INVESTIGATION OF WATERHAMMER PROBLEMS IN THE PENSTOCKS OF SMALL HYDROPOWER PLANTS

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

BY

MELİH ÇALAMAK

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

SEPTEMBER 2010

Approval of the thesis:

INVESTIGATION OF WATERHAMMER PROBLEMS IN THE PENSTOCKS OF SMALL HYDROPOWER PLANTS

submitted by **MELİH ÇALAMAK** in partial fulfillment of the requirements for the degree of **Master of Science in Civil Engineering Department, Middle East Technical University** by,

Prof. Dr. Canan Özgen ____________________
Dean, Graduate School of **Natural and Applied Sciences**

Prof. Dr. Güney Özcebe ____________________
Head of Department, **Civil Engineering**

Assoc. Prof. Dr. Zafer Bozkuş ____________________
Supervisor, **Civil Engineering Dept., METU**

Examining Committee Members:

Assoc. Prof. Dr. İsmail Aydın ____________________
Civil Engineering Dept., METU

Assoc. Prof. Dr. Zafer Bozkuş ____________________
Civil Engineering Dept., METU

Assoc. Prof. Dr. Nuri Merzi ____________________
Civil Engineering Dept. METU

Asst. Prof. Dr. Mete Köken ____________________
Civil Engineering Dept. METU

Dr. Kutay Çelebioğlu ____________________
Su-Ener. Müh. ve Enerji Hiz.San.Ltd.Şti

Date: September 13, 2010

I hereby declare that all information in this document has been obtained and presented in accordance with academic rules and ethical conduct. I also declare that, as required by these rules and conduct, I have fully cited and referenced all material and results that are not original to this work.

Name, Last name : Melih Çalamak

Signature :

ABSTRACT

INVESTIGATION OF WATERHAMMER PROBLEMS IN THE PENSTOCKS OF SMALL HYDROPOWER PLANTS

ÇALAMAK, Melih

M.S., Department of Civil Engineering

Supervisor: Assoc. Prof. Zafer BOZKUŞ

September 2010, 138 Pages

Waterhammer is an unsteady hydraulic problem which is commonly found in closed conduits of hydropower plants, water distribution networks and liquid pipeline systems. Due to either a malfunction of the system or inadequate operation conditions, pipeline may collapse or burst erratically resulting in substantial damages, and human losses in some cases. In this thesis, time dependent flow situations in the penstocks of small hydropower plants are investigated. A software, HAMMER, that utilizes method of characteristics for solving nonlinear differential equations of transient flow is used in the study. In two case studies, various operation conditions such as load rejection, load acceptance and instant load rejection are studied. The parameters and situations affecting pressure and turbine speed rises are investigated. Computed and available measured values are found to be very close. Also, differences between waterhammer responses of the Francis and Pelton turbines are revealed. Finally, specific protective measures are suggested to either diminish and/or avoid the harmful effects of waterhammer problems in small hydropower plants.

Keywords: Hydraulic Transients, Run-of-River Plants, Penstocks, Protective Measures

ÖZ

KÜÇÜK HİDROELEKTRİK SANTRALLERİN CEBRİ BORULARINDA SU DARBESİ PROBLEMLERİNİN ARAŞTIRILMASI

ÇALAMAK, Melih

Yüksek Lisans, İnşaat Mühendisliği Bölümü

Tez Yöneticisi: Doç. Dr. Zafer BOZKUŞ

Eylül 2010, 138 Sayfa

Su darbesi genellikle hidroelektrik santrallerin, su dağıtım şebekelerinin ve sıvı boru hattı sistemlerinin kapalı iletim hatlarında oluşan, zamana bağlı olarak değişen bir hidrolik problemdir. Yetersiz işletme koşullarından ya da sistemdeki bir arızadan dolayı boru hattı büyük zararla ve bazı durumlarda can kaybıyla sonuçlanabilecek şekilde çökebilir ya da patlayabilir. Bu tezde küçük hidroelektrik santrallerin cebri borularındaki zamana bağlı değişen akış durumları araştırılmıştır. Çalışmada, zamana bağlı değişen akışın doğrusal olmayan diferansiyel denklemlerini çözmek için karakteristikler metodunu kullanan HAMMER adlı bir bilgisayar programı kullanılmıştır. İki adet örnek çalışmada yük atma, yük alma ve ani yük atma gibi çeşitli işletme koşulları çalışılmıştır. Basınç ve türbin hızı artışını etkileyen değişkenler ve durumlar araştırılmıştır. Hesaplanan ve mevcut olan ölçüm değerleri çok yakın bulunmuştur. Ayrıca, Francis ve Pelton türbinlerinin su darbesi tepkilerinin farklılıkları ortaya çıkarılmıştır. Son olarak, küçük hidroelektrik santrallerdeki zararlı su darbesi etkilerini azaltan ya da ortadan kaldıran özel koruyucu önlemler önerilmiştir.

Anahtar Kelimeler: Zamana Bağlı Akım, Nehir Santralleri, Cebri Borular, Koruyucu Önlemler

To my parents Selma & Yaşar

ACKNOWLEDGMENTS

The completion of this thesis could not have been finished without professional and emotional support that my professor, my family and my girlfriend provided to me throughout my graduate studies. I am greatly indebted to my thesis supervisor Assoc. Prof. Dr. Zafer Bozkuş who suggested the topic of my study and provided kindly guidance during the development of this thesis. With his invaluable assistance, he shared his wisdom with endless patience. It has been a pleasure working with him.

My dearest family deserves the special thanks. My mother and father have always been there every part of my life. I greatly appreciate their support, care, unconditioned love and trust in me. I am really lucky to have such parents. I feel blessed having my sister Müjgan in my life. She is always there for me, not only for this study, but for anything happened in my life. I also want to thank my brother Semih who shared the good and the bad in my life. He has always been my best friend.

There is one other person in my life, my beloved one, Ece Kınık who deserves the most special attention with her matchless, endless, invaluable support and her eternal love. She has shown the greatest patience during my studies by always standing by my side without a bit of hesitation.

Finally, I would like to thank faculty members, research assistants and administrative staff of METU Water Resources Laboratory for providing me a perfect working environment.

TABLE OF CONTENTS

LIST OF TABLES

TABLES

LIST OF FIGURES

FIGURES

LIST OF SYMBOLS

A	Cross sectional area of the pipe, [m^2]
a	Wave propagation velocity through the fluid, [m/s]
a_w	Opening of the wicket gate, [%]
D	Diameter of the pipe, [m]
D_g	Radius of gyration of rotating mass in a turbine and generator couple, [m]
D_r	Diameter of the turbine runner, [m]
E	Modulus of elasticity of the pipe, [N/m^2]
e	Wall thickness of the pipe, [m]
e_g	Generator efficiency, [%]
e_h	Hydraulic efficiency, [%]
e_T	Efficiency of the turbine, [%]
f	Darcy Weisbach friction factor
G	Weight of rotating parts of turbine and generator, [kg]
g	Gravitational acceleration, [m/s^2]
H	Pressure head in the pipe in steady state flow, [m]
H_g	Gross head, [m]
H_n	Net (effective) head, [m]
K	Bulk modulus of elasticity of the fluid, [N/m^2]
L	Length of the pipe, [m]
n	Speed of the turbine, [rpm]
n_f	Runaway speed of the turbine, [rpm]

n_s	Specific speed of the turbine, [rpm]
P	Installed capacity of a hydropower plant / Power output of a turbine, [W, kW, MW]
P	Pressure, [N/m^2]
Q	Discharge, [m^3/s]
Q_{tur}	Discharge flowing through the turbine, [m^3/s]
T	Time of closure or opening, [sec]
T_f	Circumferential tensile force per unit length, [N/m]
T_r	Wave reflection time, [s]
t	Time, [s]
V	Velocity, [m/s]
V_0	Initial velocity, [m/s]
V_f	Final velocity, [m/s]
γ	Unit weight of fluid, [N/m^3]
ΔA	Change in the cross sectional area of the pipe due to waterhammer, [m^2]
ΔH	Change in the pressure head in the transient conditions, [m]
ΔP	Change in the pressure in the transient conditions, [N/m^2]
Δs	The length of pipe extension due to waterhammer, [m]
ΔV	Change in the flow velocity in the transient conditions, [m/s]
$\Delta\rho$	Change in density of the fluid due to waterhammer, [kg/m^3]
η	Overall efficiency of the power plant, [%]
ρ	Density of the fluid, [kg/m^3]
σ_f	Maximum allowable tensile stress, [N/m^2]
τ_w	Shear stress, [N/m^2]
ϕ	Angle of the runner blade, [^{0}]

ABBREVIATIONS

CS	Control Surface
CV	Control Volume
Eq.	Equation
ESHA	European Small Hydropower Association
GRP	Glass Reinforced Plastic
HP(s)	Hydropower Plant(s)
kt eq. CO_2	Kiloton Equivalent Carbon Dioxide
MOC	Method of Characteristics
OLADE	Latin American Energy Organization
PID	Proportional Integral Derivative
PRV(s)	Pressure Relief Valve(s)
PVC	Polyvinyl Chloride
rpm	Revolution Per Minute
RWCT	Rigid Water Column Theory
SHP(s)	Small Hydropower Plant(s)
SNL	Speed No Load Gate Position
TWh	Terawatt Hour
UNIDO	United Nations Industrial Development Organization
uPVC	Unplasticized Polyvinyl Chloride
W	Watt

CHAPTER 1

INTRODUCTION

1.1 General

Generation of electricity has become a hot topic due to the growing population and increasing consumption of electricity since 1990s in Turkey. Although the energy generation has been increasing, the energy demand of Turkey has grown more rapidly throughout the years. In the past, to handle this energy gap, deficient energy policies have forced Turkey to import foreign energy supplies instead of using domestic, sustainable and renewable ones. However, from early 2000s Turkey has started promoting renewable energy sources with "hydropower". Especially, small hydropower has been considered as the most economical and clean energy source among the renewable energy alternatives. Therefore, to increase the hydropower share in the generation of electricity and encourage the private sector, several laws and regulations were published. In March 2001, Electricity Market Law No. 4628 was published and later the Water Usage Right Agreement was published in March 2003. These regulations attracted entrepreneurs to build and operate a hydropower plant with a license given by the Electricity Market Regulation Authority. Then, by the publication of the Energy Law No. 5346 in May 2005, government guaranteed to buy electricity from these entrepreneurs for the duration of 10 years. Also forest land acquisition for building of small hydropower plants is simplified. Besides, with the publication of the Law No. 5784 in July 2008, there is no need to get a license to generate electricity with small hydropower plants whose installed capacity is smaller than 0.5 MW (Küçükali and Barış, 2009).

All these promoting laws and regulations have been attracting hundreds of entrepreneurs to invest in thousands of small hydropower plants, (SHPs). Nowadays, hundreds of SHPs are in operation and under construction; thousands of them are in design stages. However, the design and construction of a SHP is not the only aspect to concern. The operation study of a SHP is even more important, since energy production without failures and long delays is very crucial for the owners of these plants. All computational studies of design stage are aimed at safe and reliable operation; therefore, all operational situations should be regarded in a design stage. Steady operation of a SHP is the safest state for it as there is no change in its hydraulic variables like discharge and pressure head in the system. However, if the turbined flow changes during the hydropower operation, disturbance will occur and cause a sudden change in the state of the system. Along the hydraulic conveyance system, namely the penstock, flow parameters start to change with time. This type of flow regimes are called hydraulic transients and waterhammer, which occur during the change from one steady state to another. They can cause extremely high or low pressures in the penstock. Excessively high pressures may lead to great physical damages. Turbines, valve and several appurtenances of the penstock may be damaged. Even the penstock itself may burst dramatically, causing environmental tragedies and human losses in some cases. Moreover, extremely low pressures can lead formation of vapor cavities in the penstock or could cause the penstock to collapse.

There are huge hydropower accidents caused by waterhammer and resulted in substantial damages and loss of lives in the history. Serious failures were occurred due to the waterhammer pressures at Bartlett Dam and Oneida Station Hydroelectric Power Plant in the United States of America. Both were caused by faulty operations of valves and resulted in five losses of lives (Adamkowski, 2001). Also in 1997, the penstock of a small hydropower plant, Lapino HP in Poland, ruptured during the acceptance tests of its new governor (Adamkowski, 2001). Finally, a well known accident occurred in Japan in 1950 at Oigawa Hydropower Station. As a result of rapid valve closure, extremely high pressures occurred and caused penstock to burst. Then, resultant release of water caused extremely low pressures resulting in column separation and causing penstock to collapse. As a consequence, three workers were killed (Bergant et al., 2004).

Study of hydraulic transients in closed conduits attracts many researchers because of its complexity and significance in practice. Scientific and engineering investigations on waterhammer problems and its harmful effects are still going on to figure out underlying facts. The amount of literature on hydraulic transient concept is very impressive.

1.2 Literature Survey

The studies on fluid transients have an almost 300 years old history. The material of the following paragraph presents the historical background of waterhammer studies in the literature, and is based on Chaudhry (1987).

Their interest on the blood flow and the investigation of the propagation of sound waves in water and air have made Newton, Euler and Langrange the first researchers on hydraulic transients phenomenon in the 17th and 18th century. The celerity of waves in a canal is firstly defined by Langrange. Following this approach, the first graphical method for integrating partial differential equations for characterization of fluid behavior is developed by Monge, by the year 1789 and the literature was firstly introduced by the term "Method of Characteristics". After all these basic developments, Young investigated the pressure wave speed in a pipe for an incompressible fluid. He was the first to study fluid transients in closed conduits. Although fundamental theory of waterhammer in closed conduits had not been wholly discovered yet, preliminary studies for controlling waterhammer effects was started by Michaud in 1878. He investigated the design and use of an air chamber and safety valves. The first study of waterhammer in hydropower plants was conducted by Frizell. He was working as an engineer in Ogden Hydropower Plant in Utah when he conducted experiments on its considerably long penstock. He derived the equations for the pressure rise due to sudden stoppage of the flow and the pressure wave speed. Although Frizell and Joukowsky stated the same well known expression for the pressure rise at the same time, it was attributed to Joukowsky. The effect of branching pipes on the reflection of wave speed is also studied by him. The pressure rise expression which is known as the fundamental equation of waterhammer theory is based on Joukowsky's research. In 1897 he conducted various experiments on Moscow's drinking water supply pipes. In his well known report, he explicitly stated the wave speed equation by considering elasticity of both pipe wall and fluid. He discovered that the rapid valve closures which take less than $2L/a$ seconds was the reason for the occurrence of maximum pressures in pipes. He also studied the behavior of surge tanks, air chambers and safety valves during a transient event. In the year 1902, Allievi has brought a new approach to waterhammer analysis and defined dimensionless parameters representing valve closure characteristics and energy ratio related with the fluid and the pipe material. Then, by introducing charts showing pressure rise and drop due to valve operations, he acquired transient pressure values at the valve. The behavior of a hydraulic turbine during a transient event which is caused by load changes in a hydropower plant is firstly considered by Strowger and Kerr in 1926. They computed turbine speed changes by taking into account of turbine efficiency at various gate openings and movements.

Then, these fundamental advances in hydraulic transients inspired many scholars and researchers and led them to extend the literature on hydraulic transients in hydropower plants.

Hovey (1962) is one of the researchers who investigated the stability of hydropower plants. He studied to provide practical information and methods for controlling transients in hydropower plants by investigating the setting of dashpot times of their governors. In his study he explained the methods used in Manitoba Hydropower Station to optimize the settings of the governor. His main criteria were the damping of the turbine speed critically during load changes.

Hagihara et al. (1979) also studied the stability of hydraulic turbine units. They researched the proportional integral derivative, (PID) governor controlled hydropower plants and investigated the parameters of this type of governors on the stability. They used rigid column theory in their analytical works to calculate waterhammer effects.

Jimenez and Chaudhry (1987) included the elasticity effects, namely the elasticity of the pipe walls and the compressibility of the water column in waterhammer effects and investigated the stability of a single hydropower station unit. They derived an analytical stability criterion and verified it by a computer simulation.

Peicheng et al. (1989) conducted tests on Linzhengqu Water Power Station to show that pressure relief valves and safety membranes can replace a surge tank in a small hydropower plant. They presented that both protective measures ensure reliable and safe operation, separately. Ni et al. (1996) represented a mathematical model for analyzing hydraulic transients in a hydropower plant protected by safety membranes. Method of characteristics is employed for solving the momentum and continuity equations describing the transient flow in the penstock. The boundary condition for safety membranes is defined. Computed results were compared with measured ones and close agreement were found between them.

Souza et al. (1999) simulated transient flow in hydropower plants by considering nonlinear model of the penstock and hydraulic turbine. They developed a nonlinear-digital simulation method and analyzed both the penstock and turbine by using their electrical equivalent circuit model. Then, they simulated a literature example with their model and compared the results with those obtained by the method of characteristics, and proved the accuracy of the model.

Ramos and Almeida (2002) presented a novel technique that parameterizes the waterhammer effects in small hydro schemes to characterize the dynamic behavior of their turbines better. Their approach considered the similarity between a turbine and a dynamic orifice. The dynamic orifice technique is based on the concept of the turbine acting as a hydraulic resistive component with a dynamic discharge coefficient. They carried out an analysis and compared laboratory and field tests results. Computer model outputs were proving that the application of the technique appears to be a powerful tool in preliminary design stages.

Selek et al. (2004) simulated the transient flow in Çatalan Hydropower Plant in Turkey. They solved the governing equations of unsteady flow in the penstock by method of characteristics using various computational schemes namely, simple fixed-grid system, fixed-grid system with space-line interpolation and variable grid system. They compared the computational results of the turbine inlet pressure with those acquired from prototype test results and found that the variable-grid method of characteristics produces the results that agree best with experimental findings.
Karadzic et al. (2009) developed a novel Pelton turbine model for waterhammer analysis. They defined the boundary condition for Pelton turbine units and calculated the instantaneous head at the nozzle inlet and discharge through the

nozzle by using method of characteristics. Then, they used these values as input in the solution of the dynamic equation of the turbine unit. The solution method describing dynamic behavior of the rotating parts of a Pelton turbine during both emergency shut down and load rejection was developed. They investigated waterhammer phenomenon in Perucica hydropower plant with their computational model, and compared the calculated and measured head at turbine inlet, and turbine rotational speed. Also, a novel model describing Pelton turbine speed change during waterhammer is used. They gathered reasonable agreement between computed and field results.

Vakil and Firoozabadi (2009) studied the effects of different valve closing laws on the maximum head rise at turbine inlet. They developed a computational model that utilizes method of characteristics to solve the governing equations of unsteady pipe flow for the pressure rise, speed rise and discharge fluctuations during waterhammer. Then, a hydropower plant with a Francis turbine unit was modeled for load rejection case with various valve closing laws. Results obtained from the model were compared and validated with those obtained by a consulting company.

1.3 The Motivation and Scope of the Study

The small hydropower development of Turkey has an upward trend within the last decade. By the promotion of private sector to evaluate high gradient mountain streams, which are very suitable for SHP developments, hundreds of them constructed and thousands of them are in design stages nowadays. However, challenging design process of the hydropower development is facing a big problem: ensuring the operational safety of the SHP economically throughout its lifetime. Hydraulic transients may lead to crucial problems in SHPs as they have relatively long penstocks and equipped with small inertia turbines. Therefore, from the early design stages of a SHP, the hydraulic transients in their penstocks should be considered in order to find the economical and safer layout. The aim of the present study is to investigate time dependent flow situations in the penstocks of small hydropower plants with different system components. By using a computer program which utilizes the method of characteristic to solve nonlinear partial differential equations of transient flow, various transients scenarios will be constructed to simulate and investigate waterhammer behavior of the systems during their regular operational conditions. A series of analyses will be carried out, and computed and measured results will be compared to investigate the possible factors that affect transient flow behavior. Moreover, the effect of closing law of wicket gates on the penstock, turbine units and related equipment is studied to eliminate possible risks of damage and guide the designer in developing an optimum closing law. It is also intended to investigate the behavior of the systems with protective devices which can be replaced by frequently used and expensive measures such as, surge tanks and air chambers. The advantages and drawbacks of these devices will be discussed for further applications of them in other small hydropower plants.

