INVESTIGATION OF WATER HAMMER PROBLEMS AND POTENTIAL SOLUTIONS IN PUMP DISCHARGE LINES

A THESIS SUBMITTED TO
THE GRADUATE SCHOOL OF NATURAL AND APPLIED SCIENCES
OF
MIDDLE EAST TECHNICAL UNIVERSITY

BY

RECEP ÇAĞRI ERDEM

IN PARTIAL FULFILLMENT OF THE REQUIREMENTS
FOR
THE DEGREE OF MASTER OF SCIENCE
IN
CIVIL ENGINEERING

DECEMBER 2019

Approval of the thesis:

INVESTIGATION OF WATER HAMMER PROBLEMS AND POTENTIAL SOLUTIONS IN PUMP DISCHARGE LINES

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ABSTRACT

INVESTIGATION OF WATER HAMMER PROBLEMS AND POTENTIAL SOLUTIONS IN PUMP DISCHARGE LINES

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December 2019,109 Pages

Disturbance in boundary conditions of a hydraulic system could cause rapid change in flow velocity in confined pipe systems. Pressure wave leading to an event called water hammer may occur as a consequence of that disturbance. Water hammer could lead to catastrophic failures on the hydraulic systems. Thus, proper protection measures should be defined and installed in the system before it is put into operation. The aim of this study is to analyze a pumped discharge line and ensure its safe operation against water hammer. For this purpose, the pump discharge line is examined with a transient software called HAMMER, in which unsteady partial differential equations of the pipe flow are solved with a widely used Method of Characteristics (MOC). Pump trip (shut-down) scenarios are tested in the analyses. High pressures that could cause bursting of the pipe and low pressures which lead to cavitation are observed in unprotected version of the hydraulic system. Afterwards, air chambers, flywheels, one-way surge tank and air valve are operated in separate and combined form. Lastly, comparison of the results is done with and without protection devices.

Keywords: Water Hammer, Hydraulic Transient, Air Chamber, Air Valve, Surge Tank

ÖZ

POMPALI BORU HATLARINDA SU DARBESİ PROBLEMİNİN VE POTANSİYEL ÇÖZÜMLERİN ARAŞTIRILMASI

Erdem, Recep Çağrı
Yüksek Lisans, İnşaat Mühendisliği
Tez Danışmanı: Prof. Dr. Zafer Bozkuş

Aralık 2019, 109 Sayfa

Hidrolik sistemlerin sınır koşullarında yaşanabilecek değişimler kapalı boru sistemlerdeki akışkan hızlarında ani değişimlere sebep olabilir. Bu değişimlerin sonucu olarak su darbesi olarak adlandırılan basınç dalgalanmaları oluşabilir. Su darbesi hidrolik sistemlere kalıcı hasar verebilmektedir. Bu sebeple, sistem işletmeye alınmadan önce uygun koruma yöntemleri belirlenmeli ve sisteme monte edilmelidir. Bu çalışmanın amacı, pompalı bir boru hattının su darbesine karşı analizlerini yapmak ve güvenli işletimini sağlamaktır. Bu amaçla söz konusu hat Hammer adı verilen ve dünyada yaygın olarak kullanılan karakteristikler metodu yardımı ile borulardaki doğrusal olmayan, zamana bağlı, kısmi diferansiyel akım denklemlerini çözen bir bilgisayar programı ile incelenmiştir. Analizlerde pompanın ani durması senaryoları test edilmiştir. Sistemin korumasız halinde, boru patlamasına sebep olabilecek yüksek basınçlar ve kavitasyona sebep olabilecek düşük basınçlar gözlemlenmiştir. Daha sonra, sisteme hava kazanı, volan ilavesi, hava vanası ve denge bacası eklenerek ayrı ayrı ve bir arada olacak şekilde tekrar analizler yapılmıştır. Son aşamada ise korumalı ve korumasız durumlarda elde edilen sonuçların kıyaslamaları yapılmıştır.

Anahtar Kelimeler: Su Darbesi, Zamana Bağlı Akım, Hava Kazanı, Hava Vanası, Denge Bacası

To My Wife and To My Dear Family

ACKNOWLEDGEMENTS

I would like to express my gratitude to Prof. Dr. Zafer Bozkuş for his support. It is a real honor working with him. This thesis could not have been completed without his help.

I also would like to thank the jury members of my thesis, Prof.Dr. Mete Köken, Assoc.Prof.Dr. Kerem Taştan, Assist.Prof.Dr. Elif Oğuz and Assist.Prof.Dr. Ali Ersin Dinçer for their contributions in improving the quality of the thesis.

I really appreciate my family. My father Prof. Dr. İlhan Erdem is always a role model for me. I always feel my lovely mother Müzeyyen Erdem's love and support even we have a long distance. My brilliant and beautiful sister Bahar Erdem deserves special thanks. They have a main role on every success of my life. My beautiful wife Sema Bahar has very important effect on this thesis. I always admire her. Without her support, nothing could be accomplished. I also thank her family, Nursel Bahar and Ahmet Bahar for their support.

I want to thank Berat Alp Sarıkavak for sharing information and support.

I want to thank my chief engineer Sevi Bodur for her supports.

Onur Seltuğ and Şemsettin Cura deserves special thanks. I am grateful to have such friends from my childhood.

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CHAPTER 1

INTRODUCTION

1.1. Introduction

Water distribution systems are operated over a wide range of flow demands in modern cities. Thus, pressure change as a result of velocity change occurs even in normal conditions. In most cases, they could be controllable in normal conditions. However, change in boundary conditions such as pump trip, sudden valve closure, power failure and flow variations could cause higher pressure variations. These pressure variations are called water hammer since they sound like hammering in the pipe as they travel back and forth in the confined system. They are unexpected situations and affect hydraulic system behaviors. Thus, designers should consider both expected and unexpected conditions and plan water distribution systems considering all possible conditions.

Pumps have been used in potable water lines for decades. Their motor uses energy to rotate their impellers. Rotating impellers give energy to fluid and pumps could convey water from low head to high head. Most of the transient conditions arise from either pump start up or stoppage. In addition, valve operations associated with pumps could cause transients. Sudden pump failure could cause severe effect to the pump line. Thus, pipes must withstand positive and negative pressures. High pressure changes cause catastrophic failures. Negative pressures cause column separation in pipe systems. As a result of column separation, parts of the system may be exposed to strong vacuum conditions, that is, negative gage pressures leading to permanent damage by collapsing the pipe.

Transient conditions determine system reliability. Therefore, careful selection of protective devices has an important role at design stage. Negative pressures could be prevented by one-way surge tanks and air valves. One-way surge tank has a check valve at the bottom. It is used to separate the surge tank and the pipeline in normal conditions. The aim of the separation is to reduce height of the surge tank wall. When pressure falls below the pipeline profile, water is supplied from the surge tank into the pipeline to prevent negative pressure (Miyashiro, 1967). Air valve is another alternative for negative pressures. Air is ejected when hydraulic grade line elevation falls below pipeline profile. Moreover, flywheels could be inserted into pumps to prevent liquid column separation. Flywheels are used to increase moment of inertia of the pump motor (Yang, 2001). Air chamber is the valid and reliable solution against both positive and negative pressures, especially in pumped discharge lines. Water is transported from the chamber of the air vessel into cavity and positive pressure could be provided and maintained in the system (Stephenson, 2002). Location of the air vessel to be placed in the pipeline system has different effects against water hammer. In most applications, air chambers are placed after the check valve(s) located just downstream of the pump(s) at the upstream section of the pipeline to prevent flow reversal through the pumps. However, unsuitable locations could cause unnecessary size to prevent positive pressure, (Wang et al, 2019).

Models that are created in laboratories are very effective. However, it is difficult to study in large scale systems because they are not economical and practical. With the help of computer-aided engineering, lots of simulations could be tested more easily. Results are compared and, if needed, best protection devices could be selected (Rezaei et al., 2017)

In this study, computer program called HAMMER is used for testing the system reliability.

1.2. Literature Survey

Hydraulic transient studies have been done for many years. Equation of motion and equation of continuity which are unsteady nonlinear partial differential equations are used to describe the time dependent fluid behavior. Method of characteristics which use compatibility relations is widely used to solve these nonlinear equations, Chaudhry (1979), and Shani et al., (2017).

Nonlinear equations are complex equations. Therefore, high speed computers are widely used in transient studies. Streeter (1963) was one of the first researchers that used high speed computers. Parallel pumps with suction lines between two reservoirs were examined in his studies. Different specific speeds, diameters, pipe lengths and moment of inertias were examined in power failure scenario and results were evaluated in IBM 7090 computer. In Figure 1-1, pump station configuration used in the study is shown. H_0 is described as rated head and H_s is described as total dynamic head produced by pump at steady state condition and EGL is the energy grade line.

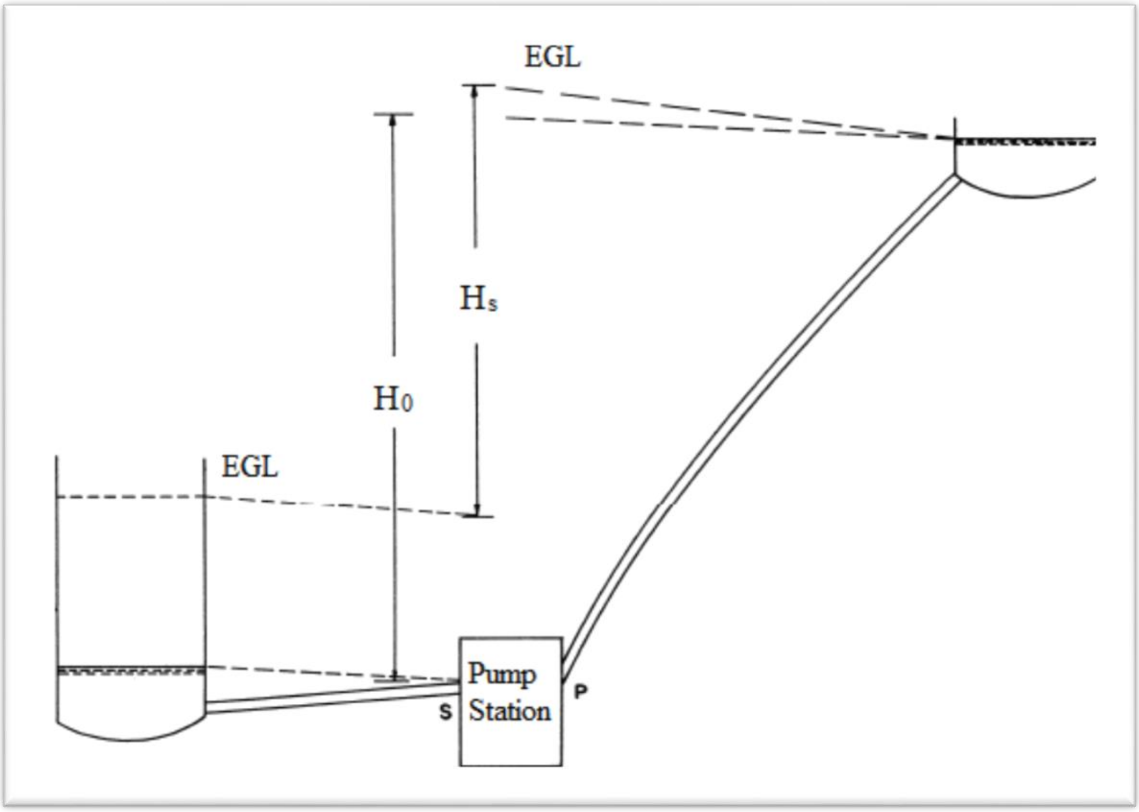


Figure 1-1: Streeter (1963) Pump Station Configuration

Liquid column separation resulted by the power failure was examined with the help of computer by Miyashiro in 1967. Method of characteristic was used to solve momentum and continuity equations. Waterhammer equations, pump related equations, one-way surge tank equations were defined. Boundary conditions such as reservoir, pump and tank are divided into parts. Equations are defined in each part accordingly. One-way surge tanks, which are given in Figure 1-2, were tested in that research. Approximate results were obtained in most calculations.

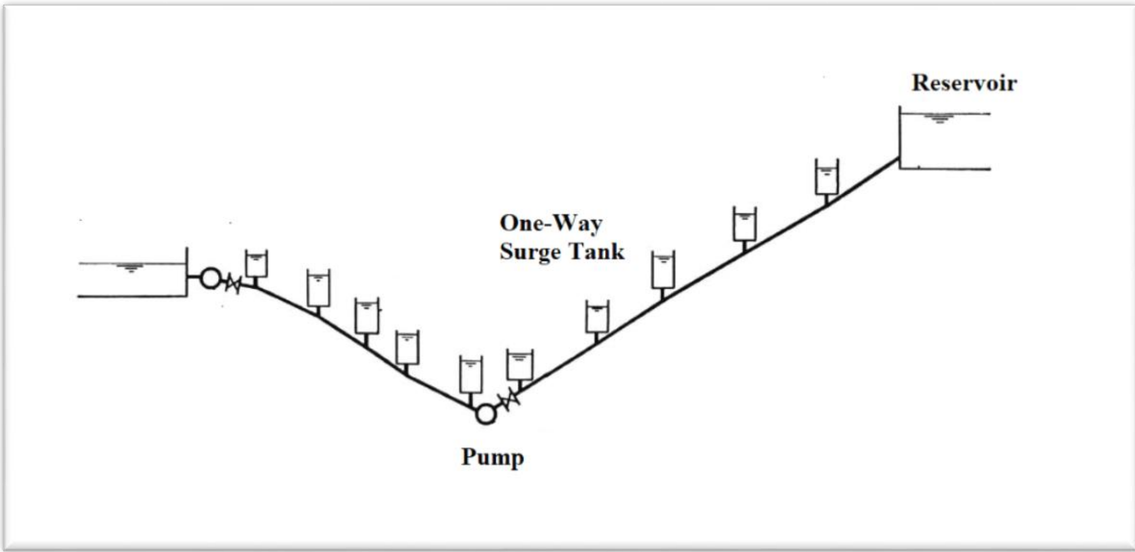


Figure 1-2: Miyashiro (1967) One Way Surge Tanks Model Tested In Computer

Power failure of a pump scenario was modeled and protection devices were suggested by Elansary (2004). Moreover, research was performed by Larsen (2012) in pump sewer mains. The effect of rapid water demand changes was analyzed by Kwon (2007), in which water distribution system in a city was simulated. It was proved that rapid change in water demand and short time valve operations could cause significant pressure difference. Choon et al. (2012) investigated waterhammer effect through the pipeline system. Dinçer (2013) examined waterhammer problem in the pumped-storage power plant's penstock. Moreover, Dursun (2013) studied protection devices against waterhammer problem in the Yeşilvadi Hydropower Plant. In addition, Dursun et al.(2015) investigated pressure relief valve's transient effect to the penstock. It was proved that rapid change in water demand and short time valve operations could cause significant pressure difference. Rezaei et al. (2017) analyzed combination of protection measures in pump discharge lines. In their studies, the effect of single and combined protection devices was compared. Alternatives were tested to understand system behavior during pump stoppage. It was seen that combination alternatives could be more effective against water hammer. Moreover, cost analysis was done and combined applications were found more economical. Flywheel, air chamber and in-line check valves were tested in that research.

Flywheels are used for increasing pump motor moment of inertia. The purpose of this is to reduce maximum angular velocity of impellers. They are chosen against liquid column separation (LCS). Column separation could occur when pressure drops below pipeline profile. Separation occurs between two water columns especially at high points. These liquid columns may collide during transient events causing high positive pressures. LCS studies have been done from the 19th century up to now (Bergant et al., 2005). Yang (2001) examined two cases in his research against LCS. It was seen that increasing moment of inertia gives positive outcome. Furthermore, Elansary (2004) found that flywheels are very successful against negative pressure arising from emergency pump shutdown. Moreover, Kavurmacıoğlu (2009) evaluated the effect of increasing pump inertia. In his study, flywheels were found effective. However, it was noted that increasing inertia is not always practical because inserting big flywheels could be necessary in some situations.

One-way surge tank is another option against negative pressures. Hu et al. (2008) studied possible locations of surge tanks. Optimization of surge tanks was done according to their location and height of the wall. Adding one-way surge tank at high points was suggested in that study. Kavurmacıoğlu and Karadoğan (2003) claimed that using one-way surge tank is very effective and economic in pump lines against transient events. They analyzed single usage of protection device such as air chamber and combination of it with one-way surge tank. It was concluded that combined usage reduces air chamber's volume. Chamani et al. (2012) concluded that size and location does not affect volume of water supplied from surge tanks. Amount of water that is used in the pipeline system is constant to avoid LCS according to their studies. Carmona et al (2019) studied pump discharge lines in Mexico. Effect of air chambers, air valves and one-way surge tanks were examined in that study.

Air valves are mainly used to prevent air existence in the flow. Bianchi et al. (2007) studied to develop equation for optimum air valve dimensions. Their validation was tested in the laboratory. It was stated that mathematical model which was developed in that research could be applicable. Ramezani et al. (2015) researched efficiency of air valves during normal operations. Proper location and size were also investigated in that study. It was seen that usage of air valve has several advantages. To illustrate, combining an air valve with an air chamber could reduce the air chamber volume. They could be effective against negative pressures. However, periodic maintenance is required for air valves in most of standards. Most of air valves are inaccessible due to their locations. Moreover, rapid air flow could create additional high pressures.

Air chamber is the most valid and reliable solution against water hammer, especially in pump discharge lines. Water is transported from the chamber of the air vessel into cavity and positive pressure could be provided and maintained in the system. (Stephenson, 2002). Deciding initial air volume and initial water volume is the most important step. Thorley (2004) developed mathematical expression and charts for determining air chamber volume. Furthermore, Stephenson (2002) also studied for theoretical expression of air chamber volume.

El-Dabaa and Khoris (2018) studied effect of height (h) and diameter (d) ratio of air chambers. It is concluded that increasing air chamber's h/d ratio has a positive impact against water hammer. Wang et al. (2019) studied optimum location of air vessel. Their study was done for long distance pipelines.

1.3. Scope of Study

Pump discharge lines are used in water distribution lines frequently. Disturbances in boundaries of the system, may change the fluid velocity, this, in turn, would cause pressure change in closed pipeline systems. Transient condition such as power failure leads to abrupt velocity and pressure change in hydraulic systems.

To prevent transient conditions, best protection device should be considered. There are alternative solutions which are used in operations. However, deciding best one would require extreme attention in some examples. Lots of unnecessary volume or device could be selected. Moreover, location of them has also importance. In Chapter 2, transient flow concept in pump discharge line is examined. In Chapter 3, brief information about protection devices is given. Software program that is used in transient calculation is explained in Chapter 4. In final chapters, case studies are evaluated.

CHAPTER 2

TRANSIENT FLOW

The main principles of transient flow are examined in this chapter. In the first part, transient flow is defined. In the second part, foundations of the water hammer which are related to basis of the physics or fluid mechanics are described. Continuity equations and momentum equations are developed. Then, nonlinear hyperbolic partial differential equations are converted to differential equations by using the method of characteristics. Equations are determined to obtain algebraic equations that could be solved in an x-t field with the boundary conditions. The theory of the pump transient is explained in the final part for the sake of completeness.

2.1. Definition of Transient Flow

Steady state flow means that flow conditions, which are pressure, velocity or discharge, do not change with time in the pipeline system. If these conditions change over time at a point in the pipeline, the flow is named as unsteady flow. Steady state flow could be defined as a unique case of an unsteady flow. Thus, unsteady flow equations can also be applied to steady state conditions. The transient flow concept is developed to describe unsteady flow condition in the pipeline or pump discharge line. When parameters at a point in the pipeline change with time, transient flow occurs in the hydraulic systems or surroundings.

The transient flow could be classified into two categories. The first one is named as quasi-steady flow. Discharges or pressure change gradually with time in quasi-steady flow. Thus, in the short time interval, the flow parameters show very close values which could be assumed as constant. The second one is named as the true transient flow. Inertia of liquid and/or flexibility of the fluid and pipe are the main parameters that affect the true transient flow development. When inertia of the pipeline has significant effect and pipe and flow elasticity has insignificant effect, the true transient flow is named as rigid-column flow. However, in addition to the inertia effect, the actual transient flow is named as water hammer, given the elasticity effects of the pipe and liquid. (Larock et al, 2000).

2.2. Water Hammer

2.2.1. General

Water hammer which means unsteady flow in the pressurized confined line could be defined as hydraulic shock basically. Sudden speed changes or direction changes lead to abrupt changes in pressure. These abrupt changes cause shock waves traveling back and forth through the whole pipeline. When the shock waves meet a solid obstacle, a hammer sound is heard. This is the reason of the term for water hammer.

Flow in the pipeline could not maintain its steady state form mostly because many factors in a confined line could affect flow conditions. Pump start up or stoppage, water demand changes, changes in a tank level or reservoir level, power failures and many unforeseen events could lead to transient condition in the pipeline system. The causes of water hammer are classified into four events. (Bentley HAMMER, 2016):

• Cavities, that arise from pump start-up, collapse suddenly and cause high pressure in the pipeline.

• Pump stoppage could lead to sudden flow velocity change in the pipeline. The hydraulic grade line drops below pipeline central axis on the discharge side and sub-atmospheric pressures occur in the pipeline. Sub-atmospheric pressures sometimes leading to the vapor pressure of the liquid could cause liquid column separation.

• Valve opening or valve closure could cause shock waves in the pipeline system. The time duration of valve opening or valve closure could cause severe effect to the hydraulic systems. When the closure time is smaller than the time passed during the pressure wave travel between the valve and the reservoir and the return movement back to the valve, it is named as sudden valve closure. Sudden valve closure could lead to sudden velocity change in the pipeline which means abrupt pressure change.

• Usage of improper devices in the pipeline that are selected for protection purposes may cause in more harm than benefit.

A transient event is generated by sudden changes in the flow conditions in the pipeline system. These events cause instability in the steady state flow condition. This imbalance in energy causes the liquid to be trapped, the pipe to elongate and expand. However, water is not easily compressed, and most of the kinetic energy produced by the imbalance caused by transients causes significant compressive forces in the system. The pressure forces rapidly spread over the entire pipeline system and change the flow and pressure characteristics in the system. The propagation of this pressure wave may cause cracks or weaken the pipeline and its supports at the most vulnerable locations.

2.2.2. Derivation of Transient Flow Equations

Transient flow in the confined pipeline is modeled with momentum equations and continuity equations. Appling the momentum equation to the control volume which is a portion of the confined system is done firstly. Then, conservation of mass equation for the liquid in the pipe is developed.

A hydraulic system, consist of a reservoir, a pipeline system and a valve at the end of the pipeline is shown in Figure 2-1 (a).

When sudden valve closure occurs at the downstream section, the nearest layer of the liquid to the valve will be in the rest position. The procedure continues for the successive layers until fluid in the entire pipeline is in the rest position. A shock wave moves towards the upstream section. The application of the momentum equation is performed on the control volume shown in Figure 2-1 (b). The absolute pressure wave velocity moving to the left due to a small change in the valve setting is a-V_0. The magnitude of the head increase at the valve is proportional to flow velocity change in the pipeline. Resultant force in the x axis is expressed in Equation 1. (Wylie et al., 1993).

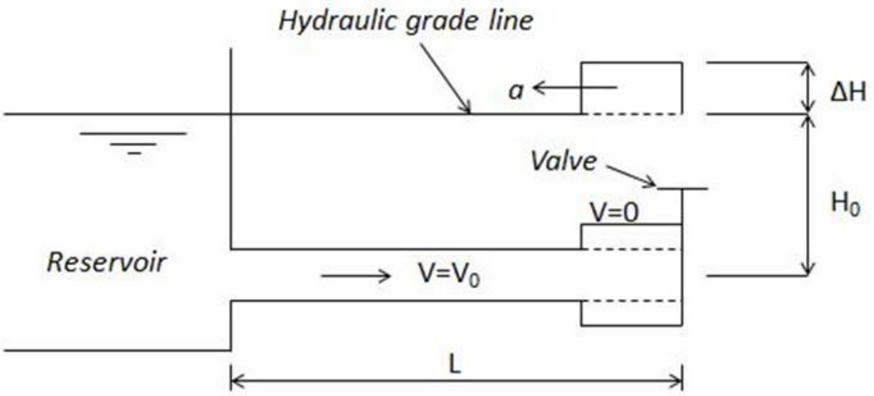


Figure 2-1(a): Head Increase in a Pipeline during Sudden Valve Closure

The momentum equation states

$$-\gamma \Delta HA = \rho A(a - V_0)\Delta V + \rho A(V_0 + \Delta V)^2 - \rho A V_0^{\ 2} \qquad (1)$$

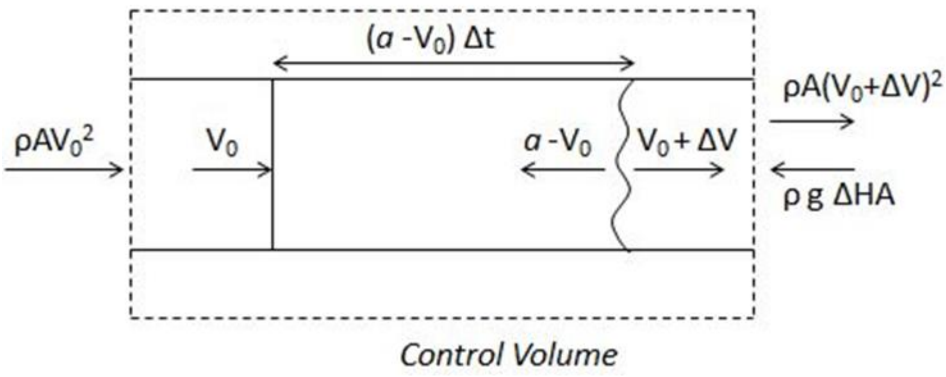


Figure 2-1(b): Momentum Equation in a Pipeline

where,

γ	:	specific weight (N/m^3)
ρ	:	density of fluid (kg/m^3)
g	:	gravitational constant (m/s^2)

A : pipe cross section area (m^2)

V_0 : initial velocity (m/s)

ΔV : change in flow velocity (m/s)

a : acoustic speed (m/s)

ΔH : incremental change in the head (m)

The velocity change is defined as (V_f - V_0). V_f could be defined as the final velocity of the liquid in the pipeline after valve operation. V_0 could be defined as the initial velocity before the valve operation. Equation for the head increase is determined as;

$$\Delta H = -\frac{a\Delta V}{g}\left(1+\frac{V_0}{a}\right) \approx -\frac{a\Delta V}{g} \tag{2}$$

The magnitude of the acoustic speed is enormously high when it is compared with the initial flow velocity, V_0. Thus, the magnitude of V_0/a is quite small when it is compared to 1 for liquids of many pipe types. Velocity is zero when the valve is closed completely.

$$\Delta V = V_f - V_0 \tag{3}$$

Then, $\Delta V = 0 - V_0 = -V_0$, and when it is inserted into Eq.(2) ΔH is described as aV_0/g. When valve closure has incremental pattern Eq.(2) could be determined as;

$$\sum \Delta H = -\frac{a}{g}\sum \Delta V \tag{4}$$

and it is valid for any movement of the valve until the pressure wave reaches the upstream end of the pipeline and returns as a reflected wave to the valve. In other words, this equation applies as long as valve operation duration satisfies $t < 2L/a$, where L is the length of the pipe.

