T3,T4
C    CONSTANT
      DELTX=XL/(N-1)
      B=WV/9.81
      AREA=DIA**2*3.14/4
      XF=FRIC*DELTX/19.62/DIA/AREA**2
C    STEADY STATE CALCULATION
      QX=(HIN-HOUT)/(XKVALVE/AREA/19.62+XF*XL/DELTX)
      QO=SQRT(QX)
      DELTT=DELTX/WV
      TIME=0
      HXX=XF*QO**2
      DO 30 I=1,N
      H(I)=HIN-HXX*(I-1)
      Q(I)=QO
   30 CONTINUE
      TIME=TIME+DELTT
  400 CONTINUE
C    CALCULATION OF HCP,HCM
      DO 50 I=1,N
      YYA=B*Q(I)/AREA
      YYB=XF*Q(I)*ABS(Q(I))
      HCP(I)=H(I)+YYA-YYB
      HCM(I)=H(I)-YYA+YYB
   50 CONTINUE
C    INNER POINTS
      DO 70 I=2,N-1
      H1(I)=(HCP(I-1)+HCM(I+1))/2
      Q1(I)=(H1(I)-HCM(I+1))/B*AREA
   70 CONTINUE
C    BOUNDRY POINTS
C    INLET
      H1(1)=HIN
      Q1(1)=(HIN-HCM(2))*AREA/B
C    OUTLET(VALVE)
      IF(TIME.GT.T1)GO TO 100
      THETA=ANGLE1
      XKVALVE=EXP((3.8-0.038*THETA)*2.3)
```

```
      GO TO 140
100 CONTINUE
      IF(TIME.GT.T2)GO TO 110
      AA1=(ANGLE1-ANGLE2)/(T1-T2)
      BB1=(ANGLE1-AA1*T1)
      THETA=AA1*TIME+BB1
      XKVALVE=EXP((3.78-0.038*THETA)*2.3)
      GO TO 140
110 CONTINUE
      IF(TIME.GT.T3)GO TO 120
      AA2=(ANGLE2-ANGLE3)/(T2-T3)
      BB2=ANGLE2-AA2*T2
      THETA=AA2*TIME+BB2
      XKVALVE=EXP((3.78-0.038*THETA)*2.3)
      GO TO 140
120 CONTINUE
      IF(TIME.GT.T4)GO TO 130
      AA3=(ANGLE3-ANGLE4)/(T3-T4)
      BB3=ANGLE3-AA3*T3
      THETA=AA3*TIME+BB3+0.0001
      XKVALVE=EXP((3.78-0.038*THETA)*2.3)
      GO TO 140
130 CONTINUE
      Q1(N)=0
      H1(N)=HCP(N-1)
      GO TO 210
140 CONTINUE
      A1=B*2*9.81*AREA/XKVALVE
      A2=2*9.81*AREA**2*(HOUT-HCP(N-1))/XKVALVE
      IF(A2.GT.0)GO TO 200
      Q1(N)=-A1/2+SQRT((0.5*A1)**2-A2)
      H1(N)=HCP(N-1)-B*Q1(N)/AREA
      GO TO 210
200 CONTINUE
      B1=-B*2*9.81*AREA/XKVALVE
      B2=2*9.81*AREA**2*(HCP(N-1)-HOUT)/XKVALVE
      Q1(N)=-B1/2-SQRT((0.5*B1)**2-B2)
      H1(N)=HCP(N-1)-B*Q1(N)/AREA
210 CONTINUE
      WRITE(8,*)TIME,H(N),Q(N)
240 CONTINUE
      IF(TIME.GT.TIMEEND)GO TO 450
      TIME=TIME+DELTT
      DO 300 I=1,N
      H(I)=H1(I)
      Q(I)=Q1(I)
300 CONTINUE
      GO TO 400
450 CONTINUE
      END
```

Appendix 3 Modified program to calculate hydraulic transient when valve is not fully closed