1.4 Organization of the Study

The study is composed of six chapters which are organized as follows:

Chapter 1 gives brief background information about the study and includes a literature survey on waterhammer phenomenon.

Chapter 2 describes the hydropower and its theory. The definition of the small hydropower plant with its main components is also included in this chapter.

General hydraulic transient concept is given in Chapter 3. Numerical solution of transient flow in closed conduits is presented. The causes that initiate waterhammer in penstocks of small hydropower plants are defined. Waterhammer responses of different turbine types are also identified. Moreover, protective measures for preventing waterhammer pressures are given in this chapter.

Chapter 4 summarizes the computer program, HAMMER, used in the study. The solution method used by the software and its capabilities are described herein.

The subject matter of Chapter 5 is case studies. Two small hydropower plants having different turbine types are modeled for transient simulations. They are subjected to disturbances that were introduced during their operation. For Case Study 1, computed waterhammer pressures are compared with field data. In Case Study 2, waterhammer effects on both penstock and turbine speed are investigated. Moreover, three protective measures are considered and their remedial effects are illustrated. Discussions about the simulations are provided at the end of the chapter.

Finally, Chapter 6 is devoted to conclusions and final remarks of the study.

CHAPTER 2

SMALL HYDROPOWER PLANTS

2.1 Hydropower

Hydropower means electricity generation from water. Water in rivers and streams has kinetic energy which is converted from potential energy while flowing from higher elevations to lower elevations. As water is available in large quantities from rain and snow, it will be sustained continuously by hydrologic cycle for unlimited time. Therefore, energy of flowing water is a renewable energy source. Among developing and renewable energy resources, such as solar, wind and biomass, hydropower is mature in technology. It is the first renewable energy source that has been utilized for electricity generation. In the ancient times, people discovered how to make use of water for power, starting with the wooden waterwheel. Today the technology of hydropower is advanced, by the help of developments in engineering techniques; the construction period is shortened, initial cost of the plant is reduced and the technically feasible potential areas of hydropower are increased. However, although it seems easy to design a hydropower plant, there are many things to take into consideration starting from preliminary studies of the project. Selection of the optimum topography, design of the penstock, turbine and generator, their operation and maintenance, environmental impact assessment of the hydropower plant, resettlement or relocation of people or any other living things, and financing are the main concerns of a hydropower project. Even if it is an old technology, problems encountered during construction and operation of a small hydropower are still being studied by the researchers because of its complexity. Waterhammer problem in hydropower plants is one of these problems.

2.2 Terminology and Theory of Hydropower

The fundamental theory of hydropower can be easily understood by defining these concepts: *installed capacity, gross and net (effective) heads, hydraulic and overall efficiency, demand and load*.

The *installed capacity* of a hydroelectric power plant is a characteristic that shows the maximum power which can be produced by its generators. It is primarily a function of the volume of flowing water per unit time through the turbine and the gross head. The *gross head, H_g*, is the vertical difference between the elevations of tailwater and headwater surfaces of the plant when it is not in operation (see Figure 2.1). Greater the gross head and volume of water means greater spin applied to the turbine and greater output of electricity. Gross head is determined by the conditions of flow in the stream. During design process of a hydropower plant, extreme variation in gross head from maximum to minimum is considered.

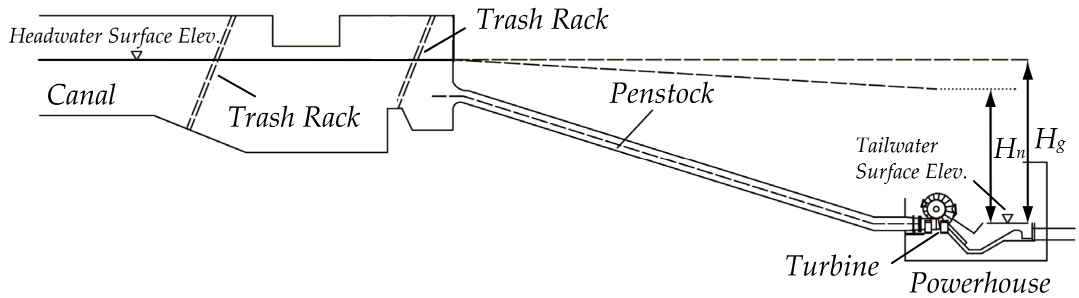


Figure 2.1 Description of the Gross and Net Heads

The gross head in a hydropower plant may be natural or created artificially by constructing a dam to raise the water level. However, the whole of the gross head cannot be utilized in generation of power because of the hydraulic and machinery losses. Hydraulic losses are the frictional losses in conduits (tunnels or penstocks), minor losses at the intake entrance, trash racks, expansions, contractions, bends in conveyance and losses in the turbine. The *net or effective head, H_n*, is the head which is utilized for energy production and it is equal to the difference between the gross head and sum of all losses from headwater to tailwater. The ratio of net head to the gross head is called *hydraulic efficiency*. The multiplication of the hydraulic efficiency with turbine and generator efficiencies yields the *overall efficiency* of the hydropower plant.

The installed capacity of a hydropower plant can be determined from:

$$P = \gamma Q_{tur} H_g \eta \tag{2.1}$$

where;

$$\eta = e_h e_g e_T \tag{2.2}$$

and

P:	Installed capacity, [W]	η:	Overall efficiency of the power plant
γ :	Unit weight of water, [N/m³]	e_h :	Hydraulic efficiency
Q_{tur} :	Turbine discharge, [m³/s]	e_g :	Efficiency of the generator
H_g :	Gross head, [m]	e_T :	Efficiency of the turbine

Finally, two key words of hydropower have to be known for perfect understanding of the theory. *Demand* is the needed or consumed electricity instantly in a system and *load* refers to instantly produced electrical energy in a system. A basic condition of system operation is that the electricity cannot be stored. This means that electricity must be consumed immediately while it is being generated. Therefore, there must be a balance between load and demand. In the case of an unbalanced supply and demand, system is exposed to over or under frequency.

2.3 Description and Categorization of Small Hydropower Plants

Small hydropower plants have a key role in countries' energy development strategies in the world. Its vast and reliable potential and cost effective technology make it one of the most common renewable energy sources. Today thousands of small hydropower plants are spread around the world. The description and categorization of it vary with countries. They are generally categorized by their installed capacity. The top limits of installed capacity range between 1 MW and 50 MW. Other categories for SHP classification are the gross head, and the layout of powerhouse. On the basis of their experience, countries have determined the upper installed capacity limits of SHPs as given below.

Table 2.1 SHP classification (adapted from Laguna et al., 2006; Penche, 2004; Jiandong, et al., 1997; Yüksek et al., 2007)

Name of country or organization	P, [MW]<
Canada	25
China	25
ESHA	10
France	12
Greece, Belgium, Portugal, Spain, Ireland	10
Italy	3
Japan	10
Norway	10
OLADE	5
Poland	15
Sweden	1.5

Table 2.1 cont.'d

Name of country or organization	**P, [MW]<**
Turkey	**50**
UNIDO	10
United Kingdom	20
USA	30

These upper limits for small hydropower plants show the degree of development of the country. It is also related with the share of hydropower in energy sector in the country. As a consequence of this, different countries have different description of small hydropower.

Since most of the SHPs are run of river schemes in the world, they indicate the run of river type of hydropower plants (HPs) commonly. A run of river HP diverts some quantity of the river flow by diversion weir to drive the turbines. Then it returns water to the river at a downstream location after power generation. They use water just as it comes and does not store it. Since the water is not stored, energy is generated by the natural flow regime. Their electricity generation is for the base load throughout the year because they are only in operation when water sustained by natural flow. Contrary to the dams with hydropower plants, the gross head is supplied by the topography of the site, not by an artificial body. Here is the basic logic behind the run of river schemes (see Figure 2.2);

- A diversion weir is constructed across the river to raise the water level up and direct it to the intake structure which has a canal at its end.
- After water has passed the intake, a canal carries it to a relatively small reservoir, called forebay.

- Forebays may store relatively small volumes of water. It regulates and distributes the flow and sustains pressurized flow for penstock.
- Penstocks, usually made of steel, are pipes that carry pressurized flow from forebay to turbines.
- Pressurized flow runs the turbines to convert hydropower energy into mechanical energy. By means of a shaft, that mechanical energy is converted to electrical energy by a generator.
- The mechanical equipment is kept in powerhouse to protect, inspect and maintain them easily.
- Finally after giving up its energy, the water is discharged by a draft tube and tailwater channel and returns to its natural bed at a downstream location.

2.3.1 Advantages of Small Hydropower Plants on Large Scale Hydropower Schemes

A run of river scheme is very common type of hydropower plants because (Gagnon et al., 2002):

- The unit cost per kilowatt hour is very low compared to a HP with storage.
- Its energy payback ratio, which is the ratio of energy generated during the operation to the energy spent during the construction and operation, is higher than large hydropower schemes.
- There is very little or no resettlement and relocation problem of living things, because direct land requirement of run of river schemes is very low with respect to hydropower plants with reservoirs.

- It has very little impact on the hydrology of the project area.
- Since it emits very low greenhouse gases, 2 kt eq. CO_2 /TWh in quantity (while a large hydro scheme emits 15 kt eq. CO_2 /TWh), it is very environment friendly.

2.3.2 Small Hydropower Development in Turkey

According to World Energy Council, Turkey's economically feasible hydropower potential is 130 TWh/yr and 6.7 % of that potential can be developed by the small hydropower plants whose installed capacity is smaller than 10 MW (WEC, 2007). As can be seen from Table 2.2, as of the year 2006, there are 105 SHPs in operation with a 952.81 MW installed capacity in Turkey.

Table 2.2 Number of SHPs in Turkey with their Installed Capacity and Annual Generation (DSI, 2006)

	In Operation	Under Construction	Final Design Comp.	Feasibility Study Comp.	Master Plan Comp.	Preliminary Study Comp.
Number of SHPs	105	24	5	134	65	238
Installed Capacity of the SHP, [MW]	953	473	54	2107	1215	2390
Average Annual Generation of the SHP, [TWh]	3.682	1.654	0.206	8.649	5.096	9.691

2.4 Main Components of Small Hydropower Plants

A typical SHP consists of the following components:

- A diversion weir and intake structure,
- Hydraulic conduits and their facilities which can be canal, tunnel, penstock, draft tube, tailwater channel, gates and valves,
- A forebay, in other words, headpond,
- A powerhouse including turbine and generator units and a draft tube,
- A switchyard with transformer,
- Connection to main transmission line.

The following components have various types of themselves, but only types that related with SHPs are introduced herein.

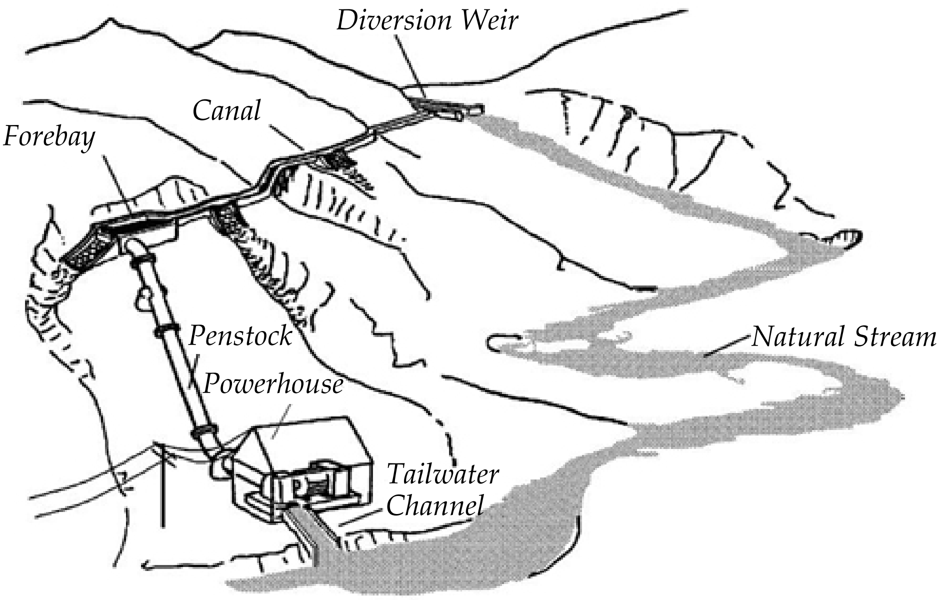


Figure 2.2 Scheme of a Small Hydropower Plant (Paish, 2002)

2.4.1 Diversion Weir and Intake Structure

Diversion weir is a kind of an obstacle or barrier for raising the water level in a river, to divert the water. Main purpose of diversion weirs is to get the required amount of water from the river, for the most of time, by means of an intake. The level of water is controlled by the spillway and it is limited by the elevation of spillway crest. Thus, required amount of water can be diverted from river. Also, construction of a diversion weir is considerably reduces the entrainment of sediment into the intake.

Diverted water for the hydropower generation is drawn from the river to the canal by means of the intake structure. Intake structures are constructed for;

- Preventing the entrainment of sediment into the canal,
- Keeping away floating objects (ice or debris materials) from canal,
- Minimizing the head losses at the entrance of the canal.

While designing the diversion weir and intake structure, following requirements should be considered:

- The height of the weir and the dimensions of the intake must be chosen in such a way that, the desired amount of water is diverted and ensured for hydropower generation for any regime of the river.
- The peak discharges of the river should not cause any problems on both weir and intake. Before designing these structures, topographical and hydrological data should be processed in order to determine the dimensions of them.

- As SHPs are in mountainous regions generally, both structures should allow maintenance free operation or can be simply repaired in case of any disorder.

- All rivers transport sediment in the form of suspended and/or bed load. However, canal flow must be free of both of them. Therefore, dimensions of the settling basin should be determined carefully.

A common type of diversion weir and intake structure is given in Figure 2.3. It is the diversion weir with lateral intake. Other combinations of them are diversion weir with frontal intake and diversion weir with drop (bottom) intake (Yanmaz, 2006).

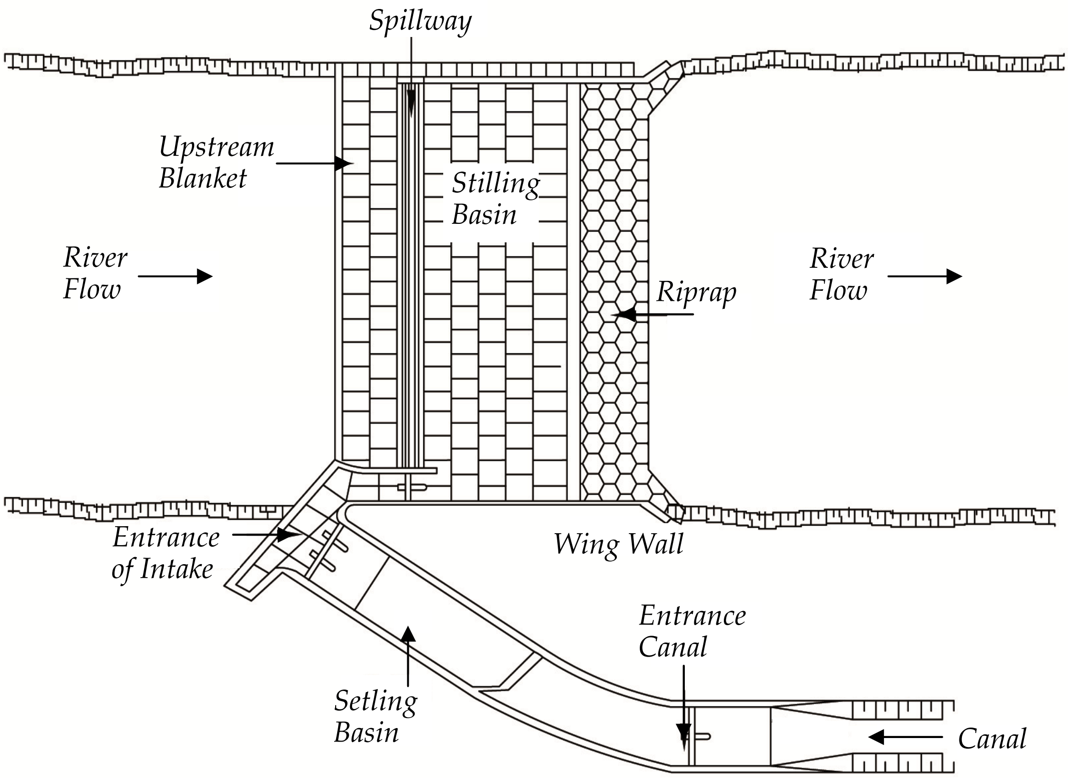


Figure 2.3 Plan View of an Overflow Spillway and a Lateral Intake

2.4.2 Canal or Tunnel

By the time water has passed the settling basin of the intake, it enters an open channel conduit called canal, which transports it with free surface flow to the forebay. Their cross section may be both trapezoidal and rectangular and dimensions are designed for the maximum discharge of the plant. Minimum permissible velocity check is done for preventing silting in the canal in dry seasons. Also, maximum permissible velocity check should be done for preventing excessive head losses in the canal.

The slope of the canal is very mild compared with the natural stream to keep the gross head of the water. In some regions, canal may be covered to prevent it from small landslides or ice loads.

If the topographical formation of the site is not suitable technically or economically for canals, free surface tunnels can be constructed for the conveyance of the water.

2.4.3 Forebay

The forebay is a structure which collects, regulates and distributes the water that is conveyed by the canal to penstocks.

It serves as a small reservoir and temporarily stores water for the following cases:

- When the load of the plant increased suddenly, the water in the penstock accelerates and the water level just above the penstock inlet drops down. If the submergence height is not enough above the inlet, an unwanted situation, hydraulic suction occurs and air enters the penstock. Only canal itself cannot provide that flow to the penstock. Forebay supplies that submergence depth for the penstock inlet.

- If a complete load rejection occurs in the system, waterhammer phenomenon occurs while the canal still supplies water for the plant. Forebay may act as a surge tank and stabilize the surge effects of waterhammer.

The forebay has an inlet structure to direct water to the penstock. That inlet should have trash racks to prevent the entrainment of silt and debris materials that might damage the wicket gates, runners or nozzles of turbines (Linsley et al., 1992).

The dimensions of the forebay are designed by the following considerations:

- The width of the forebay should be greater than the total width of the inlet and the required minimum size of the trash rack. Also the average velocity requirement for the settlement of the harmful particles in the forebay should be provided.

- The length of the forebay must be greater than the overflow spillway crest length. That spillway is designed for the maximum discharge of the canal.

- Finally, submergence depth of the inlet should be adequate enough to prevent hydraulic suction during transient events in dry seasons.

Profile of a typical forebay structure with its components is given in Figure 2.4. In the figure, HPL is the highest pool level, NPL refers the normal pool level and LPL represents the lowest pool level.