Wave speed expression can be obtained by applying the continuity equation for the same pipeline under the same conditions. As the rapid valve closure causes pressure head increase in the pipeline, referring to Fig. 2-2, pipe is stretched with the length ΔS. The magnitude of the stretch depends on a pipe support. It is assumed that the stretch of the pipe takes place in *L/a* seconds or the velocity is $\Delta Sa / L - V_0$. Thus, ΔV is equal to $(\Delta S\ a/L) - V_0$. During the time elapsed, L/a, the mass of liquid entering the pipe is $\rho A V_0 L/a$. This mass is stored in the pipe as the cross-sectional area A increases in the expanding pipe. Furthermore, the compression of the liquid causes a higher density of the liquid mass. Eq.5 is obtained from the continuity principle;

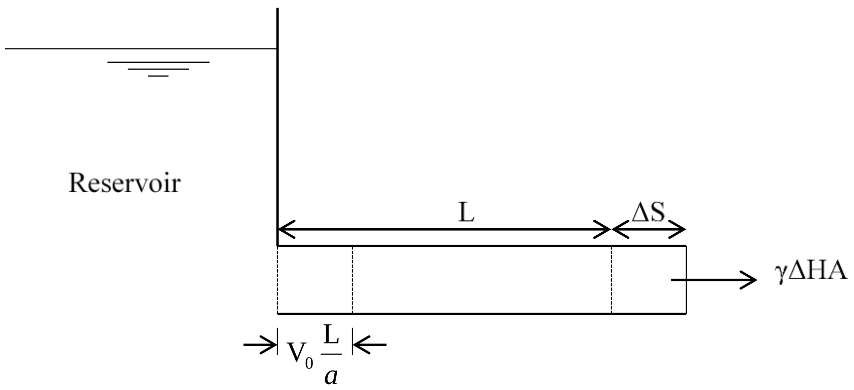


Figure 2-2: Continuity Relations in Pipeline

$$\rho A V_0 \frac{L}{a} = \rho L \Delta A + \rho A \Delta S + L A \Delta \rho \tag{5}$$

Eq.5 is simplified with $\Delta V = \Delta S a / L - V_0$ and the following equation is obtained;

$$-\frac{\Delta V}{a} = \frac{\Delta A}{A} + \frac{\Delta \rho}{\rho} \tag{6}$$

To remove ΔV, Eq. 4 may be inserted and if the wave speed is left alone on the left,

$$a^2 = \frac{g\Delta H}{\frac{\Delta A}{A} + \frac{\Delta \rho}{\rho}} \tag{7}$$

If the extension of the pipe is prevented by the pipe supports, ΔS = 0. At this point we may bring the bulk modulus of elasticity, defined by;

$$K = \frac{\Delta p}{\frac{\Delta \rho}{\rho}} = -\frac{\Delta p}{\frac{\Delta \forall}{\forall}} \tag{8}$$

where $\Delta\forall/\forall$ could be defined as relative change in the original volume. Then,

$$a^2 = \frac{\frac{K}{\rho}}{1 + \frac{K}{A}\frac{\Delta A}{\Delta p}} \tag{9}$$

The thick pipe wall could prevent any change in the cross-sectional area of the pipe. Thus, the pressure increase mostly causes to compression of the fluid in the pipeline and this compression could cause the density increase. $\Delta A/\Delta p$ becomes quite small and acoustic speed becomes $a \approx \sqrt{K/\rho}$

On the other hand, for highly flexible pipes, increase in pressure caused by the transients is often in agreement with the increased cross-sectional area of the pipe. Thus, 1 is small and insignificant compared to other terms in the denominator.
Then, acoustic wave velocity for highly flexible pipes

$$a \approx \sqrt{\frac{A\Delta p}{\rho\Delta A}} \tag{10}$$

Finally, acoustic wave speed for thin walled pipes is

$$a = \frac{\sqrt{\frac{K}{\rho}}}{\sqrt{1 + \left[\left(\frac{K}{E}\right)\left(\frac{D}{e}\right)\right]C_1}} \tag{11}$$

where C_1 is a constant that shows the effect of pipe constraint conditions.

When the pipe is anchored at its upstream end; $C_1 = 1 - \mu/2$, when the pipe is anchored throughout against axial movement; $C_1 = 1 - \mu^2$, when the pipe is anchored with expansion joints throughout; $C_1 = 1$, in which μ is Poisson's ratio.

2.2.3. Derivation of Continuity and Momentum Equations and their Solution

The aim of the water hammer analysis is to determine velocity-pressure pair or discharge-piezometric head pair in the pipeline at any time during the transient event. Conservation of mass equations and continuity equations are applied to determine these parameters. The law of conservation of mass is used to described the continuity equation. According to Newton's second law of motion, the time rate of the momentum change of the system is equal to the sum of the forces exerted on the system by its environment. The parameters for continuity and momentum equations are shown in the Figure 2.3. The liquid is assumed as slightly compressible and the pipe wall is elastic, and the flow is one dimensional in the control volume of Figure 2.3.

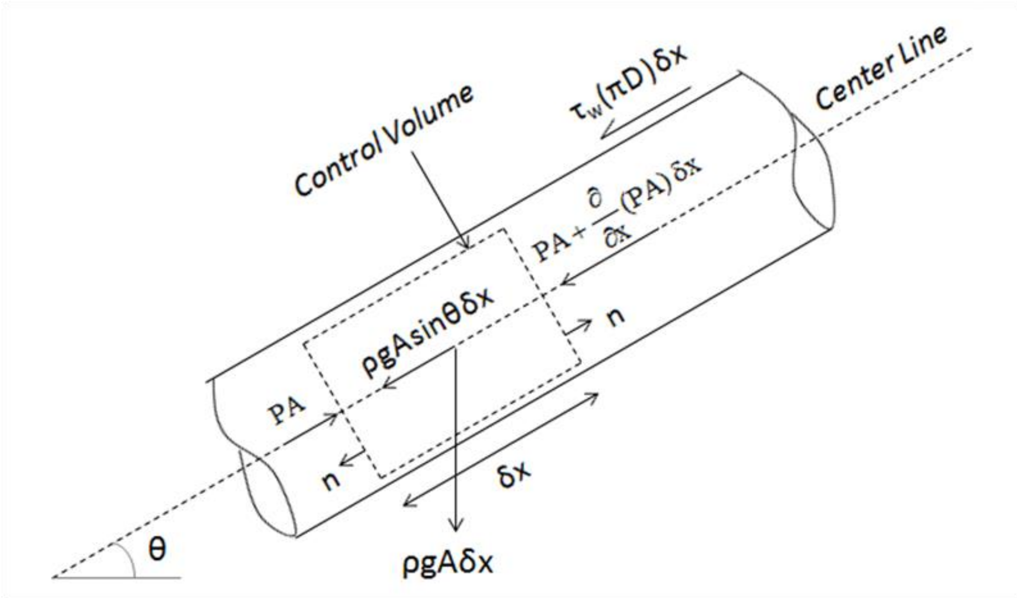


Figure 2-3: Continuity Relations in Pipeline

Continuity and momentum equations are described respectively as;

$$\frac{\partial P}{\partial t} + V\frac{\partial P}{\partial x} + \rho a^2 \frac{\partial V}{\partial x} = 0 \tag{12}$$

$$\frac{\partial V}{\partial t} + V\frac{\partial V}{\partial x} + \frac{1}{\rho}\frac{\partial P}{\partial x} + g\sin\theta + \frac{4\tau_w}{\rho D} = 0 \tag{13}$$

where,

ρ	:	fluid density (m^3/s)
P	:	Pressure (N/m^2)
V	:	flow velocity (m/s)
a	:	wave speed (m/s)
τ_w	:	wall shear stress (N/m^2)
D	:	pipe diameter (m)
g	:	gravitational acceleration constant (m/s^2)

These non-linear partial differential equations could not be solved in closed form. Thus, a numerical solution method of choice should be applied to for their solution. Among the potential methods, the methods of characteristics (MOC) is selected. For one-dimensional fluid transient problems, the characteristics method is much better than other methods in many respects, such as accurate simulation of orthogonal wave fronts, display of wave propagation, ease of programming and efficiency of calculations. (Chaudhry, 1979). The software Bentley HAMMER used in this thesis to analyze the transient event in the pump discharge line is also based on the Method of Characteristics.

The continuity and momentum equations defined in Eq.(12) and Eq.(13) will be solved by using the MOC. The dependent variables are pressure and velocity and independent variables are time and space in these equations. (Wylie et al., 1993).

Momentum equation is labeled by L_2 and continuity equation by L_1 to simplify the procedure. In addition, $\frac{4\tau_\omega}{\rho D}$ $+g\sin\theta$ is described as F. Multiplier is defined as λ ,

$$L_1 + \lambda L_2 = 0 \tag{14}$$

$$\frac{\partial P}{\partial t} + V\frac{\partial P}{\partial x} + \rho a^2 \frac{\partial V}{\partial x} + \lambda\left(\frac{\partial V}{\partial t} + V\frac{\partial V}{\partial x} + \frac{1}{\rho}\frac{\partial P}{\partial x} + F\right) = 0 \tag{15}$$

Eq.(15) could be re-arranged as;

$$\left[\frac{\partial P}{\partial t} + \left(V + \frac{\lambda}{\rho}\right)\frac{\partial P}{\partial x}\right] + \lambda\left[\frac{\partial V}{\partial t} + \left(V + \frac{\rho a^2}{\lambda}\right)\frac{\partial V}{\partial x}\right] + \lambda F = 0 \tag{16}$$

From calculus we know that:

$$\frac{d\theta}{dt} = \frac{\partial\theta}{\partial t} + \frac{\partial\theta}{\partial x}\frac{dx}{dt} \tag{17}$$

First term in Eq.(16) is $\frac{dP}{dt}$ if $V+\frac{\lambda}{\rho}=\frac{dx}{dt}$, and similarly the second term is equal to $\frac{dV}{dt}$ if $V+\frac{\rho a^2}{\lambda}=\frac{dx}{dt}$. Consequently, Eq.(14) becomes

$$\frac{dP}{dt}+\lambda\frac{dV}{dt}+\lambda F=0 \tag{18}$$

With the condition that

$$\frac{dx}{dt}=\left(V+\frac{\lambda}{\rho}\right)=\left(V+\frac{\rho a^2}{\lambda}\right) \tag{19}$$

λ is found from Eq.(19);

$$\lambda=\pm\rho a \tag{20}$$

λ is inserted into Eq.(19);

$$\frac{dx}{dt}=V\pm a \tag{21}$$

The velocity of the pressure wave is quite high when it is compared with the flow velocity. Thus, V term that is described in Eq.(21) could be omitted. λ value is obtained in Eq.(20) and it is inserted into Eq.(18), C^+ and C^- equations are obtained for both plus and minus signs;

$$C^+\left\{\begin{array}{ll}\frac{1}{\rho}\frac{dP}{dt}+a\frac{dV}{dt}+aF=0 & (22)\\[2ex] \frac{dx}{dt}=+a & (23)\end{array}\right.$$

$$C^- \begin{cases} \dfrac{1}{\rho}\dfrac{dP}{dt} - a\dfrac{dV}{dt} - aF = 0 & (24) \\[2em] \dfrac{dx}{dt} = -a & (25) \end{cases}$$

Thus, two partial differential equations are transformed into four ordinary differential equations using two real λ values. The acoustic wave speed depends on, the properties of the pipe and the fluid. Thus, the pipe or fluid properties remain constant until pipe and fluid properties change. Consequently, the characteristic equations given in Eq.(23) and Eq. (25) draws straight lines with "+*L/ a*" and "-*L/ a*"slope in the xt plane, which is an independent variable plane (Figure 2-4). These lines are named as "characteristic lines" and Eq.(22) and Eq.(24) are applicable to the appropriate characteristic line.

With reference to Figure 2-4, a pipe N is divided into equal parts. On the x axis of xt plane, it is seen that each length of extension is Δx. And the time interval is Δt in the y axis. According to the Courant condition, the time step size Δt must be less than or equal to $\Delta x/a$. At each time step, characteristic equations for N + 1 nodes must be solved. Eq.(23) is described in the line between A and P and the Eq.(25) is described between P and B. It is assumed that the dependent variables V and H are known at point A. Then the compatibility equation at point P can be written in terms of dependent variables by integrating the equation. Then, compatibility equation at point *P* could be written in terms of dependent variables by integrating Eq.(22) which is valid on the C^+ line, between the limits *A* and *P*. In the same manner, Eq.(24) is valid on the C^- line, and by integration compatibility equation along the *BP* characteristic line a second equation in terms of the same two unknowns at point *P* is gathered. Simultaneous solution of these equations gives the unknowns at point *P* at the particular time.

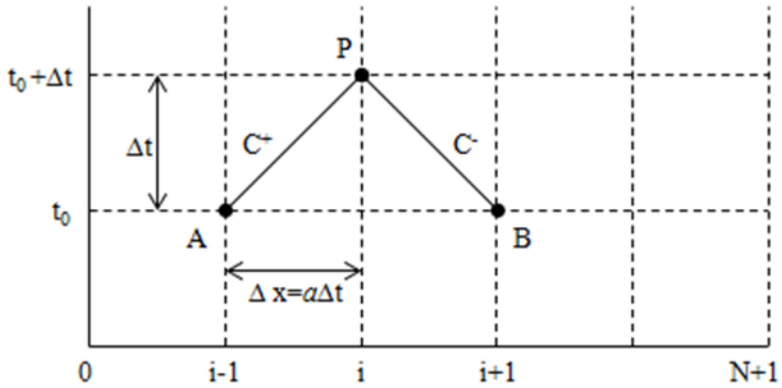


Figure 2-4: Characteristic Lines

In order to facilitate the integration of the compatibility equations, the shear stress defined by Darcy-Weisbach can be applied in the transient flow.

$$\tau_w = \frac{\rho f V|V|}{8} \tag{26}$$

Therefore,

$$F = g\sin\theta + f\,\frac{V|V|}{2D} \tag{27}$$

Then, by multiplying C^+ compatibility equation by $a\frac{dt}{g} = \frac{dx}{g}$, and by introducing the pipeline area to write the equation in terms of discharge , the equation may be placed in a form suitable for integration along the C^+ characteristic line.

$$\int_{H_A}^{H_P} dH + \frac{a}{gA}\int_{Q_A}^{Q_P} dQ + \frac{f}{2gDA^2}\int_{X_A}^{X_P} Q|Q|dx = 0 \tag{28}$$

A similar procedure can be applied to obtain the following equations;

$$C^+ : H_P = H_A - B(Q_P - Q_A) - RQ_A|Q_A| \tag{29}$$

$$C^- : H_P = H_B + B(Q_P - Q_B) + RQ_B|Q_B| \tag{30}$$

where $B = \dfrac{a}{gA}$ and $R = \dfrac{f\Delta x}{2gDA^2}$

Generally,

$$C^+ : H_{Pi} = C_P - BQ_{Pi} \tag{31}$$

in which;

$$C_P : H_{i-1} + BQ_{i-1} - RQ_{i-1}|Q_{i-1}| \tag{32}$$

$$C^- : H_{Pi} = C_M + BQ_{Pi} \tag{33}$$

in which;

$$C_M : H_{i+1} - BQ_{i+1} + RQ_{i+1}|Q_{i+1}| \tag{34}$$

2.3. Transients caused by Pumps

Pump operations could cause devastating consequences to the pump discharge line, especially during a pump start-up or a sudden pump stoppage. The total inertia of the pump rotating parts are generally smaller than the liquid inertia in the pump discharge line. This leads to decrease in pump speed after a power failure. The flow provided by the pump is reduced. Positive and negative pressures occur in the pipeline. Liquid column separation could be seen when hydraulic grade line drops below the pipeline profile. High pressures are seen when the columns are rejoined. Thus, careful examination should be done for the pump discharge line in the design stage. The worst case scenario should be taken into account and necessary protection device should be selected if needed.

2.3.1. Sequence of Events during Pump Power Failure

When the power supply to the pump motor suddenly stops, the only energy left to move the pump in the forward direction is the kinetic energy of the motor and the rotating elements of the pump and the entrained water in the pump. Since this energy is generally small compared to what is required to flow towards the discharge head, the reduction in pump speed is quite rapid.

The lower the pump speed, the lower the water flow in the discharge line adjacent to the pump.

As a result of these rapid flow changes, sub-normal water pressures are generated in the discharge line of the pump. These sub-normal pressure waves move rapidly from the discharge line to the discharge outlet where a wave reflection occurs. After a short time, the speed of the pump is reduced to a point where no water will flow into the existing head. If the pump does not have a control valve, the flow in the pump is reversed, although the pump can still rotate in the forward direction. After a short time, the pump acting as a turbine achieves the runaway speed.

Three effects must be considered to determine the transient hydraulic conditions and the discharge line in the pump after a power failure in the pump motor; that is, pump and motor inertia, pump characteristics and water wave phenomena in the discharge line.

The effect of pump and motor inertia is derived from the inertia equation. This equation describes the relationship between pump speed and torque at a given time in terms of kinetic energy of the rotating system.

The pump characteristics are derived from a complete pump characteristic diagram. This diagram describes how pump torque and speed change to the head and how it is discharged throughout the operating range as a pump, energy distributor and turbine.

Waterhammer effects are obtained from the waterhammer equations. These equations explain the relationships between flow and pressure during transient flow conditions under the influence of waterhammer waves. HGL in the pump discharge line is given in the Figure 2-5. Entrance head loss coefficient is defined as K_e and valve head loss coefficient is defined as K_v. Moreover, pump curves with different valve closure situations are shown in Figure 2-6.

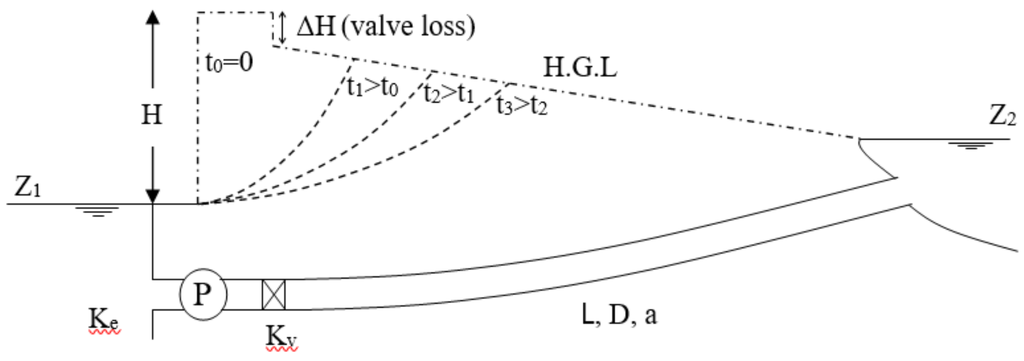


Figure 2-5: HGL in Pump Discharge Line

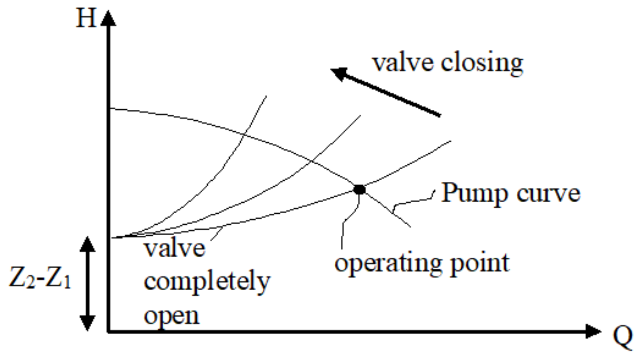


Figure 2-6: Pump Curves for Closing Valve Situations

System demand equation is obtained by applying the energy equation,Eq.(35).

$$H = (Z_2 - Z_1) + \left(K_e + K_v + f\frac{L}{D} + 1 \right)\frac{V^2}{2g} \tag{35}$$

$$H = (Z_2 - Z_1) + bQ^2 \tag{36}$$

where

$$b = \left(K_e + K_v + f\frac{L}{D} + 1 \right)\frac{1}{2gA^2} \tag{37}$$

Pump

efficiency

$$\eta = \frac{\gamma Q H}{T w} = \frac{\gamma Q H}{P} \tag{38}$$

Eq.(38) is defined for pump.

2.3.2. Dimensionless Pump Characteristics

Pump functional relation could be defined as f (P, D, w, Q, gH, μ, ρ) = 0. Buckingham-π theorem is used to obtain following parameters;

$$\pi_1 = \frac{\rho w D^2}{\mu} = \mathrm{Re} \tag{39}$$

$$\pi_2 = \frac{Q}{wD^3} = C_Q \tag{40}$$

$$\pi_3 = \frac{gH}{w^2 D^2} = C_H \tag{41}$$

$$\pi_4 = \frac{P}{\rho w^3 D^5} = \frac{T}{\rho w^2 D^5} = C_p \text{ or } C_T \tag{42}$$

where;

T is the shaft torque

P is power,

w is the angular velocity of the shaft.

D = diameter of machine

μ = viscosity

gH = energy added per unit mass

π_1= Reynolds Number

π_2= Flow Coefficient

π_3= Head Coefficient

π_4= Torque Coefficient

Head coefficient vs flow coefficient graph is given in Figure 2-7. Addition of efficiency and torque coefficients to Figure 2-7 is shown in Figure 2-8.

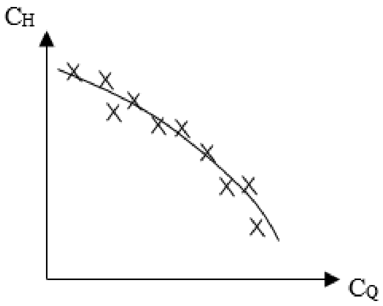


Figure 2-7: Head vs Flow Coefficients

Note that pump efficiency can be expressed also as follows:

$$\frac{C_Q C_H}{C_T} = \frac{\rho g Q H}{T w} = \eta \tag{43}$$

2.3.2.1. Definition of Dimensionless Pump Variables

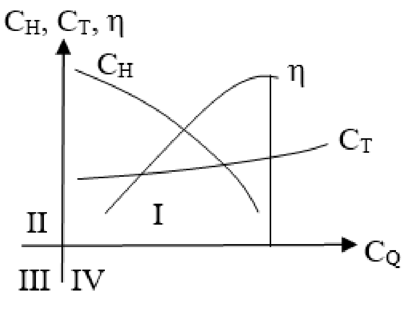


Figure 2-8: Head, Efficiency and Torque vs Flow Coefficients

$$Re = \frac{\rho w D^2}{\mu}$$

$$C_Q = \frac{Q}{wD^3}, \qquad C_H = \frac{gH}{w^2 D^2}$$

$$C_T = \frac{T}{\rho w^2 D^5} \quad \eta = \frac{C_Q C_H}{C_T}$$

Flow: $v = \dfrac{Q}{Q_R}$, **Speed:** $\alpha = \dfrac{w}{w_R}$ **or** $= \dfrac{N}{N_R}$

Head: $h = \dfrac{H}{H_R}$**,** **Torque:** $\beta = \dfrac{T}{T_R}$

R refers to "rated conditions" (Best Efficiency Point)

Typically $\eta_R = \eta_{max} = .80 \Rightarrow Q_R, H_R$ from curves

Then,

$$C_Q = \left(\frac{Q_R}{w_R D^3}\right)\frac{v}{\alpha} = C_{QR}\,\frac{v}{\alpha} \tag{44}$$

$$C_H = \left(\frac{gH_R}{w_R^2 D^2}\right)\frac{h}{\alpha^2} = C_{HR}\,\frac{h}{\alpha^2} \tag{45}$$

$$C_T = \left(\frac{T_R}{\rho w_R^2 D^5}\right)\frac{\beta}{\alpha^2} = C_{TR}\,\frac{\beta}{\alpha^2} \tag{46}$$

One should plot the following functions for a given pump.

$$\frac{h}{\alpha^2} = f_n\left(\frac{v}{\alpha}\right) \quad \text{vs.} \quad \frac{\beta}{\alpha^2} = f_n\left(\frac{v}{\alpha}\right)$$

This function is difficult to plot. A formulation given below is used, instead.

$$\frac{h}{\alpha^2 + v^2} = WH(x), \qquad \frac{\beta}{\alpha^2 + v^2} = WB(x) \tag{47}$$

This form is preferred for digital computers in solving transient problems.

$$x = \pi + \tan^{-1}\left(\frac{v}{\alpha}\right) \tag{48}$$

Four zones of pump operation can be defined, which are given in Table 2-1 and shown in Figure 2-9, using the parameters developed above,

Table 2-1: Pump Operation Zones

Zone	Comment	h	v	α	β
I	Normal pumping	>0 or <0	>0	>0	>0 or <0
II	Turbine pumping, or Reversal speed dissipation	>0 or <0	>0	<0	<0
III	Turbine	>0 or <0	<0	<0	>0 or <0
IV	Energy dissipation	>0	<0	>0	>0

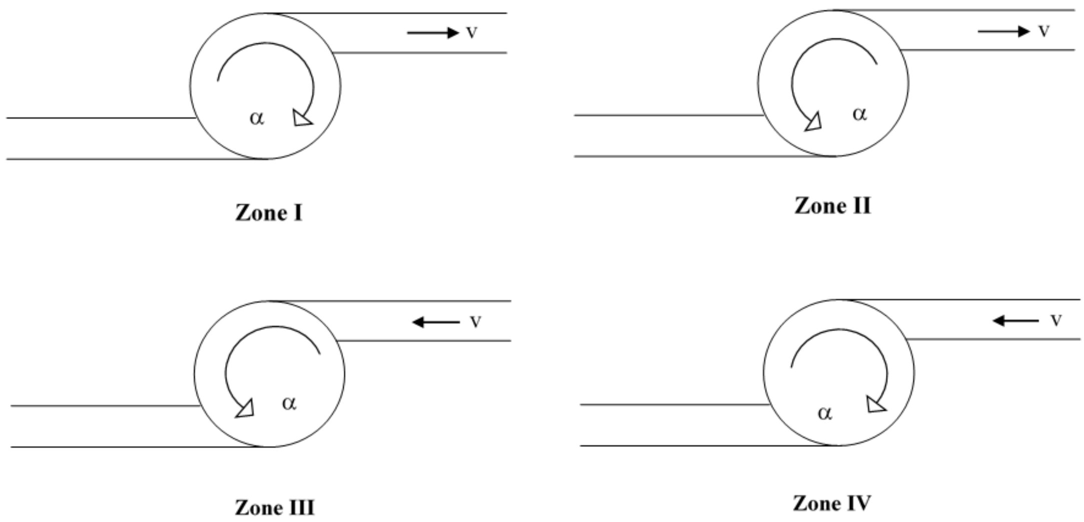


Figure 2-9: Pump operation zones showing the directions of pump impeller and flow velocity parameters

2.3.3. Head Balance Equation and Calculation of Speed Change

For a power failure, two equations are solved simultaneously for each time interval with help of the Characteristic Method. The equations are the head equilibrium equation over the pump and the relief valve and the torque-angular deceleration equation for rotating masses.

2.3.3.1. Derivation of Head-Balance Equation

Pump boundary condition according to C^- is shown in Figure 2-10.

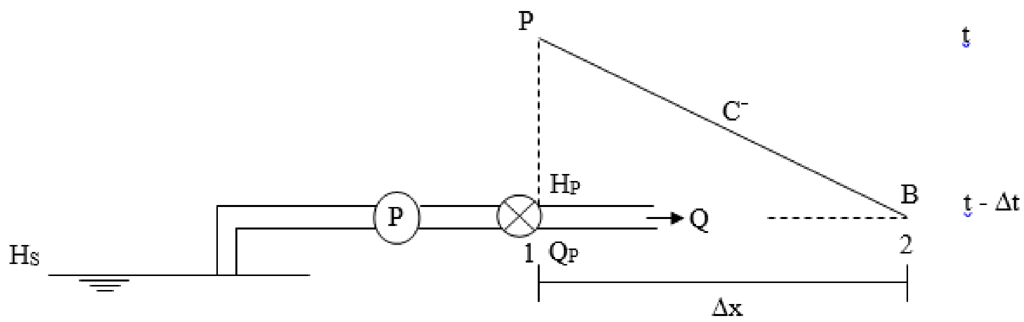


Figure 2-10: Pump Boundary Condition

where;

H_p= head at station 1 in pipe

H_s=suction head

H= dynamic pump head

ΔH= head drop across valve

Q_p= Instantaneous discharge in pump at time t

$$H_S + H - \Delta H = H_P \quad (49)$$

At time t;

$$H_P = C_M + BQ_P \quad (50)$$

$$\Delta H = \frac{\Delta H_0}{\tau^2}\left(\frac{Q_P}{Q_0}\right)^2 \quad (51)$$

in which ΔH_o = head loss at the valve initially, Q_o = initial discharge ($\neq Q_R$)

The following has been previously defined $H = H_R h$, $Q_P = Q_R v$

$$\Delta H = \frac{\Delta H_0}{Q_0^2}\frac{Q_R^2}{\tau^2}v^2 = \frac{K_R}{\tau^2}v|v| \quad (52)$$

To account for both directions.

$$H_s + H - \Delta H = H_p \quad (53)$$

$$H_S + H_R h - \frac{K_R}{\tau^2}v|v| = C_M + (B\,Q_R)v \quad (54)$$

$\tau = f_n(t)$ and h, v are unknown parameters.

2.3.3.2. Calculation of Speed Change

$$T_s - T = I\frac{dw}{dt} \tag{55}$$

where;

Ts= driving torque on the shaft

T=resisting fluid torque

$$I = \frac{WR^2}{g} \tag{56}$$

where;

I = rotational moment of inertia of all rotating parts (pump, motor, shaft, and fluid in impeller

W = weight of rotating parts + entrained fluid

R = radius of gyration of rotating mass

w = angular velocity in rad / sec.