```
C    THIS PROGRAM WILL CALCULATE THE HYDRAULIC TRANSIENT IN PIPE
LINE
      DIMENSION H(1:1000),H1(1:1000),Q(1:1000),HCP(1:1000),Q1(1:1000),
     &HCM(1:1000)
      OPEN(UNIT=7,FILE='MOSABIN.DAT',STATUS='OLD')
      OPEN(UNIT=8,FILE='MOSABOUT.DAT',STATUS='OLD')
      READ(7,*)XL,DIA,HIN,HOUT
      READ(7,*)N,FRIC,WV,XKVALVE0
      READ(7,*)TIMEEND,M
      READ(7,*)ANGLE1,ANGLE2,ANGLE3,ANGLE4,T1,T2,T3,T4
C     CONSTANT
      DELTX=XL/(N-1)
      B=WV/9.81
      AREA=DIA**2*3.14/4
      XF=FRIC*DELTX/19.62/DIA/AREA**2
C     STEADY STATE CALCULATION
      QX=(HIN-HOUT)/(XKVALVE/AREA/19.62+XF*XL/DELTX)
      QO=SQRT(QX)
      DELTT=DELTX/WV
      TIME=0
      HXX=XF*QO**2
      DO 30 I=1,N
      H(I)=HIN-HXX*(I-1)
      Q(I)=QO
   30 CONTINUE
      TIME=TIME+DELTT
  400 CONTINUE
C     CALCULATION OF HCP,HCM
      DO 50 I=1,N
      YYA=B*Q(I)/AREA
      YYB=XF*Q(I)*ABS(Q(I))
      HCP(I)=H(I)+YYA-YYB
      HCM(I)=H(I)-YYA+YYB
   50 CONTINUE
C     INNER POINTS
      DO 70 I=2,N-1
      H1(I)=(HCP(I-1)+HCM(I+1))/2
      Q1(I)=(H1(I)-HCM(I+1))/B*AREA
   70 CONTINUE
C     BOUNDRY POINTS
C     INLET
      H1(1)=HIN
      Q1(1)=(HIN-HCM(2))*AREA/B
C   OUTLET(VALVE)
      IF(TIME.GT.T1)GO TO 100
      THETA=ANGLE1
      XKVALVE=EXP((3.8-0.038*THETA)*2.3)
```

```
      GO TO 140
100 CONTINUE
      IF(TIME.GT.T2)GO TO 110
      AA1=(ANGLE1-ANGLE2)/(T1-T2)
      BB1=(ANGLE1-AA1*T1)
      THETA=AA1*TIME+BB1
      XKVALVE=EXP((3.78-0.038*THETA)*2.3)
      GO TO 140
110 CONTINUE
      IF(TIME.GT.T3)GO TO 120
      AA2=(ANGLE2-ANGLE3)/(T2-T3)
      BB2=ANGLE2-AA2*T2
      THETA=AA2*TIME+BB2
      XKVALVE=EXP((3.78-0.038*THETA)*2.3)
      GO TO 140
120 CONTINUE
      IF(TIME.GT.T4)GO TO 130
      AA3=(ANGLE3-ANGLE4)/(T3-T4)
      BB3=ANGLE3-AA3*T3
      THETA=AA3*TIME+BB3+0.0001
      XKVALVE=EXP((3.78-0.038*THETA)*2.3)
      GO TO 140
130 CONTINUE
      THETA=ANGLE4
      XKVALVE=EXP((3.78-0.038*THETA)*2.3)
      GO TO 210
140 CONTINUE
      A1=B*2*9.81*AREA/XKVALVE
      A2=2*9.81*AREA**2*(HOUT-HCP(N-1))/XKVALVE
      IF(A2.GT.0)GO TO 200
      Q1(N)=-A1/2+SQRT((0.5*A1)**2-A2)
      H1(N)=HCP(N-1)-B*Q1(N)/AREA
      GO TO 210
200 CONTINUE
      B1=-B*2*9.81*AREA/XKVALVE
      B2=2*9.81*AREA**2*(HCP(N-1)-HOUT)/XKVALVE
      Q1(N)=-B1/2-SQRT((0.5*B1)**2-B2)
      H1(N)=HCP(N-1)-B*Q1(N)/AREA
210 CONTINUE
      WRITE(8,*)TIME,H(N),Q(N)
240 CONTINUE
      IF(TIME.GT.TIMEEND)GO TO 450
      TIME=TIME+DELTT
      DO 300 I=1,N
      H(I)=H1(I)
      Q(I)=Q1(I)
300 CONTINUE
      GO TO 400
450 CONTINUE
      END
```

Appendix 4 Modified program to calculate the maximum and minimum pressure