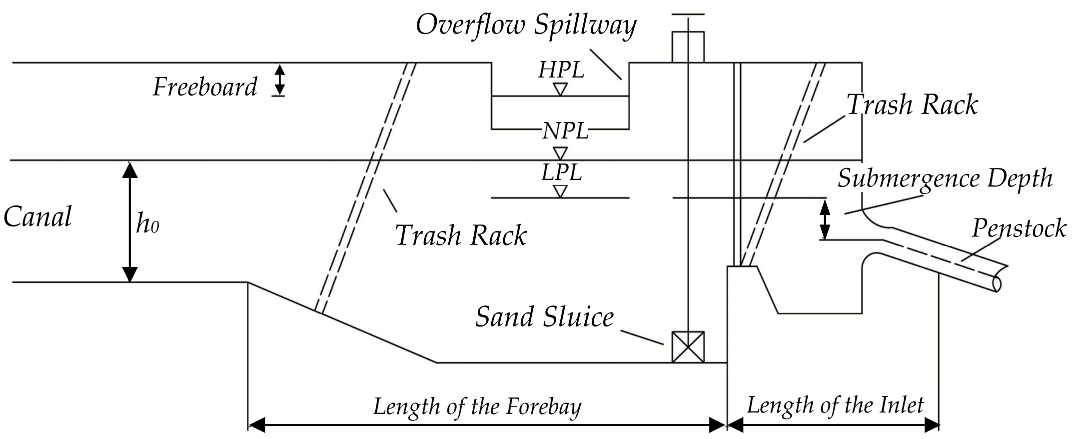
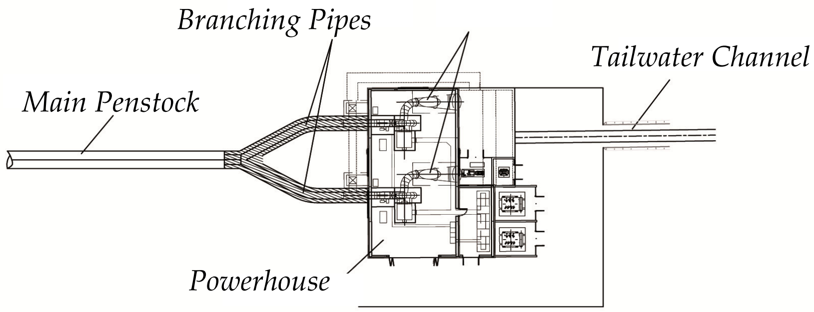


Figure 2.4 Profile of the Forebay (Jiandong et al., 1997)

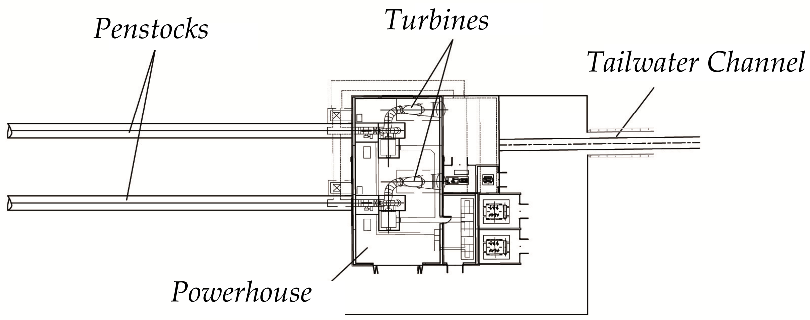
2.4.4 Penstock

Penstock is a pressurized conduit that conveys water from forebay to the turbine. There are different materials that penstocks can be made from, such as steel, wood-stave, reinforced concrete, asbestos cement, polyvinyl chloride (PVC), polyethylene, GRP and plastic. But most commonly used material is steel because of its strength for high pressures and long level of service.

In a SHP, there are two possible penstock layouts. One of them is branching layout which consists of one main pipe that branches at its lower end to feed every single turbine unit, Figure 2.5 (a), and the other one is separate layout in which every turbine has its own penstock, Figure 2.5 (b). Both of these layouts can be installed over the ground or buried. The topographical conditions of the site, material type of the pipe and environmental regulations determine the type of the installation. The layout selection is carried out after having done an economical analysis by considering the cost of pipe, construction site and all other parameters.



(a)



(b)

Figure 2.5 Layout Schemes of Penstocks (a) Branching Layout, (b) Separate Layout

The structural design of a penstock is complex. It consists of the determination of the material of the pipe, determination of economical diameter, loading of steady state and transient pressures, weight of the pipe and the water, selection of wall thickness, the type and spacing of supports, design of anchorages and expansion joints. The material is selected according to local availability and production technology, condition of transportation, installation, jointing and climate.

For a given discharge, the diameter of a pipe may vary between wide limits. However, by considering friction losses and waterhammer effects, velocity in the pipe must be in a predetermined range. Small incremental change in the friction loss may significantly affect the net head and hence, the energy production. Also a doubled velocity in the steady state condition increases the pressure rise double in case of the sudden stoppage of the flow. So, for successful operation, all possible conditions should be determined to optimize the diameter. For pre-assumed larger, smaller and intermediate values of diameter, frictional losses, wall thicknesses, transient pressures, cost of the penstock and revenues from the energy production should be calculated. There is usually one size that gives the minimum cost. However, some uncertainties may exist.

The rate of water flow may suddenly change in the penstock due to opening or closing of gates and valves, changes in load, blockage of the turbine by an obstruction or fluctuations of water surface in the forebay. This results in a sudden change in velocity and movement of excessive amount of water and pressure wave in the pipe with an acoustic speed. That phenomenon is called waterhammer and causes excessive changes in pressure above or below the normal pressure through the pipe. The pressures caused by waterhammer may be several times greater than steady state pressure. If waterhammer occurs in a system, the outcome can be very costly and even deadly. Pipeline may collapse or burst erratically resulting in substantial damages, and loss of lives in some cases. Therefore, the wall thickness must resist the summation of maximum steady state and dynamic pressures. Free body diagram of a semicylinder pipe is shown in Figure 2.6. The radial tensile force on pipe wall, regardless of the support situations of the pipe is represented in the figure.

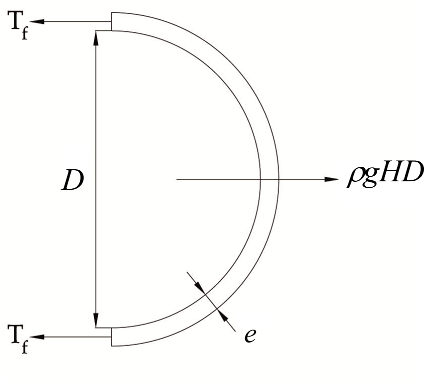


Figure 2.6 Cross-Section of a Semicylinder Pipe with Acting Forces

Radial tensile force, T_f, and its relation between maximum allowable tensile stress, σ_f, are described in Eq. (2.3) and Eq. (2.4), respectively.

$$2T_f = \rho g H D \tag{2.3}$$

$$\sigma_f = T_f / e \tag{2.4}$$

The wall thickness can be determined from:

$$e = \frac{\gamma (H + \Delta H) D}{2\sigma_f} \tag{2.5}$$

where;

e :	Wall thickness of the pipe, [m]	ΔH :	Change in the pressure head in the transient conditions, [m]
γ :	Unit weight of fluid, [N/m³]	D :	Diameter of the pipe, [m]
H :	Pressure head in the pipe in steady state condition, [m]	σ_f :	Maximum allowable tensile stress, [N/m²]

In long penstocks with varying pressure heads, diameter can be selected a constant value while wall thickness is changing through the length of the pipe.

If the penstock is above the ground, it must have supports throughout its layout, and these supports should allow longitudinal movements of the pipe. Commonly, there are two types of supports, namely ring girders and saddle supports. Ring girders are for long span elevated penstocks. They are welded steel plate rings (see Figure 2.7 a). Loads are transferred from penstock to the ring girder and support legs which are attached to the bearing plates transferring the load to the concrete foundation (McStraw, 1996). Saddle supports are typically for shorter span distances. They are reinforced concrete blocks that transfer the load directly to the ground (see Figure 2.7 b).

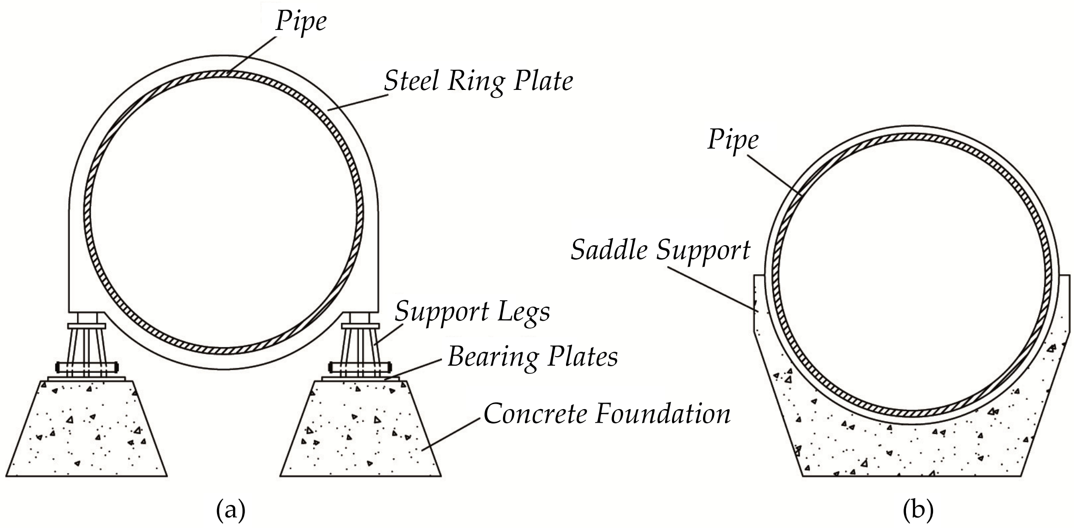


Figure 2.7 Schemes of (a) Ring Girder and (b) Saddle Support

Anchor blocks are reinforced concrete structures and designed to withstand resultant of all forces on the penstock. They are installed at angle joints and between every two expansion joints in the pipeline (see Figure 2.8). A penstock may have three different types of anchorage. These are anchorage at pipe's upstream end only, anchorage throughout against axial movement of the pipe and anchorage with expansion joint through the pipe (Wylie et al., 1993). The type of the anchorage affects the magnitude of the speed of wave which travels along the pipeline during a transient event.

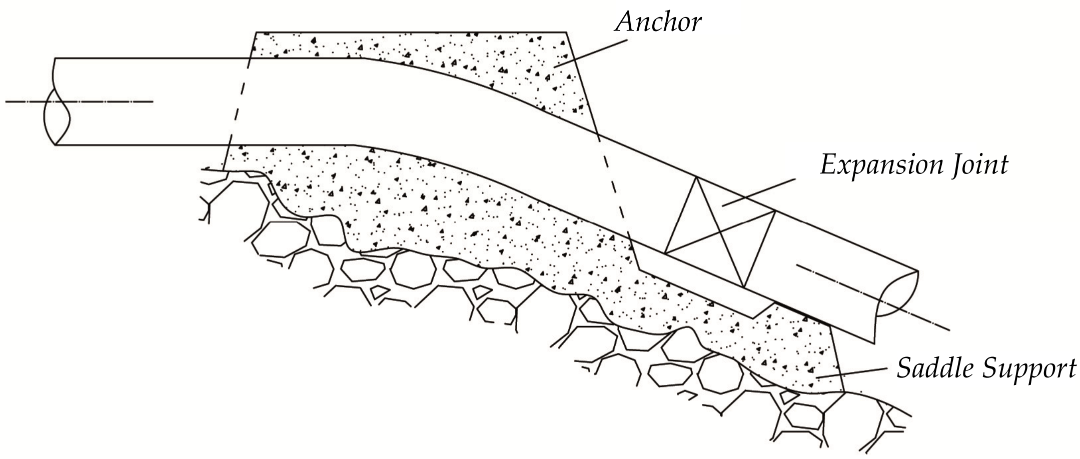


Figure 2.8 Profile of an Anchor

A penstock over the ground may be exposed to extremes of temperature. That may cause the pipeline both expand or contract during its lifetime. Usually, if the plant does not operate continuously during a year or stops working for maintenance, penstock may suffer from expansion and contraction in hard weather conditions. These events cause excessive longitudinal stresses in the pipe unless expansion joints provided. There are two general types of expansion joints, i.e., the slip joint and the diaphragm joint (Creager and Justin, 1950). They are placed between two anchors in order to reduce the movement of the whole pipe.

2.4.5 Turbine

Hydraulic turbines transform the energy of water into rotating mechanical energy. They have buckets, gates or blades which rotate around an axis. The rotating part of them is called runner. They may be grouped according to different aspects. According to principles of their water flow action and structural properties, there are two types of turbines: impulse and reaction turbines. In *impulse turbines*, potential energy of the water is transferred into kinetic energy with a high velocity jet discharging from the orifice of a nozzle. The free flow of the jet into the atmosphere strikes the bowl shaped buckets of the runner. Types of impulse turbines are Pelton, Turgo, and Michell-Banki which is known also as crossflow turbine. Buckets and the runner of a Pelton turbine are shown in Figure 2.9. *Reaction turbines* utilize both pressure and velocity of the water which completely fills the runner. Water enters from a spiral case and passes through the wicket gates located around the runner. Francis, Kaplan, Bulb and Gorlov turbines are well known reaction turbines. Blades of the runner of a Francis turbine are presented in Figure 2.10.

According to flow direction in the runner, turbines can be divided into axial, diagonal, radial, tubular and crossflow turbines. Moreover, the orientation of the shaft is another class for grouping of turbines. In that class, horizontal and vertical shaft turbines exist.

Fundamental definitions are made below for the sake of a good understanding of the turbine concept:

The *speed*, n, of a turbine is the number of rotations of the runner per unit time.

The *specific speed*, n_s, value is the speed of a geometrically similar turbine with unit head and power output under similar operating conditions. It is constant for similar turbines and operating conditions.

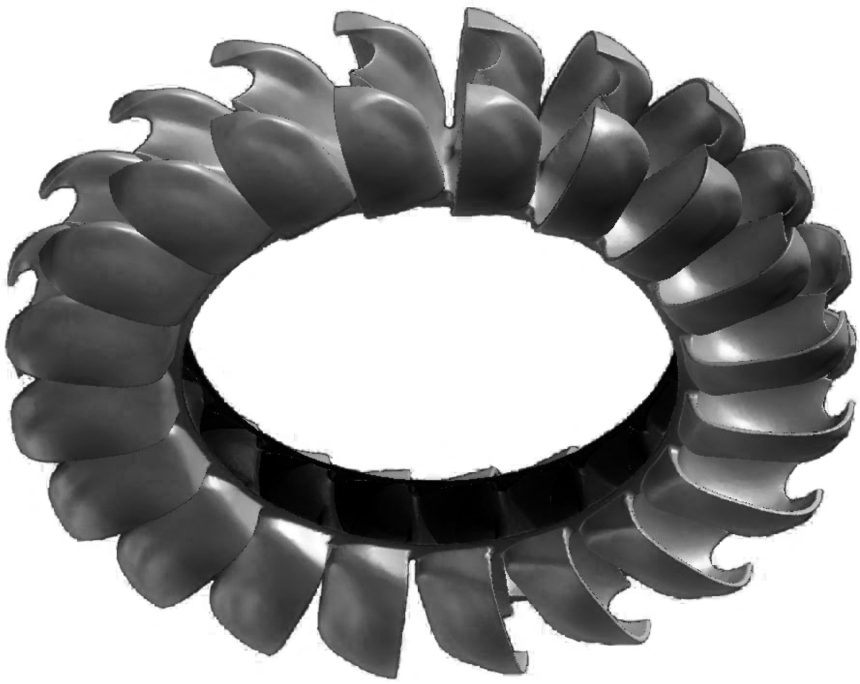


Figure 2.9 Buckets of a Pelton Turbine Runner

Figure 2.10 Blades of a Francis Turbine Runner

The *runaway speed*, n_f, is the increased maximum speed that a turbine can withstand mechanically. It usually occurs when the wicket gates or nozzles cannot be closed during a load rejection. Turbine rotational speed may rise to a maximum value and the system could not be operated safely if it is not controlled by a governor.

The *efficiency*, e_T, of a turbine is the ratio of the converted mechanical energy to the supplied energy from the water. There are some friction losses and hydraulic leakages in a turbine; therefore, whole energy of water cannot be converted. Efficiency curves are supplied by the turbine manufacturers.

Additional definitions for the reaction turbines may be useful:

Wicket gate controls, changes, regulates the discharge and power output of the turbine by opening and closing. Its another important mission is to protect the turbine from runaway by stopping the operation during load rejection.

The *spiral case* supplies uniform flow through the inlet of the wicket gate of the reaction turbine. Wicket gates have to be exposed axially symmetric flow for the stability of the turbine and that flow is guaranteed by it.

The performance of a turbine is determined by two general parameters (Jiandong et al., 1997):

- The geometrical parameters such as diameter of the runner, D_r, opening of the wicket gate, a_w (for a Francis turbine), angle of the runner blade, ϕ (for a Kaplan turbine) and nozzle opening (for a Pelton turbine),
- The kinetic parameters which show the operating conditions of the turbine such as, specific speed, n_s, discharge Q_{tur}, net head H_n, and efficiency, e_T.

There is a relationship between these parameters and that relationship is named as turbine characteristics. To define turbine characteristics, steady state model tests are held by manufacturers and obtained results are presented in graphical forms (Chaudhry, 1987). Turbine characteristic curves are presented in Figure 2.11 and Figure 2.12.

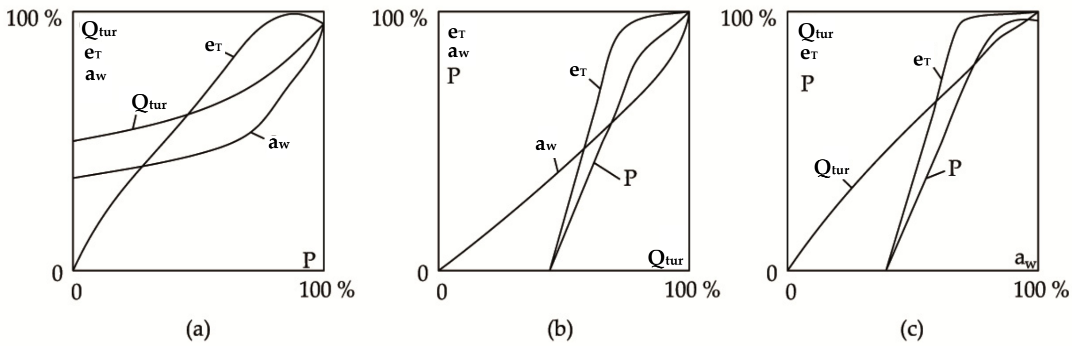


Figure 2.11 The Turbine Characteristics Curves (a) Output, (b) Discharge, (c) Opening, (Jiandong et al., 1997)

In general, there are five types of characteristics curves (Jiandong et al., 1997):

- If diameter of the runner, D_r, speed, n, and net head H_n are constant, the relationship between power output, P, efficiency, e_T, discharge Q_{tur} and opening of wicket gate, a_w, can be showed by three curves: output, discharge and opening curves.

  The relationships between turbine parameters in these three curves are defined below.

  For output curves, e_T=f(P), Q_{tur}=f(P), a_w=f(P); similarly, for discharge curves, e_T=f(Q_{tur}), P=f(Q_{tur}), a_w=f(Q_{tur}); and for opening curves, e_T=f(a_w), P=f(a_w), Q_{tur}=f(a_w).