For pump trip, i.e. loss of power $T_S = 0$;

$$-T = \frac{WR^2}{g}\frac{dw}{dt} \tag{57}$$

let $w = \frac{2\pi}{60}N$ where N=speed in rev/min.

$$w = \frac{2\pi}{60}N_R\alpha \tag{58}$$

$$\beta = \frac{T}{T_R} \tag{59}$$

substituting them gives;

$$-\beta = \left[\frac{WR^2}{g}\frac{N_R}{T_R}\frac{\pi}{30}\right]\frac{d\alpha}{dt} \tag{60}$$

Assume $\frac{d\alpha}{dt} \cong \frac{\alpha-\alpha_0}{\Delta t}$ and $\beta \cong \frac{1}{2}(\beta + \beta_0)$, β_0 and α_0 refer to conditions at time t- Δt

$$\frac{\beta+\beta_0}{2} \cong \left[\frac{WR^2}{g}\frac{N_R}{T_R}\frac{\pi}{30}\right]\frac{(\alpha_0-\alpha)}{\Delta t} \tag{61}$$

Eq. (61) can be expressed as:

$$\beta + \beta_0 = C_{31}(a_0 - a) \tag{62}$$

in which;

$$C_{31} = \frac{WR^2}{g}\frac{N_R}{T_R}\frac{\pi}{15\Delta t} \tag{63}$$

$$H_S + H_R h - \frac{K_R}{\tau^2}\mathrm{v}|\mathrm{v}| = C_M + (B\,Q_R)\mathrm{v} \tag{64}$$

In which,

$$h = (\alpha^2 + \mathrm{v}^2)WH(x) \tag{65}$$

Eq.(65) becomes

$$H_R(\alpha^2 + v^2)WH(x) - \frac{K_R}{\tau^2}v|v| - (BQ_R)v + (H_S - C_M) = 0 \tag{66}$$

or FH(α,v)=0 which is the Head-balance equation in terms of v and α. Eq.(62) becomes with

$$\beta = (\alpha^2 + v^2)WB(x) \tag{67}$$

$$(\alpha^2 + v^2)WB(x) + \beta_0 = C_{31}(\alpha_0 - \alpha) \tag{68}$$

or FT(α,v)=0

which is the speed-change equation in v and α

Newton-Raphson method can be used to solve these equations. Figure 2-11 is an example for the use of WH(x), WB(x) functions in a pump shutdown simulation, (Lima and Junior, 2017).

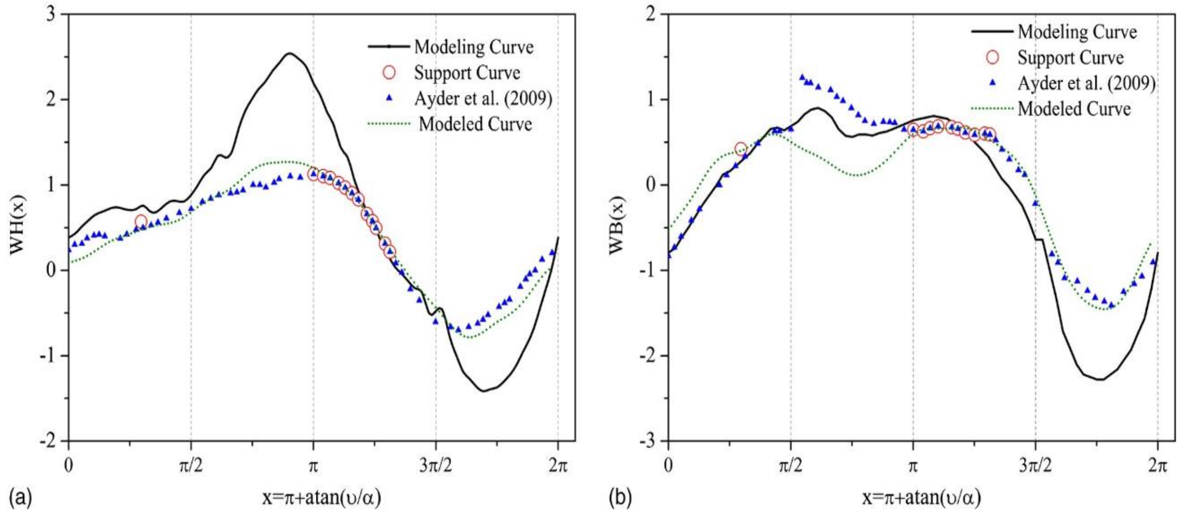


Figure 2-11: Lima and Junior (2017) Complete Curves For a Radial Flow Pump with a Pump Shutdown Simulation

CHAPTER 3

PROTECTION DEVICES

3.1. Air Chambers (Air Vessels)

Air chambers are commonly used in pump discharge lines. There is an air at the upper part and liquid at the remaining part of the air chamber. Orifice is used between vessel and pipeline. The aim of the orifice is to limit inflow and outflow. Outflow should be provided from chamber to pipeline to suppress negative pressures. Furthermore, inflow should be limited to decrease air chamber size. Therefore, orifice size should be selected considering these limitations. Figure 3.1 shows some air chamber (air vessel) types. To restrict inflow, differential type orifice may be selected. More head loss is created for inflow with help of its shape. Optimum inflow/outflow head loss ratio is given in the Table 3-1. Air vessels have advantages over other alternatives such as surge tanks. They require less volume than surge tanks. Excessive wall height may be required for surge tanks in some cases. Moreover, air chambers could be used in cold climates. Additional heaters may be provided to solve freezing problems. However, there are some disadvantages of the air chamber. For instance, additional equipment may be provided such as a compressor with a generator. Furthermore, regular maintenance is required.

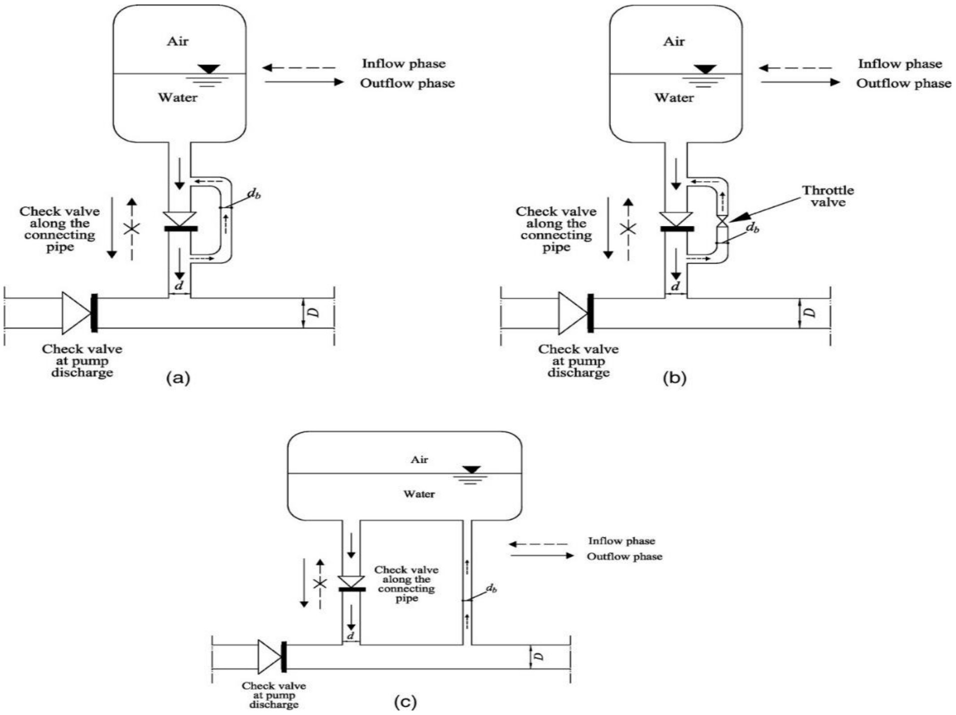


Figure 3-1: Different Air Chamber Connection Types, (a) Vertical cylindrical air chamber with bypass line, (b) Vertical cylindrical air chamber with throttle valve in bypass line, (c) Horizontal cylindrical air chamber with separate inlet-outlet pipe.

Table 3-1: Orifice Head Loss Ratio For Inflow and Outflow (Thorley, 2004)

Type	Inflow/Outflow
Head Loss Ratio in Chamber	2.5

Usage of chambers against pump failure were studied by various researchers. Charts were developed to determine air chamber size. In addition, Graze and Horlacher (1989) formed design charts which are used in this thesis. Their charts which are given in Figure 3-2 and Figure 3-3 are developed in non-dimensional form. (Thorley, 2004).

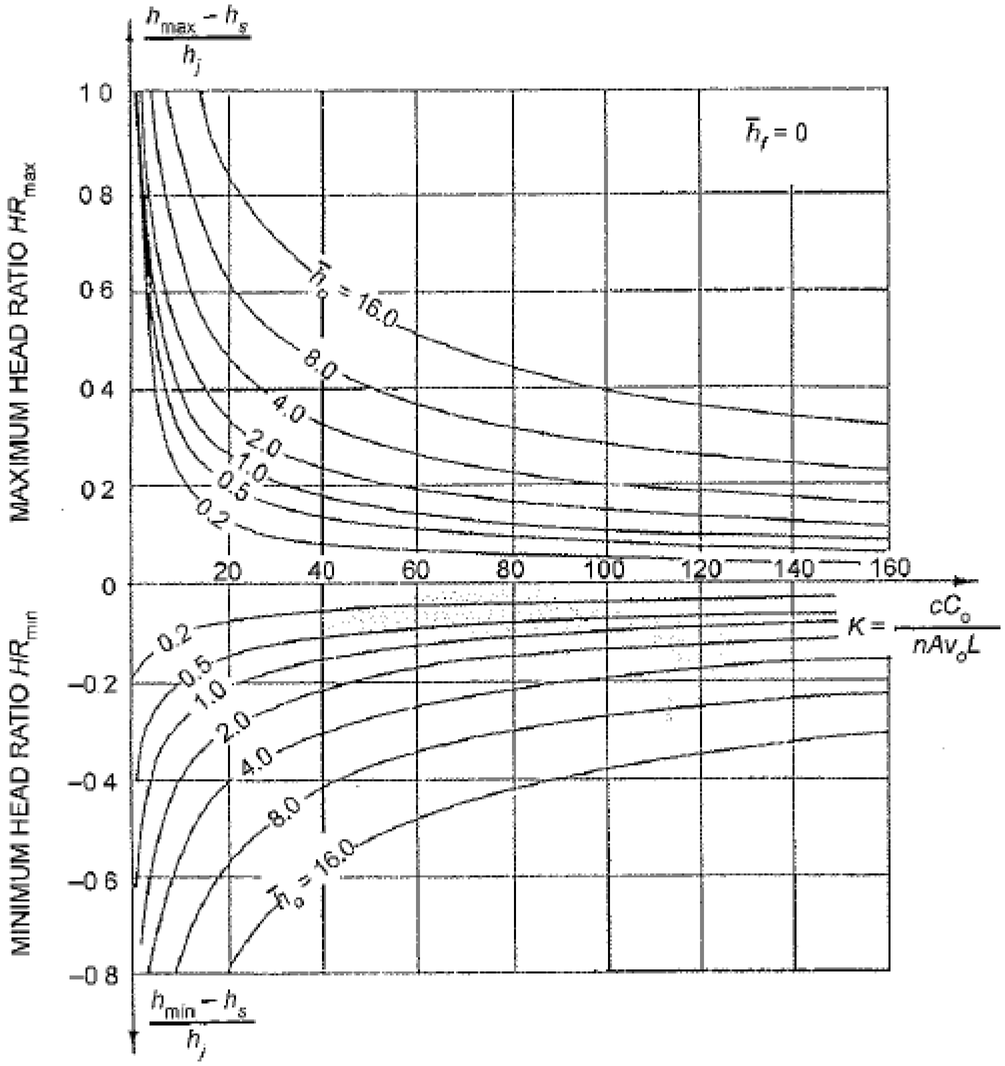


Figure 3-2: Maximum Head Ratio and Minimum Head Ratio w/ K value; $\bar{h}_f$=0 (Thorley, 2004)

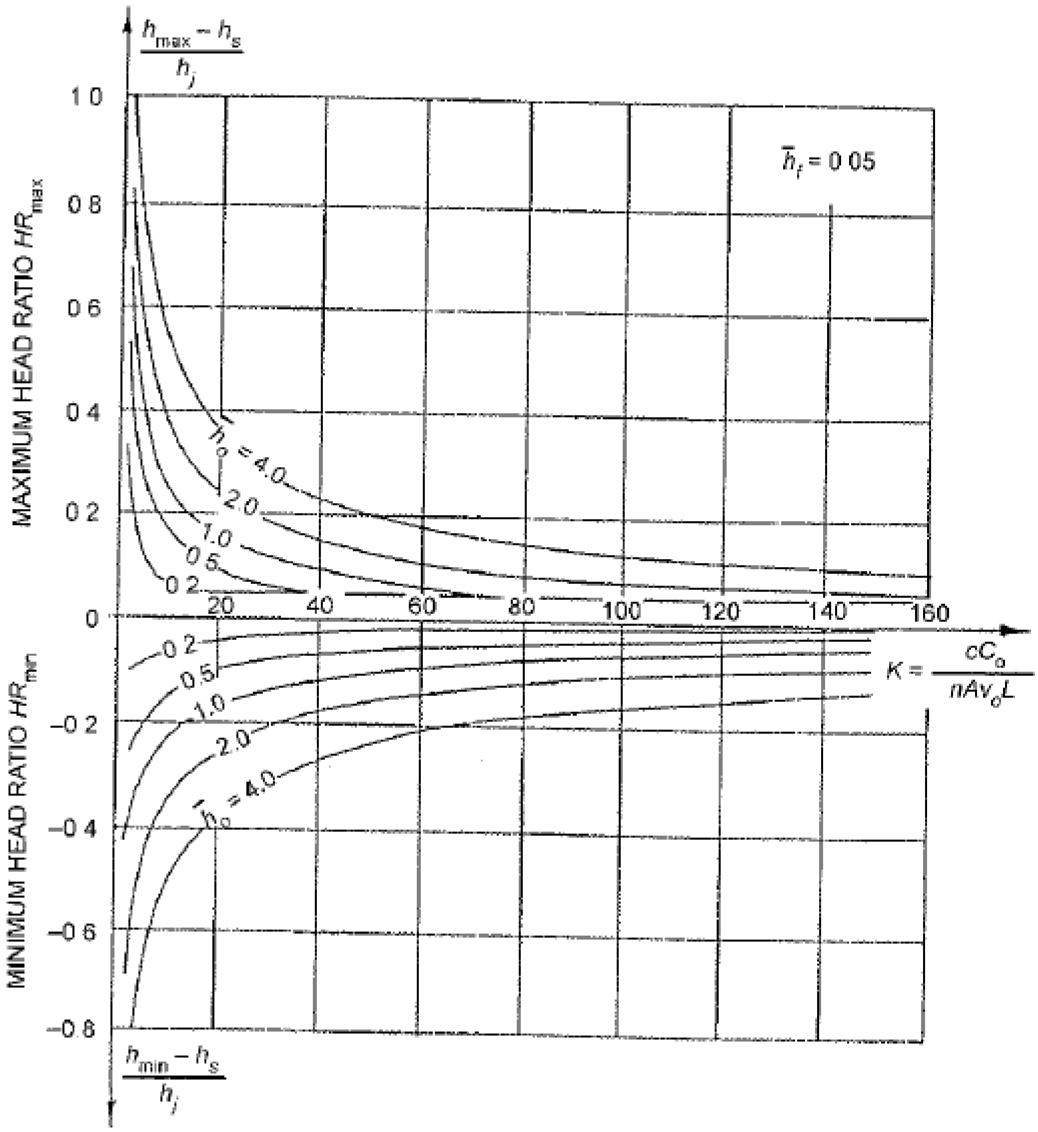


Figure 3-3: Maximum Head Ratio and Minimum Head Ratio w/ K value; $\bar{h}_f$=0.05 (Thorley, 2004)

Friction parameter is defined:

$$\bar{h}_f = \frac{h_L}{h_j} \tag{69}$$

Joukowsky equation could be described as;

$$h_j = \frac{cv_0}{g} \tag{70}$$

$$\bar{h}_0 = \frac{h_0^*}{h_j} = \frac{h_0 + h_{atm}}{h_j} \tag{71}$$

Maximum and minimum heads could be described;

$$HR_{max} = \frac{h_{max} - h_s}{h_j} \tag{72}$$

$$HR_{min} = \frac{h_{min} - h_s}{h_j} \tag{73}$$

Air chamber parameter (K) that is used in charts could be described;

$$K = \frac{cC_0}{nALv_0} \tag{74}$$

$$C' = C_0 \left(\frac{h_0^*}{h_{min}^*} \right)^{1/n} \tag{75}$$

$$V_T = C' * safety\ factor \tag{76}$$

where;

V_T= Air Chamber Volume (m^3)

h_S= Pump Static Head (m)

C_0= Initial Air Volume (m^3)

C'= Expanded Air Volume (m^3)

h_{max}= Maximum Allowable Head In Pump Discharge Line (m)

h_{min}= Minimum Allowable Head In Pump Discharge Line (m)

h_0= Sum of the static and friction head (m)

n= Constant from 1 to 1.4

Safety Factor=1.2 or 1.25

c=Wave speed (m/s) (Thorley, 2004)

Results are obtained and K value is written from a suitable chart. Following K value, air chamber volume is found from Eq.(76).

Stephenson (2002) also developed mathematical equations for certain H_{max}/H_0 and H_{min}/H_0 ratios to determine air chamber volume;

$$S' = \frac{gS_0H_0}{ALv_0} \tag{77}$$

which;

S_0= Initial air volume in air chamber

S= Air Chamber Volume

H_0= Pump Static Head

A=Pipe Area

When H_{min}/H_0 and H_{max}/H_0 ratios are found, S' is determined according to Figure 3-4 and Table 3-2. Then, air volume is found with Eq.(77). H_{min}/H_0 vs $\frac{gSH_0}{ALv_0}$ chart is formed which is given in Figure 3-5. Instead of S_0, S is used in formulation and air chamber volume is determined. Then, the inlet and outlet pipe ratios are determined according to parameters given in the Table 3-3. Figure 3-6 can also be examined to understand the inlet and outlet pipelines in the air chamber.

Table 3-2: H_{max}/H_0 and H_{min}/H_0 Ratios Given In Stephenson (2002) Chart

Ratio	Value	S'
H_{max}/H_0	2.2	0.5
H_{max}/H_0	1.4	1
H_{max}/H_0	1.2	8
H_{min}/H_0	0.7	8
H_{min}/H_0	0.4	1
H_{min}/H_0	0.2	0.5

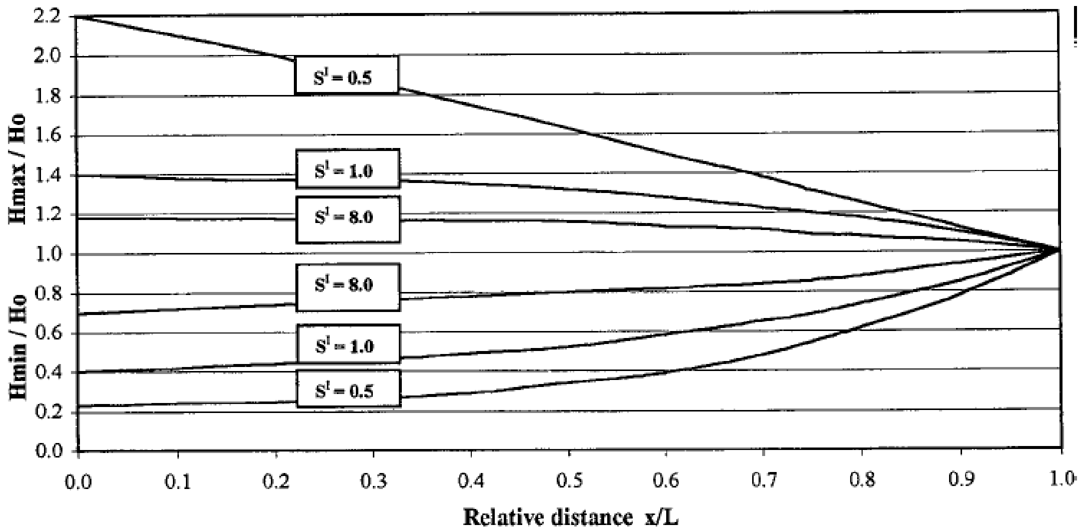


Figure 3-4: S' Graph w/ H_{min}/H_0 and H_{max}/H_0

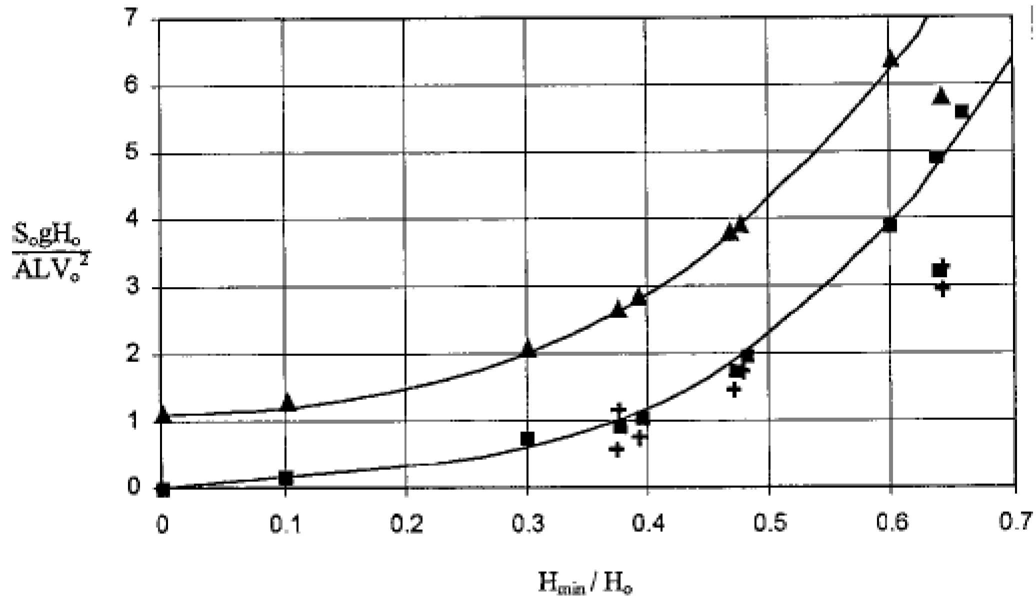


Figure 3-5: H_{min}/ H_0 vs $\frac{gSH_0}{ALv_0}$ Graph

Table 3-3: Inlet and Outlet Ratios Suggested (Stephenson, 2002)

Ratio	Value
Chamber Outlet/Pipe Diameter	0.25 to 0.5
Chamber Inlet/Pipe Diameter	0.15 to 0.4

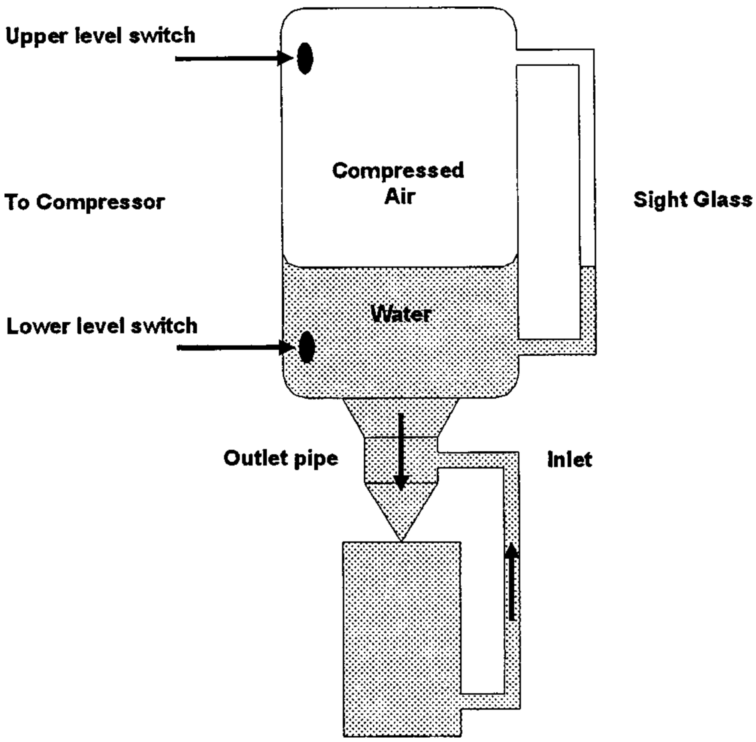


Figure 3-6*:* Air Chamber (Stephenson, 2002)

Air chambers are placed usually near the pumps. Location has effect on the volume of the vessels in long distance pipelines. Their volume could be huge to overwhelm massive pressures especially in high discharge and head systems in long distance pipes. There are several researches to investigate optimal location. Wang et al. (2019) studied to formulate optimum location of the air chamber. Their schematic is given in Figure 3-7.

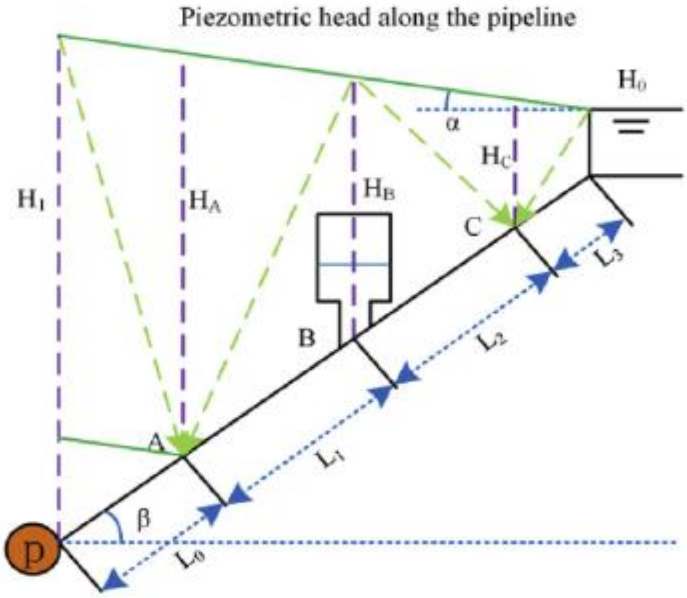


Figure 3-7: Hydraulic System Schematic Diagram (Wang et al, 2019)

$$L_1 = (\mathrm{H_A} - \mathrm{H_B})\frac{cosa}{\sin(a+\beta)} \tag{78}$$

L_0 is defined as a pipe section where no negative pressure is seen in that area. L_1 and L_2 are protection length before and after air chamber. L_3 is the minimum length before reservoir. Slopes are defined as α and β meaning angle of hydraulic and pipe lines respectively. All length could be calculated according to Eq.(78).

3.2. Air Valves

Air could cause some negative effects to hydraulic systems. Dissolved air in fluid leads to pumping unnecessary water. Air existence could cause additional head loss. Furthermore, air is capable of increasing corrosion potential of steel pipes. Moreover, liquid column separation is observed in pump lines. Air valves could be used to prevent such conditions. (Stephenson, 1997)

Relief valves and vacuum valves are widely used in hydraulic systems. Spring is placed in relief valves. When high pressures occur, relief opens.

Vacuum valves are used to prevent negative pressures. When pressure drops below vapor pressure, air influx is provided by them.

Orifice size should be determined carefully. Small diameter may cause providing insufficient air to pipeline. However, enormous diameters could lead to poor resistance to air outflow. When air is exhausted, velocity suddenly becomes zero. Sudden velocity change causes high heads in hydraulic systems. (Bianchi et al, 2007) Diameters from 50 mm to 200 mm are suggested for vacuum valves.

3.3. Flywheels

Following power failure, reverse flow occurs in a very short time interval. Vacuum condition could be observed during transient condition. Flywheels are mainly used against this vacuum condition. Inertia could be defined as resistance to velocity change. Therefore, resistance magnitude of pump rotating part is an important parameter in hydraulic systems.

Inertia should be calculated including all rotating parts. Rotating parts could be

- Motor
- Pump Impeller
- Shaft
- Flywheel

Shaft inertia could be ignored because it has small value when comparing other parameters.

Motor inertia could be calculated as;

$$I_m = 118 \left(\frac{P}{N}\right)^{1.48} \tag{79}$$

where;

P= Power (kW)

N= Speed (rpm)

Impeller inertia could be calculated as;

$$I_p = 1.5 * 10^7 \left(\frac{P}{N^3}\right)^{0.9556} \tag{80}$$

$$P = \frac{\rho * Q * h * g}{3.6 * 10^6 * \eta} \tag{81}$$

Flywheels inertia could be calculated as;

$$I = k * m * r^2 \tag{82}$$

where

m= flywheel mass (kg)

r= inside diameter of cylinder (m)

k=constant depends on flywheel type

3.4. One-Way Surge Tanks

One-way surge tanks are used to suppress negative pressures in hydraulic systems. There is a short pipe between pipeline and surge tank and check valve is placed in there. When pressure in a pipeline is lower than water level in surge tank, water is provided from one-way surge tank. Schematic of one-way surge tank in the pump discharge line is shown in Figure 3-8.