```
C    THIS PROGRAM WILL CALCULATE THE MAXIMUM AND THE MINMUM
PRESSURE IN PIPE LINE
     DIMENSION H(1:1000),H1(1:1000),Q(1:1000),HCP(1:1000),Q1(1:1000),
    &HCM(1:1000),HMAX(1:1000),HMIN(1:1000)
     OPEN(UNIT=7,FILE='MOSABIN.DAT',STATUS='OLD')
     OPEN(UNIT=8,FILE='MOSABOUT.DAT',STATUS='OLD')
     OPEN(UNIT=9,FILE='MOSABMAX.DAT',STATUS='OLD')
     READ(7,*)XL,DIA,HIN,HOUT
     READ(7,*)N,FRIC,WV,XKVALVE0
     READ(7,*)TIMEEND,M
     READ(7,*)ANGLE1,ANGLE2,ANGLE3,ANGLE4,T1,T2,T3,T4
C    CONSTANT
     DELTX=XL/(N-1)
     B=WV/9.81
     AREA=DIA**2*3.14/4
     XF=FRIC*DELTX/19.62/DIA/AREA**2
C    STEADY STATE CALCULATION
     QX=(HIN-HOUT)/(XKVALVE/AREA/19.62+XF*XL/DELTX)
     QO=SQRT(QX)
     DELTT=DELTX/WV
     TIME=0
     HXX=XF*QO**2
     DO 30 I=1,N
     H(I)=HIN-HXX*(I-1)
     Q(I)=QO
  30 CONTINUE
     DO 40 I=1,N
     HMAX(I)=H(I)
     HMIN(I)=H(I)
  40 CONTINUE
     TIME=TIME+DELTT
 400 CONTINUE
C    CALCULATION OF HCP,HCM
     DO 50 I=1,N
     YYA=B*Q(I)/AREA
     YYB=XF*Q(I)*ABS(Q(I))
     HCP(I)=H(I)+YYA-YYB
     HCM(I)=H(I)-YYA+YYB
  50 CONTINUE
C    INNER POINTS
     DO 70 I=2,N-1
     H1(I)=(HCP(I-1)+HCM(I+1))/2
     Q1(I)=(H1(I)-HCM(I+1))/B*AREA
  70 CONTINUE
C    BOUNDRY POINTS
C    INLET
     H1(1)=HIN
     Q1(1)=(HIN-HCM(2))*AREA/B
```

```
C     OUTLET(VALVE)
      IF(TIME.GT.T1)GO TO 100
      THETA=ANGLE1
      XKVALVE=EXP((3.8-0.038*THETA)*2.3)
      GO TO 140
  100 CONTINUE
      IF(TIME.GT.T2)GO TO 110
      AA1=(ANGLE1-ANGLE2)/(T1-T2)
      BB1=(ANGLE1-AA1*T1)
      THETA=AA1*TIME+BB1
      XKVALVE=EXP((3.78-0.038*THETA)*2.3)
      GO TO 140
  110 CONTINUE
      IF(TIME.GT.T3)GO TO 120
      AA2=(ANGLE2-ANGLE3)/(T2-T3)
      BB2=ANGLE2-AA2*T2
      THETA=AA2*TIME+BB2
      XKVALVE=EXP((3.78-0.038*THETA)*2.3)
      GO TO 140
  120 CONTINUE
      IF(TIME.GT.T4)GO TO 130
      AA3=(ANGLE3-ANGLE4)/(T3-T4)
      BB3=ANGLE3-AA3*T3
      THETA=AA3*TIME+BB3+0.0001
      XKVALVE=EXP((3.78-0.038*THETA)*2.3)
      GO TO 140
  130 CONTINUE
      THETA=ANGLE4
      XKVALVE=EXP((3.78-0.038*THETA)*2.3)
      GO TO 210
  140 CONTINUE
      A1=B*2*9.81*AREA/XKVALVE
      A2=2*9.81*AREA**2*(HOUT-HCP(N-1))/XKVALVE
      IF(A2.GT.0)GO TO 200
      Q1(N)=-A1/2+SQRT((0.5*A1)**2-A2)
      H1(N)=HCP(N-1)-B*Q1(N)/AREA
      GO TO 210
  200 CONTINUE
      B1=-B*2*9.81*AREA/XKVALVE
      B2=2*9.81*AREA**2*(HCP(N-1)-HOUT)/XKVALVE
      Q1(N)=-B1/2-SQRT((0.5*B1)**2-B2)
      H1(N)=HCP(N-1)-B*Q1(N)/AREA
  210 CONTINUE
      WRITE(8,*)TIME,H(N),Q(N)
      DO 240 I=1,N
      IF(H1(I).LT.HMAX(I))GO TO 240
      HMAX(I)=H1(I)
  240 CONTINUE
      DO 250 I=1,N
      IF(H1(I).GT.HMIN(I))GO TO 250
      HMIN(I)=H1(I)
```