- If D_r, H_n and a_w are constant, speed characteristic curve can be obtained (see Figure 2.12 a).
- When D_r, n, a_w are kept constant, relationship between H_n, e_T and P can be acquired (see Figure 2.12 b).

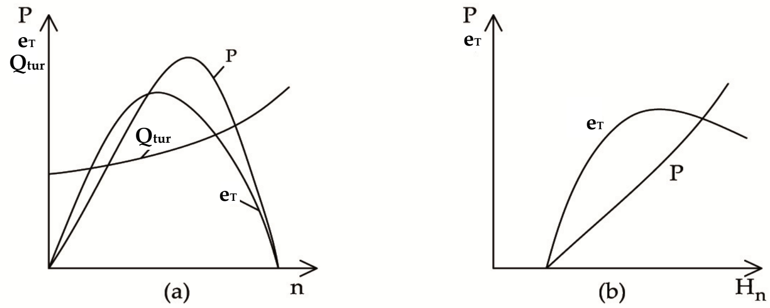


Figure 2.12 (a) Speed Characteristic and (b) Head Characteristic Curves (Jiandong et al., 1997)

All different types of turbines are designed to operate under different conditions. The performance and economy of the system depend on the selection of the turbine. Therefore most appropriate development must be chosen under existing conditions. The selection of the turbine depends on the net head and discharge of the system, and each type of turbine has its own limitations in application. In Table 2.3 application heads and corresponding turbine types are given. Also in Figure 2.13, envelopes of various types of turbines are given. After determination of the type, main parameters of the turbine are selected from the turbine characteristic curves. The design of the turbines is not in the scope of this study and transient analysis and models of turbines will be discussed in the following chapter.

Table 2.3 Application Heads of Impulse and Reaction Turbines (Paish, 2002)

Turbine Type	**Head Classification**		
	High	**Medium**	**Low**
Impulse	Pelton Turgo	Pelton Crossflow Turgo	Crossflow
Reaction		Francis	Francis Kaplan

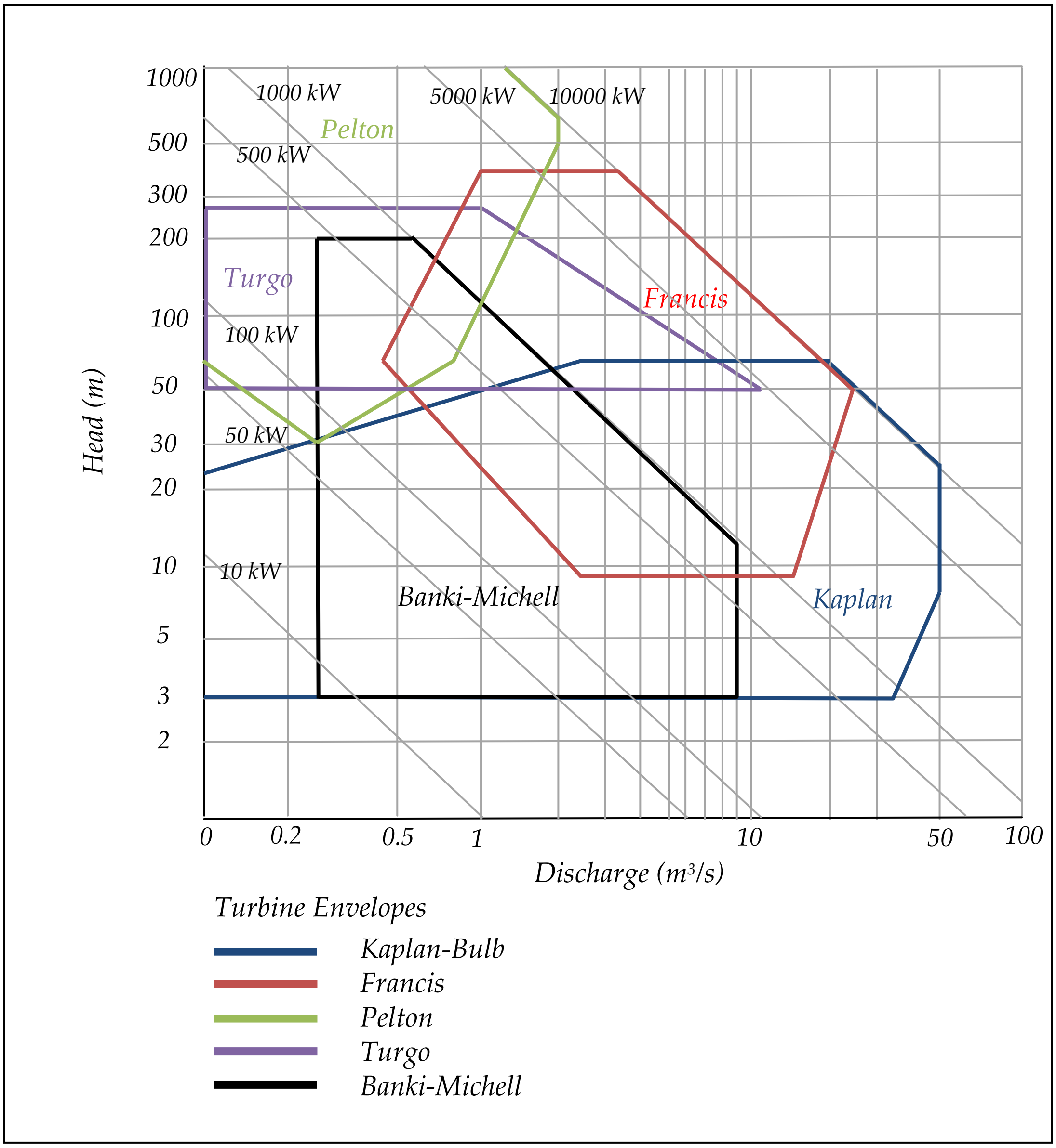


Figure 2.13 Turbine Application Limitations for Design Heads (Penche, 2004)

The determination of number of turbine units in a hydropower plant depends on many factors. More number of units means high flexibility in generation and high revenue. However, it comes with high initial, operation and maintenance costs. Therefore, an optimization study must be done for determining the number of turbine units and minimizing the cost of the SHP.

2.4.6 Governor

Hydraulic turbine governors are used to minimize the amplitude of the turbine speed deviation from its synchronous speed during a transient event by closing or opening turbine wicket gates or nozzles. It consists of speed sensing device and a servo mechanism for opening and closing. There are both mechanical and electrical speed sensing devices which detect the deviation from the reference speed. Servo motors can apply great forces to supply the required movement. They are usually electrical or electronic devices. Also motors that use hydraulic, pneumatic or magnetic principles can be provided for that purposes.

2.4.7 Generator

In hydropower plants, mechanical energy is converted to electrical energy by generators. The phenomenon producing an electrical current in a conductor, discovered by Michael Faraday, involves moving a copper coil through a stationary magnetic field (Warnick, 1984). Generators have two main parts. Rotor is the rotating part (moving copper coil) which is driven by the turbine and stator is the stationary part (magnetic field).

2.4.8 Powerhouse

Powerhouses of hydropower plants provide protective housing for mechanical and electrical equipments. It usually consists of superstructure and substructure. Superstructure contains cranes and control units. Substructure consists of supports made of reinforced concrete and steel for spiral case, turbine, generator and the draft tube.

2.4.9 Draft Tube and Tailwater Channel

Draft tube is used to direct the water from turbine to tailwater channel. The tailwater channel is for discharging the water that has passed from turbines and draft tube, to the natural stream. Depending on topographical conditions water may be directly discharged into stream or with an open channel.

CHAPTER 3

HYDRAULIC TRANSIENTS IN SMALL HYDROPOWER PLANTS

3.1 General Concept of Hydraulic Transients

3.1.1 Transient Flow

The term *steady* in a flow means that velocity, pressure and discharge do not change with time, at a point, in a flow field. If the mean values of these flow parameters vary with time, the flow is said to be *unsteady*. Steady flow equations are derived from unsteady flow equations by neglecting the time dependent terms. So it can be said that steady flow is a special case of unsteady flow. Transient flow, which is unsteady flow, represents the change in flow conditions between two successive steady states. One of these steady states may be the rest state. Any change in the conveyance or control of the fluid of a hydraulic component responded by the transient flow in the system.

Generally, transient flow can be classified in two types: quasi-steady flow and the true transient flow. In *quasi-steady* flow the variation of flow parameters are gradual and over short time intervals. An observer judges the flow as steady. Drawdown in the water level of a reservoir or a tank can be considered as quasi-steady. However, *true transient* flow is characterized by the fluid inertia, elasticity of the fluid and pipe.

Waterhammer is a type of the true transient flow, and elastic properties of both pipe and fluid are considered with inertial effects.

3.1.2 Waterhammer

If steady state velocity in a pipe system rapidly changes, a hydraulic transient, pressure surge occurs. The sound of the moving water which is being suddenly stopped in a pressurized pipe is like the hammering sound; therefore, this phenomenon is called as waterhammer. Actually the cause of that sound is the travelling surge pressure, which has nearly acoustic speed. Either an acceleration or deceleration of the flow adds extra stress of dynamic loads to the pipe, valves, supports and other system equipments. Typical events that result in such changes in a pipe flow are as follows:

- Variation in valve opening
- Mechanical failures in flow control equipments or power failures
- Human errors in operation
- Emergency shutdown of the units
- Sudden changes in water surface elevation of reservoirs, forebays or pressure tanks
- Filling or emptying of penstocks for maintenance
- Vibration of valves or impellers of turbines and pumps

These changes cause the conversion of the kinetic energy of the fluid into the elastic energy and hence waterhammer. When these excitations occur, the velocity of the liquid column cannot adjust itself to the new situation due to the inertia of the fluid. This inertia builds up a transient pressure and with this transient pressure, pipe and the fluid are deformed. Then surge pressure travels with the sound speed through the pipe and harms every part of the pipe where it reaches (Lüdecke and Kothe, n.d.). Then, the surge pressure dampens gradually; therefore, it threats the system for a long time and during this transient state destructive situations, like *resonance*, may occur. Before defining the resonance, the terms *period* and *frequency* should be known. *Period* is the time interval at which transient flow conditions are repeated. It is expressed in seconds. *Frequency* is cycles per second for a periodic flow. When the frequency of the transient flow coincides with the natural frequency of the pipeline, the phenomenon called *resonance* occurs in the system. If the pipeline is not properly supported and/or anchored, the whole or some part of it may be exposed to destructive resonant vibrations. For the buried penstocks, the harmful effects of resonance can be ignored while carrying dynamic analysis of the structure.

In the year 1897, Nikolai Joukowsky, a Russian scientist, conducted a series of experiments on Moscow drinking water supply system and published his both experimental and theoretical studies in 1898 (Lüdecke and Kothe, n.d.). He developed that the pressure change in a fluid caused by an instantaneous excitation can be calculated by:

$$\Delta P = \pm \rho a \Delta V \tag{3.1}$$

$$\Delta V = V_f - V_0 \tag{3.2}$$

where;

ΔP :	Change in the pressure, [N/m²]	ΔV :	Change in the flow velocity, [m/s]
ρ :	Density of the fluid, [kg/m³]	V_f :	Final velocity, [m/s]
a :	Pressure wave propagation velocity through the fluid, [m/s]	V_0 :	Initial velocity, [m/s]

This equation can only be applied for rapid closures of valves and wicket gates which take place shorter than the wave reflection time, T_r. It is the time needed for pressure wave to travel up and down entire length of the pipeline. If the length of the pipe is “L” meters and the wave propagation velocity is “ a ” meters per second, the wave reflection time is equal to $2L/a$ seconds.

Changes in flow parameters in a pipe system may cause serious consequences if it is not properly designed for all operational conditions. As it is not possible to avoid pressure transients while operating a pipeline system, it should be designed according to controllable pressure limits. Otherwise, during a transient event, when pressure rises, pipe may burst, connections may be damaged and some water can be lost, water quality may be affected because of high shear stresses resulting in detachment of protective inner pipe material and turbomachinery, valves and other equipments can be damaged. If pressure falls, pipe may collapse, buckle or disintegrate, groundwater or wastewater may be drawn into the pipe from connections. Another thing that may be considered during the design is protective measures. The consequences of waterhammer cost much more than the cost of the preventive and control measures. These measures for SHPs will be discussed in following topics.

3.1.3 Numerical Simulation of Waterhammer

3.1.3.1 Derivation of Wave Propagation Equation

Wave speed definition is derived by considering a control volume in a pipe section and a pipe which are exposed to an excitation. First, the unsteady momentum equation is applied to the control volume and then, continuity concept is regarded in the pipe. In the control volume, after the initiation of transient flow, it is accepted that the wave is moving to the left from right with an absolute speed of $(a - V_0)$. The change of velocity, ΔV, is accompanied by a change in pressure, ΔP. With that change, a force is exerted in negative x direction with a magnitude of ΔPA. Writing the momentum equation in x direction shows the resultant force and it is equal to the time rate of change of momentum in this direction (see Figure 3.1).

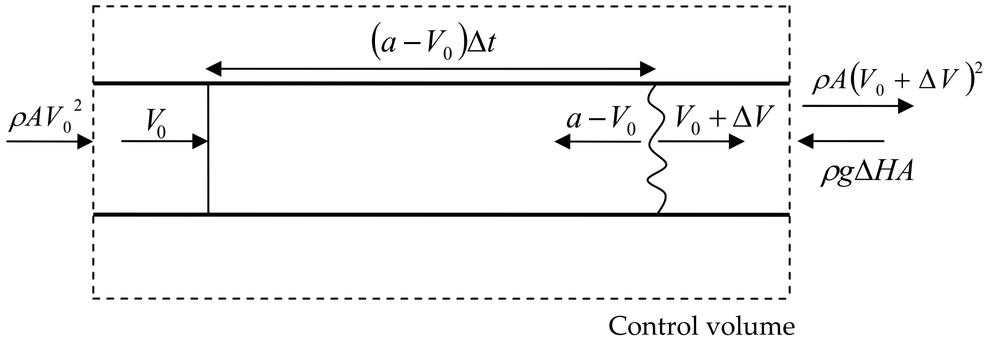


Figure 3.1 Selected Control Volume for the Unsteady Momentum Equation Application (Wylie et al., 1993)

Mathematically momentum equation means:

$$\sum \vec{F} = \frac{\partial}{\partial t} \int_{C.V.} \vec{V} \rho d\forall + \int_{C.S.} \vec{V} \rho (\vec{V}.\vec{n}) dA \tag{3.3}$$

Using the Eq. (3.3), it can be written that

$$-\gamma \Delta H A = \rho A (a - V_0) \Delta V + \rho A (V_0 + \Delta V)^2 - \rho A V_0^2 \tag{3.4}$$

where;

γ :	Unit weight of fluid, [N/m^3]	V_0 :	Initial velocity, [m/s]
ρ :	Density of the fluid, (ρ / g), [kg/m^3]	ΔV :	Change in the flow velocity, [m/s]
g :	Gravitational acceleration, [m/s^2]	a :	Pressure wave propagation velocity through the fluid, [m/s]
A :	Cross sectional are of the pipe, [m^2]	ΔH :	Change in the pressure head, [m]

The term "ΔV^2" is very small compared with other terms, therefore it can be ignored and Eq. (3.4) can be simplified as:

$$\Delta H = -\frac{a\Delta V}{g}\left(1+\frac{V_0}{a}\right) \cong -\frac{a\Delta V}{g} \tag{3.5}$$

As the speed of sound in water is much greater than the steady flow velocity, $\frac{V_0}{a}$ is near to zero and can be neglected. If the flow stopped completely, $\Delta V = -V_0$ and $\Delta H = aV_0 / g$.

Since $\Delta P = \rho g \Delta H$, pressure change in the pipe,

$$\Delta P = -\rho a \Delta V \quad \text{(Recall equation 3.1)} \tag{3.6}$$

To determine the numerical value of wave speed, continuity equation is applied to a pipe.

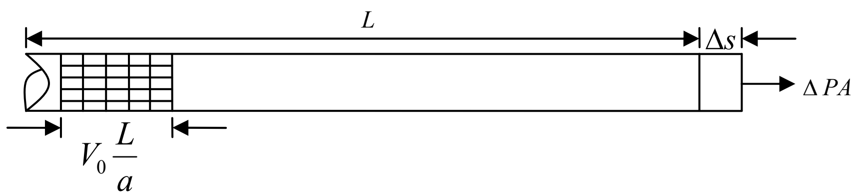


Figure 3.2 Continuity Concept in the Pipe during Sudden Stoppage of the Flow

If the flow suddenly stops, the transient pressure will be accommodated within the pipe by increasing its cross sectional area, and by filling the extra volume due to pipe extension, and by compressing the liquid and changing its density. While these changes are occurring in the system, a mass enters the pipe that has not been affected by the transient excitation yet. After sudden stoppage of the flow, during

L/a seconds, the amount of that mass is $\rho AV_0 \frac{L}{a}$. In equation form, this concept can be expressed as:

$$\rho AV_0 \frac{L}{a} = \rho L \Delta A + \rho A \Delta s + LA \Delta \rho \tag{3.7}$$

where;

L :	Length of the pipe, [m]	Δs :	Length of pipe extension, [m]
ΔA :	Change in the pipe cross sectional area, [m²]	$\Delta \rho$:	Change in density of the fluid, [kg/m³]

Since the flow is suddenly stopped and pipe has stretched in length, depending on how it is supported, the final velocity of the flow, V_f, will be $\Delta s \frac{a}{L}$. Hence, the velocity change ΔV is equal to $\left(\Delta s \frac{a}{L} - V_0 \right)$. By use of that in Eq. (3.7) to eliminate V_0,

$$-\frac{\Delta V}{a} = \frac{\Delta A}{A} + \frac{\Delta \rho}{\rho} \tag{3.8}$$

Then, using Eq. (3.5) to eliminate ΔV,

$$a^2 = \frac{g \Delta H}{\frac{\Delta A}{A} + \frac{\Delta \rho}{\rho}} \tag{3.9}$$

The bulk modulus of elasticity, K, of a fluid is defined as

$$K = \frac{\Delta P}{\Delta \rho / \rho} = -\frac{\Delta P}{\Delta \forall / \forall} \tag{3.10}$$

On the basis of Eq. (3.10), Eq. (3.9) becomes,

$$a^2 = \frac{K/\rho}{1 + \left(\frac{K}{A}\right)\left(\frac{\Delta A}{\Delta P}\right)} \tag{3.11}$$

Finally, by using the relation between circumferential tensile stress and strain, Eq. (3.11) appears in a simpler form:

$$a = \sqrt{\frac{K/\rho}{1 + \left(\frac{K}{E}\right)\left(\frac{D}{e}\right)}} \tag{3.12}$$

3.1.3.2 Momentum Equation

Momentum equation is also known as the equation of motion. It is derived from a control volume of conical tube which is shown in Figure 3.3. The main goal of the numerical simulation of waterhammer is to determine the velocity, V or discharge, Q and pressure, P or head, H at any point at any time during a transient event. Therefore, both momentum and continuity equations are written in terms of centerline pressure, P(x,t) and centerline velocity, V(x,t). In this concept x and t are independent variables, P and V are dependent variables.