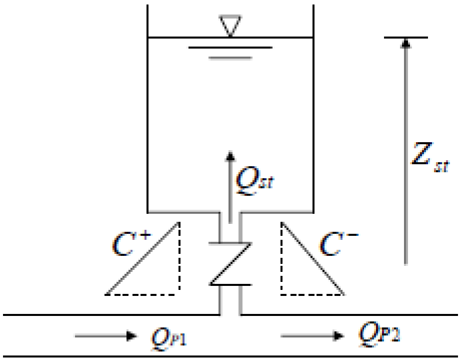


Figure 3-8: Model of One-way Surge Tank In a Pump Discharge Line (Hu et al. 2008)

From Figure 3-4, continuity equation is written as;

$$Q_p = Q_{st} + Q_{p2} \tag{83}$$

$$H_p = Z_{st} + R_k Q_{st} |Q_{st}| \tag{84}$$

$$\frac{dZ_{st}}{dt} = \frac{Q_{st}}{A_{st}} \tag{85}$$

CHAPTER 4

COMPUTER SOFTWARE

4.1. Brief Information about Software Programs Essentials

Various software programs exist to simulate transient conditions. Method of characteristic is widely used for solving hydraulic equations. Gray (1953) is the pioneer of using the method of characteristic in hydraulic equations and Streeter and Lai (1962) and Streeter and Wylie (1967) made important contributions. (Thorley, 2004) With the help of this method, internal conditions and boundary conditions could be estimated in different time and space. In addition, other methods are also formed. Some of them are acceptable but they are still insufficient in some boundary conditions.

In computer solution, some assumptions are made:

- Mach number is taken as very small values. It means velocity in a pipe is very low when it is compared with acoustic wave speed.
- Compressibility is accepted as negligible. In other words, density could be taken as constant.
- Pipe-wall interaction has small values when it is compared with total head loss.
- For elastic modulus and bulk modulus, one constant is taken.

Using proper data has an importance on software program. Devices should be well defined before running hydraulic transients. Additional information could also be described when needed. In general,

- Pipeline characteristics are diameters, lengths, elevation, pipeline profile, head losses, bends, reductions, valves, reservoirs, pumps, tanks, air chambers.

- Fluid parameters are acoustic speed, temperature, pressure, ingredients, density and bulk modulus.

Transient conditions like sudden valve closure or pump failure are also included to the system. (Thorley, 2004)

4.2. HAMMER Software Program

Hammer program is developed to evaluate pipelines and pump lines in transient conditions. All pipe networks could define in that program. HAMMER is a capable of,

- ➢ Defining existing models, creating planned pipelines and running transient simulations
- ➢ Defining pipe, junction, pump, reservoir, valve conditions,
- ➢ Transferring of field data from other program forms to HAMMER
- ➢ Describing transient conditions, time intervals, different scenarios and different cases
- ➢ Describing protection measures, surge tanks, air chambers, valves
- ➢ Analyzing hydraulic system reactions with numerical results and graphical animations. (Bentley HAMMER, 2016).

Due to these advantages, HAMMER program is chosen for this thesis.

HAMMER software program has also some disadvantages. Firstly, some parameters are not included in the transient simulations. To illustrate, initial liquid volume in the air chamber is not a parameter for transient solver. Liquid volume always starts from zero. Secondly, it takes time to get used to the software program.

However, with its advantages; HAMMER is one of the best software programs that could be used in hydraulic transient calculations. Thus, HAMMER program is chosen for this thesis.

4.2.1. Working Environment in Hammer

Program has several shortcuts on its interface. Some of them are given in the Figure 4-1. Layouts could be modified according to user-request. Shortcuts on Figure 4-1 are respectively; Select, Pipe, Junction, Hydrant, Tank, Reservoir, Customer Meter, SCADA Element, Periodic Head Flow, Pump, Variable Speed Pump, Pump Station, Turbine, Valves, Valve With Linear Area Change, Check Valve, Orifice Between Pipes, Discharge To Atmosphere, Surge Tank, Air Chamber, Air Valve, Surge Valve, Rupture Disk, Isolation Valve, Spot Elevation, Border, Text and Line.

- **Pipes** are main framework of water distribution systems. Diameter, pipe material, roughness coefficient, length, minor losses, check valves and acoustic speed are defined in pipe properties.
- **Junctions** are used for mainly connection of pipes, defining elevations and demands.
- **Reservoirs** are defined for symbolizing water resources such as dams. Water surface elevations could be defined in different time intervals.
- **Pumps** are defined according to their design conditions. Pump definition is an important part of the transient simulation. Pump stoppage could be defined in HAMMER with "shut down after time delay" option. Transient time interval is described in calculation options. Variable speed pumps could also be defined with user-defined curve or fixed curve.
- **Surge Tanks, Air Chambers and Air Valves** are selected for protection purposes. Initial gas volume, initial liquid volume, elevation, inlet diameter, loss ratio and coefficient and chamber type could be defined in **air chambers.** In addition to this, animation graph of air volume and liquid volume with respect to time could be seen in transient graphs. Operating elevation, type of tank, inlet diameter and head loss coefficient could be defined in **surge tanks.** Different kind of valves could be selected in **air valves.**

Figure 4-1: Shortcuts in HAMMER

Elevation and pipeline distance could be taken automatically from other sources with help of ModelBuilder. It is used for synchronizing purposes of the raw data that are described in AutoCAD, MicroStation or ArcGIS. However, diameter, pipe material and acoustic speed should be still defined in each pipe. FlexTables are used for this purpose. All hydraulic elements such as pumps, pipes, valves, reservoirs, surge tanks and hydropneumatic tanks exist in FlexTables shown in Figure 4-2. With help of GlobalEdit, selected parameters could be changed totally. Moreover, all parameters are checked in that tables. For developing different cases, Scenarios and Alternatives are used. Alternatives include many options. In that section, several topologies could be activated. To illustrate, various air chambers are defined and selected one(s) could be run in different scenarios.

In properties section which is given in Figure 4-3;

- Efficiency, moment of inertia, best efficiency point, head-flow relation, transient conditions, pump types and flow patterns are defined for pumps.
- Elevations are defined in reservoirs.
- Demands and elevations are defined for junctions.
- Length, diameter, wave speed, head loss and minor losses are described for pipes.

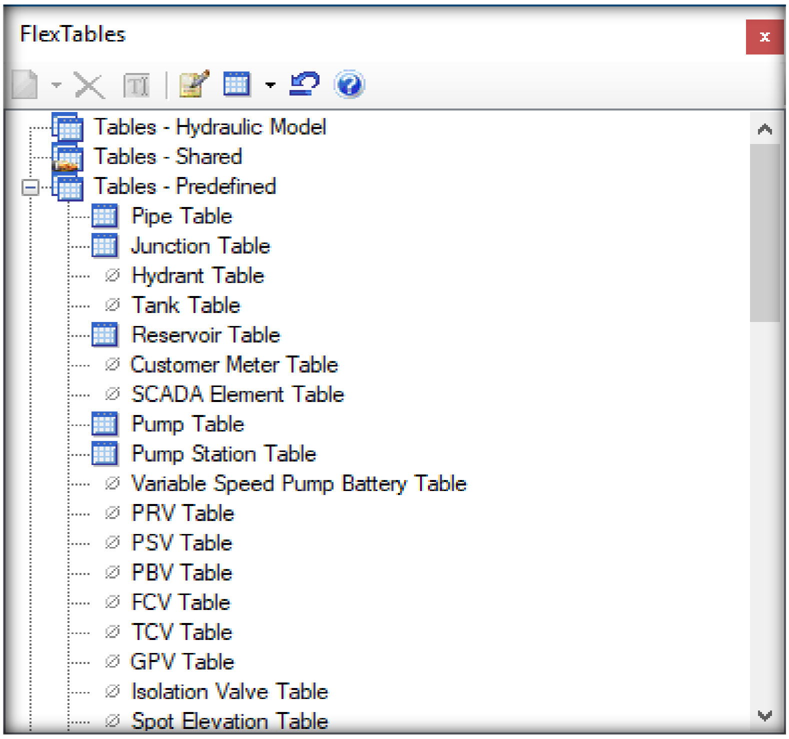


Figure 4-2: FlexTables

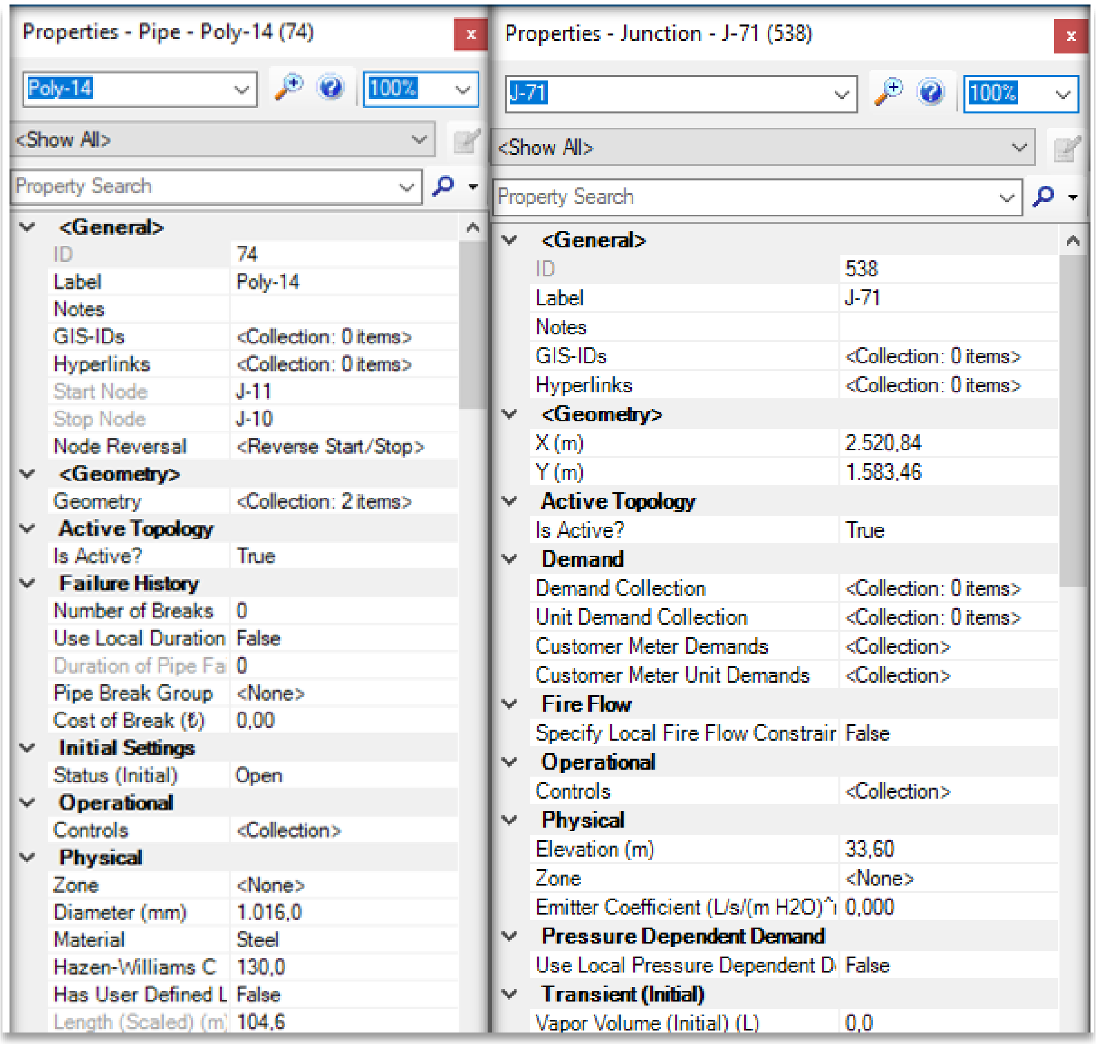


Figure 4-3: Pipe and Junction Properties

After all hydraulic system parameters are set, validation could be done. Errors could be detected before running system. In Hammer Software, calculation is done according to steady state and transient state, respectively. Initial condition could be controlled with “Compute Initial Conditions”. Moreover “Compute” option is used for transient calculations. Results are seen from “Calculation Summary” and “Transient Calculation Summary” tables. Errors are showed in User Notifications section. The program interface is shown in Figure 4-4. Steady state-transient state options and results are given in Figure 4-5 and Figure 4-6, respectively.

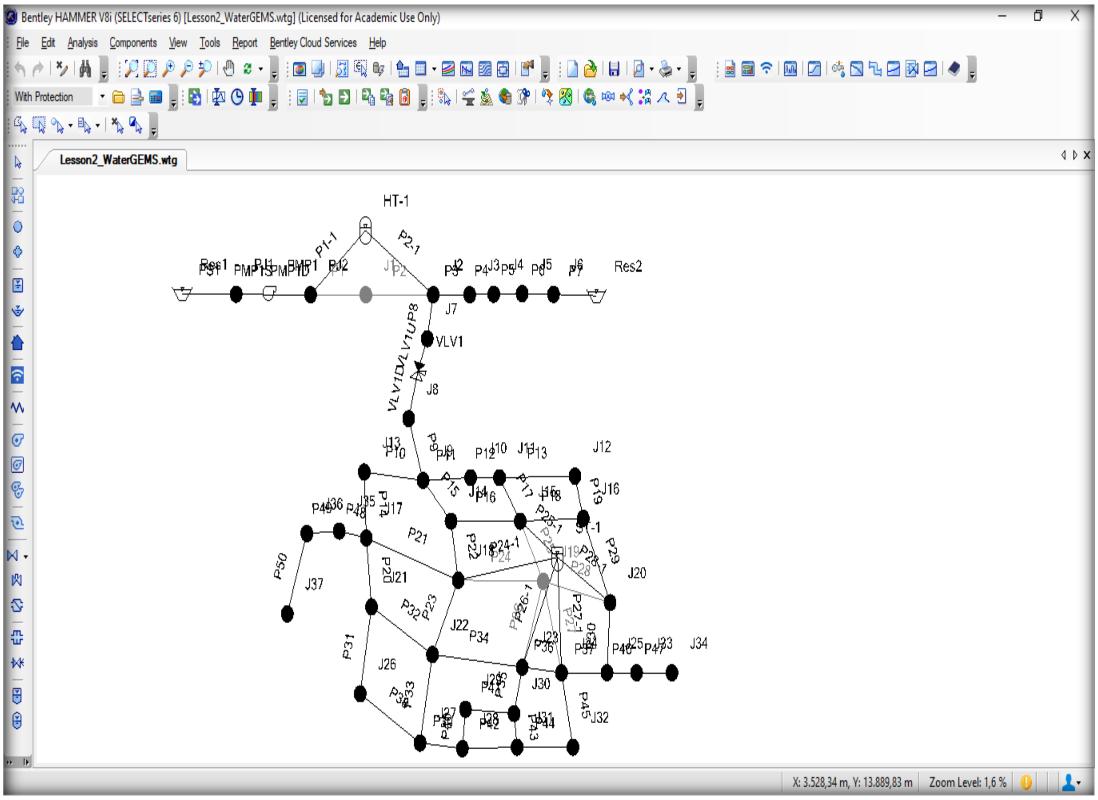


Figure 4-4: Hammer Interface

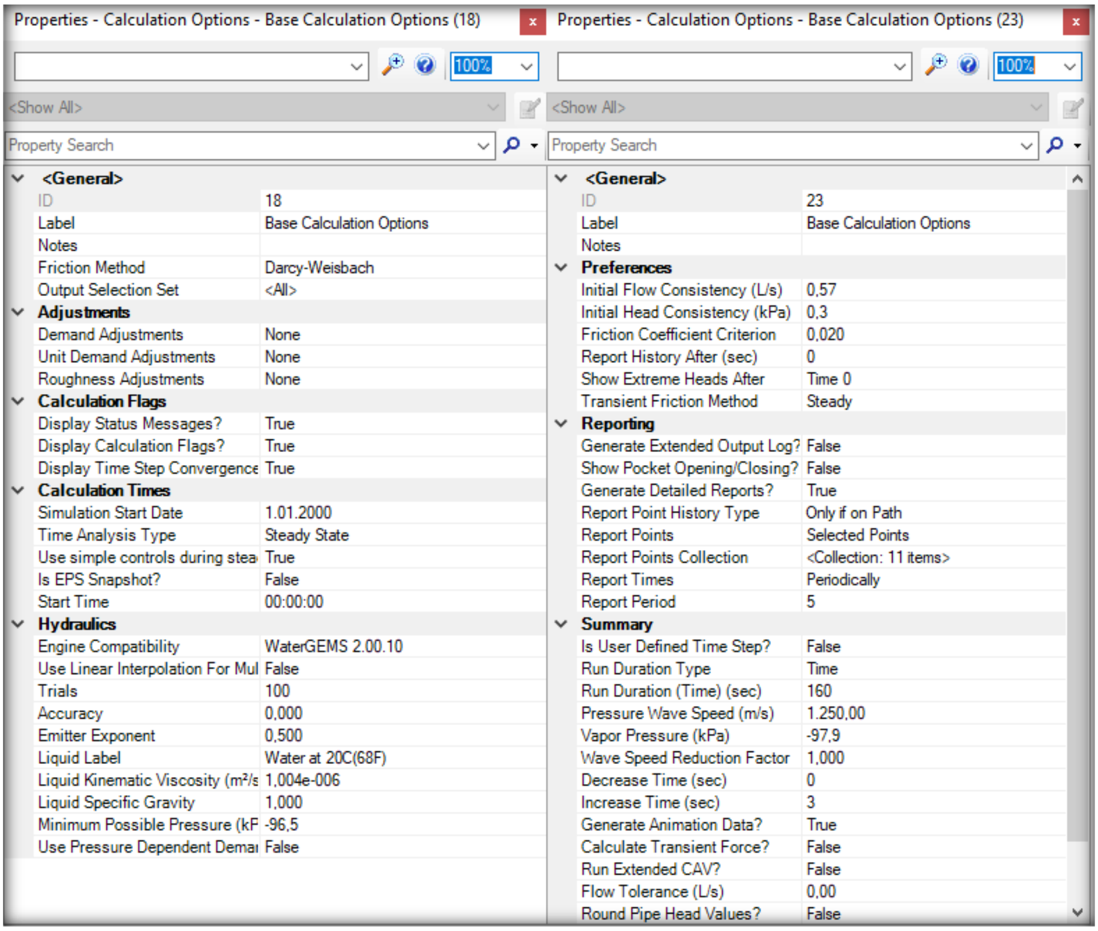
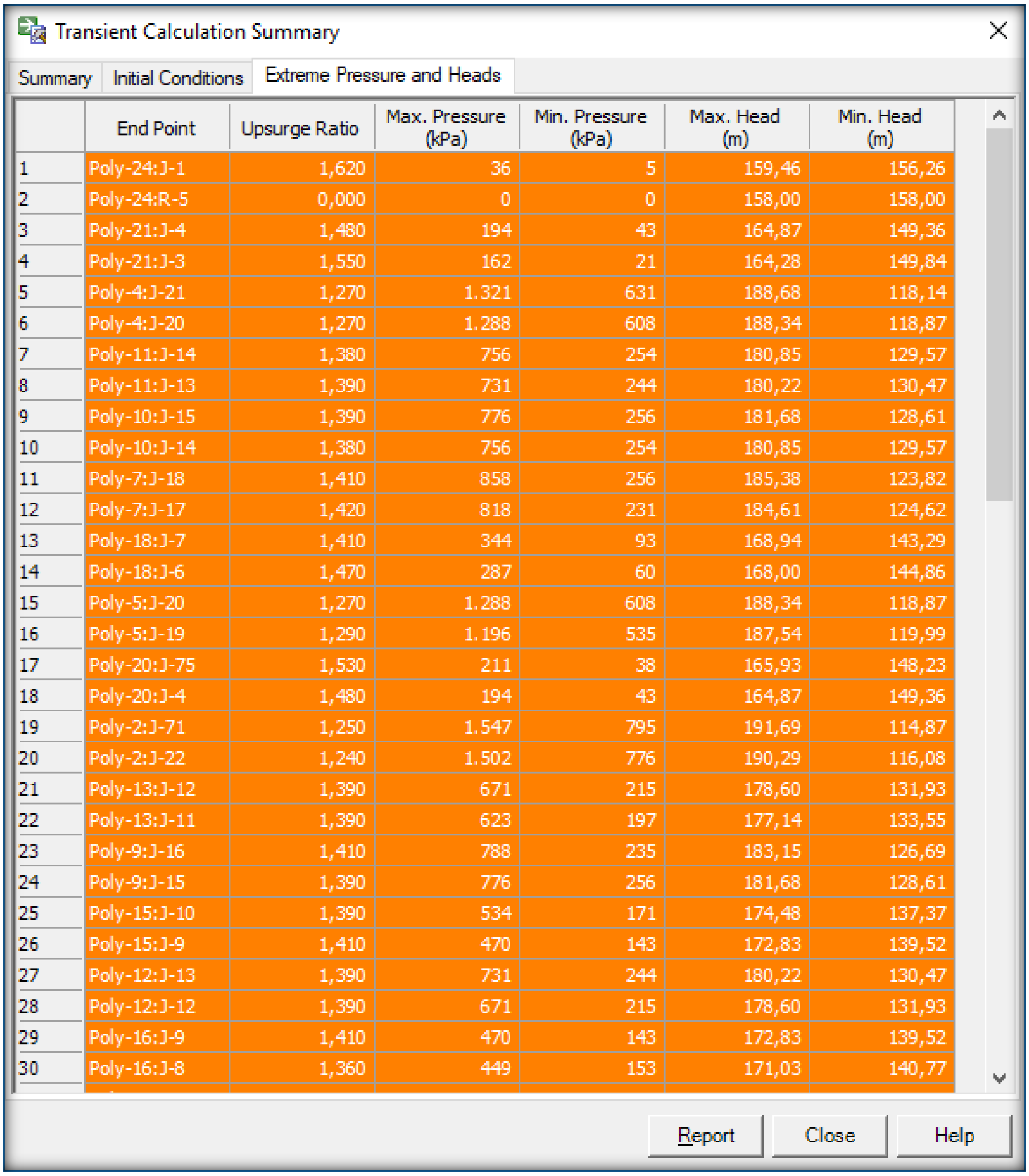


Figure 4-5: Steady State and Transient Calculation Solvers

After pipe and fluid characteristics are set and steady state and transient state options are defined, results could be examined in different scenarios with numerical values or graphs. According to results, protection devices could be selected.



	End Point	Upsurge Ratio	Max. Pressure (kPa)	Min. Pressure (kPa)	Max. Head (m)	Min. Head (m)
1	Poly-24:J-1	1,620	36	5	159,46	156,26
2	Poly-24:R-5	0,000	0	0	158,00	158,00
3	Poly-21:J-4	1,480	194	43	164,87	149,36
4	Poly-21:J-3	1,550	162	21	164,28	149,84
5	Poly-4:J-21	1,270	1.321	631	188,68	118,14
6	Poly-4:J-20	1,270	1.288	608	188,34	118,87
7	Poly-11:J-14	1,380	756	254	180,85	129,57
8	Poly-11:J-13	1,390	731	244	180,22	130,47
9	Poly-10:J-15	1,390	776	256	181,68	128,61
10	Poly-10:J-14	1,380	756	254	180,85	129,57
11	Poly-7:J-18	1,410	858	256	185,38	123,82
12	Poly-7:J-17	1,420	818	231	184,61	124,62
13	Poly-18:J-7	1,410	344	93	168,94	143,29
14	Poly-18:J-6	1,470	287	60	168,00	144,86
15	Poly-5:J-20	1,270	1.288	608	188,34	118,87
16	Poly-5:J-19	1,290	1.196	535	187,54	119,99
17	Poly-20:J-75	1,530	211	38	165,93	148,23
18	Poly-20:J-4	1,480	194	43	164,87	149,36
19	Poly-2:J-71	1,250	1.547	795	191,69	114,87
20	Poly-2:J-22	1,240	1.502	776	190,29	116,08
21	Poly-13:J-12	1,390	671	215	178,60	131,93
22	Poly-13:J-11	1,390	623	197	177,14	133,55
23	Poly-9:J-16	1,410	788	235	183,15	126,69
24	Poly-9:J-15	1,390	776	256	181,68	128,61
25	Poly-15:J-10	1,390	534	171	174,48	137,37
26	Poly-15:J-9	1,410	470	143	172,83	139,52
27	Poly-12:J-13	1,390	731	244	180,22	130,47
28	Poly-12:J-12	1,390	671	215	178,60	131,93
29	Poly-16:J-9	1,410	470	143	172,83	139,52
30	Poly-16:J-8	1,360	449	153	171,03	140,77

Figure 4-6: Transient Results in Numeric Values

CHAPTER 5

CASE STUDY

In this chapter, transients in the pump discharge line extending from Çokal Dam Pump Station to Water Treatment plant is examined. As protection measures air chamber, one-way surge tank, air valve and flywheel will be used in the simulations.

5.1. Information about Pipeline

The aim of Çokal Dam is to store water for agricultural purposes and drinking water supply. In Figure 5-1, pipeline profile is given. Pump station is at the starting point and water treatment plant is at the end point. In addition, physical characteristics of the dam are given in the Table 5-1.

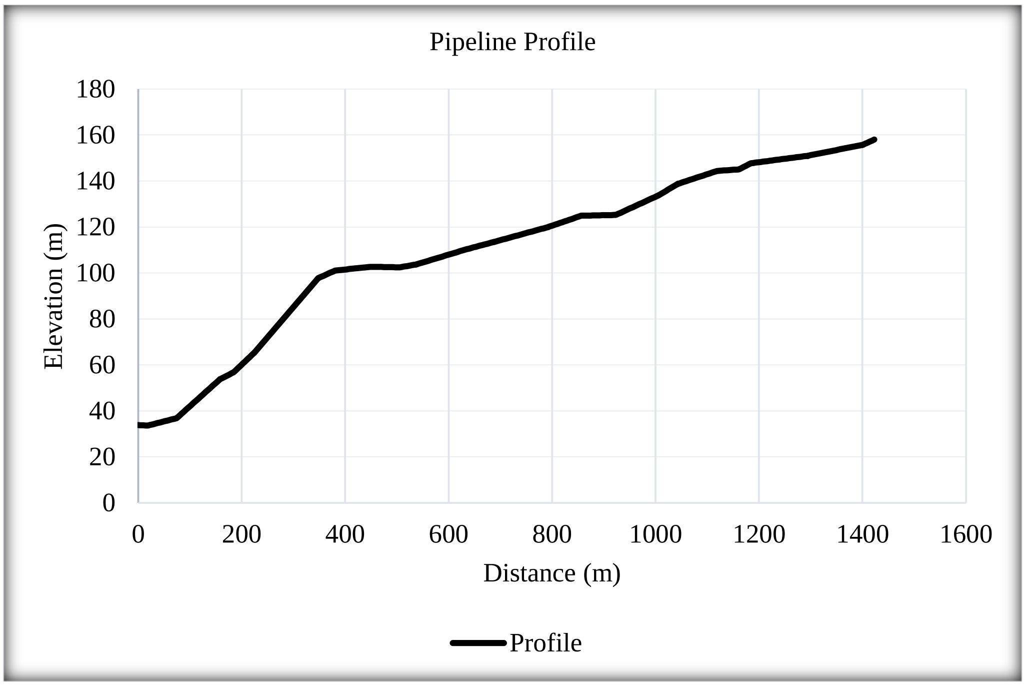


Figure 5-1: Pipeline Profile between Pump Station and Water Treatment Plant

Table 5-1: Physical Characteristic of the Çokal Dam

Parameters	Elevation
Crest Elevation	85 m
Minimum Water Elevation	43.74 m
Normal Water Elevation	80 m
Maximum Water Elevation	83.68 m
Maximum Flow Used for Irrigational Purposes	6700 l/s
Maximum Flow For Drinking Water Supply	1150 l/s
Minimum Flow For River	250 l/s
Capacity of Water Intake Structure	8100 l/s

Water distribution system is planned starting from Çokal Dam to Gelibolu, Eceabat, Evreşe and Şarköy locations. Some settlements are located at seaside. However, others are located at high elevations that affect the system hydraulics. This situation determines water treatment plant location. Designers considered that location of the treatment plant should be at high elevations. Pipeline and pump characteristics are given in Table 5-2, Table 5-3 and Table 5-4.