```
250 CONTINUE
   IF(TIME.GT.TIMEEND)GO TO 450
   TIME=TIME+DELTT
   DO 300 I=1,N
   H(I)=H1(I)
   Q(I)=Q1(I)
300 CONTINUE
   GO TO 400
450 CONTINUE
   DO 260 I=1,N
   WRITE(9,*)I,HMAX(I),HMIN(I)
260 CONTINUE
   END
```

Appendix 5 Program (model) for pump stop (SOFTSTOP.FOR)

```
C    THIS PROGRAM COMPUTES THE FLUID TRANSIENT IN A PIPELINE
C    WITH A PUMP EQUIPPED WITH SOFT STOP AND A CHECK VALVE. THE
C    SOFT STOP IS DESCRIBED BY A PREDETERMINED SLOWDOWN OF THE
C    ROTATIONAL SPEED OF THE PUMP. SUTER DIAGRAM
      DIMENSION H(1:200),H1(1:200),Q(1:200),Q1(1:200),SUT1(1:89),
     &SUT2(1:89),HCP(1:200),HCM(1:200)
      COMMON /SOFTSTOP/XNR0,XNR1,XNR2,XNR3,XNR4,T0,T1,T2,T3,T4,
     &A1,A2,A3,A4,B1,B2,B3,B4
      OPEN(UNIT=7,FILE='SUT35UT.DAT',STATUS='OLD')
      OPEN(UNIT=8,FILE='SOFTIN.DAT',STATUS='OLD')
      OPEN(UNIT=9,FILE='SOFTUT.DAT',STATUS='OLD')
      READ(8,*)XLAENGD,FRIK,DIA,HIN,HUT
      READ(8,*)VAGH,N,M,EPS
      READ(8,*)HR,QR,XNRR
      READ(8,*)T0,T1,T2,T3,T4,TIDSLUT
      READ(8,*)XNR0,XNR1,XNR2,XNR3,XNR4
      DO 10 I=1,88
      READ(7,*)SUT1(I),SUT2(I)
   10 CONTINUE
C
C    CONSTANTS
C
      A1=(XNR0-XNR1)/(T0-T1)
      B1=XNR1-A1*T1
      A2=(XNR1-XNR2)/(T1-T2)
      B2=XNR2-A2*T2
      A3=(XNR2-XNR3)/(T2-T3)
      B3=XNR3-A3*T3
      A4=(XNR3-XNR4)/(T3-T4)
      B4=XNR4-A4*T4
      HDLIN=HIN/HR
      HDLUT=HUT/HR
      APIPE=3.14*DIA**2/4
      B=VAGH/9.81/APIPE
      DELTX=XLAENGD/(N-1)
      DELTT=DELTX/VAGH
      R=FRIK*DELTX/19.62/DIA/APIPE**2
      XKONST1=B*QR/HR
      XKONST2=HIN/HR
      XKONST3=FRIK*XLAENGD/DIA*QR**2/HR/APIPE**2/19.62
C
C    CALCULATION OF STEADY STATE
C
      ALFA0=XNR0/XNRR
      V0=1.
   90 CONTINUE
      X=3.1416+ATAN2(V0,ALFA0)
      IHEL=X/3.1416*44
```