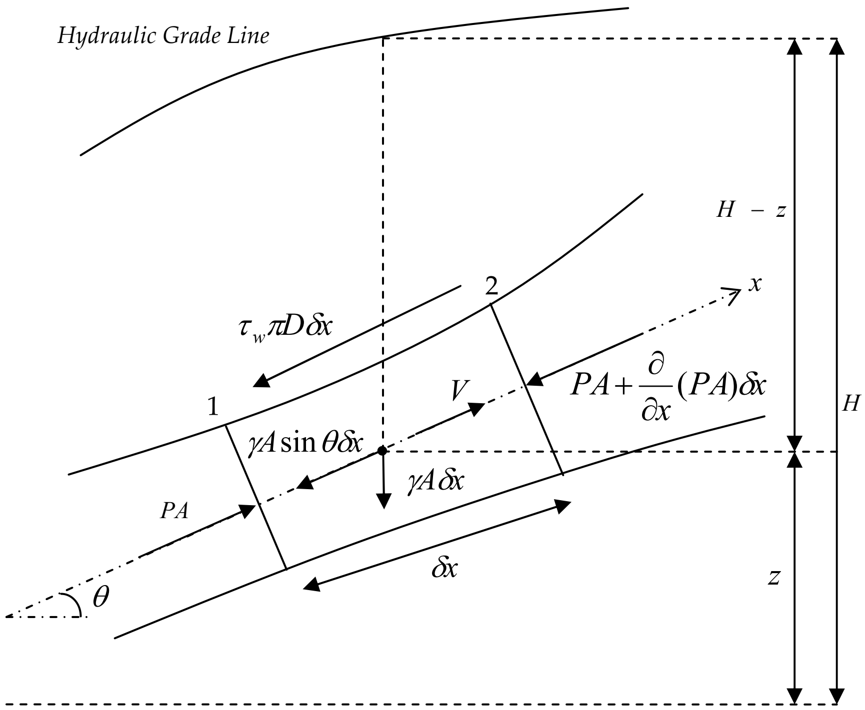


Figure 3.3 Notation for Momentum Equation

With reference to the figure, sum of all forces exerted on the control volume (CV) is equal to the summation of time rate of change of momentum in the CV and momentum flux through the control surface (CS) 1 and 2. Sum of all forces in x direction is

$$\sum F_x = PA - \left[PA + \frac{\partial}{\partial x}(PA)\delta x \right] - \tau_w \pi D \delta x - \rho g A \sin\theta \delta x \tag{3.13}$$

The rate of change of linear momentum in the CV is

$$\frac{\partial}{\partial t} \int_{C.V.} \rho V d\forall = \frac{\partial}{\partial t}(\rho V A)\delta x \tag{3.14}$$

The linear momentum flux through the CS is

$$\frac{\partial}{\partial x}(\rho V^2 A)\delta x \tag{3.15}$$

Then, the complete momentum equation becomes

$$-\left(A\frac{\partial P}{\partial x}+\tau_w \pi D+\rho g A\sin\theta\right)\delta x=\frac{\partial}{\partial t}(\rho VA)\delta x+\frac{\partial}{\partial x}(\rho V^2 A)\delta x \tag{3.16}$$

After dividing Eq. (3.16) by δx and then rearranging, it can be divided by ρA and can be simplified.

$$\frac{\partial V}{\partial t}+V\frac{\partial V}{\partial x}+\frac{1}{\rho}\frac{\partial P}{\partial x}+g\sin\theta+\frac{4\tau_w}{\rho D}=0 \tag{3.17}$$

This equation is called equation of motion.

3.1.3.3 Continuity Equation

Continuity equation is derived from the application of the law of conservation of mass. The same control volume used in the derivation of momentum equation, which is shown in Figure 3.4, is considered. The fluid inside is single phase liquid and compressible, conduit walls are elastic and cross section is constant. Therefore, due to pressure changes, control volume may stretch. It is assumed that the flow is one dimensional and pressure is uniform in the control surface. Continuity equation states that the time rate of change of mass inside the control volume is equal to the net mass flux across the entire control surface sections.

$$\frac{\partial}{\partial t}\int_{C.V.}\rho d\forall+\int_{C.S.}\rho(\vec{V}.\vec{n})dA=0 \tag{3.18}$$

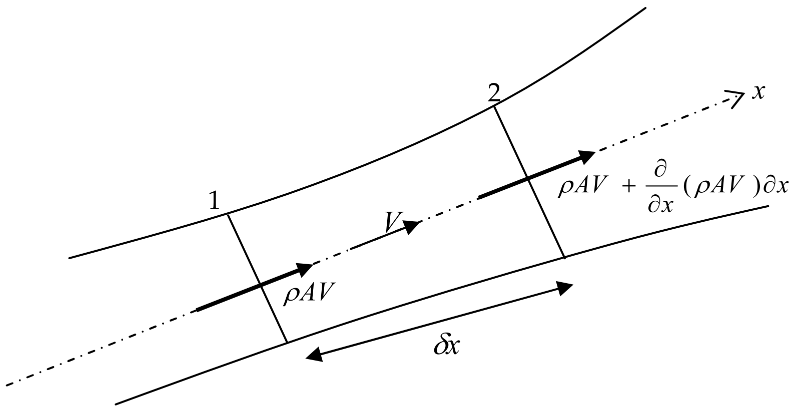


Figure 3.4 Notation for Continuity Equation

Regarding that the ρ is constant in the control surface

$$\frac{\partial}{\partial t}(\rho A)\delta x + \left[\rho AV + \frac{\partial}{\partial x}(\rho AV)\delta x\right] - \rho AV = 0 \tag{3.19}$$

After simplifications, Eq. (3.19) becomes

$$\frac{\partial}{\partial t}(\rho A) + \frac{\partial}{\partial x}(\rho AV) = 0 \tag{3.20}$$

The differentiation of Eq. (3.20) by parts, and then substitution of the definition of bulk modulus of elasticity of the fluid, given in Eq. (3.10), and the definition of modulus of elasticity of the pipe into this equation yields

$$\left(\frac{1}{K} + \frac{D}{Ee}\right)\left(\frac{\partial P}{\partial t} + V\frac{\partial P}{\partial x}\right) + \frac{\partial V}{\partial x} = 0 \tag{3.21}$$

Substituting Eq. (3.11) into Eq. (3.21) and simplifying the resulting equation give

$$\frac{\partial P}{\partial t} + V\frac{\partial P}{\partial x} + \rho a^2 \frac{\partial V}{\partial x} = 0 \tag{3.22}$$

This is the general form of the continuity equation.

3.1.4 Solutions of Waterhammer Equations with Method of Characteristics

The equations describing transient flow in closed conduits (Eqs. 3.17 and 3.22) are non linear partial differential equations. These two equations can be transformed into four ordinary differential equations by method of characteristics. Then, these latter equations can be integrated to yield finite difference equations which can be conveniently handled numerically (Wylie et al., 1993).

To start with, the term $g\sin\theta + \frac{4\tau_w}{\rho D}$ in Eq. (3.17) is defined as F. Then, continuity and momentum equations can be identified as L_1 and L_2

$$L_1 = \frac{\partial P}{\partial t} + V\frac{\partial P}{\partial x} + \rho a^2 \frac{\partial V}{\partial x} = 0 \tag{3.23}$$

$$L_2 = \frac{\partial V}{\partial t} + V\frac{\partial V}{\partial x} + \frac{1}{\rho}\frac{\partial P}{\partial x} + F = 0 \tag{3.24}$$

where

$$F = g\sin\theta + \frac{4\tau_w}{\rho D} \tag{3.25}$$

Linear combination of Eqs. (3.23) and (3.24) can be considered as

$$L = L_1 + \lambda L_2 = 0$$

By writing Eqs. (3.23) and (3.24) in that combination form

$$\left[\frac{\partial P}{\partial t} + \left(V + \frac{\lambda}{\rho}\right)\frac{\partial P}{\partial x}\right] + \lambda\left[\frac{\partial V}{\partial t} + \left(V + \frac{\rho a^2}{\lambda}\right)\frac{\partial V}{\partial x}\right] + \lambda F = 0 \tag{3.26}$$

Then, from calculus

$$\frac{\partial P}{\partial t} + \left(V + \frac{\lambda}{\rho}\right)\frac{\partial P}{\partial x} = \frac{dP}{dt} \qquad \text{if} \qquad V + \frac{\lambda}{\rho} = \frac{dx}{dt} \tag{3.27}$$

$$\frac{\partial V}{\partial t}+\left(V+\frac{\rho a^2}{\lambda}\right)\frac{\partial V}{\partial x}=\frac{dV}{dt} \quad \text{if} \qquad V+\frac{\rho a^2}{\lambda}=\frac{dx}{dt} \tag{3.28}$$

So Eq. (3.26) appears in a simpler from

$$\frac{dP}{dt}+\lambda\frac{dV}{dt}+\lambda F=0 \tag{3.29}$$

The definition of unknown multiplier can be made by using the constraints in Eqs. (3.27) and (3.28)

$$\lambda=\pm\rho a \tag{3.30}$$

Now by substituting the values of the λ into the constraints in Eqs. (3.27) and (3.28) and ignoring the small flow velocity compared with acoustic speed

$$\frac{dx}{dt}\cong\pm a \tag{3.31}$$

This equation demonstrates the change in position of wave related to the change in time. If two values of λ is substituted into Eq. (3.29), it leads to two sets of equations which are called *characteristic* C^+ and C^- equations.

$$\frac{1}{\rho}\frac{dP}{dt}+a\frac{dV}{dt}+aF=0 \qquad \text{if} \qquad \frac{dx}{dt}=+a \tag{3.32}$$

$$\frac{1}{\rho}\frac{dP}{dt}-a\frac{dV}{dt}-aF=0 \qquad \text{if} \qquad \frac{dx}{dt}=-a \tag{3.33}$$

It should be noted that Eqs. (3.32) and (3.33) are valid if their constraints are satisfied. This provides the elimination of one independent variable, x and conversion of non linear partial differential equations of transient flow into ordinary differential equations. However, this simplification comes with a price. Equations (3.17) and (3.22) are valid everywhere in x-t plane; whereas, Eqs. (3.32) and (3.33) are valid only along their straight lines which are described by their constraints.

In x-t plane which is shown in Figure 3.5, two straight lines having slopes of $\pm 1/a$ represent the validation of Eqs. (3.32) and (3.33). These are called C^+ and C^- characteristic lines. Physically they represent the followed path of the transient disturbance. This discussion can be applied into a pipe by dividing it into reaches and forming nodes. If it is divided into N equal reaches, there will be N+1 formed nodes to be solved for each time step. The time step can be calculated as $\Delta t = \Delta x / a$.

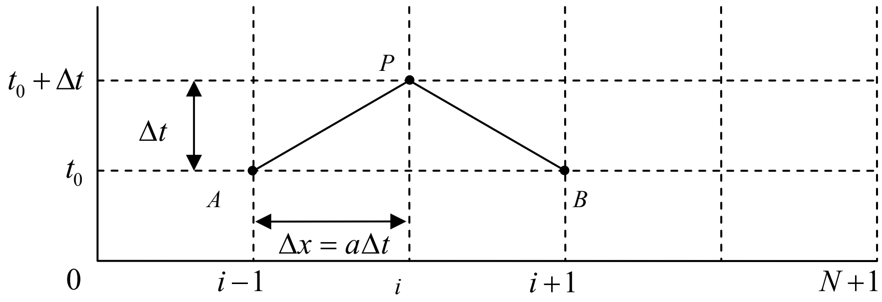


Figure 3.5 Characteristic Lines in Time Space Domain

If the dependent flow parameters H and V at points A and B are known, Eqs. (3.32) and (3.33) can be integrated along the line AP and BP, respectively. These integrations yield two equations with two unknowns, H and V at point P. Then, by solving these two equations simultaneously, dependent flow parameters of point P can be gathered.

It is assumed that shear stress in transient flow is the same with that of steady flow and following Darcy-Weisbach definition for shear stress can be used for simplification.

$$\tau_w = \frac{\rho f V|V|}{8} \tag{3.34}$$

therefore,
$$F = g\sin\theta + f\frac{V|V|}{2D} \tag{3.35}$$

Also, the relation between pressure and head can be given as, $P = \rho g(H - \sin\theta)$. Now by multiplying Eq. (3.32) by $a\frac{dt}{g} = \frac{dx}{g}$ and by introducing the pipeline area to write the equation in terms of discharge in place of velocity and applying the manipulations which are given above, the equation can be placed in a form suitable for integration along the C^+ characteristic line.

$$\int_{H_A}^{H_P} dH + \frac{a}{gA}\int_{Q_A}^{Q_P} dQ + \frac{f}{2gDA^2}\int_{X_A}^{X_P} Q|Q|dx = 0 \tag{3.36}$$

The integration of Eq. (3.34) and a similar integration of Eq. (3.33) along the C^- line, following equations are obtained.

$$H_P - H_A + \frac{a}{gA}(Q_P - Q_A) + \frac{f\Delta x}{2gDA^2}Q_A|Q_A| = 0 \tag{3.37}$$

$$H_P - H_A - \frac{a}{gA}(Q_P - Q_B) + \frac{f\Delta x}{2gDA^2}Q_B|Q_B| = 0 \tag{3.38}$$

Above two compatibility equations are basic algebraic relations that describe the transient pipe flow in a pipeline. Both these equations can be solved for H_P and following equations can be written.

$$C^+ : H_P = H_A - B(Q_P - Q_A) - RQ_A|Q_A| \tag{3.39}$$

$$C^- : H_P = H_B + B(Q_P - Q_B) - RQ_B|Q_B| \tag{3.40}$$

in which

$$B = \frac{a}{gA} \qquad \text{and} \qquad R = \frac{f\Delta x}{2gDA^2}$$

In general form

$$C^+ : H_{Pi} = C_P - BQ_{Pi} \qquad \text{and} \qquad C_P = H_{i-1} + BQ_{i-1} - RQ_{i-1}|Q_{i-1}| \tag{3.41}$$

$$C^- : H_{Pi} = C_M + BQ_{Pi} \qquad \text{and} \qquad C_M = H_{i+1} - BQ_{i+1} + RQ_{i+1}|Q_{i+1}| \tag{3.42}$$

3.2 Waterhammer in Small Hydropower Plants

A hydraulic turbine that is connected to a generator feeding an electricity grid has to be operated at a constant rotational speed to generate electricity at a constant frequency. Any change in frequency will result in a change in generator and turbine rotational speeds. The disturbances that change turbine speed is followed by the governor action which tries to keep the turbine at synchronous speed by closing or opening wicket gates in reaction turbines or by a change in the position of jet deflector and closing or opening of needle valves in impulse turbines. Both of these immediate actions cause changes in turbined flow parameters. That means, there will be a transient state which will generate waterhammer pressures in the penstock and spiral case of hydropower plants, resulting in excessive pressure rises and drops.

Waterhammer is a very important constraint in small hydropower plants as they are generally equipped with small inertia turbines and long penstocks. As they are usually installed in mountainous areas, they have long penstocks in order to increase the available gross head over the turbine. They also have small installed capacities, and small inertia turbines.

Hydraulic transient is the key factor that determines the operational safety in a small hydropower plant. Any disturbance in the system will influence the overall response and the stability. The cases that cause system safety to fail are the pipe burst or collapse, vapor and air pocket formation, water column separation, air entrance into the pipe and overtopping of water from forebay walls. Severe hydraulic transients lead to these accidents resulting in both economical and deadly problems. Moreover, they affect the plant's reliability and production quality.

In light of these facts, from preliminary and feasibility studies to final design stage, waterhammer phenomenon should be considered in design of the safest and the most economic layout. The algorithm of the design process in the transient analysis of a small hydropower plant can be illustrated as follows (Ramos et al., 2000):

Preliminary and Feasibility Studies and Early Design Phases

Preliminary waterhammer analysis for basic situations.

Objective: to guarantee a feasible and economic solution without special protection devices .

↓

Detailed Design Studies for Tendering and Contracting

Detailed transient analysis and studies including the selected protection systems in order to obtain the hydraulic response to usual and extreme turbine operational conditions.

Objective: to determine the main parameters of penstocks and/or tunnels, to specify the main component characteristics of surge control equipments and to determine the closure times of gates and/or valves.

↓

Final Studies for Construction and Operation

Detailed transient analysis and computer simulations including the characteristics of the selected equipment and final specifications of the civil works.

Objective: to verify the safety level of the hydro system and to specify operation rules.

While applying this methodology, these minor steps are followed to accomplish a complete transient study:

- Determination of possible excitations that will start hydraulic transients
- Definition of physical origin of the disturbance, and characterization its mathematical model
- Evaluation of the maximum and the minimum pressure transients
- Selection and analysis of protection devices that control transient pressure in an acceptable manner if required
- Determination of plant's operational procedures.

There are three possible conditions that a small hydropower plant can be operated, and they must be considered during design studies and transient analysis of a SHP (Ramos et al., 2000):

- In *normal operating conditions,* expected and usual disturbances that cause waterhammer pressures are considered. They have maximum safety factors, and they are the most likely to occur events.
- In *emergency operating conditions,* probable but unexpected disturbances are regarded. They have average safety factors.
- In *exceptional operating conditions,* unexpected and highly unlikely to occur disturbances are taken into consideration. As they may cause severe damages to the system, the factor of safety of these conditions is very small.

Table 3.1 lists the events that cause waterhammer with related operational conditions.

Table 3.1 Events with Operational Conditions for Design Studies and Transient Analysis of a SHP (adapted from Ramos et al., 2000)

Normal Operating Conditions	• Transition between steady state flow conditions for different turbine discharges. • Load rejection followed by closure of the flow control equipment during operation of the turbine. • Penstock filling and emptying situations.
Emergency Operating Conditions	• Mechanical failure on closure mechanism of the turbine wicket gate or on jet deflector. • Rapid closure of the flow control equipment after an instant load rejection or due to malfunction. • Turbine runaway condition. • Rapid start up of the plant. • Failure on the waterhammer protection device.
Exceptional Operating Conditions	• Complete failure of the turbine wicket gate closure mechanism or jet deflector and needle valves with a instant stoppage of the flow ($T_r < 2L/a$). • Column separation in the penstock. • Resonance or oscillatory flow in the penstock. • Sudden change in the elevation of the forebay. • Seismic waves on the surface of the forebay.

All these operational conditions should be considered to design the safest and reliable layout. Detailed transient analyses of the systems ensure no or negligible waterhammer damages to the penstocks or other hydraulic equipments of SHPs.

3.2.1 Causes of Waterhammer in Small Hydropower Plants

Besides general operations, disturbances and changes that produce waterhammer, which were previously mentioned, following changes in the system cause transient state conditions in the penstocks of small hydropower plants:

- Load rejection
- Load acceptance
- Load variation

If the operational faults and mechanical failures are not regarded, severity of a hydraulic transient in a small hydropower plant is directly dependent on the electric grid which the plant is connected to. A unit that is synchronized to a large electric grid may not be exposed to severe load fluctuations. On the other hand, a system that is connected to an isolated grid may be imposed much more from load oscillations and exposed to waterhammer effects.

3.2.1.1 Load Rejection

The turbine generator system is connected to a distribution grid, which can be national or isolated. Any failure in transmission lines of this grid, any failure in acceptance of electrical load and any sudden drop in power demand will cause *load rejection* in the system. When the produced electrical load is rejected from the grid, the external load on the system is removed. That causes a rapid increase on frequency of the grid and rotational speeds of the turbine and generator. To prevent the turbine from reaching its runaway speed and keep the speed rise with an acceptable manner, wicket gates or needle valve must respond quickly. This action may result in high transient pressures in penstocks.