Table 5-2: Dimensions of Steel Pipe Between Pump Station and Treatment Plant

Pipe Parameters	Dimensions
Outside Diameter	1016 mm
Total Wall Thickness	18.8 mm
Inside Diameter	978.4 mm
Length	1433.5 m

Table 5-3: Pipe Lengths

Pipe Name	Length (m)
Poly-1	29.6
Poly-2	55.3
Poly-3	83.5
Poly-4	26.8
Poly-5	40.0
Poly-6	123.3
Poly-7	32.7
Poly-8	68.2
Poly-9	57.0
Poly-10	31.4
Poly-11	28.2
Poly-12	65.2
Poly-13	56.9
Poly-14	104.6
Poly-15	64.4
Poly-16	66.7
Poly-17	83.2
Poly-18	36.6
Poly-19	75.9
Poly-20	42.0
Poly-21	23.8
Poly-22	110.0
Poly-23	106.2
Poly-24	22.0

Table 5-4: Pump Parameters

Pump Parameters	Magnitudes/Specification
Pump Type	Double Suction
Flow	287.5 l/s
Pump Head	126 m
Efficiency	83.2 %
Power	500 kW
Net Positive Suction Head	4.31 m
RPM	1488
Weight	3700 kg

4 parallel pumps are operated and totally 1150 l/s water is pumped in a confined line. Pump with a frequency converter is selected in the design stage. Head values could be set with the help of the frequency converter.

Modulus of elasticity of steel is taken as 207 GPa. (American Iron and Steel Institute, 2013). In addition, bulk modulus of water is taken as 2.07 GPa. (Rao, B.C., 2009).

Wave speed could be calculated as;

$$a = \sqrt{\frac{K \,/\, \rho}{1 + (K \,/\, E)(D \,/\, e)}} \tag{86}$$

From Eq.(86);

$$a = \sqrt{\frac{2.07 * 10^{9}/1000}{1 + (2.07 * 10^{9} \,/\, 2.07 * 10^{11})(978.4 \,/\, 18.8)}}$$

Result of the wave speed is given in Table 5-5. Tensile forces occur during transient event. The magnitude of lateral stress depends on the circumferential tensile force and the pipe wall thickness which is shown in Figure 5-2.

Table 5-5: Wave Speed

Pipe Diameter	Wave Speed
1016 mm (Inside diameter is 978.4 mm)	1166.81 m/s

Maximum allowable head is calculated as follows;

$$\sigma = \frac{\gamma * H * D}{2e} \tag{87}$$

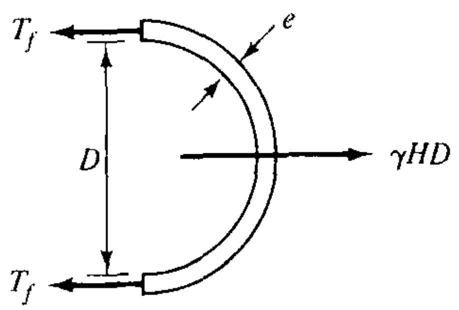


Figure 5-2: Forces On Pipe During Transient

$$60 * 10^6 \frac{N}{m^2} = \frac{9810 \frac{N}{m^3} * \mathrm{H} * 0.978\mathrm{m}}{2 * 0.018\, m}$$

$$\mathrm{H}_{max} = 225 \text{ m}$$

5.2. Preliminary Analyses

Emergency pump shutdown for all 4 pumps is simulated with Hammer. The simulation time duration is determined as 100 seconds for transient flow conditions, which is pretty long time duration. Shutdown starts at 5th second and the graph is given in the Figure 5-4. Unprotected system and different alternatives are simulated. Time step is chosen according to the suggested value that is calculated by program based on the Courant condition that could be checked from Figure 5-3.

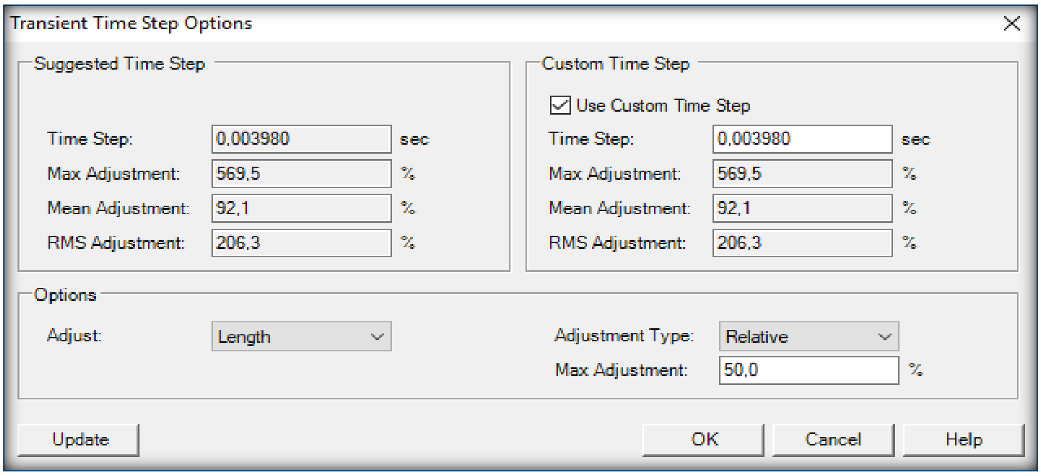


Figure 5-3: Transient Time Step Options

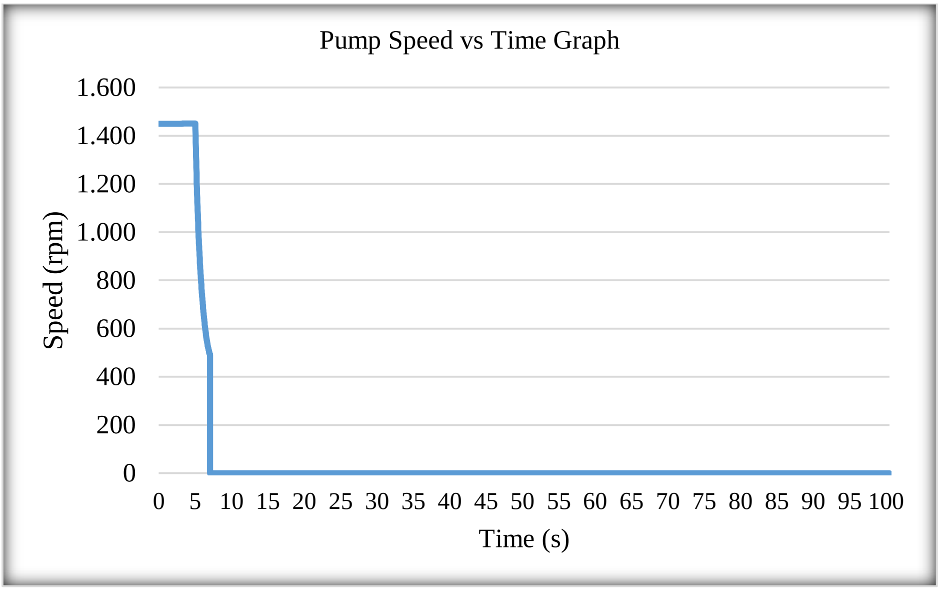


Figure 5-4: Pump Speed Variation With Respect To Time

All cases that were simulated are given in Figure 5-5.

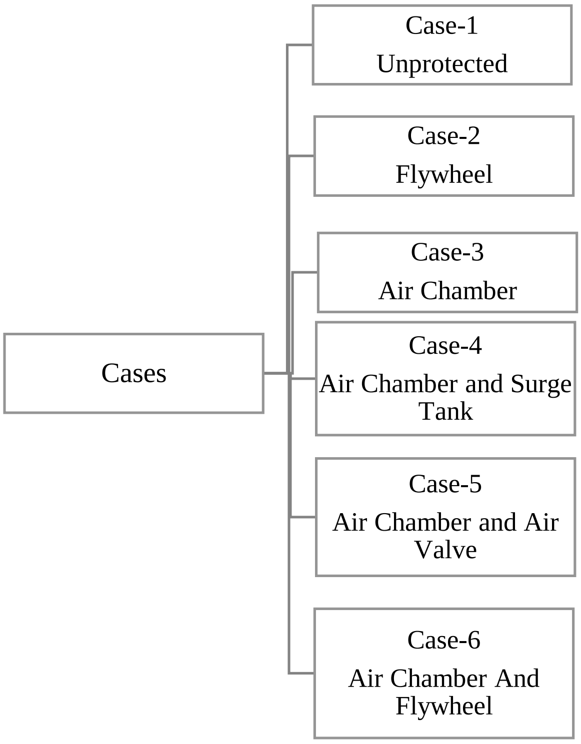


Figure 5-5: All Cases

5.3. Unprotected Case

Pump discharge line is examined with its suggested properties. Cavitation problem is observed in the confined line that could be seen from Figure 5-6. It is observed that first air volume in the pump discharge line is seen at 90 m elevation which is at 320 m distance. In addition, maximum air volume is at 144.3 m elevation and 1180 m distance. The highest observed air volumes are given in detail in the Table 5-6.

Hydraulic grade line drops below pipeline profile nearly for the entire pipeline except for the first 153.4 m. In addition, highest pressures are seen in the first 150 m that could be seen from Figure 5-7. In addition, elevations of the highest pressures are given in detail at Table 5-7. It is concluded that, protection device should be considered in the pump discharge line because pipe line could not withstand these pressures.

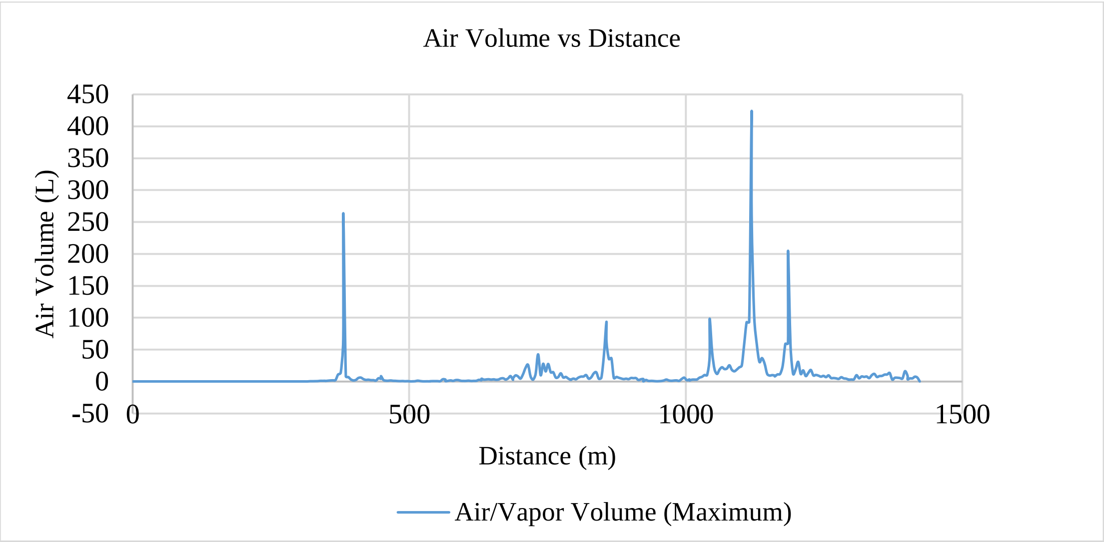


Figure 5-6: Vapor Volume In a Pipeline

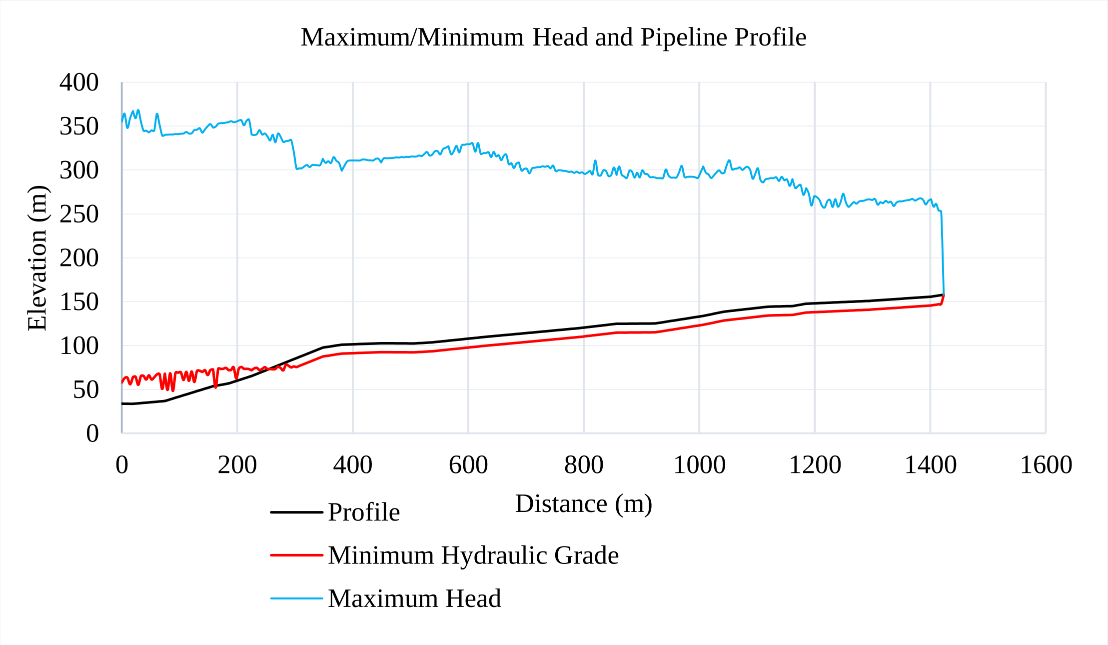


Figure 5-7: Maximum and Minimum HGL In Pump Discharge Line

Table 5-6: Elevations According To Maximum Vapor/Air Volume

Elevation (m)	Vap. Grade Line (m)	Min. HGL (m)	Initial HGL (m)	Max. HGL (m)	Max. Air Volume (L)
144.30	134.30	134.30	158.50	290.10	420.0
101.00	91.00	91.00	159.80	299.20	263.5
144.30	134.30	134.30	158.50	290.10	259.7
147.70	137.70	137.70	158.40	279.40	204.7
144.40	134.40	134.40	158.50	290.80	102.1
138.70	128.70	128.70	158.70	296.70	98.3
124.90	114.90	114.90	159.00	294.40	93.2
144.00	134.00	134.00	158.50	289.50	93.1
143.70	133.70	133.70	158.50	285.90	93.0
101.00	91.00	91.00	159.80	299.20	67.5
124.90	114.90	114.90	159.00	294.40	62.3
147.70	137.70	137.70	158.40	279.40	59.7

Table 5-7: Elevations According To Maximum Hydraulic Grade Line

Elevation (m)	Vap. Grade Line (m)	Min. HGL (m)	Initial HGL (m)	Max. HGL (m)
34.10	24.10	55.50	160.30	368.50
33.60	23.60	64.30	160.40	367.30
33.60	23.60	64.30	160.40	367.30
33.70	23.70	63.00	160.40	364.30
36.00	26.00	67.30	160.30	364.10
33.60	23.60	56.10	160.40	359.30
33.90	23.90	64.60	160.40	359.00
64.40	54.40	73.30	160.00	357.40
61.50	51.50	75.70	160.00	356.70
60.60	50.60	73.80	160.10	356.40
63.40	53.40	73.70	160.00	356.10
57.70	47.70	72.40	160.10	355.70

5.4. Flywheel

In this alternative, inserting flywheels to each pump is evaluated. Motor and impeller moment of inertia are calculated:

$$I_m = 118\left(\frac{500}{1488}\right)^{1.48}$$

$$I_m = 24.18 \text{ kg.m}^2$$

$$I_p = 1.5 * 10^7 \left(\frac{500}{1488^3}\right)^{0.9556}$$

$$I_p = 4.57 \text{ kg.m}^2$$

$$I_t = 28.76 \text{ kg.m}^2$$

Total inertia is simulated with 28.76 kg.m^2 (unprotected case) , 100 kg.m^2,160 kg.m^2 and 300 kg.m^2. When 100 kg.m^2 is simulated, no air volume is seen but negative pressures are observed in the pump discharge line. Then, 160 kg.m^2 is run. No negative pressure and no cavitation problem are observed in the pump discharge line when inertia is greater than or equal to 160 kg.m^2.

Results that are given at Figure 5-8 and Figure 5-9 reveal that negative pressures could be suppressed with flywheels. Yang (2001) also suggested to increase inertia from 45.5 kg.m^2 to 320 kg. m^2 to his client in one of his case to conserve liquid form in the pipeline.

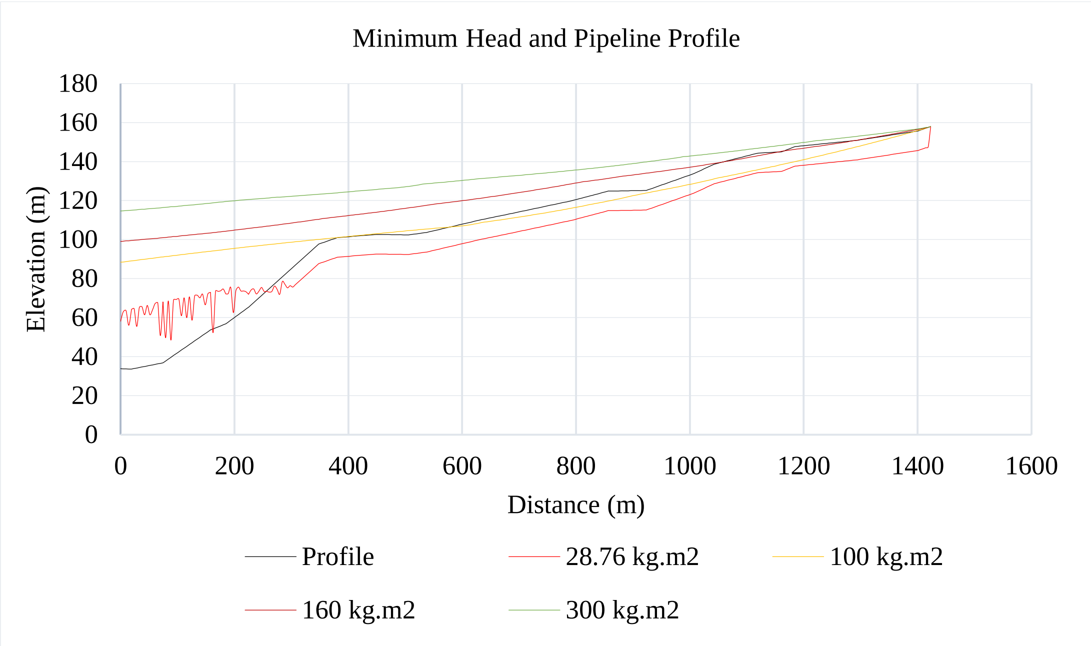


Figure 5-8: Minimum Hydraulic Grade Line With Different Inertia Values

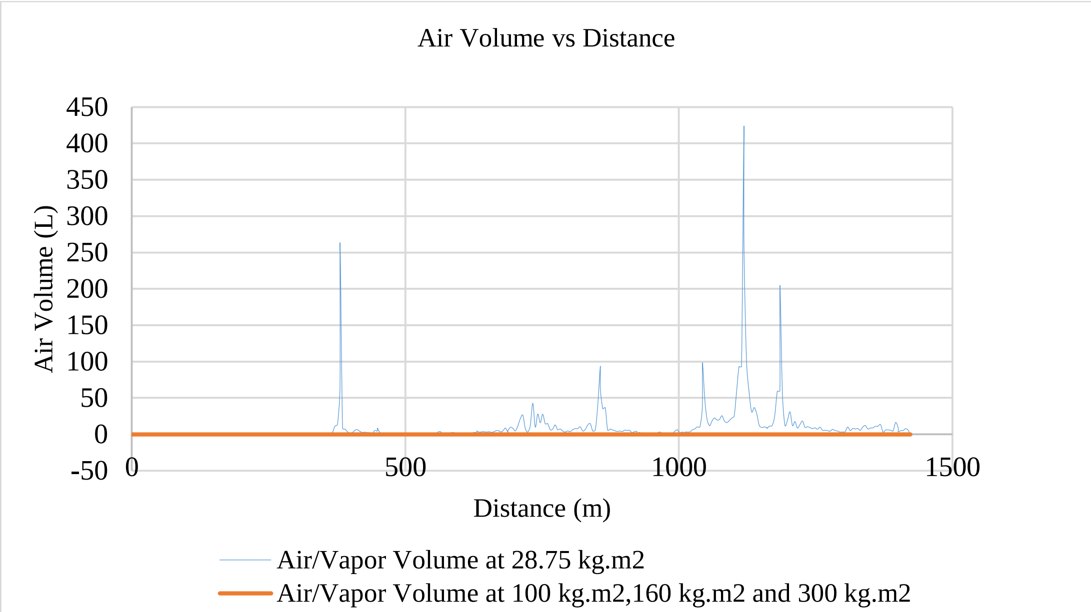


Figure 5-9: Air Volume with Different Moment of Inertia

5.5. Protection with an Air Chamber

In this case, air chambers are tested with different volume. It is seen that air volume ratio in hydropneumatic tank for this hydraulic system is between 0.35 to 0.55

according to Thorley (2004) air chamber design calculations and graph. Outlet diameter of the chamber should be chosen from 0.25 to 0.5 of the main pipe diameter. In addition, inlet diameter is taken between 0.1 to 0.4 ratio. (Stephenson, 2002) Moreover, head loss ratio of outflow to inflow could be taken as 0.4. Outflow diameter should be large enough to avoid low head. In addition, velocity should not be higher than 10 m/s in transient condition. Furthermore, inflow diameter should be small to decrease air volume size (Chaudhry, 1979). It is suggested that check valve should be placed between pump and air chamber. (Wylie et al, 1993).

When calculation was done, it was seen that initial gas volume, water volume and outlet orifice diameter have remarkable effects on minimum and maximum heads. Increasing initial gas volume reduce maximum pressure. However, enlargement on gas volume causes less liquid space which effects minimum pressure negatively. Thus, gas and liquid volume effects whole system stability. Furthermore, orifice diameter has also effect on air chamber volume. The smallest possible orifice diameter means low cost. However, when diameter is excessively small, pipeline system behaves that there is no air chamber in confined line. On the other hand, bigger diameter could not decrease high pressures effectively. Thus, optimum orifice diameter should be determined carefully. Various trials were done to find optimum diameter in each case.

5.5.1. Air Chamber with a Volume of 6.5 m^3

In this alternative, air chamber having a volume of 6.5 m^3 is selected as protective measure in hydraulic system. Dimensions were calculated according Stephenson (2002) guide which is given in Figure 3-4 and Thorley's (2004) Air Chamber volume estimation which is shown in Figure 3-3 and Figure 3-4.

$S' = \frac{gS_0H_0}{ALv_0}$=1 is taken (Figure 3-4)

where;

g=9.81 m/s^2

H_0=126.6 m

$A = 0.75\ m^2$

$L = 1433\ m$

$V_0 = 1.46\ m/s$

$S_0 =$ **$1.84\ m^3$ initial air volume**

S=3 (Figure 3-5)

$V' = 1.84 * 3 = 5.52\ m^3$

$V = 1.2$(safety constant)$* V' = 6.6\ m^3$

According to Thorley; (Figure 3-3)

$h_{atm} = 10\ m$

$h_s = 126.6\ m$

$h_L = 3.47\ m$

$h_0 = 130.07\ m$

$a = 1166.81\ m/s$

$V_0 = 1.46\ m/s$

$g = 9.81\ m/s^2$

$$h_j = \frac{1166.81\frac{m}{s} * 1.46\ m/s}{9.81\frac{m}{s^2}} = 173.7$$

$$\bar{h}_f = \frac{3.47\ m}{173.7\ m} = 0.01998$$

$$\bar{h}_0 = \frac{h_0^*}{h_j} = \frac{130.07\ m + 10\ m}{173.7\ m} = 0.807$$

$h_{max} = 225\ m$

$h_{min} = 80\ m$

$K = \frac{cC_0}{nALv_0}$=3 is determined from the $\bar{h}_f$=0.05 chart; (Figure 3-3)

n=1

C_0=4.27 m^3 Initial Air Volume

$C' = 4.27\ \text{m}^3 * \left(\frac{130.07\ m+10\ m}{80\ m+10\ m}\right)^{1/1}$=6.23 m^3

V= 6.5 m^3 Air Chamber Volume

Trials were done (1.84 m^3 -4.27 m^3) to determine initial air volume and it was seen 2.3 m^3 air volume gives optimum results. Air chamber specifications are given in Table 5-8. The location of the pump station was determined as the nearest possible point considering that the highest pressures are at the closest points to the pump station. The outlet orifice diameter was tested according to ratio given in Table 3-3. The outlet orifice having 190 mm diameter gave the best results in the simulation.

Significant changes are observed in the pipeline. Maximum head and maximum air volume decreases in remarkable percentage which could be checked from Table 5-9 and Table 5-10. However, negative pressures are seen from 307 m to 1433 m and cavitation problem is observed that could be seen from Figure 5-10 and Figure 5-11. When pump stoppage occurs at the 5^{th} second, flow is supplied from the air chamber and liquid volume decreases. Then, air volume increases in seconds due to reduced liquid volume given in Figure 5-12.