```
      AA0=SUT1(IHEL+1)
      AA1=SUT2(IHEL+1)
      XX1=ALFA0**2+V0**2
      FSTEAD=XX1*(AA0+AA1*(3.1416+ATAN2(V0,ALFA0)))-HDLUT
     &+HDLIN-XKONST3*V0**2
      DFSTEAD=ALFA0**3*AA1/XX1+2*V0*AA0+2*V0*AA1*3.1416+2*V0*AA1
     &*ATAN2(V0,ALFA0)+V0**2*AA1*ALFA0/XX1-XKONST3*2*V0
      DELTV0=-FSTEAD/DFSTEAD
      V0=V0+DELTV0
      ADELTV0=ABS(DELTV0)
      IF(ADELTV0.GT.EPS)GOTO 90
      X=3.1416+ATAN2(V0,ALFA0)
      IHEL=X/3.1416*44
      XX1=ALFA0**2+V0**2
      AA0=SUT1(IHEL+1)
      AA1=SUT2(IHEL+1)
      Q0=V0*QR
      H(1)=HR*XX1*(AA0+AA1*X)
      H1X=HUT-HIN+XKONST3*HR*V0**2
      DO 95 I=1,N
      H(I)=H(1)-(H(1)-HUT)/(N-1)*(I-1)
      Q(I)=Q0
   95 CONTINUE
      TID=0
      WRITE(9,*)TID,H(M),Q(M),XNR0
      TID=TID+DELTT
C
C     TRANSIENT
C
  140 CONTINUE
      DO 150 I=1,N
      HCP(I)=H(I)+B*Q(I)-R*Q(I)*ABS(Q(I))
      HCM(I)=H(I)-B*Q(I)+R*Q(I)*ABS(Q(I))
  150 CONTINUE
C
C     INNER POINTS
C
      DO 160 I=2,N-1
      H1(I)=(HCP(I-1)+HCM(I+1))/2
      Q1(I)=(H1(I)-HCM(I+1))/B
  160 CONTINUE
C
C     DOWNSTREAM BOUNDARY CONDITION
C
      H1(N)=HUT
      Q1(N)=(HCP(N-1)-H1(N))/B
C
C     UPSTREAM BOUNDARY CONDITION
C
      IF(Q(1).LT.0.0000001)GOTO 550
      CALL XSOFT(TID,XNRX)
```

```
      ALFA1=XNRX/XNRR
      V=Q(1)/QR
  500 X=3.1416+ATAN2(V,ALFA1)
      IHEL=X/3.1416*44
      AA0=SUT1(IHEL+1)
      AA1=SUT2(IHEL+1)
      XX1=ALFA1**2+V**2
      F1=XKONST2+XX1*(AA0+AA1*X)-HCM(2)/HR-XKONST1*V
      DF1=ALFA1**3*AA1/XX1+2*V*AA0+2*AA1*3.1416*V+2*V*AA1*ATAN2
     &(V,ALFA1)+V**2*AA1*ALFA1/XX1-XKONST1
      DELTV=-F1/DF1
      ADELTV=ABS(DELTV)
      V=V+DELTV
      IF(ADELTV.GT.EPS)GOTO 500
      IF(V.LT.EPS)GOTO 550
      Q1(1)=V*QR
      X=3.1416+ATAN2(V,ALFA1)
      IHEL=X/3.1416*44
      AA0=SUT1(IHEL+1)
      AA1=SUT2(IHEL+1)
      XX1=ALFA1**2+V**2
      H1(1)=HR*XX1*(AA0+AA1*X)
      GOTO 560
  550 CONTINUE
      Q1(1)=0
      H1(1)=HCM(2)
  560 CONTINUE
      WRITE(9,*)TID,H1(M),Q1(M),XNRX
      DO 570 I=1,N
      H(I)=H1(I)
      Q(I)=Q1(I)
  570 CONTINUE
      TID=TID+DELTT
      IF(TID.LT.TIDSLUT)GOTO 140
      END
C     SUBROUTINE XSOFT
C
      SUBROUTINE XSOFT(TIME,XNRTIME)
      COMMON /SOFTSTOP/XNR0,XNR1,XNR2,XNR3,XNR4,T0,T1,T2,T3,T4,
     &A1,A2,A3,A4,B1,B2,B3,B4
      IF(TIME.GT.T0)GOTO 600
      XNRTIME=XNR0
      GOTO 650
  600 CONTINUE
      IF(TIME.GT.T1)GOTO 610
      XNRTIME=A1*TIME+B1
      GOTO 650
  610 CONTINUE
      IF(TIME.GT.T2)GOTO 620
      XNRTIME=A2*TIME+B2
      GOTO 650
```

```
620 CONTINUE
   IF(TIME.GT.T3)GOTO 630
   XNRTIME=A3*TIME+B3
   GOTO 650
630 CONTINUE
   IF(TIME.GT.T4)GOTO 640
   XNRTIME=A4*TIME+B4
   GOTO 650
640 CONTINUE
   XNRTIME=XNR4
650 CONTINUE
   RETURN
   END
```