Load rejection must be ended up as quickly as possible with the stoppage of the turbine generator system to limit the speed rise of the units. These types of stopping operations, that follow a load rejection, are called *regular stop* of turbines. The time required for the closure of the gates or nozzles is determined after a detailed transient analysis. In the application, closure is controlled by the governor.

The rejection is called *instant load rejection* if the electrical load on the turbine instantaneously falls to zero by the disconnection of generator from turbine, and the operation followed by this type of rejection is named as *instant stop* of turbines.

Contrary to load rejection, if the demand from the grid is greater than the supplied power, the turbine will slow down and the frequency of the grid will fall down. This action will result a brown out in the grid.

3.2.1.2 Load Acceptance

When the turbine and generator couple gets connected to the electrical grid or starts operating, *load acceptance* occurs. Generally, during *regular start* of turbines, wicket gates are opened to speed no load gate position before they get connected to the grid. Speed no load (SNL) gate position is the minimum gate opening that provides synchronous speed to the turbine with zero power output. When the generator starts producing electricity, wicket gates have to be opened as quickly as possible from SNL position to meet the power demand. This action causes waterhammer pressures in the penstocks of SHPs. Load acceptance is followed by the formation of low pressures in the penstock. Although transient pressures generated by load acceptance are relatively less severe than those resulting from load rejection, they must be considered during transient analysis to simulate the existence of vapor formation.

3.2.1.3 Load Variation

The basic characteristic of the electrical load is that the demand is not constant, but a function of time. It varies with day hours in a day, weekdays in a week and seasons in a year. This situation may create difficulties in the operation of SHPs. The balance between supply and demand must be provided to prevent the change in the frequency and the turbine unit speed. In case of the load variation, the wicket gate opening is adjusted by the governor to control the turbine flow so that the balance between supply and demand can be provided. This change in flow accelerates or decelerates the water column inside the penstock resulting in pressure variations.

The waterhammer pressures generated by the load variation may not be significant compared with those resulting from load rejection or other disturbances from the view of hydraulic design.

3.2.2 Waterhammer Response and Modeling of Different Turbine Types

3.2.2.1 Pelton Turbine

The power output and unit speed of a Pelton turbine is adjusted by the needle valve(s). Discharge through the turbine is controlled by opening or closing of the needle valve(s), which takes place in the mouth of the nozzle(s), and with the position of the jet deflector. Figure 3.6 is showing the parameters directly affecting the turbine flow: nozzle diameter, d_m and maximum needle stroke, $s_{\max}$. Contrary to reaction turbines, flow in the penstock of the power plant is only function of the position of the needle valve; it is not dependent on the turbine speed.

Closure operation in a Pelton turbine is directed by the nozzle and the deflector simultaneously. When load rejection occurs or turbine disconnects from electrical grid, needle(s) start to close gradually to SNL position. Simultaneously, nozzle(s) are directed from the wheel rapidly. Therefore, the disturbance source causing transient events in the penstock is the needle movement in a Pelton turbine. As in all computational models of hydraulic transients, the definition and determination of the boundary condition and characteristics equation of the Pelton turbine must be done. However, these are not well defined in the literature. Waterhammer analysis of hydropower plants with Pelton turbines is generally simplified with the representation of the turbine with a valve.

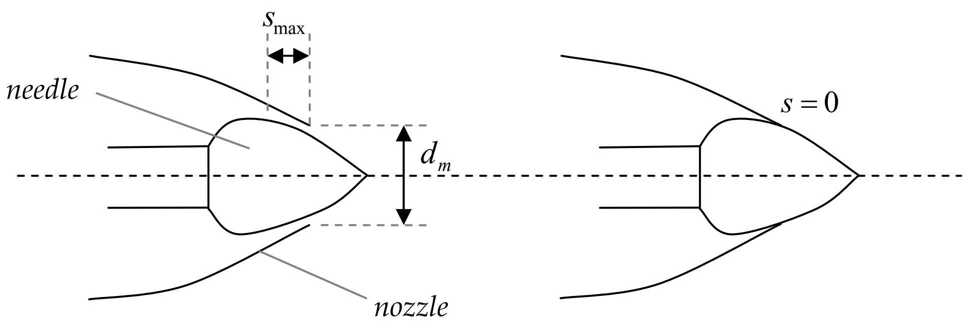


Figure 3.6 Scheme of a Pelton Turbine Nozzle and Needle Valve (Karadzic et al., 2009)

In the computational model, head loss variation, upstream boundary conditions, nozzle characteristics, instantaneous head loss across the nozzle and discharge are taken into consideration. If the reference point for the hydraulic grade line is taken at the valve, instantaneous discharge through the nozzle can be written as follows:

$$Q_P = K_Q A_m \sqrt{2g\Delta H} \tag{3.43}$$

In Eq. (3.43), K_Q is the nozzle discharge coefficient, A_m is the nozzle discharge area and ΔH is the head drop across the needle valve. The relationship between the normalized nozzle discharge coefficient, K_Q and the normalized s/d_m value is shown in Figure 3.7. In the figure, $(K_Q)_{\max}$ is the maximum nozzle discharge coefficient occurring at $(s/d_m)_{\max}$.

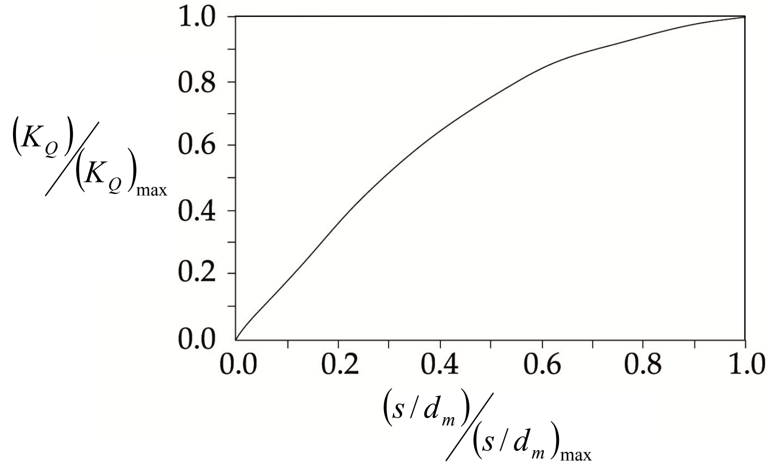


Figure 3.7 Pelton Turbine Nozzle Discharge Coefficient (Karadzic et al., 2009)

The dimensionless nozzle opening parameter that expresses the needle closing law is as follows:

$$\tau = \frac{s}{s_{\max}} \tag{3.44}$$

In Eq. (3.44), s is the ratio of the needle stroke. Computational model can be simulated through the solution of upstream boundary condition equations and Eq. (3.43) by method of characteristics. Different needle closure scenarios can be specified to control the transient pressures along the penstock in Pelton turbines.

3.2.2.2 Francis Turbine

Modeling of a Francis turbine includes relatively more parameters that represent turbine characteristics. In addition to the instantaneous head and discharge equations, dynamic equations of the turbine unit rotating parts must be solved for its computational simulation. If various transient regimes are considered for the waterhammer analysis, the characteristic curves of the turbine showing the relations between its parameters must be made available.

Although governed turbines have dynamic response equations for their governors, hydraulic transient analysis of them does not need governor's internal operational equations.

A Francis turbine at the end of a single penstock is described with the following equations (Wylie et al., 1993):
Characteristic head and torque curves equations:

$$H_P = H_R\left(\alpha^2 + \nu^2\right)\left(A_0 + A_1 x\right) \tag{3.45}$$

$$T = T_R\left(\alpha^2 + \nu^2\right)\left(B_0 + B_1 x\right) \tag{3.46}$$

in which H_P is the instantaneous head, T is the torque applied to the unit, the subscript R is the indication of rated values, α is the dimensionless speed ratio, υ is the dimensionless discharge ratio, A_0, A_1, B_0 and B_1 are operational coefficients and $x = \tan^{-1}(v/a)$.
The turbine torque equation:

$$\frac{T}{T_R} - \frac{P_G}{\alpha}\frac{C_1}{T_R\omega_R} = \frac{\omega_R}{T_R} I \frac{d\alpha}{dt} \tag{3.47}$$

where P_G is the power absorbed by the generator, ω is the angular turbine speed, I is the polar moment of inertia of rotating fluid and turbine and C_1 is the unit conversion coefficient.

Solving these equations with C^+ characteristic equations yields discharge and head values of the turbine.

3.2.3 Protective Measures for Waterhammer in Small Hydropower Plants

During design studies of a small hydropower plant, the results of the transient analysis may force the designer to select big safety factors not to cause both economical and deadly problems during the lifetime of the system. However, the selection of the pipe material, its dimensions and profile etc. according to maximum and minimum waterhammer pressures might not be economical. Therefore, remedial structures, devices or control methods are used to prevent unwanted transient conditions such as column separation, turbine overspeed and excessive pressures. As every system has unique characteristics, the methods for controlling transients vary with different operating conditions. The determination of the appropriate surge protection measure for a SHP can be done only after carrying out optimization, effectiveness and dependability studies. Hence, many alternatives which give considerable response are regarded and the best one is selected (Wylie et al., 1993 and Chaudhry, 1987).

The protective measures can respond by different ways: by supplying or removing water or by dissipation of energy (Ramos et al., 2000). Common protective measures for SHPs are as follows:

- Surge Tank
- Air Chamber
- Valves
- Flywheel
- Safety Membrane

The method of characteristics can be used for simulating and modeling of these structures.

3.2.3.1 Surge Tank

A surge tank is an open reservoir that allows the absorption and control of waterhammer pressures and unsteady discharges by fluctuating them. During a transient event, it reflects the pressure waves and supplies or stores excess water. The friction inside the surge tank dampens the mass oscillation of water column. This leads to a considerable reduction in surge pressures. So, the length and wall thickness of the penstock can be reduced. The size of a surge tank is determined according to the amount of maximum water that must be supplied to the pipe. It can have different sizes, shapes and connection types that control the flow. The general types of surge tanks are simple, orifice, differential and one-way. Schematic representations of them are shown in Figure 3.8.

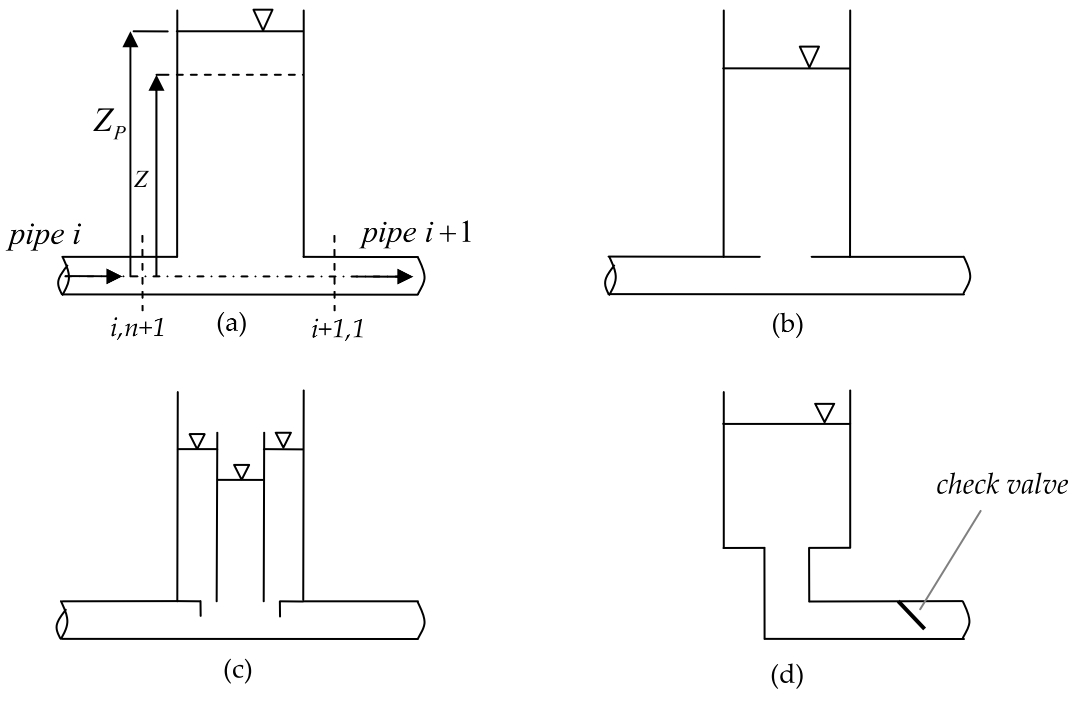


Figure 3.8 Schemes of Different Types of Surge Tanks: (a) Simple, (b) Orifice, (c) Differential, (d) One Way

The computational modeling of a simple surge tank can be done by using following characteristics equations (Chaudhry, 1987):

C^+ equation for section (i, n+1)

$$Q_{P_{i,n+1}} = \frac{C_p - H_{P_{i,n+1}}}{B} \tag{3.48}$$

C^- equation for section (i+1,1)

$$Q_{P_{i+1,1}} = \frac{H_{P_{i+1,1}} - C_M}{B} \tag{3.49}$$

The continuity equation at the surge tank connection

$$Q_{P_{i,n+1}} = Q_{P_{i+1,1}} + Q_{P_T} \tag{3.50}$$

where Q_{P_T} is the discharge into the surge tank that can be both positive or negative according to its direction; the subscripts *i* and *i+1* refer to the pipe numbers and the subscripts *1* and *n+1* refer to the section numbers.

If the minor losses are ignored at the junction and in the entrance of the surge tank, following equation can be written for piezometric heads;

$$H_{P\,i,n+1} = H_{P\,i+1,1} = Z_P \tag{3.51}$$

The heights of the liquid column in the tank at the beginning and at the end of the time step are presented with Z and Z_P, respectively.

For the liquid surface elevation at the end of the time step, if the time step size is small, it can be written that

$$Z_P = Z + \frac{1}{2}\frac{\Delta t}{A_s}\left(Q_{P_S} + Q_S\right) \tag{3.52}$$

in which Q_{P_S} and Q_S are the discharges at the end and at the beginning of the time step, respectively and A_s is the cross sectional area of the surge tank.

3.2.3.2 Air Chamber

An air chamber is a vessel and acts like a surge tank; but, it is not open to atmospheric pressure, and it has relatively small dimensions contrary to surge tanks. The chamber partially filled with water with topped air (see Figure 3.9). The filling process is performed with a pump and while being filled, the air inside the chamber is compressed with a compressor. This entrapped air absorbs the excessive surge energy. Compressibility of air prevents too large tank dimensions and provides attenuation of waterhammer pressures. As it can easily force the water out of vessel into the pipe, it is more effective than other protective measures during the formation of negative surge pressures in the penstock (Stephenson, 2002).

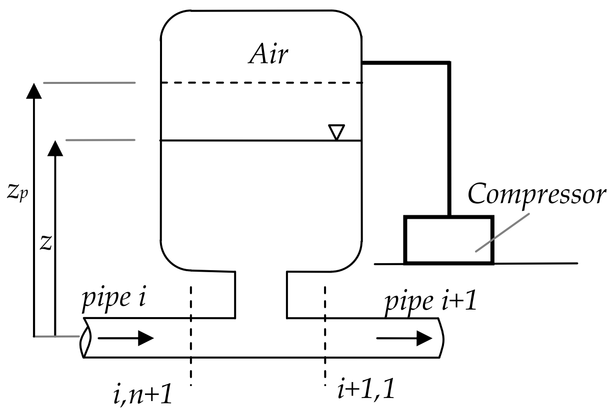


Figure 3.9 Air Chamber

This protective element can be modeled and simulated computationally on the base of following equations (Chaudhry, 1987):

C^+ equation for section (i, n+1)

$$Q_{P_{i,n+1}} = \frac{C_p - H_{P_{i,n+1}}}{B} \tag{3.53}$$

C^- equation for section (i+1,1)

$$Q_{P_{i+1,1}} = \frac{H_{P_{i+1,1}} - C_M}{B} \tag{3.54}$$

The continuity equation at the surge tank connection

$$Q_{P_{i,n+1}} = Q_{P_{i+1,1}} + Q_{P\,orf} \tag{3.55}$$

where $Q_{P\,orf}$ is the discharge through the orifice that can be both positive or negative according to its direction.

If the minor losses are ignored at the junction

$$H_{P\,i,n+1} = H_{P\,i+1,1} \tag{3.56}$$

Head loss through the orifice of the chamber can be obtained by using the following equation:

$$h_{Porf} = C_{orf} Q_{Porf} \left| Q_{Porf} \right| \tag{3.57}$$

in which C_{orf} is coefficient of orifice losses and h_{Porf} is the head loss in the orifice for a flow of Q_{Porf} .

If the air inside the chamber is assumed as a perfect gas, its behavior can be expressed with polytropic relation:

$$H^{*}{}_{Pair} \forall^{m}{}_{Pair} = C_2 \tag{3.58}$$

in which $H^{*}{}_{Pair}$ and $\forall^{m}{}_{Pair}$ are the absolute head and volume of the entrapped air, respectively; C_2 is a constant and determined from the steady state conditions of the air chamber. Following equations can be written for the entrapped air volume inside the chamber:

$$H^{*}{}_{Pair} = H_{Pi,n+1} + H_b - z_p - h_{Porf} \tag{3.59}$$

$$\forall^{m}{}_{Pair} = \forall_{air} - A_c (z_p - z) \tag{3.60}$$

$$z_p = z + \frac{1}{2}(Q_{orf} + Q_{Porf}) \frac{\Delta t}{A_c} \tag{3.61}$$

where H_b is barometric pressure head; z and z_p are the elevations of the liquid surface in the chamber at the beginning and at the end of the time step, respectively;

$\forall_{air}$ is the volume of air at the beginning of time step; A_c is horizontal cross sectional area of the chamber, and Q_{orf} is the orifice flow at the beginning of the time step. The solution of equations (3.53) to (3.61) yields the head and discharge values at the junction.

3.2.3.3 Valves

As a protective device, valves are used to discharge the water from the pipeline when the pressure exceeds a certain limit or to draw air into the pipeline to prevent the formation of vapor cavities. Commonly used valves are pressure relief valves (PRVs) in SHPs. These valves are placed near to turbines or powerhouses. By means of a spring or weight, they are loaded to open automatically when the pressure inside the penstock exceeds a prescribed pressure limit. When valve opens, it allows the outlet of pipe flow into the atmosphere and attenuates the maximum surge pressures (see Figure 3.10). A relief valve is relatively more efficient device for power plants with high heads (Ramos et al., 2000).

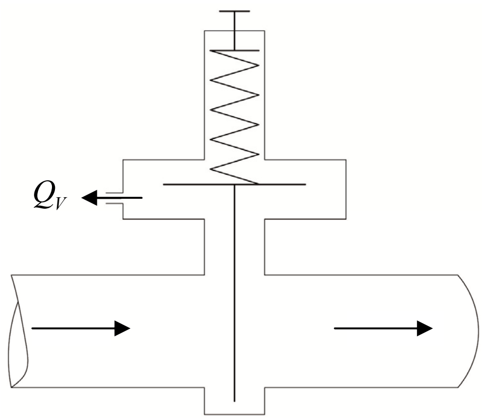


Figure 3.10 Pressure Relief Valve

A pressure relief valve coupled with a turbine can be modeled and simulated computationally on the base of turbine characteristics, hydraulic and following equations (Ramos et al., 2000):

C^+ equation for the penstock

$$H_P = C_P - BQ_P \tag{3.62}$$

where Q_P and H_P discharge and piezometric head of the penstock flow, respectively. Characteristic equation of the valve element can be written as,

$$Q_V = C_{valve}\sqrt{H_0} \tag{3.63}$$

in which Q_V is the discharge passing through the valve; C_{valve} is the valve discharge coefficient and H_0 energy loss through the valve.