Table 5-8: Air Chamber (6.5 m^3) Specifications

Air Chamber Specifications	Magnitude
Elevation	34 m
Volume	6.5 m^3
Initial Liquid Volume	4 m^3
Initial Gas Volume	2.3 m^3
Chamber Outlet Orifice Diameter	190 mm
Minor Loss Coefficient (Outflow)	2.5
Ratio of Losses (Inflow/Outflow)	2.5
Distance to Pump Station	23.63 m

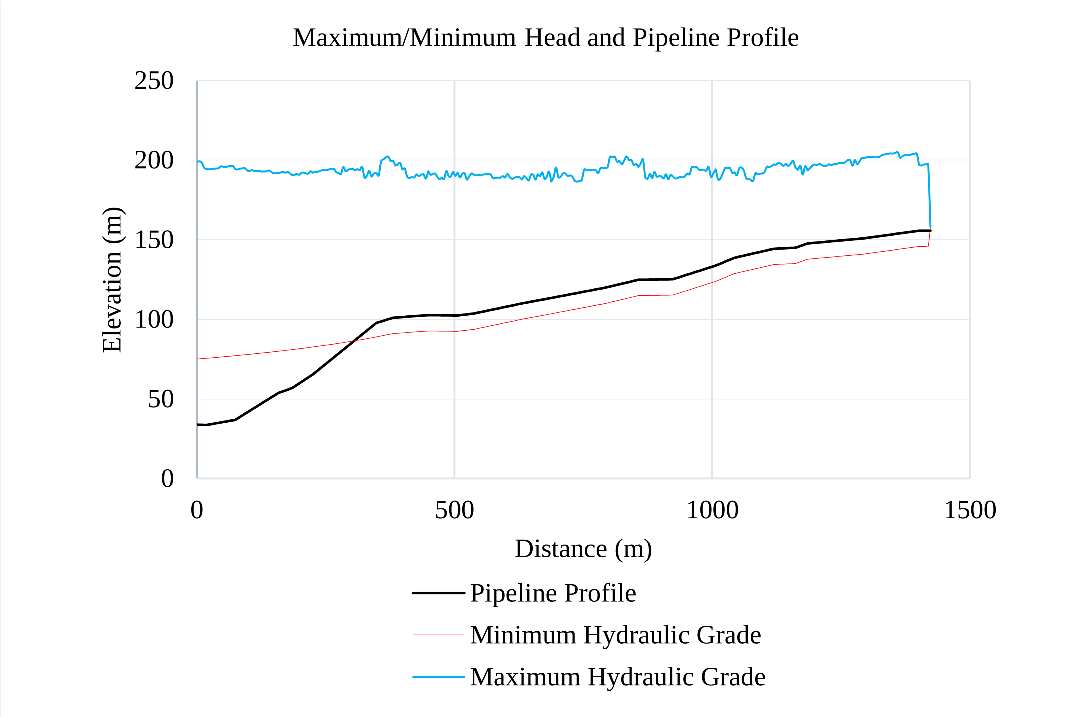


Figure 5-10: Maximum and Minimum Heads in the Air Chamber (6.5 m3)

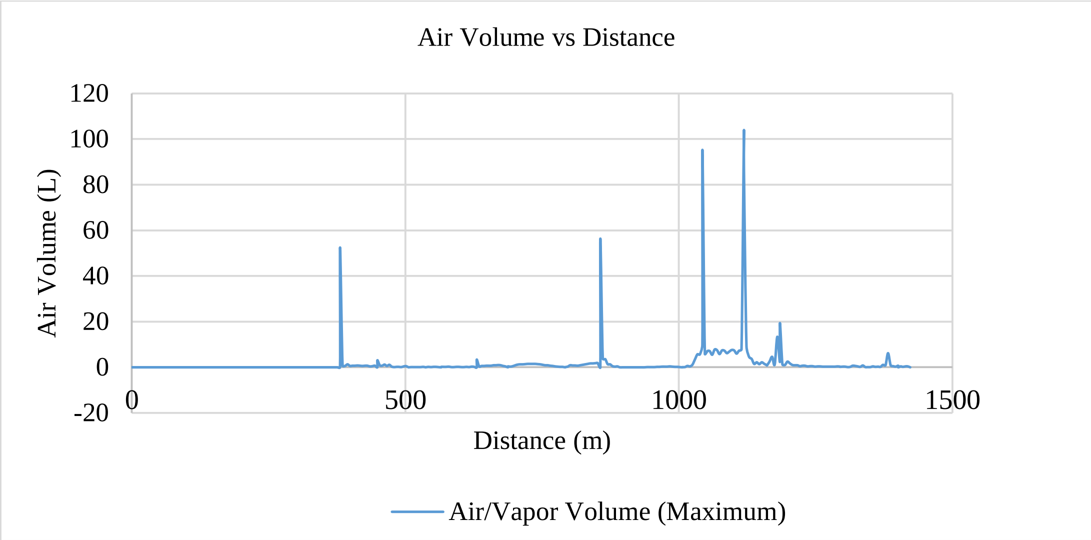


Figure 5-11: Air Volume in the Pipeline (6.5 m^3)

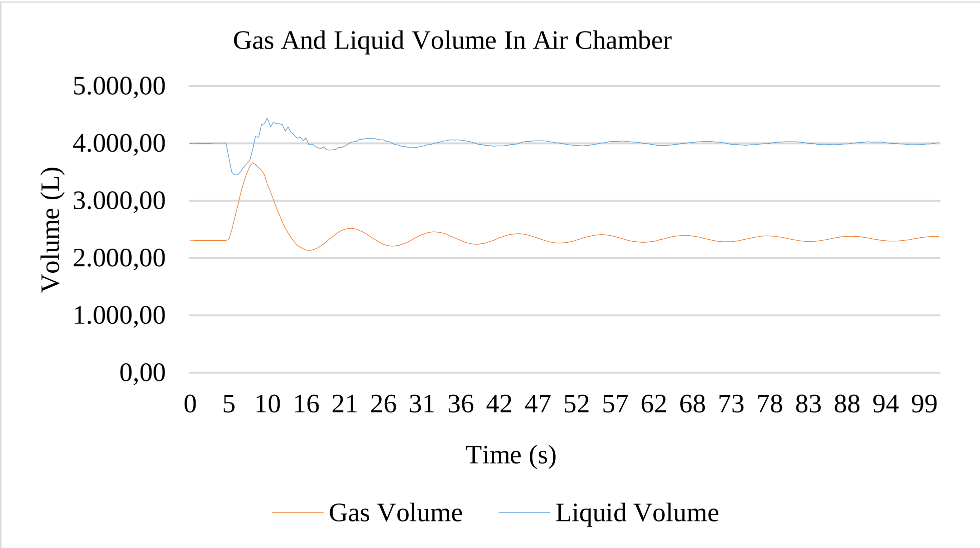


Figure 5-12: Gas and Liquid Variations in the Air Chamber (6.5 m^3)

Table 5-9: Comparison of the Air Chamber (6.5 m^3) with No Protection Case

Properties	Magnitude	Elevation
Max Head- No Protection	368.5 m	34.1 m
Max Head - Air Chamber (6,5 m^3)	204.9 m	153.9 m
Max Air Vol.- No Protection	420 L	144.3 m
Max Air Vol.- Air Chamber (6,5 m^3)	101.7 L	144.3 m

Table 5-10: Head and Air Volume Decrease in the Pipeline with the Air Chamber (6.5 m^3)

Changing Parameter	Decrease in Percentage
Max Head	%44.3
Max Air Volume	%75.7

Cavitation could cause severe damage to the pipeline. Thus, it is concluded that air chamber with a volume of 6.5 m^3 is inadequate against liquid column separation.

5.5.2. Air Chamber with a Volume of 10 m^3

10 m^3 air chamber is calculated according to Thorley (2004) design calculation. Figure 3-2 is used in calculations. Air chamber specifications are given in Table 5-11.

h_{max}=225 m

h_{min}=85 m

$K = \frac{cC_0}{nALv_0}$=5 is determined from the $\bar{h}_f$=0 (Figure 3-2)

n=1

C_0=6.67 m^3 Initial Air Volume

$C' = 6.67\ \text{m}^3 * \left(\frac{130.07\ m+10\ m}{85\ m+10\ m}\right)^{1/1}$=9.84 m^3

V= 10 m^3 Air Chamber Volume

The results of the calculations began to be tested in Hammer and the most appropriate results were determined in the simulation that is given in the Table 5-11. The outlet orifice having 190 mm diameter was tested and it was understood that the diameter should be increased. Positive results were obtained with a diameter of 300 mm.

Maximum heads are in acceptable levels and vapor/air volume decreases in the pipeline profile that could be controlled from Figure 5-13. Cavitation which means negative pressure is still observed but the air volume inside the pump discharge line is drastically reduced that could be seen from Figure 5-14 and Table 5-12. It is also observed that maximum head and maximum air volume occur at almost same elevations that is in the Table 5-12. In addition, maximum head is decreased in certain percentage given in Table 5-13. However, after 810.7 m, minimum heads drop below pipeline profile. In hydropneumatic tank, air volume fluctuates between 8 m^3 to 4.5 m^3. In addition, liquid flow variations are between 3.25 m^3/s to 4.75 m^3/s that is shown in the Figure 5-15.

Table 5-11: Air Chamber (10 m^3) Specifications

Air Chamber Specifications	Magnitude
Elevation	34 m
Volume	10 m^3
Initial Liquid Volume	4 m^3
Initial Gas Volume	5.5 m^3
Chamber Outlet Orifice Diameter	300 mm
Minor Loss Coefficient (Outflow)	2.5
Ratio of Losses (Inflow/Outflow)	2.5
Distance to Pump Station	23.63 m

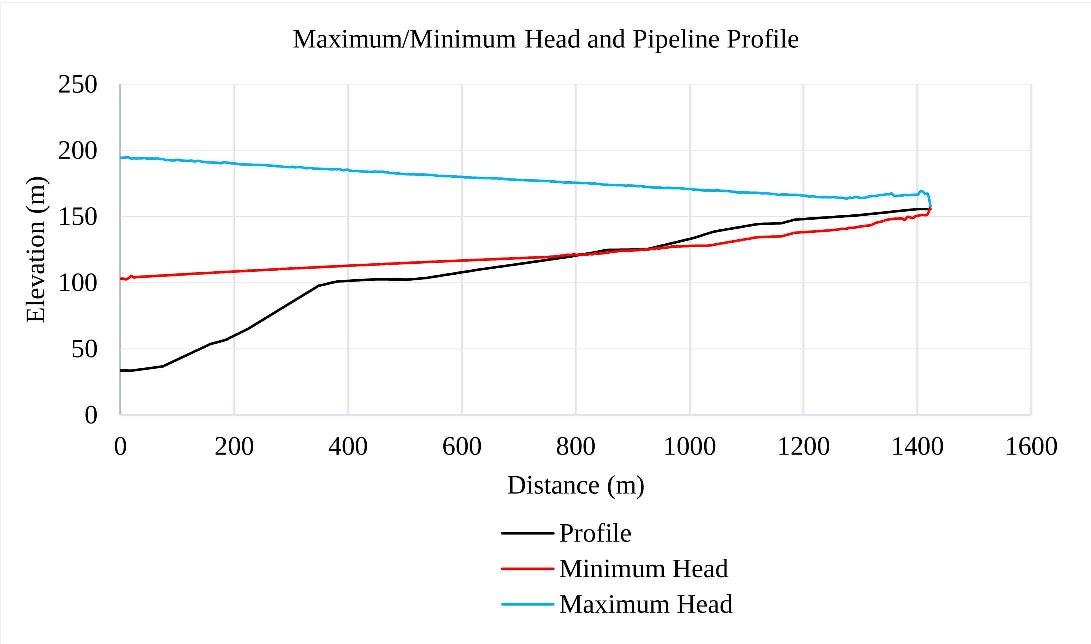


Figure 5-13:Maximum and Minimum Heads in the Air Chamber (10 m^3)

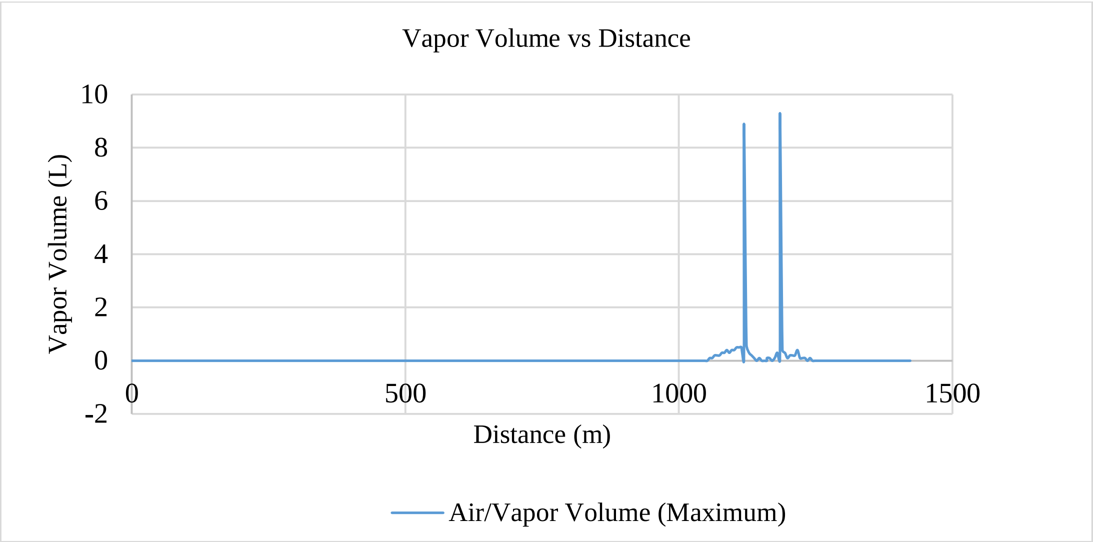


Figure 5-14: Air Volume in the Pipeline (10 m^3)

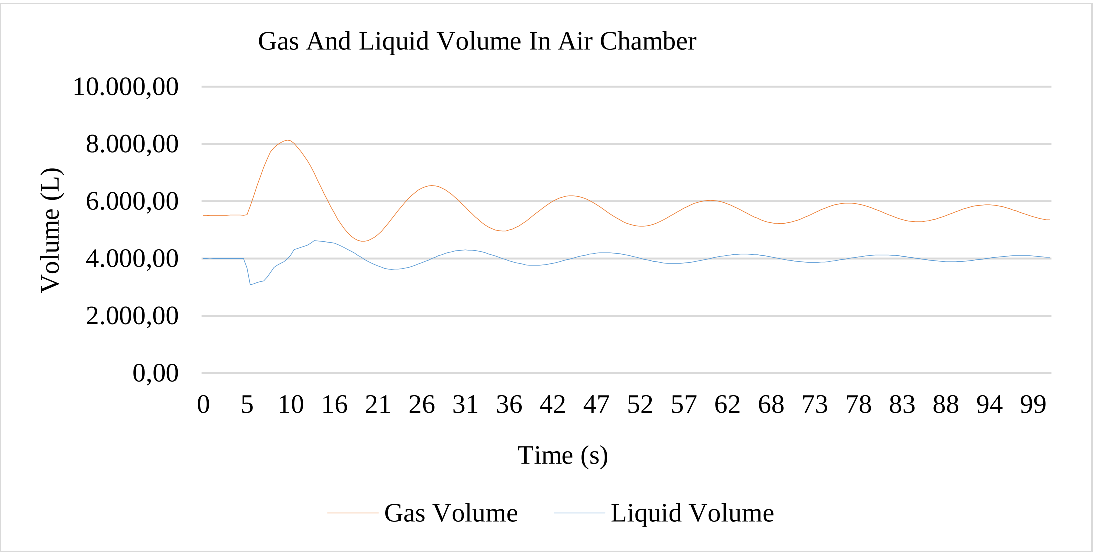


Figure 5-15: Gas and Liquid Volume Variations in the Air Chamber (10 m^3)

Table 5-12: Comparison of Air Chamber (10 m^3) with No Protection Case

Properties	Magnitude	Elevation
Max Head- No Protection	368.5 m	34.1 m
Max Head -10 m^3 Air Chamber	194.6 m	33.6 m
Max Air Vol.- No Protection	420 L	144.3 m
Max Air Vol.-10 m^3 Air Chamber	9.3 L	147.7 m

Table 5-13: Head and Air Volume Decrease in the Pipeline with Air Chamber (10 m^3 and 6.5 m^3)

Changing Parameter	Decrease in Percentage (10 m^3)	Decrease in Percentage (6.5 m^3)
Max Head	%47.19	%44.3
Max Air Volume	%97.7	%75.7

In general, maximum head and max vapor volume decreases in significant amount when 10 m^3 air chamber placed to the pump discharge line. However, negative pressures are seen in the last 420 m of the pipeline. In order to avoid negative pressure problem, firstly air chamber having a volume of 10 m^3 is simulated in different location according to Wang et al. (2019) calculation.

5.5.2.1. Air Chamber Having a Volume of 10 m^3 in Different Location

With help of Eq.(78) that is formulated by Wang et al (2019), location of air chamber is determined. Pipeline characteristic which was used in calculation is given in the Table 5-14.

As it can be seen from Figure 5-16 and 5-17, there is no significant difference when air chamber is located 160.3 m away from the pump station. In addition, maintenance and control could be done more easily when location is close to the pump station.

Table 5-14: Hydraulic Characteristic of the Pipeline

Pipeline Characteristic	Magnitude
Downstream Water Level	158 m
Upstream Water Level	39.4 m
Pipe Centerline Elevation	115.1 m
Pipe Centerline Distance	711,69 m
Pump Head	126 m
Total Pipe Distance	1433 m
Flow	1150 l/s

from Figure 3-7;

$\sin\alpha=0,005$

$\sin\beta=0,06$

$\cos\alpha=1$

$\cos\beta=0,998$

Total distance is calculated as 160,3 m away from the pump station from Eq. (78) and Figure 3-7.

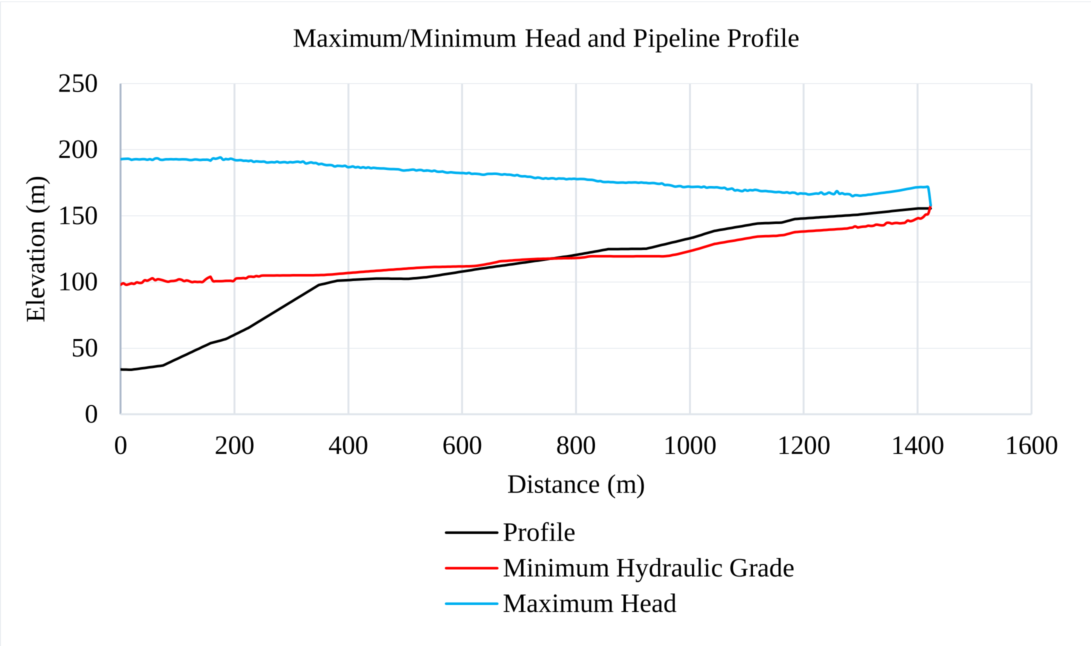


Figure 5-16: Maximum and Minimum Heads in the Air Chamber (10 m^3)

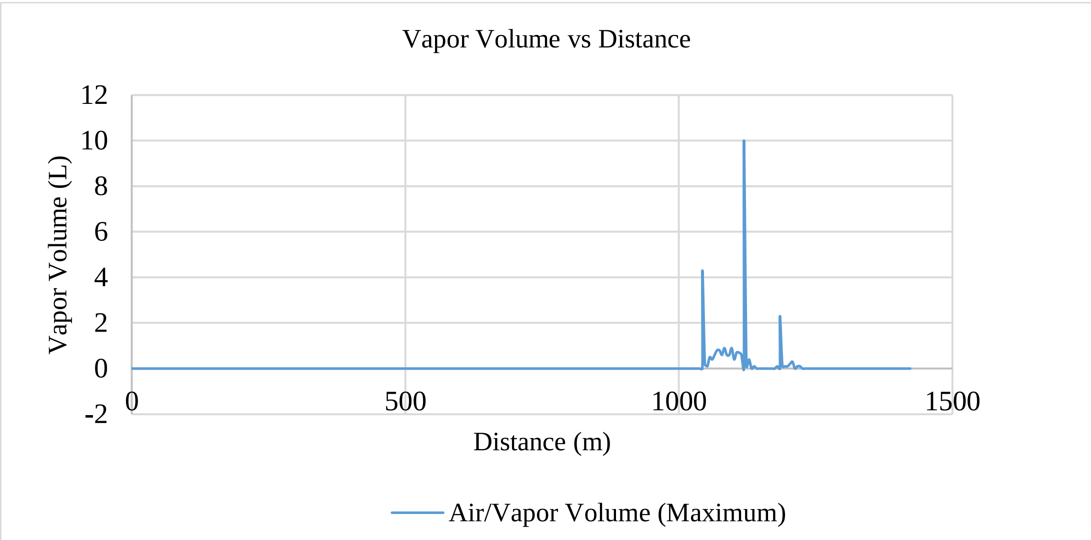


Figure 5-17: Air Volume in the Pipeline (10 m^3) according to Eq. 26

Wang et al. (2019) studied for long distance pipe and this pipeline could be categorized as short distance pipe according to their research. In their research, pump discharge line with a length of 11.4 km was examined and this pipe line has a length of 1.4 km. Thus, it is concluded that the best location is the nearest location to the pump station for these short distance pipelines. For this confined line, increasing air chamber volume must be done.

5.5.3. Air Chamber with a Volume of 12 m^3

12 m^3 air chamber is selected according to the following design considerations. h_{max}=225 m

h_{min}=85 m

$K = \frac{cC_0}{nALv_0}$=5 is determined from the $\bar{h}_f$=0 (Figure 3-2)

n=1

C_0=6.67 m^3 Initial Air Volume

$C' = 6.67\ \text{m}^3 * \left(\frac{130.07\ m+10\ m}{85\ m+10\ m}\right)^{1/1}$=9.84 m^3

V= 1.2*C$'$=12 m^3 Air chamber volume

Air chamber specifications are given in Table 5-15. Initial air volume is found as 6.67 m^3. Trials were done in the Hammer software program and similar results were obtained by simulation that could be controlled from Table 5-15 and Figure 5-20. The outlet orifice diameter was determined as 300 mm and head loss ratio was determined according to Table 3-1.

Minimum heads drop below pipeline after 810.7 m distance from the pump station that could be checked from Figure 5-18. Thus, some improvement should be made to the pump discharge line. Liquid volume is between 3.5 m^3 to 5.1 m^{3}and air volume fluctuates from 9 m^3 to 5 m^3 shown in Figure 5-20 when transient condition occurs. Vapor volume is seen in the pump discharge line shown in Figure 5-19.

Table 5-15: Air Chamber Specifications (12 m^3)

Air Chamber Specifications	Magnitude
Elevation	34 m
Volume	12 m^3
Initial Liquid Volume	4.5 m^3
Initial Gas Volume	6.26 m^3
Chamber Outlet Orifice Diameter	300 mm
Minor Loss Coefficient (Outflow)	2.5
Ratio of Losses (Inflow/Outflow)	2.5
Distance to Pump Station	23.63 m

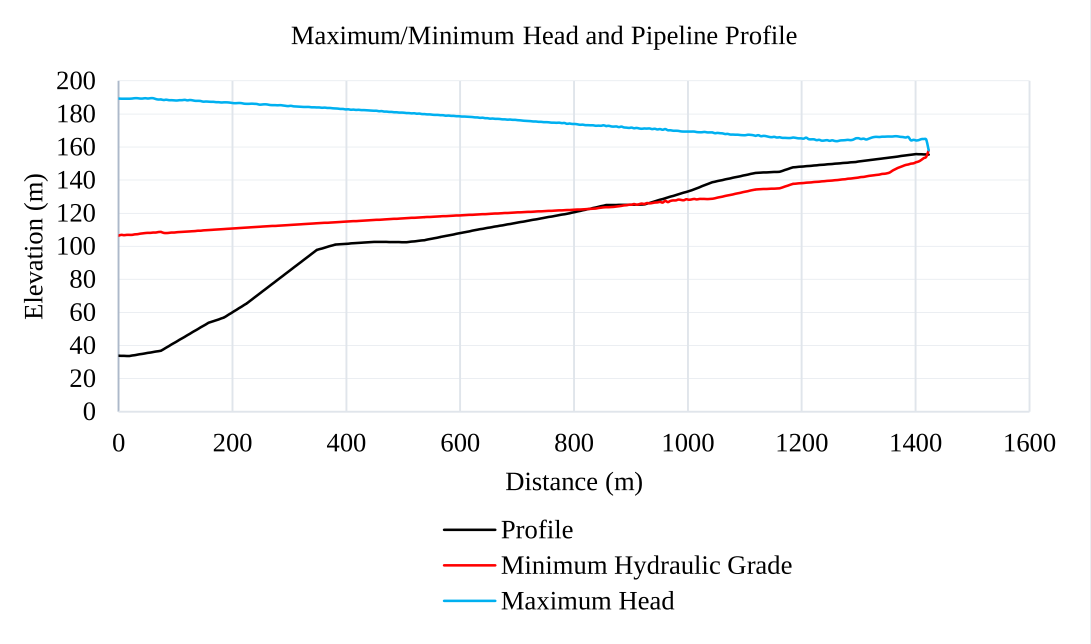


Figure 5-18: Maximum and Minimum Heads in the Air Chamber (12 m^3)

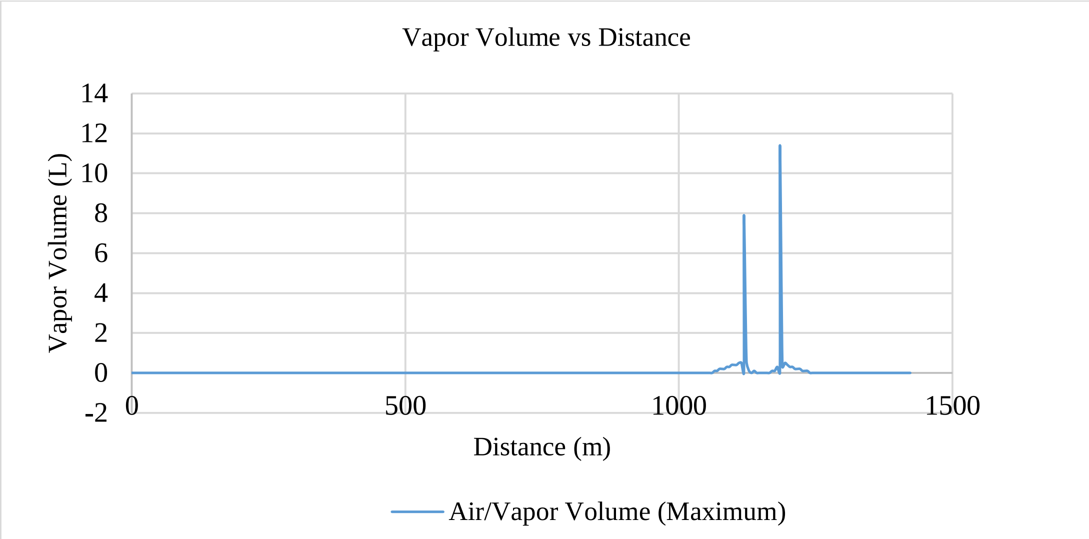


Figure 5-19: Air Volume in the Pipeline (12 m^3)

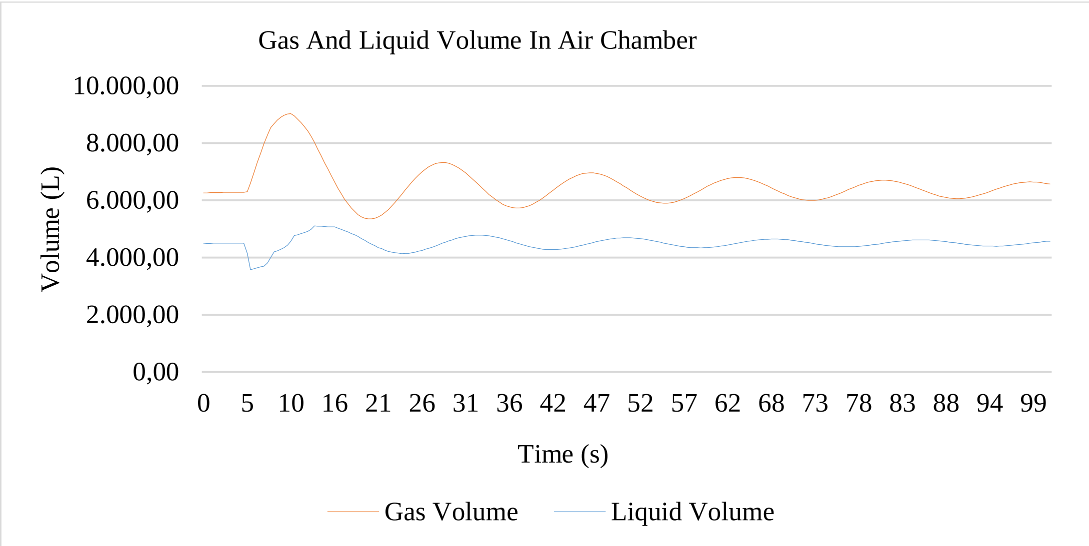


Figure 5-20: Gas and Liquid Volume Variations in the Air Chamber (12 m^3)

When Table 5-16 and Table 5-17 are examined, it is seen that maximum head in the pipeline decreased when it is compared with the air chamber having a volume of 10 m^3. However, air volume in the pump discharge line increased. The reason could be the increasing air volume inside the air chamber.

Table 5-16: Max Head and Max Air Volume in Pipeline W/Air Chambers

Properties	Magnitude	Elevation
Max Head- 12 m^3 Air Chamber	189.5 m	34.10 m
Max Head -10 m^3 Air Chamber	194.6 m	33.6 m
Max Air Vol.-12 m^3 Air Chamber	11.4 L	147.7 m
Max Air Vol.-10 m^3 Air Chamber	9.3 L	147.7 m

Table 5-17: Head and Air Volume Change with Air Chambers

Changing Parameter	Change in Percentage
Max Head (10 m^3 to 12 m^3)	%2.62 Decrease
Max Air Volume(10 m^3 to 12 m^3)	%22.58 Increase

To sum up, maximum head decreases and maximum air volume increases with increasing chamber volume. Increasing air volume in the pipeline could be suppressed by changing chamber outlet parameters. Thus, firstly tank inlet/outlet diameter was changed.

5.5.3.1. Air Chamber Having a Volume of 12 m^3 w/Changing Outlet Diameter

The diameter is increased to 400 mm to supply more water from the air chamber to hydraulic system. It is aimed to remove cavitation problem that is seen in 300 mm chamber outlet pipe diameter. All parameters are given in Table 5-18. All parameters except diameter in Table 5-18 are the same with those of Table 5-15.

It is seen that negative pressures are still seen in the pipeline when Figure 5-21 is examined. However, no air volume is observed in the pump discharge line (see Figure 5-22). The reason is that negative pressures do not reach to the water vapor pressure. When Figure 5-23 and Figure 5-25 are examined, more air and more water are supplied to the pipeline from the air chamber. However, as it could be understood from the Figure 5-24, maximum head increased in the pump discharge line.

Table 5-18: Air Chamber Specifications (12 m^3) W/ 400 mm Outlet

Air Chamber Specifications	Magnitude
Elevation	34 m
Volume	12 m^3
Initial Liquid Volume	4.5 m^3
Initial Gas Volume	6.26 m^3
Chamber Outlet Orifice Diameter	400 mm
Minor Loss Coefficient (Outflow)	2.5
Ratio of Losses (Inflow/Outflow)	2.5
Distance to Pump Station	23.63 m

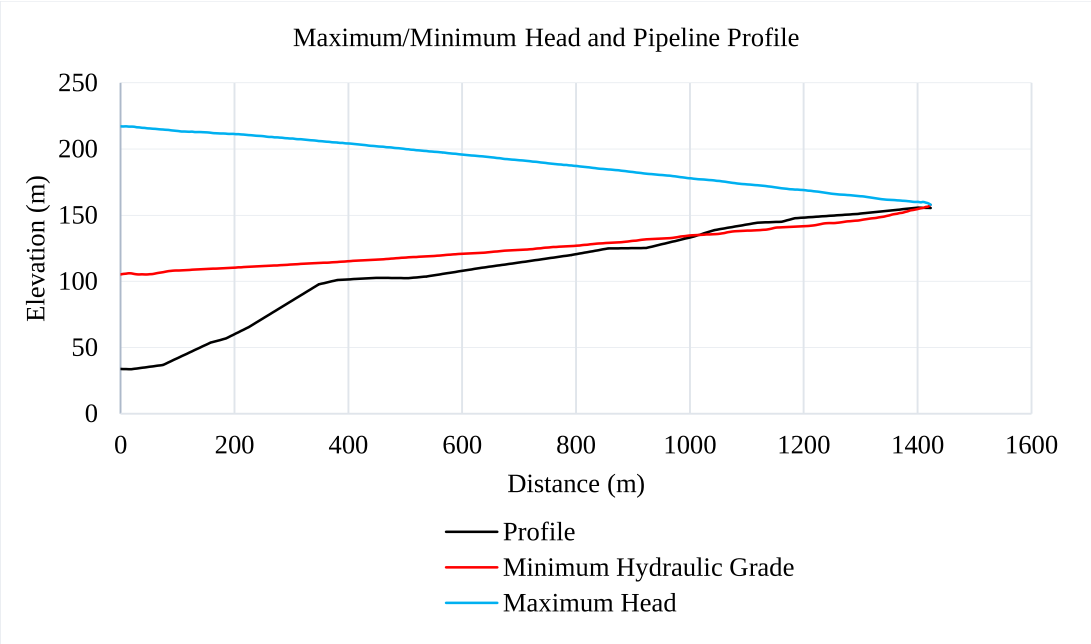


Figure 5-21: Maximum and Minimum Heads in the Air Chamber (12 m^3) W/400 mm orifice

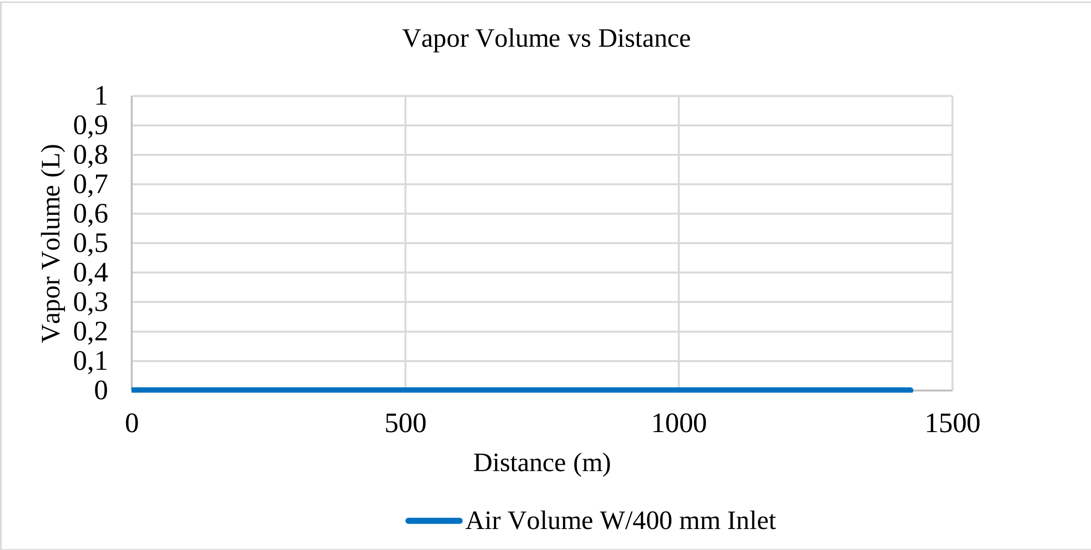


Figure 5-22: Air Volume in the Pipeline (12 m^3) W/400 mm Diameter

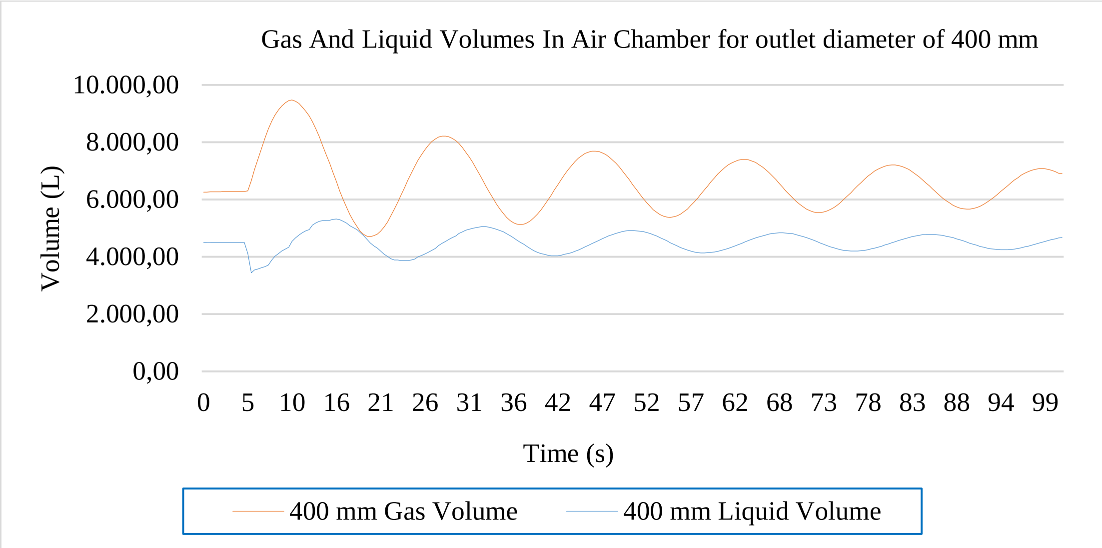


Figure 5-23: Gas and Liquid Volume Variations in the Air Chamber (12 m^3) w/ 400 mm Diameter

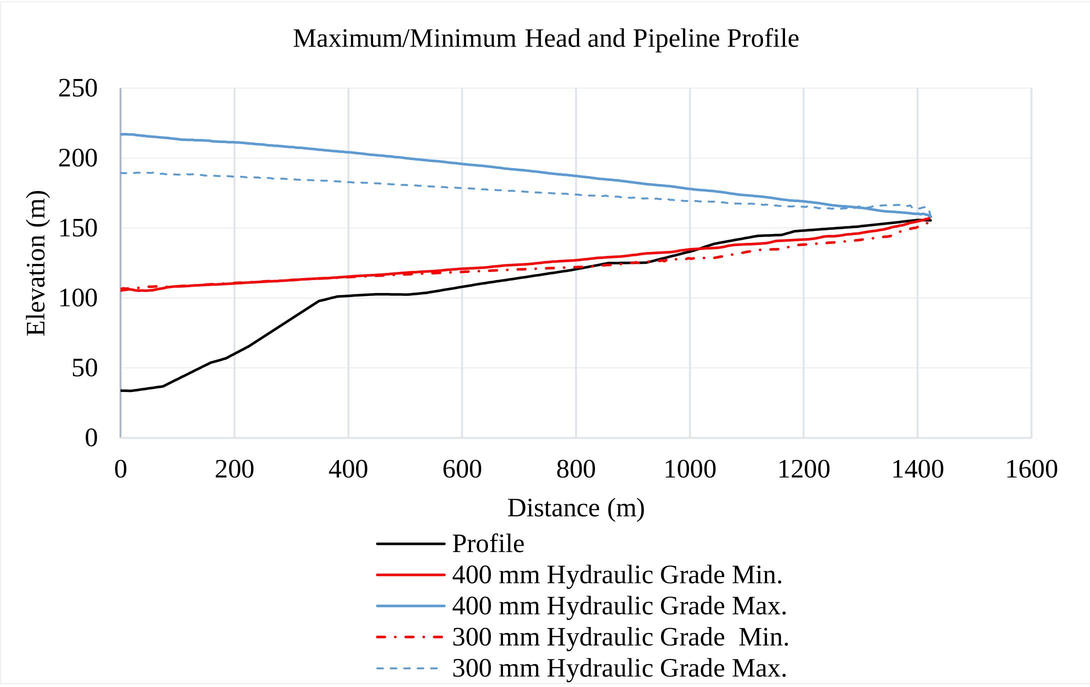


Figure 5-24: Maximum and Minimum Head Comparison w/ Different Diameters

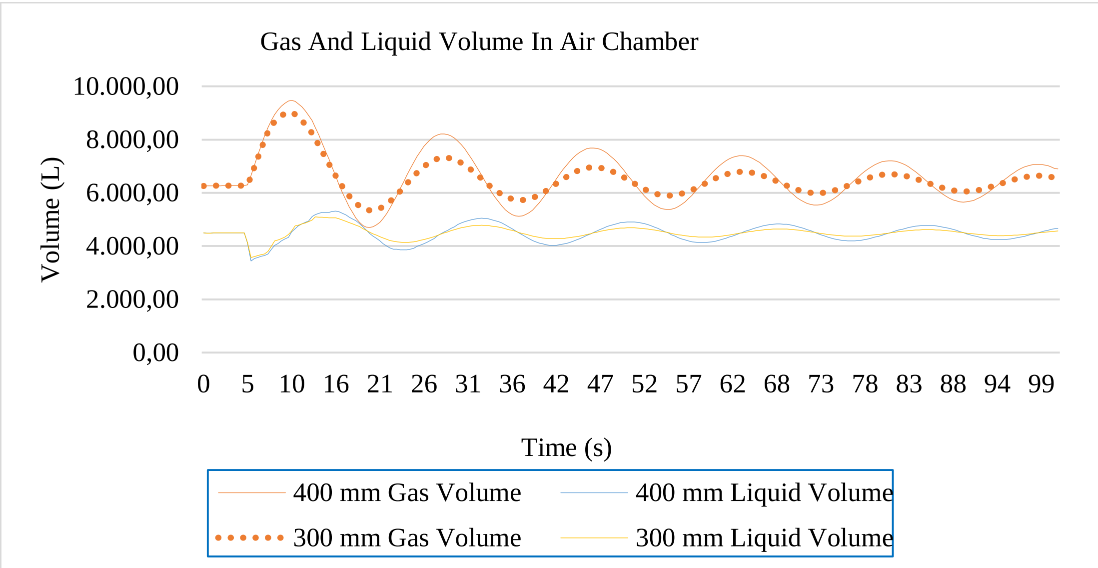


Figure 5-25: Comparison of gas and liquid volumes with various outlet diameters from the chamber

When orifice diameter is increased, cavitation problem is removed. Negative pressures are still seen from 1020,4 m to 1433 m. However, no vapor volume is seen in hydraulic system because negative pressures do not reach the water vapor pressure. On the other hand, maximum head increases up to 217 m. Thus, increasing inlet head loss and decreasing outlet head loss was simulated.

5.5.3.2. Air Chamber Having a Volume of 12 m^3 W/ Different Head Loss Coefficients

In this case, outlet orifice head loss coefficient is increased from 2.5 to 15 and outlet coefficient is decreased from 2.5 to 0.5. It is aimed to decrease maximum head and increase minimum head in the pump discharge line system. Air chamber specifications are given in Table 5-19.

When Figure 5-26 is examined, no remarkable change is seen between this case and previous case in both minimum and maximum head. The meaning of coefficients in the HAMMER are given in Table 5-20 and Table 5-21. When coefficient and ratio are defined for inflow or outflow, their products are used in the calculation.

Table 5-19: Air Chamber Specifications (12 m^3) w/ Different Head Losses

Air Chamber Specifications	Magnitude
Elevation	34 m
Volume	12 m^3
Initial Liquid Volume	4.5 m^3
Initial Gas Volume	6.26 m^3
Chamber Outlet Orifice Diameter	400 mm
Minor Loss Coefficient (Outflow)	0.5
Ratio of Losses (Inflow/Outflow)	15
Distance to Pump Station	23.63 m

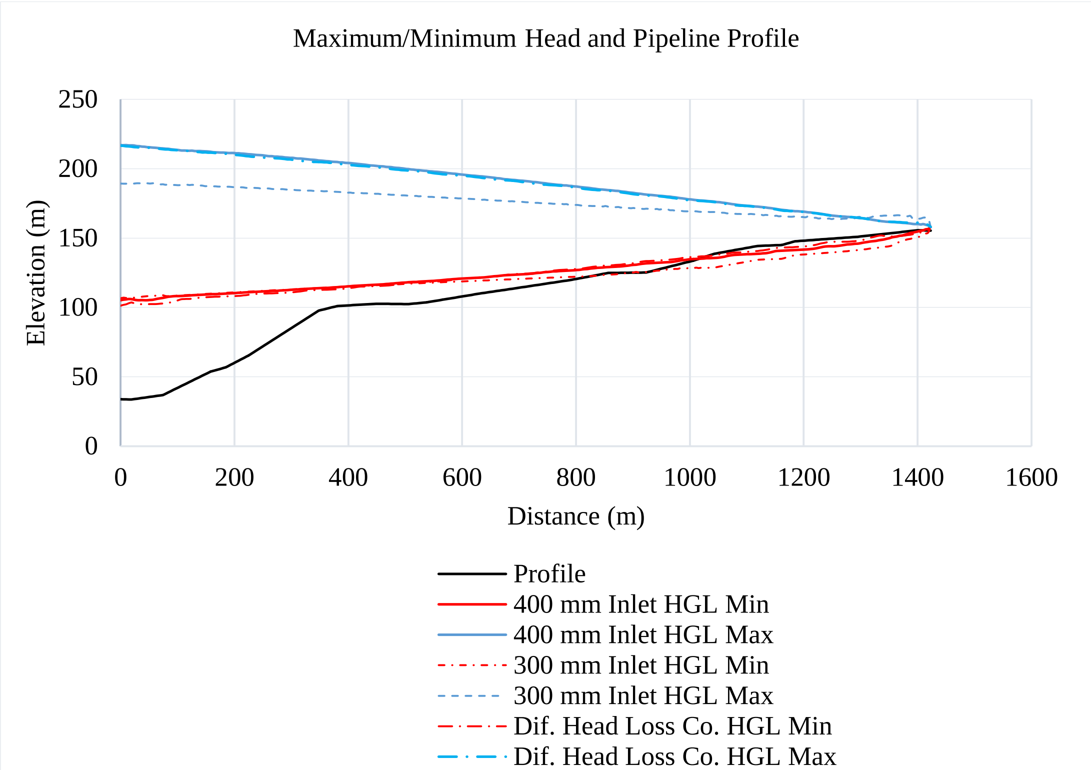


Figure 5-26: Maximum and Minimum Head Comparison of Air Chamber with Different Specifications

Table 5-20: Minor Loss Coefficients As Selected 2.5

Head Loss Coefficient	**Magnitude**
Minor Loss Coefficient (Outflow)	2.5
Ratio of Losses (Inflow/Outflow)	2.5
Minor Loss Coefficient (Inflow)	6.25

Table 5-21: Minor Loss Coefficient as selected 0.5 for Outflow and 15 for Ratio

Head Loss Coefficient	**Magnitude**
Minor Loss Coefficient (Outflow)	0.5
Ratio of Losses (Inflow/Outflow)	15
Minor Loss Coefficient (Inflow)	7.5

5.5.4. Air Chamber with a Volume of 15 m^3

$K = \frac{cC_0}{nALv_0}$=5 is determined from the $\bar{h}_f$=0 chart;

n=1.4

C_0=9.34 m^3 Initial Air Volume

$$C' = 9.34 \text{ m}^3 * \left(\frac{130.07\ m + 10\ m}{85\ m + 10\ m}\right)^{1/1} = 12.33 \text{ m}^3$$

V= 1.2*C$'$=14.8 m^3 Air chamber volume

It was seen that 8 m^3 initial air volume gave optimum results. Thus, initial air volume was decreased from 9 m^3 to 8 m^3. Other parameters that were used in the simulation are given in the Table 5-22.

When Figure 5-27 and Figure 5-28 are examined, it is seen that maximum head decreased in a certain amount. However, negative pressure and cavitation problem were still seen in the pump discharge line as it could be checked from Figure 5-29. In addition, liquid volume and gas volume variations are given in Figure 5-30.

Table 5-22: Air Chamber Specification (15 m^3)

Air Chamber Specifications	**Magnitude**
Elevation	34 m
Volume	15 m^3
Initial Liquid Volume	6 m^3
Initial Gas Volume	8 m^3
Chamber Outlet Orifice Diameter	300 mm
Minor Loss Coefficient (Outflow)	2.5
Ratio of Losses (Inflow/Outflow)	2.5
Distance to Pump Station	23.63

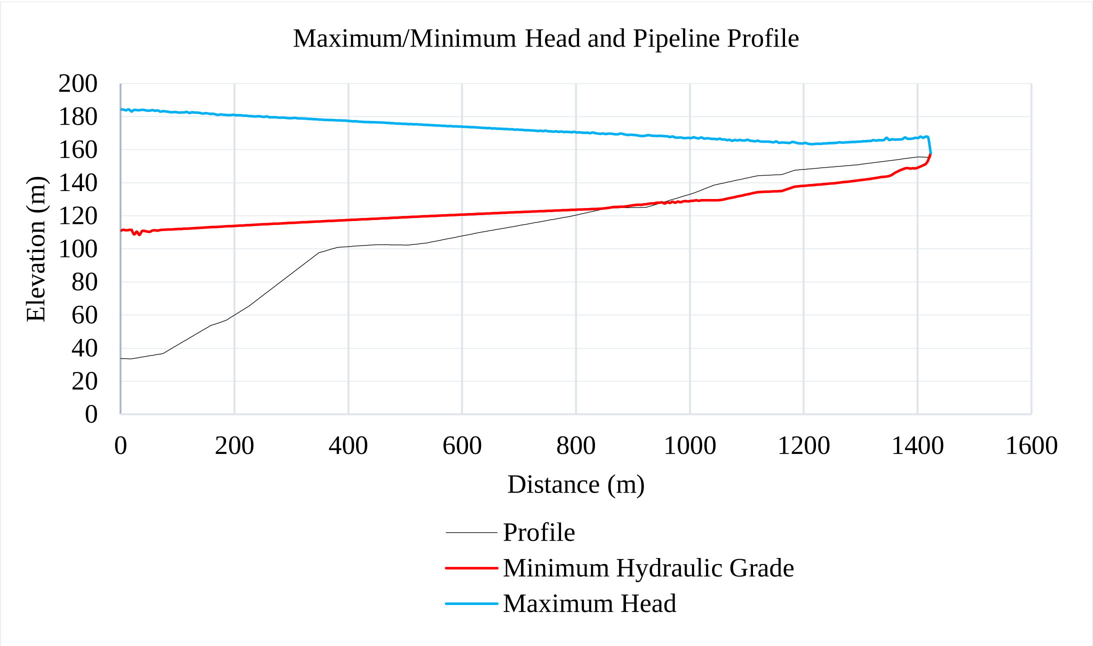


Figure 5-27: Maximum and Minimum Heads in the Air Chamber (15 m^3)

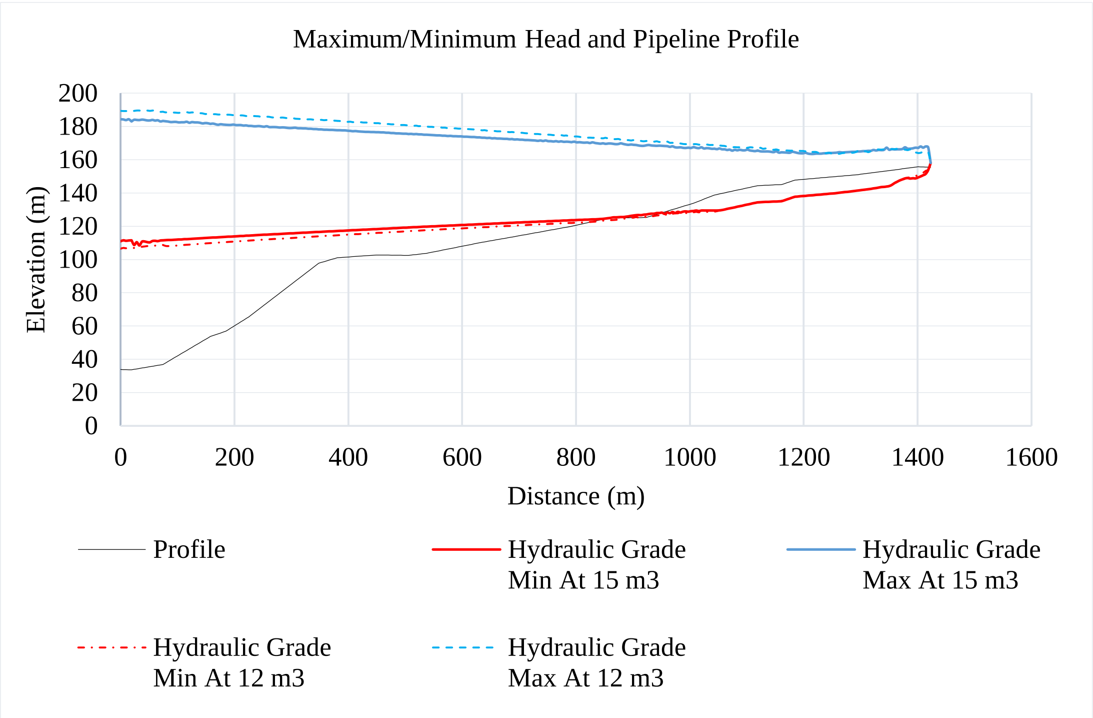


Figure 5-28: Comparison of Air Chambers with a volume of 15 m^3 and 12 m^3

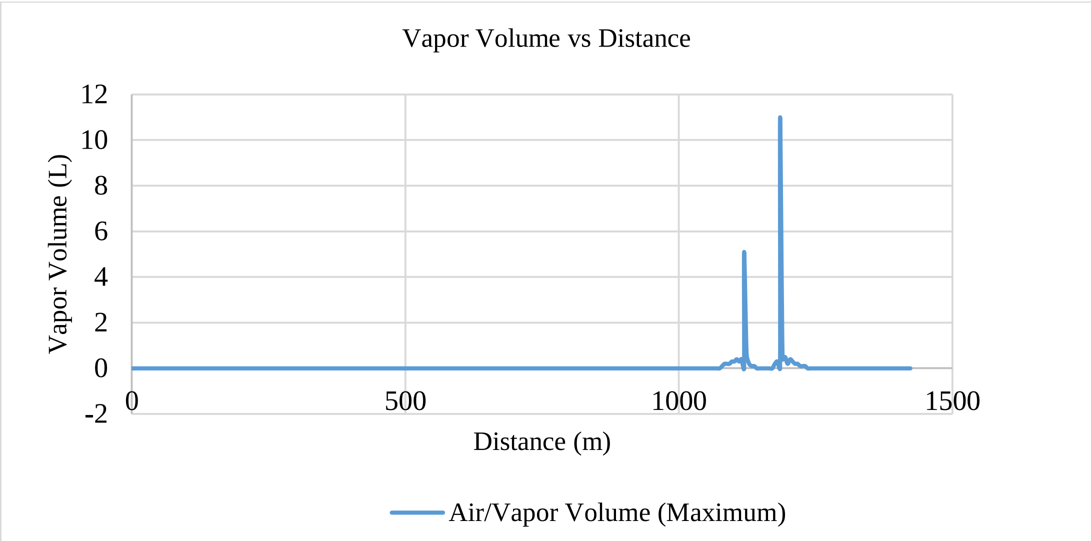


Figure 5-29: Air Volume in the Pipeline (15 m^3)

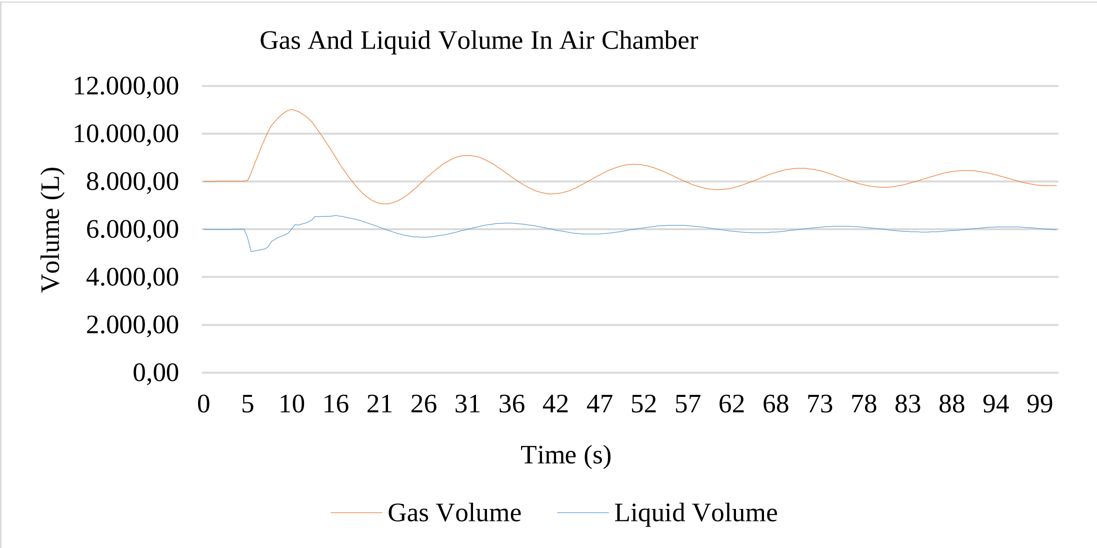


Figure 5-30: Gas and Liquid Variations in the Air Chamber (15 m^3)

Maximum hydraulic grade is 184,30 m. However, air volume is seen at 1185 m and 1119,2 m distance, that is, there is a cavitation problem in the confined line. After 951,1 m distance, hydraulic grade line falls below pipeline elevations. Thus air chamber volume was increased to 20 m^3.

5.5.5. Air Chamber with a Volume of 20 m^3

$K = \frac{cC_0}{nALv_0}$=7 is determined from the $\bar{h}_f$=0 chart;

n=1.4

C_0=12.15 m^3 Initial Air Volume

$C' = 12.15 \text{ m}^3 * \left(\frac{130.07\ m+10\ m}{85\ m+10\ m}\right)^{1/1}$=16.25 m^3

V= 1.2*C'=20 m^3 Air chamber volume

Initial diameter is increased from 300 mm to 370 mm. This increased diameter and all other possible diameters that are in the range given in Table 3-3 was simulated for all cases presented here. However, only this case gave positive results. Air volume inside air chamber was determined as 10 m^3 after trials. Air chamber specifications were given in Table 5-23.

As it could be observed from Figure 5-32, no air volume is observed in the pump discharge line because negative pressures do not reach to the vapor volume. Change in minimum head could be observed when Figure 5-31 is compared with Figure 5-24 or Figure 5-28. Moreover, volume variations inside the air chamber is given in Figure 5-33.

In 20 m^3 case, minimum hydraulic grade falls below pipeline profile after 1047,7 m. However, no vapor volume is observed in transient results because head difference falls approximately 6 m below the pipeline at maximum difference and this means negative pressures do not reach vapor pressure. Moreover, maximum head that is observed in the pipeline is 193 m. Results are summarized in the Table 5-24.