3.2.3.4 Flywheel

Flywheel is a mechanical surge protection device that increases polar moment of inertia of the rotating parts of the system. It has considerably large moment of inertia which allows the control of waterhammer pressures by slowly reducing the turbine speed, the time to reach its runaway speed and increasing the stoppage time of the unit (see Figure 3.11).

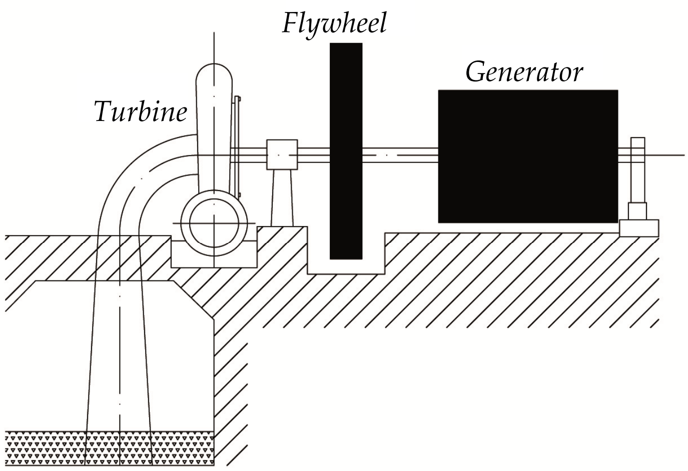


Figure 3.11 Scheme of Flywheel and Turbine-Generator System

There may be one disadvantage that makes flywheel rare in application: the increase in the moment of inertia of rotating parts may complicate the start up of the unit. The characteristic equation of the flywheel device, that can be solved simultaneously with turbine characteristics and hydraulic equations, can be given as (Ramos et al., 2000)

$$B_H - B_R = I \frac{2\pi}{60} \frac{dn}{dt} \tag{3.64}$$

where

$$I = I_{tur} + I_g + I_f \tag{3.65}$$

in which B_H is the hydraulic torque, B_R is the resistant electromagnetic torque, I is the total rotating mass inertia, I_{tur}, I_g, I_f are rotating mass inertia of turbine, generator and flywheel, respectively and n is speed of turbine.

3.2.3.5 Safety Membrane

Another protective measure which is not very common but practical one is safety membrane. They are also called as rupture disk. The device is used in many SHPs in China. It is made of a material that is weaker than the penstock's material. Usually aluminum is used for the membrane in steel penstocks. Like other surge protection devices, these weaker controlled points are placed near to the turbines. If the waterhammer pressure in the penstock rises over the design pressure, the safety membrane bursts and surge pressure is eliminated by discharging some quantity of water through the orifice of the membrane. If this discharge is incapable of keeping the pressure rise in a prescribed manner, more membranes may be used to protect the penstock and turbine. The manufacture, operation and maintenance of safety membranes are easier and cheaper than other transient control devices (see Figure 3.12).

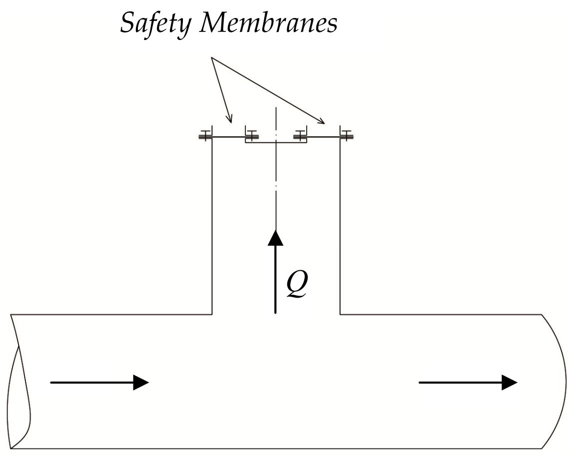


Figure 3.12 Installation of Safety Membranes on a Penstock

As a boundary condition, safety membrane can be modeled computationally with the following equation (Ni et al., 1996):

$$Q = \varphi \frac{\pi}{4} d_m N \sqrt{2gH} \tag{3.66}$$

where Q is the discharge through the safety membrane, N is the number of ruptured membranes, d_m is the diameter of the membrane, H is the transient state head and φ is the velocity coefficient.

CHAPTER 4

COMPUTER SOFTWARE

4.1 Overview of the Computer Software Used in the Study

Hydraulic transient analysis of a SHP requires detailed and complex studies, and it is a critical work. The aim of designing the safest and the most economical structure leads the engineer to consider lots of operating alternatives for a SHP. Computational and mathematical model of these alternatives consist of a number of boundary conditions, and hence many number of characteristics equations to simulate the transient behavior of the SHP and its components. Many computer codes have been developed to overcome the solution of huge number of equations with method of characteristics (MOC). However, the solution of equations itself is only a part of the transient analysis. Therefore, some computer softwares have developed to simulate a complete transient analysis in hydraulic structures.

HAMMER, developed by Bentley, is one of these waterhammer and transient analysis softwares. It helps designers to analyze the transient states of hydropower plants, complex pumping systems and piping networks. It has both a built-in steady state engine and hydraulic transient engine which uses MOC. HAMMER can compute initial steady state operating conditions of the systems, like discharge and hydraulic grades at desired points. The outputs of the steady state engine are the inputs of hydraulic transient engine and that makes the software a standalone hydraulic transient solver. The main aim of the software is to compute the hydraulic transient results along a pipe and develop cost effective surge control measures. HAMMER is developed to prevent catastrophic failures, reduce operation and maintenance costs, eliminate costly over design, minimize service interruptions and determine transient pipe forces (Bentley HAMMER, 2010).

HAMMER uses MOC for every system simulation by default. Only if it is enabled for specific reaches, for highly elevated points of pipelines, rigid column theory is used to track the formation of the air pocket and resulting water column separation and acceleration accurately.

The boundary conditions for each tool have already been defined in the software. To set up a hydraulic model, only the following data are required in addition to the information required for a steady state model (Bentley HAMMER V8i Edition User's Guide, n.d.):

- The material type, dimensions and characteristics of the pipe to determine the acoustic wave propagation velocity.
- The type and conditions of the fluid to determine the vapor pressure.
- Turbine characteristics, inertia and specifications for hydropower plants.
- Pump characteristics, inertia and specifications for pumping stations.
- Operational patterns, head loss and discharge characteristics of valves.
- The dimensions and characteristics of surge protection equipments.

As a transient analysis tool, the software can be used for a wide variety of pumping, hydropower and piping systems. In every model, same main steps are followed to set them up:

- *Drawing and creating the model* is the first and a user friendly step of the simulation with the drawing tools and interface. A model can be created either drawing it in HAMMER medium or exporting it from other mediums such as WaterCAD, WaterGEMS, EPANET. Also background layers as pictures can be exported behind the model in order to relate the components of it with their real appearance.

- *Defining properties of the model* is the most important stage of the simulation. All the required data for steady and transient state analysis of the model are defined in this step. Physical, geometrical, transient, operational properties and initial status of the elements should be described. Also, the water quality can be defined in this step.

- *Building up scenarios* allows creating and analyzing unlimited numbers of simulations that have various element properties and calculation options on the same model. Determination of cost effective surge protection measures and comparison of results of different excitations can be actualized by modifying the conditions of the model with scenario and alternative managers.

- *Computation* is a two step process. First step is the computation of steady state conditions and the second one is the run of transient analysis.

- *Viewing results and reports* can be both performed in graphical and tabular forms. Steady state results can be shown only in tabular forms; however, time history transient results of selected points can be viewed in both ways. Time dependent head, pressure, flow and vapor pocket volume values of desired elements and profiles, and also the components of transient forces at selected points can be drawn in a graph or represented in a table. Moreover, the animation of transient head and pressure envelopes can be played with transient results viewer tool.

4.2 The Interface, Main Window and Tools of the Software

The main form of the software consists of menu commands, user friendly toolbars, lists of frequently used functions, a drawing pane and user notifications window. Figure 4.1 demonstrates the main window of the software. Frequently used tools for setting up a hydraulic model and their functions and properties are described herein.

The standard toolbar consists of managerial commands of the software like new, open, save and print a/the project.

The edit toolbar includes delete, undo, redo and find actions.

The view toolbar contains the commands that manage the appearance of the main window. It consists of tables, panels, graphs and profiles. Commonly used ones are listed below.

- *Background Layers* allow create and manage background pictures illustrating the project appearance.
- *Flex Tables* allow create and manage tabular definitions, results and reports of elements and components in the model.
- *Profiles* allow create and manage longitudinal profiles in which the transient results are preferred to be viewed or animated.
- *Properties* command is for editing the general, geometrical, physical, operational and transient definitions of the elements and components of the model.

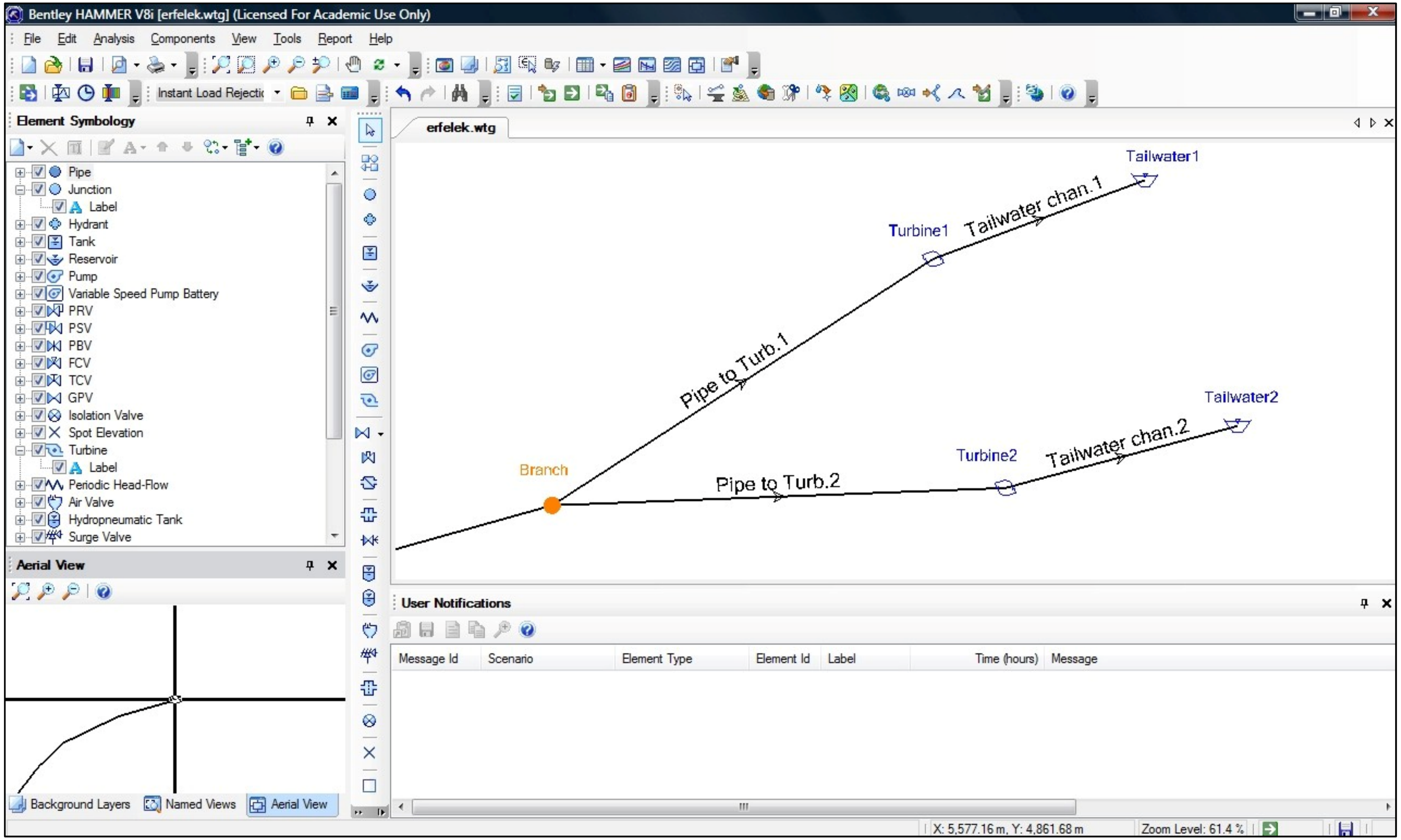


Figure 4.1 Main Window of the Software and its Interface

4.2.1 Tools for Drawing, Creating and Defining the Model

Drawing pane is one of the main parts of HAMMER and already active if a new project has started. Schematic and scaled drawing modes can be preferred to create a model. Every element and component that takes place in layout of a model is drawn by selecting it from the layout toolbar and placing into the drawing pane by clicking. This toolbar contains every single hydraulic element that a pumping station, hydropower plant and piping network may have. Following elements contained by the toolbar are commonly used in SHPs. The properties windows of four of them are illustrated in Figure 4.2.

- *Reservoir* is used to represent the forebay and the free water surface of the tailwater in a SHP. The elevation of the water surface can be defined with a fixed or variable hydraulic grade pattern. However, during the transient simulation, the defined elevation of the water surface does not change with the pressure wave movement. In other words, HAMMER does not explicitly model the forebay's damping and stabilizing role in transient states.

- *Pipe,* which is the main component of the hydraulic model, takes place between two nodes or hydraulic elements. As physical properties, diameter, friction factor, length, minor loss coefficients and optionally pipe material must be assigned to run a steady state analysis. The definition of the wave speed is a prerequisite for transient analysis. The tool *wave speed calculator* can carry out related computations for pipes.

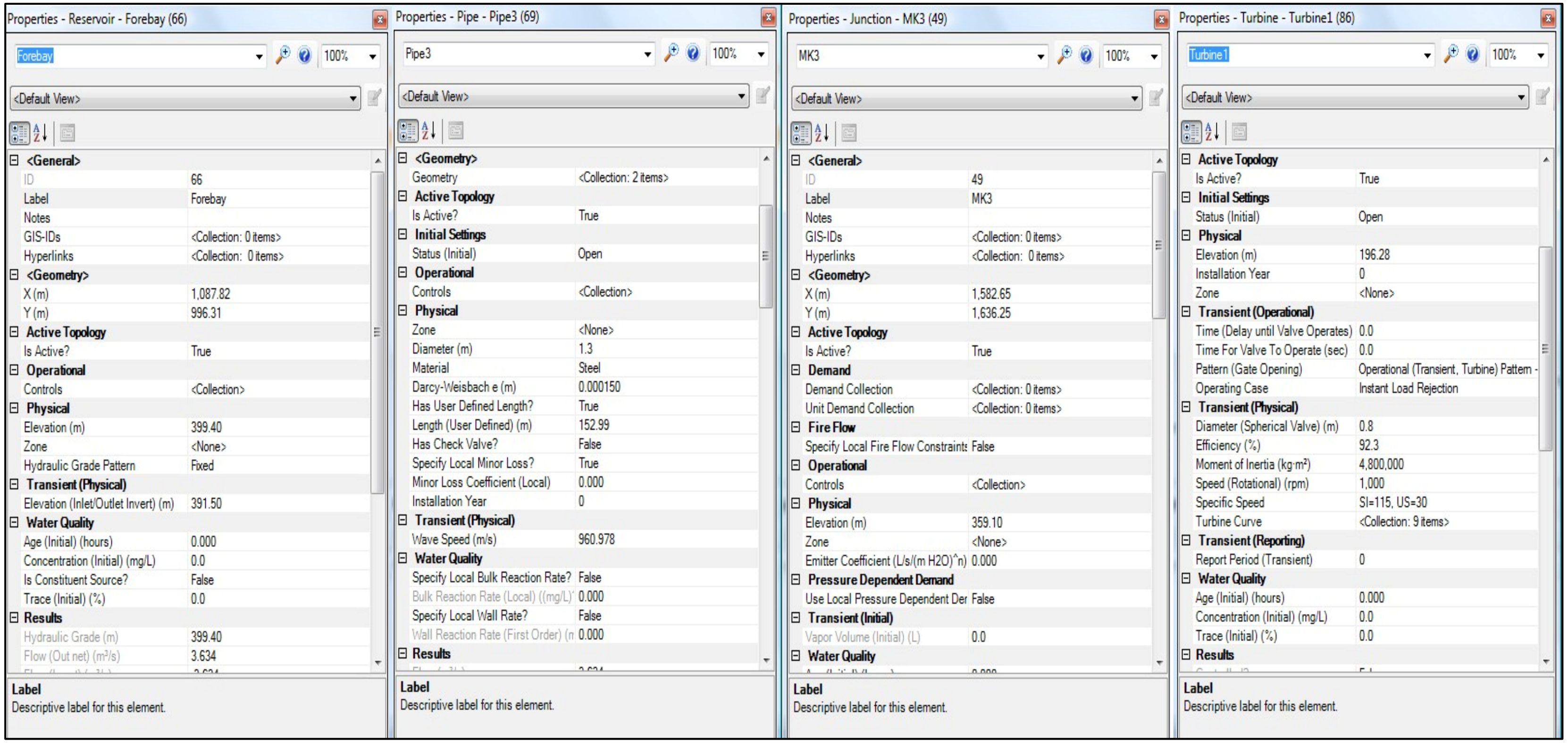
- *Junction* is used to connect two pipes that have different physical or transient properties. They are also used for branching pipe sections in a model. Only assigning its elevation is sufficient to run a transient analysis.

- *Turbine* element in HAMMER represents only reaction turbines. Impulse turbines are modeled with a valve or "discharge to atmosphere" element. It is defined with its elevation, efficiency, moment of inertia, rotational speed, rated head and flow. Also, its head and flow curve should be assigned. There are pre-defined four operating cases that initiate transient state in HAMMER: instant load rejection, load rejection, load acceptance and load variation.

- *Valves* are used in the upstream of turbines in SHPs. The main purpose of installing a valve is to provide the required installation and maintenance condition to the turbine. Flow of the turbine is only adjusted by wicket gates or nozzles of the turbine. Therefore, valves may not be regarded as a boundary condition for transient simulations in SHPs. A typical valve can be modeled with "general purpose valve" element in the software.

- The definition of a s*urge tank* is quite in detail in HAMMER. There are four different surge tank types that can be simulated in the software: simple, orifice, differential and one way surge tanks. Also, the opportunity of simulation of an overflow spillway is provided. Properties including operating types, initial, minimum and maximum elevations in the tank, cross section type and area, orifice diameter and head loss coefficient in the tank can be identified.

- Numerical simulation of fluctuations in a *hydropneumatic tank (air chamber)* can be modeled with two different ways: constant area approximation and gas law model. According to the method chosen, the input data varies. The effective volume of the tank and hydraulic grade line elevation of the maximum and minimum water volumes are needed when the chamber is modeled with constant area approximation method. If the gas law model is preferred, total volume, initial pressure or hydraulic grade line of the tank must be specified. As default properties, tank inlet diameter, head loss ratio and minor loss coefficient are defined for the simulation.



(a) (b) (c) (d)

Figure 4.2 Properties Windows for (a) Reservoirs, (b) Pipes, (c) Junctions, (d) Turbines

- There are two *check valve* types that can be modeled in HAMMER: surge anticipator valve (SAV) and pressure (surge) relief valve (PRV). A check valve may act as only a SAV, only a PRV and both SAV and PRV according to the design. The opening of a PRV is initiated with surge pressure that exceeds the prescribed limit. Since the computational model of the PRV comprised of a spring and a plate (see Figure 3.10), simulation of the valve opening requires the spring constant and check valve diameter.