Table 5-23: Air Chamber Specifications (20 m^3)

Air Chamber Specifications	Magnitude
Elevation	34 m
Volume	20 m^3
Initial Liquid Volume	9 m^3
Initial Gas Volume	10 m^3
Chamber Outlet Orifice Diameter	370 mm
Minor Loss Coefficient (Outflow)	2.5
Ratio of Losses (Inflow/Outflow)	2.5
Distance to Pump Station	23.63 m

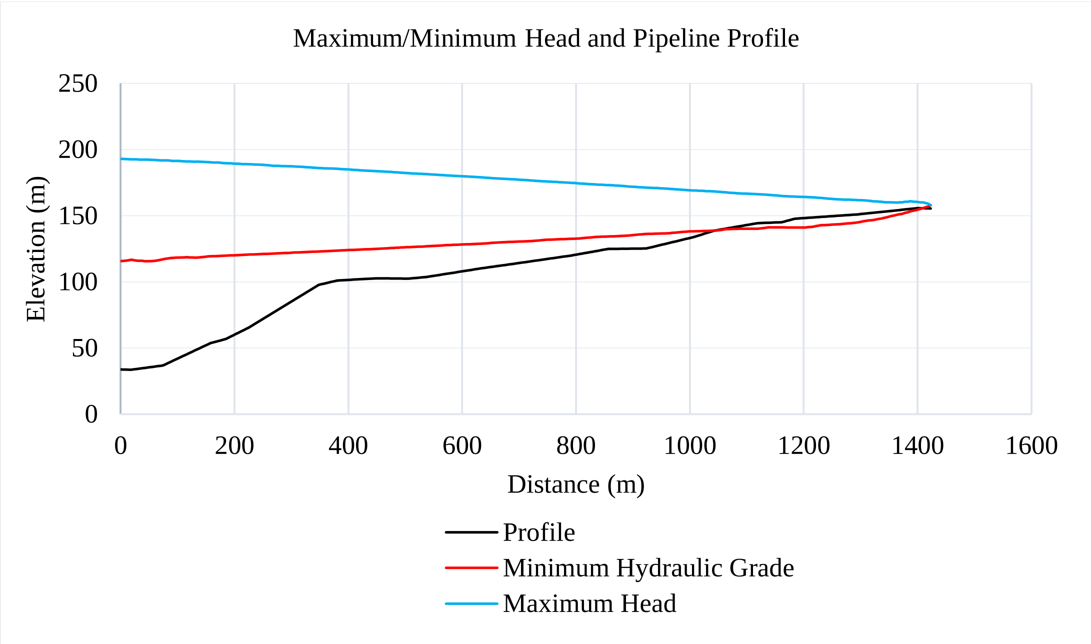


Figure 5-31: Maximum and Minimum Heads in the Air Chamber (20 m^3)

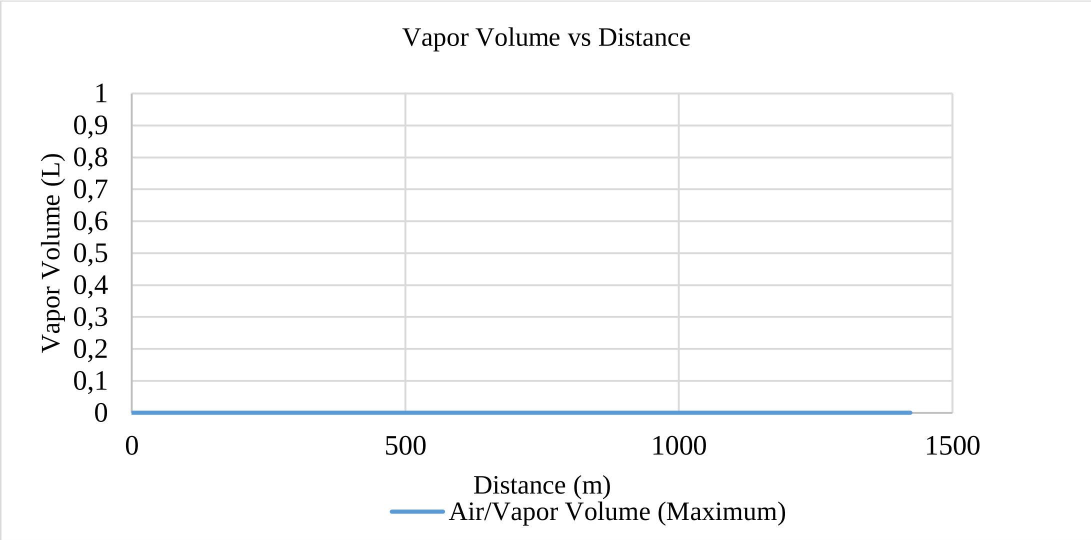


Figure 5-32: Air Volume in the Pipeline (20 m^3)

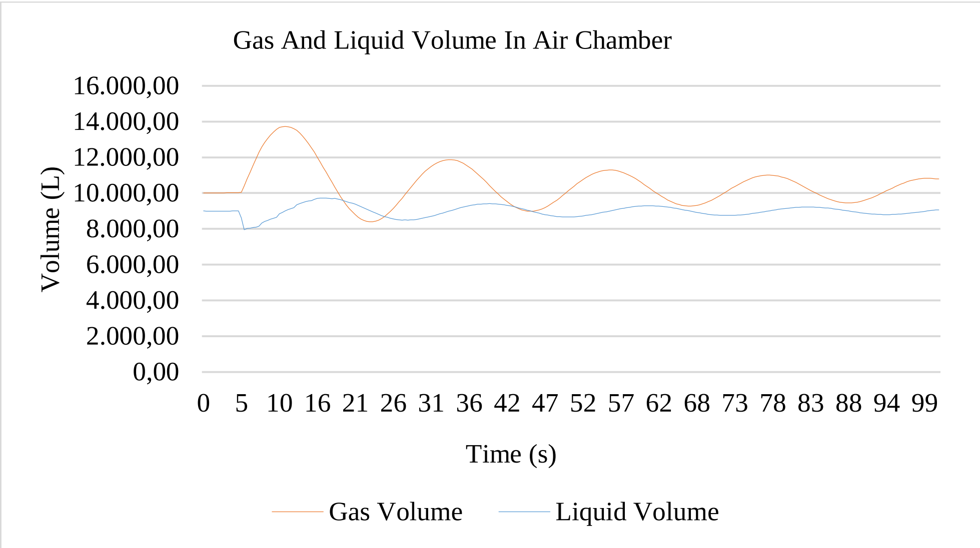


Figure 5-33: Gas and Liquid Variations in the Air Chamber (20 m^3)

Table 5-24: Max Head and Max Air Volume for All Air Chambers

Properties	Magnitude	Elevation
Max Head- 20 m^3 Air Chamber	193 m	33,8 m
Max Head- 12 m^3 Air Chamber	189,5 m	34,10 m
Max Head -10 m^3 Air Chamber	194,6 m	33,6 m
Max Air Vol.-20 m^3 Air Chamber	-	-
Max Air Vol.-12 m^3 Air Chamber	11,4 L	147,7 m
Max Air Vol.-10 m^3 Air Chamber	9,3 L	147,7 m

It is concluded that there is a remarkable decrease in maximum head and cavity form with help of the air chamber. However, single air chamber is inadequate against negative pressures in some elevations. Combination measures should be evaluated.

5.6. Combination of an Air Valve and Air Chamber

It is seen that maximum air volume is mostly seen at 144.34 m and 147.7 m elevations in all cases. It is determined that 147.7 m elevation gives better results. Thus, inserting air valve into the pipeline at 147.7 m elevation is evaluated. Vacuum breaker type air valve is used. Two cases are examined.

5.6.1. Air Chamber Having a Volume 15 m^3 with an Air Valve

Location of the air valve; air chamber and air valve specifications are given in Table 5-25 and Table 5-26.

It is seen that results show some improvements against negative pressure. However, negative pressures are still seen in the pipeline that could be checked from Figure 5-34 and Figure 5-35. Thus, air chamber volume was increased to 20 m^3.

Table 5-25: Air Valve Specification

Air Valve Specifications	Magnitude
Elevation	147.7 m
Distance from Pump Station	1185 m
Orifice Diameter	200 mm(Stephenson, D., 1997)

Table 5-26: Air Chamber Specification (15 m^3)

Air Chamber Specifications	Magnitude
Elevation	34 m
Volume	15 m^3
Initial Liquid Volume	6 m^3
Initial Gas Volume	8 m^3
Chamber Outlet Orifice Diameter	300 mm
Minor Loss Coefficient (Outflow)	2.5
Ratio of Losses (Inflow/Outflow)	2.5
Distance to Pump Station	23,63 m

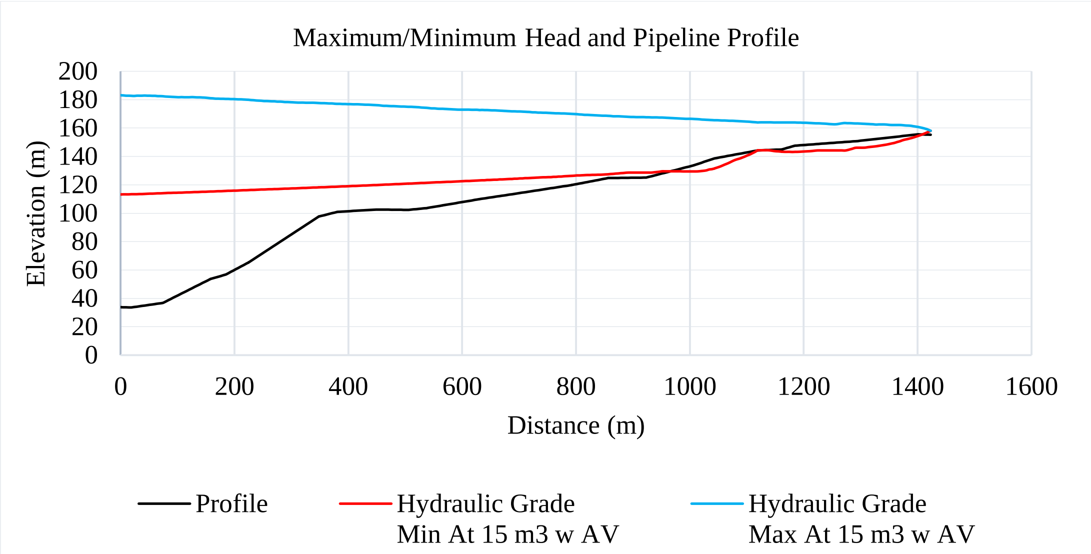


Figure 5-34: Maximum and Minimum Head Air Valve with Air Chamber Having Volume 15 m^3

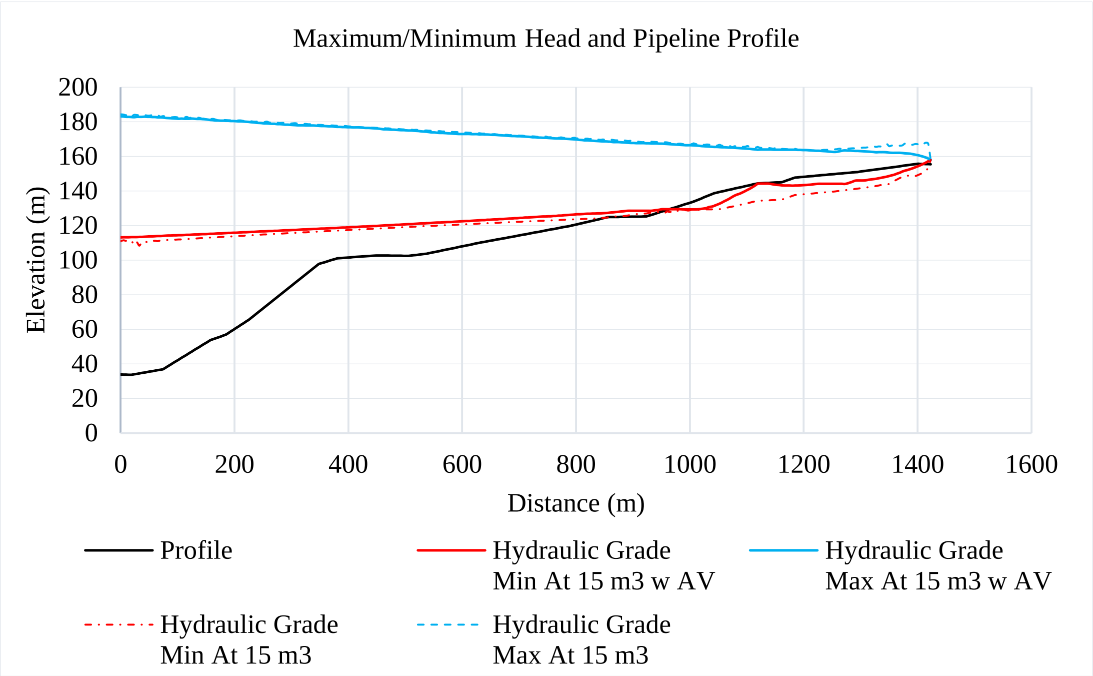


Figure 5-35: Maximum and Minimum Head Comparison

5.6.2. Air Chamber Having a Volume 20 m^3 with an Air Valve

Air valve and air chamber specifications are given in Table 5-27 and Table 5-28.

It is noted that this case gives favorable results when maximum flow is supplied from the dam. No negative pressure is seen and all pressures are in reasonable levels that could be observed from Figure 5-36. Furthermore, highest pressure observed in the pipeline is 188.7 m which means %48.79 decrease.

Table 5-27: Air Valve Specifications

Air Valve Specifications	Magnitude
Elevation	147,7 m
Distance from Pump Station	1185 m
Orifice Diameter	200 mm(Stephenson, D., 1997)

Table 5-28: Air Chamber Specifications (20 m^3)

Air Chamber Specifications	Magnitude
Elevation	34 m
Volume	20 m^3
Initial Liquid Volume	9 m^3
Initial Gas Volume	10 m^3
Chamber Outlet Orifice Diameter	370 mm
Minor Loss Coefficient (Outflow)	2.5
Ratio of Losses (Inflow/Outflow)	2.5
Distance to Pump Station	23,63 m

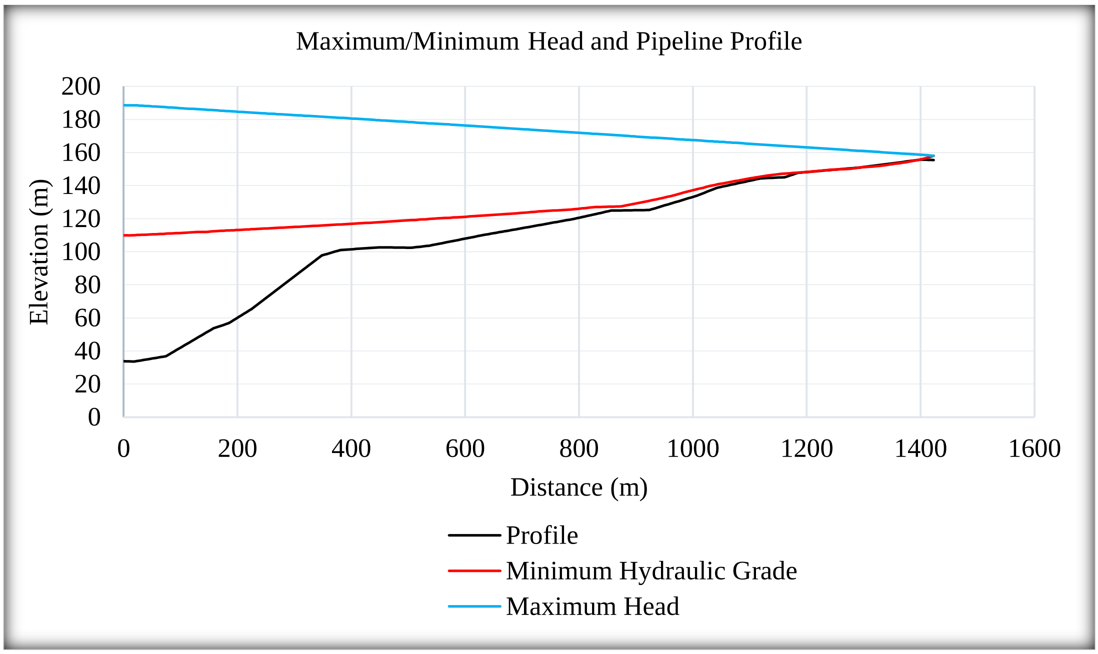


Figure 5-36: Maximum and Minimum Head with an Air Valve And Air Chamber(20 m^3)

In addition to minimum hydraulic grade line, that is, water level is considered as 39.24 m for Çokal Dam, maximum water level which is 79.18 m should be tested to see system response. It is observed that pressures are in acceptable values in all conditions. Air valve and air chamber having 20 m^3 volume gives best results up to now.

5.7. Combination of the Air Chamber and the One-Way Surge Tank

In this case, 12 m^3 air chamber is combined with the one-way surge tank. Trials are done via HAMMER.

Thorley (2004) concluded that optimum location is the vicinity of peak points. Kavurmacıoğlu and Karadoğan also claimed the same conclusion in their research. Peak points could be 144.34 m and 147.7 m elevations due to potential cavitation problems at those points. After trials, it is understood that the best location is 147.7 m elevation.

Wylie (1993) stated that the size of the one-way surge tank should be enough to fill voids that are formed due to liquid column separation. The worst case scenario which is unprotected case is evaluated and total of 3.835 m^3 air volume is seen in the pipeline (Figure 5-6). Sizing is done to suppress liquid column separation.

There is a check valve between the surge tank and pipeline. With the help of a check-valve, one-way surge tank could be used only against negative pressures. Thus, surge tank water elevation could be below water elevation in the pipeline.

The location of the one-way surge tank is 1119.2 m away from the pump station. One-way surge tank specification are given in Table 5-29 and air chamber specification is given in Table 5-30.

This combination is also simulated with maximum water level in dam. After simulation, it is concluded that one-way surge tank could be very effective against negative pressure (Figure 5-37). In addition, with help of surge tank, air chamber volume is reduced from 20 m^3 to 12 m^3. Moreover, maximum head decreases to 190.7 m which means %48.25 drop.

Table 5-29: One-Way Surge Tank Specifications

Surge Tank Specifications	Magnitude
Elevation (Maximum)	152.5 m
Elevation (Minimum)	151.3 m
Elevation (Base)	150 m
Height of Surge Tank	2.5 m
Diameter of Surge Tank	1.5 m
Diameter of Orifice	800 mm
Head Loss Coefficient	2.5

Table 5-30: Air Chamber Specification

Air Chamber Specifications	Magnitude
Elevation	34 m
Volume	12 m^3
Initial Liquid Volume	4.5 m^3
Initial Gas Volume	6.26 m^3
Chamber Outlet Orifice Diameter	300 mm
Minor Loss Coefficient (Outflow)	2.5
Ratio of Losses (Inflow/Outflow)	2.5
Distance to Pump Station	23.63 m

Total air volume is calculated as 3.835 m^3 in the pipeline. To suppress this air volume, one-way surge tank is designed to store minimum 4 m^3 water. It is seen that 3.67 m^3 water is used from one-way surge tank.

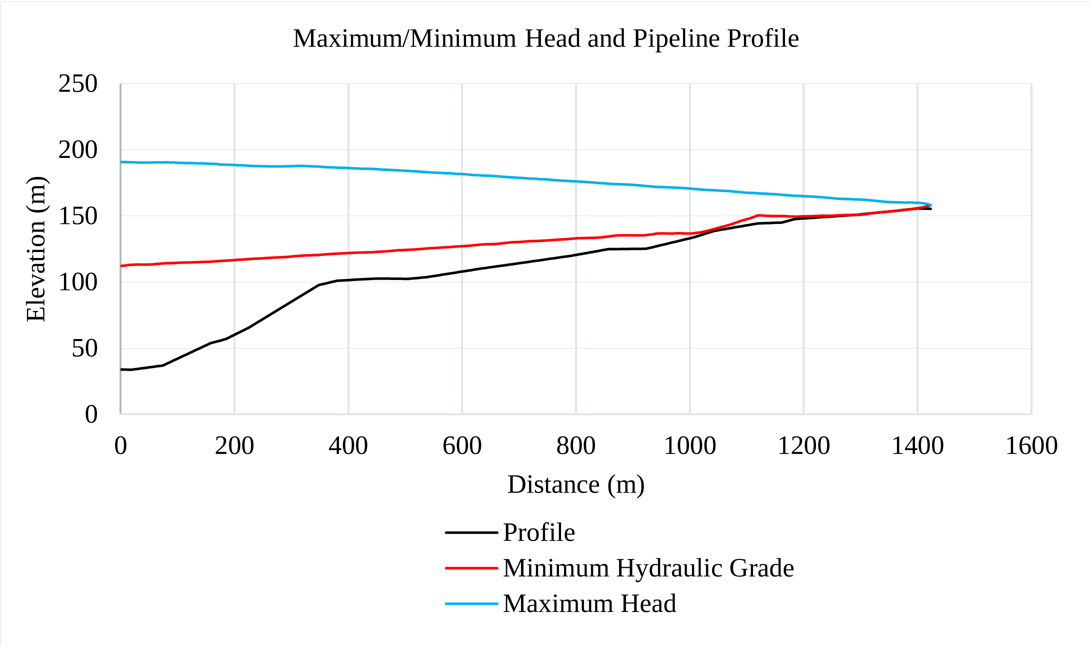


Figure 5-37: Maximum and Minimum Head Air Chamber and One-Way Surge Tank

5.8. Combination of the Air Chamber with Flywheel

Flywheel is used against negative pressure. Moreover, air chamber is a powerful device against high pressures. Combination of these device could be beneficial for both hydraulic and economic way. Air chamber having a volume of 7.5 m^3 is simulated with different moment of inertia. It is observed that 140 kg.m^2 gives effective results. Air chamber specifications are given in Table 5-31.

It is concluded that this combination is effective against positive and negative pressure. Maximum head decreases from 368.5 m to 187.1 m, that is, %49.22 decrease (see Figure 5-38). In addition, liquid column separation problem is prevented. No negative pressure is seen in the pipeline as it could be seen from Figure 5-39.

Table 5-31: Air Chamber Specifications (7.5 m^3)

Air Chamber Specifications	Magnitude
Elevation	34 m
Volume	7.5 m^3
Initial Liquid Volume	3.68 m^3
Initial Gas Volume	3.7 m^3
Chamber Outlet Orifice Diameter	250 mm
Minor Loss Coefficient (Outflow)	2.5
Ratio of Losses (Inflow/Outflow)	2.5
Distance to Pump Station	23.63 m

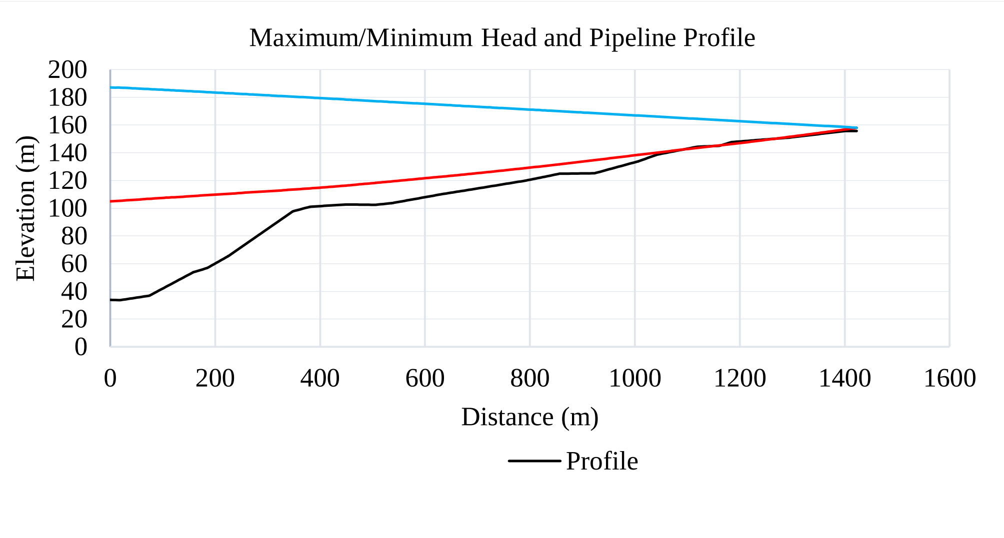


Figure 5-38: Maximum and Minimum Heads in the Pipeline w/Air Chamber (7.5 m^3) and Flywheel

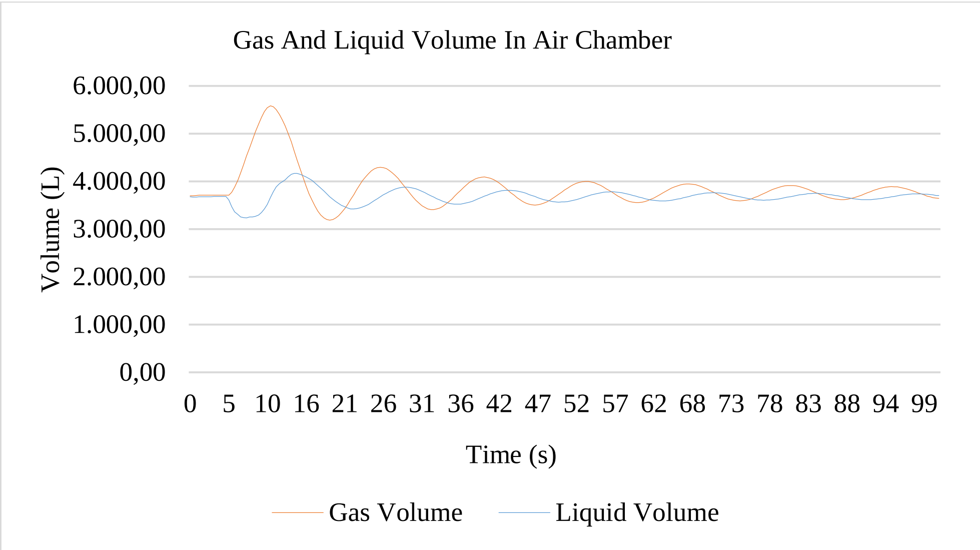


Figure 5-39: Liquid and Gas Variations in the Air Chamber (7.5 m^3)

After all alternatives are evaluated, following table is obtained.

Table 5-32: Performance Analysis of Protection Devices

Protection Device	Maximum Pressure (m)	Air Volume	Negative Pressure
The Unprotected Case	368.5	Observed	Observed
Flywheel (160 kg.m^2 total inertia)	216.7	Not Observed	Not Observed
6.5 m^3 Air Chamber	204.9	Observed	Observed
10 m^3 Air Chamber	194.6	Observed	Observed
12 m^3 Air Chamber	189.5	Observed	Observed
15 m^3 Air Chamber	184.3	Observed	Observed
20 m^3 Air Chamber	193	Not Observed	Observed
15 m^3 Air Chamber w/Air Valve	183.2	Observed	Observed
20 m^3 Air Chamber w/Air Valve	188.7	Not Observed	Not Observed
12 m^3 Air Chamber w/OWST	190.7	Not Observed	Not Observed
7.5 m^3 Air Chamber w/Flywheel	187.1	Not Observed	Not Observed

CHAPTER 6

CONCLUSIONS

In this study, a pump discharge line was examined with single and combined devices. Liquid column separation is observed and high pressures are seen in the unprotected form. Alternatives including flywheel, air chamber, air valve and one-way surge tank were simulated. System performance was tested during transient event that occurs due to power failure. Air chambers with different volume, outlet diameter (the diameter of the pipe segment supplying water from the air chamber into the main pipeline) and head loss coefficients are simulated.

Air chamber is used to suppress both negative and positive pressures. It is seen that inlet/outlet diameter ratio, initial air volume and initial liquid volume have an important effect on the transient behavior of the liquid. However, greater volume (chamber having a volume of 20 m^3 at least) which means higher cost could be needed when air chamber is chosen as a single protection device. Thus, it is concluded that single application is inadequate for that hydraulic system.

One-way surge tank, air valve, air chamber and flywheels are effective only against liquid column separation. Thus, single usage of them is unsatisfactory also.

Combined forms could have crucial advantages against water hammer. Last three alternatives, namely, 1) Air Chamber having a volume of 20 m^3 with the Air Valve, 2) Air Chamber having a volume of 12 m^3 with the One-Way Surge Tank and 3) Air Chamber having a volume of 7.5 m^3 with the Flywheel have satisfactory results against water hammer. Third combination gives the best results hydraulically. Moreover, air chamber volume has the smallest value in that alternative. Other two alternatives could also be considered. However, in those alternatives roughly twice or three times bigger chamber volume is needed when compared with the last alternative. Moreover, one-way surge tank or air valve at high elevations must be used. Because of that they need regular maintenance and their locations would be difficult to access. In addition, cold climates could be problematic due to freezing conditions in those elevations.

Overall, protecting pump discharge lines against water hammer should be well examined at the design stage. Different alternatives should be evaluated properly to ensure the system stability.

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