Also, following elements are used and included in the layout toolbar to create and define a model:

Safety membrane (rupture disk), hydrant, tank, periodic head-flow element, pump, check valve and air valve.

4.2.2 Tools for Creating and Editing Scenarios

Scenarios are used to simulate a model with different operation and calculation alternatives. By creating different scenarios, the effect of any change in the options can be seen. Also, input values of two scenarios can be compared to identify differences. The scenarios toolbar is composed of calculation options, alternatives and scenario controls.

The steady state and transient solvers of the software allow a variety of changes in parameters that directly affect the output data. *Calculation options* manage these user preferred changes. For both steady and transient state solvers, wide variety of parameters can be configured. For the steady state solver, steady friction method, demand and roughness adjustments, calculation flags preferences and some definitions about the fluid may be configured. Transient state solver contains more detailed and enhanced configurations. Under this option, output results can be limited. Moreover, specific report points of the model and computation time interval may be defined to have a report in relatively smaller file size. Another important parameter for a proper simulation of a model is the run duration time of the transient analysis. It is configured under the calculation options. A fundamental parameter for the transient analysis, absolute vapor pressure, is also defined to simulate the vaporization of the liquid and the formation of the vapor pockets.

The transient state solver provides the opportunity to simulate the model with three different friction methods: steady, quasi-steady and unsteady. According to the characteristics of the model and by use of engineering judgment, preferred method can be defined from calculation options.

In addition, generation of the animation of the model's head and pressure envelopes and usage of rigid column theory in the simulation can be activated.

Alternatives are main components of scenarios. Various combinations of alternatives compose a scenario, and they hold the input data in categories. These categories contain topological, physical, operational, transient, demand, etc. properties for each element in the model. The data entered at the beginning of the model definition is stored in base alternative categories. New alternatives should be created to see the system behavior under different conditions and properties.

4.2.3 Tools for Computation

Computation of a HAMMER project includes two main analyses: steady state and transient. Before running these analyses, the tool *validate* can be used to detect the data errors and modeling problems. This process may end up with a notification window that shows warning messages about the model. Some of them may be fatal and may prevent running the analysis.

Transient analysis of a model requires the initial conditions of its components. If they are specified manually, there is no need to run steady state analysis. If not, the tool *compute initial conditions* starts steady state calculations.

Once the steady state data is acquired, the transient analysis can be performed by using the *compute* tool.

4.2.4 Tools for Viewing Results and Reports

There are many ways to see the results of the transient analysis. However, checking some brief information about the calculation may be beneficial before viewing detailed results. After the computation step, a window pops up and gives very useful and significant information about transient calculation. On summary tab, time step, total time of the simulation and other predetermined constants are shown in tabular form. Moreover, initial heads and transient extreme pressures and heads throughout the pipe are given in two different tabs and tables.

Transient results can be illustrated in graphs in the form of time series and profiles. Plotting graphs and playing animations can be performed by using *transient results viewer*. Time history results of hydraulic grade line, pressure, flow, transient forces, velocity and vapor volume of selected points in the model are plotted with that tool. Also, the animation of head and pressure envelopes and development of surge pressures from beginning to the end of the transient simulation can be observed in real time.

Another option that allows viewing maximum and minimum head, pressure and vapor values of pipes and nodes with color highlighting is *transient thematic viewer.* This tool colorizes pipes or nodes with respect to their selected transient state parameter by using color maps and gradient. According to specified sets of constraints that define ranges, every element is colored to see the existence of any risky element which may cause damage in the model.

Finally, *reports* are the data sheets that contain transient analysis results and properties of elements in tabular form. Various types of reports can be used to see the details of scenarios, project and pipe inventory, vapor and air pocket formation, time history results of flow, head and vapor volume of desired points.

CHAPTER 5

CASE STUDIES

This chapter is comprised of two case studies which are classified as small hydropower plants. They are same in type, similar in their capacity; however, different in their turbine types. The aim of selecting different turbine types is to analyze, discuss and compare their transient behavior with the hydropower plants' transient simulation model.

The first case study has two Pelton turbine units and comes with its real time operating data and measured transient pressure results at the turbines' inlets. The operation of the SHP for several transient cases is simulated by using the software. The computed and measured results are compared and reasonable agreements are found between them. The disagreements between the results are also explained with the possible reasons.

The second case study is composed of series of analysis that investigate the pressure rise and drop at the end of the penstock and speed rise behavior of the Francis turbine units due to three disturbances during operation. The effects of closing curve on the pressure rise at the end of the penstock and turbine rotational speed are investigated. Moreover, flywheels with different polar moment of inertias, a pressure relief valve and safety membranes are placed as protective measures and their effects on transient behavior of the system are studied separately.

5.1 Case Study 1: Çakırlar Small Hydropower Plant

In this case study, a small hydropower plant with two Pelton turbine units is exposed to real time disturbances and consequently resulting transient pressures are considered. After setting up the simulation model of the SHP, for load rejection, load acceptance and instant load rejection cases, pressure histories of the turbine inlets are calculated and the computed results are compared with the measured ones.

5.1.1 Background Information about the Hydropower Plant

Çakırlar Hydropower Plant, a run of river type SHP, is located at Artvin province in the eastern Black Sea region of Turkey (Figure 5.1). It gathers the water from four individual streams: Kunsu, Eğrisu, Suludüz and Köpürten. Then, by dropping the water from 463.43 m, it produces the electricity with its 16.49 MW installed capacity. The plant has been in operation since August 2009. The operation works of the plant have been conducted by GAMA Energy Incorporated.

The project is comprised of four diversion weirs built across different streams. There are two main transmission lines carrying water to the forebay. In the first transmission line, at the upstream, water diverted from Kunsu Weir is conveyed through the transmission line to the Eğrisu Weir. Then, it is transmitted to the Suludüz Weir with the water of Eğrisu River. A settling basin (i.e. settling basin 1) located on the transmission line, between Eğrisu and Suludüz weirs, settles down the sediment particles up to a certain size. Then, the water diverted from the Suludüz Weir joins to the first transmission line and after passing from settling basin 2, the collected water from three streams reaches to the forebay. The second transmission line, which has one settling basin, just conveys the water that is diverted from the Köpürten River and transmits it to the forebay. Water is collected at the forebay and kept with a constant elevation during operation. In other words, the hydropower plant is operated with the elevation regulation policy.

The main penstock has a length of 1108.97 m and consists of 16 pipes. Through it, the wall thickness varies from 10 mm to 20 mm. Moreover, the main pipe diameter is constant and 1150 mm. To feed the two Pelton turbine units, the main pipe branches just upstream of the powerhouse. The branched pipes are contracting in diameter in the flow direction, and the diameters of two successive pipes in one branch are 800 mm and 600 mm, respectively. The body of the main penstock lies over the ground surface through its route. However, after the branching junction, it goes under the ground. In the computation of the wave speed of individual segments of the penstock, this layout conditions are regarded. The schematic layout of the penstock and its as built characteristics are given in Figure 5.2 and Table 5.1, respectively.

There are two Pelton turbine units that have vertical axes. Each one has a 600 mm diameter inlet globe valve in their upstream sections. However, flow of the turbine units is controlled by the four jet deflectors and nozzles.

After turbining the flow, by the tailwater channel, water is directly discharged to the Kabaca Weir reservoir area.

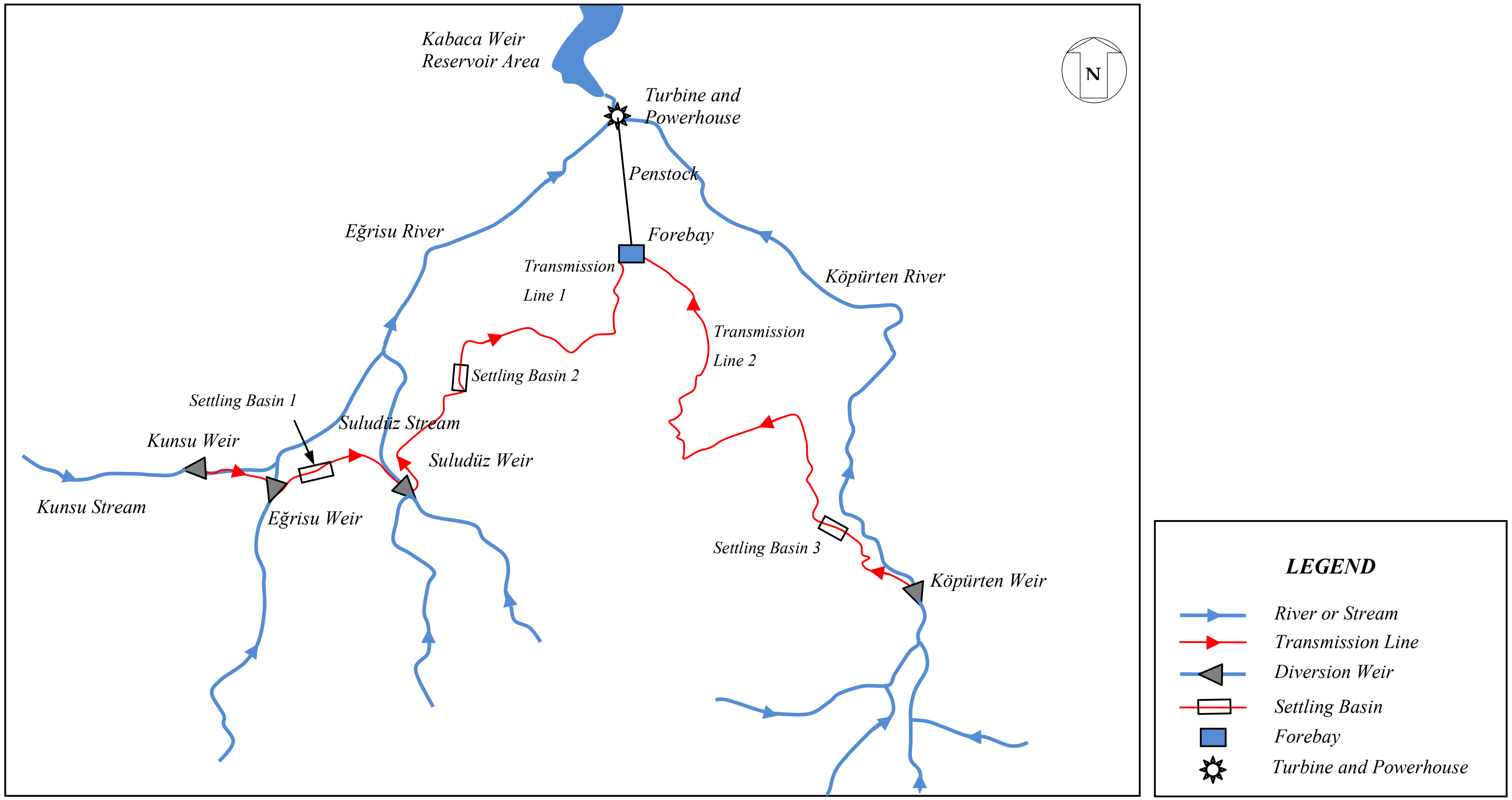


Figure 5.1 General Plan View of the Çakırlar Hydropower Plant (Not Scaled)

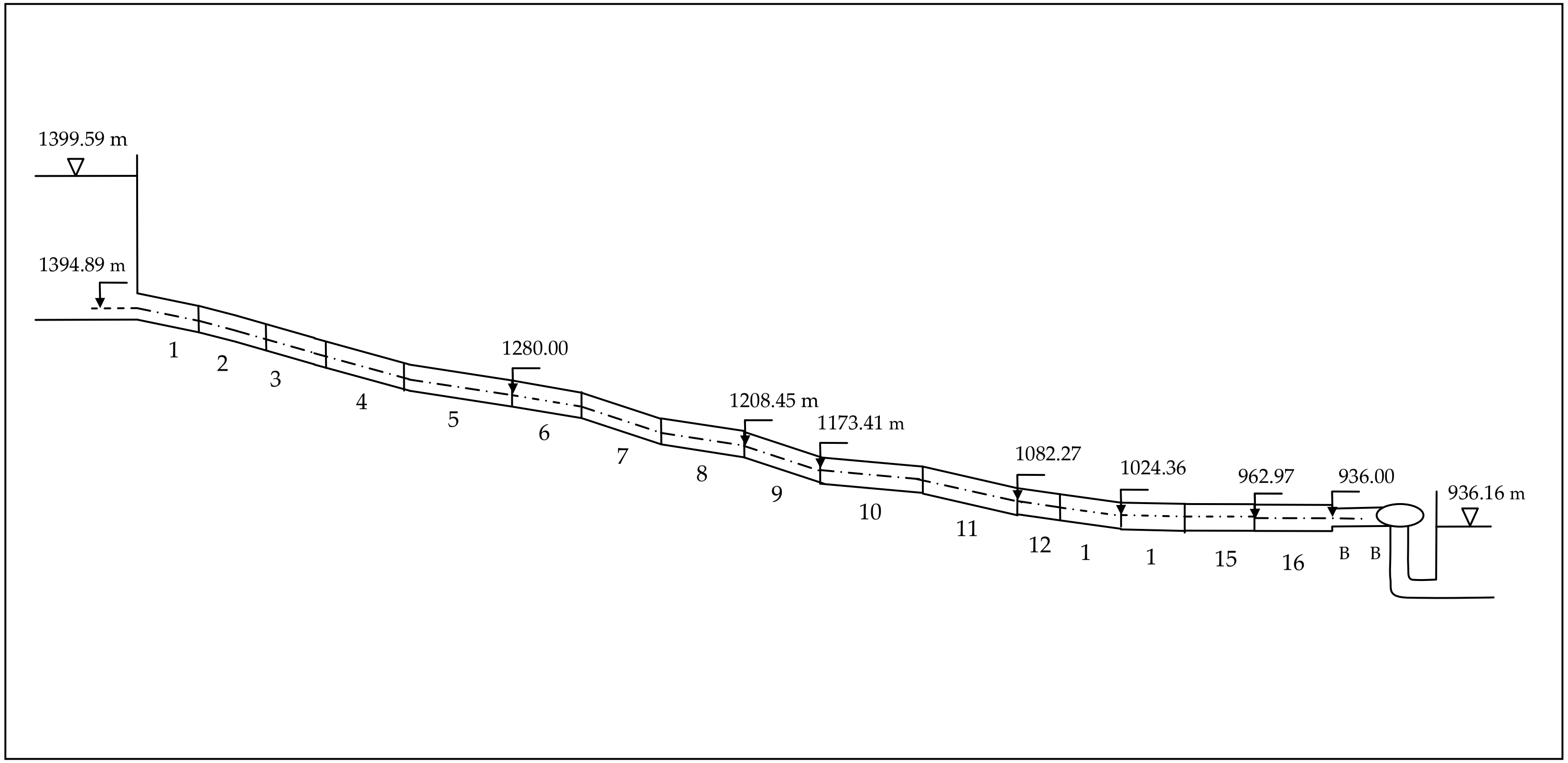


Figure 5.2 Schematic Layout of the Penstock of the Çakırlar SHP

(All elevations are above the sea level.)

Table 5.1 As Built Penstock Properties of the Çakırlar SHP

Pipe Number	Pipe Length (m)	Wall Thickness (mm)	Diameter (mm)	Wave Speed (m/s)
1	56.72	10	1150	994.77
2	62.83			
3	57.98			
4	36.03			
5	103.96			
6	88.97	12		1043.63
7	80.75			
8	92.12			
9	78.80	14		1083.30
10	81.53	16		1116.20
11	91.32			
12	35.71	18		1143.98
13	80.66			
14	82.35	20		1167.76
15	44.04			
16	35.20			
Branch 1	7.70	16	800	1216.74
Branch 2	7.95	16	600	1269.36

Table 5.2 Basic Characteristics of the Pelton Turbine Units

Type	Vertical Axis Pelton
No. of Identical Turbine Units	2
Number of Jet Deflectors	4
Turbine Output (kW)	2 x 8244
Rated Speed (rpm)	750
Rated discharge (m^3/s)	2 x 2.08
Nominal Gross Head (m)	463.43
Nominal Net head (m)	449.33
Runner diameter (mm)	1170

In the steady state computational model of the hydropower plant, following discharge, net head and output tabulation, provided by the manufacturer, is used. For transient analysis and computations of the hydropower plant, jet deflectors and nozzles of the Pelton turbine units are characterized with needle valves in the computational model. As a downstream boundary condition, needle valves are the best elements for modeling of the Pelton turbine units according to the both literature and the computer software.

Table 5.3 Turbine Discharge, Net Head and Output Table (GAMA Energy Inc., 2008)

Turbine Discharge (m^3/s)	**Net Head (m)**	**Jet Number**	**Efficiency (%)**	**Turbine Output (kW)**
2.08	449.3	4	89.99	8244
1.87	452.0	4	90.24	7485
1.66	454.4	4	90.33	6695
1.46	456.5	3	90.10	5870
1.25	458.4	3	90.17	5056
1.04	459.9	2	89.91	4215
0.83	461.2	2	90.02	3386
0.62	462.2	2	89.45	2529
0.42	462.9	1	88.77	1675

5.1.2 Transient Analysis and Comparisons of Measured and Computed Transient Results of the Hydropower Plant

In this case study, for the following scenarios, transient state pressures in the penstock are computed:

- Regular stop of the turbines separately during load rejection
- Simultaneous start of the turbines during load acceptance
- Instant stop of the turbines at the same time during instant load rejection

During the operation, listed actions above are observed and by the help of built-in instruments of the system, waterhammer data are measured and recorded. The numerical results for the transient pressures just upstream of the turbine are compared with the measured data for all scenarios.

5.1.2.1 Scenario A: Regular Stop Procedure of the Turbines during Load Rejection

In this scenario, the computational model of the Çakırlar SHP is set up for a specific operating case that has occurred on May 4th 2010. The load on the plant was rejected two successive times. In the first phase, the turbine unit 1 was stopped, and similarly in the second phase second turbine was stopped in their regular closing times. The system is analyzed and pressure rise at the turbine inlet is computed for both turbines separately. Then the measured waterhammer pressures are compared with the computed ones.

Just before the transient state has occurred, turbine unit 1 and 2 were working steadily with the discharges of 1.54 m^3/s and 2.04 m^3/s, respectively. At 01:39:17 p.m., the first phase of the load rejection occurred and the first turbine started to close. This regular closure lasted 60 seconds with nearly three main strokes. The closing time of the needles, $T_c = 60\,s$, was much longer than the wave reflection time, $T_r = 2L/a \cong 2.2\,s$. The initial characteristics of the Pelton turbine units before the first phase of load rejection are given in Table 5.4.

Table 5.4 Initial Characteristics of the Pelton Turbine units for Scenario A for the First Phase

	Turbine 1	**Turbine 2**
Discharge (m^3/s)	1.54	2.04
Net Head (m)	450.20	449.78
Turbine Output (kW)	6220	8110
Closing Time of the Nozzles (sec)	60	Fixed

The closing law and comparison of computed and measured pressure histories at the turbine inlet are shown in Figure 5.3 (a) and (b), respectively. As the closure is very slow, in both computational and measured results of the system, the pressure head in front of the turbine unit 1 mildly increases. The rise and oscillation trends of the pressure are very similar in the model and observed field data. The maximum pressure head is observed and computed as 461.5 m and 462.3 m, respectively in front of the first turbine.

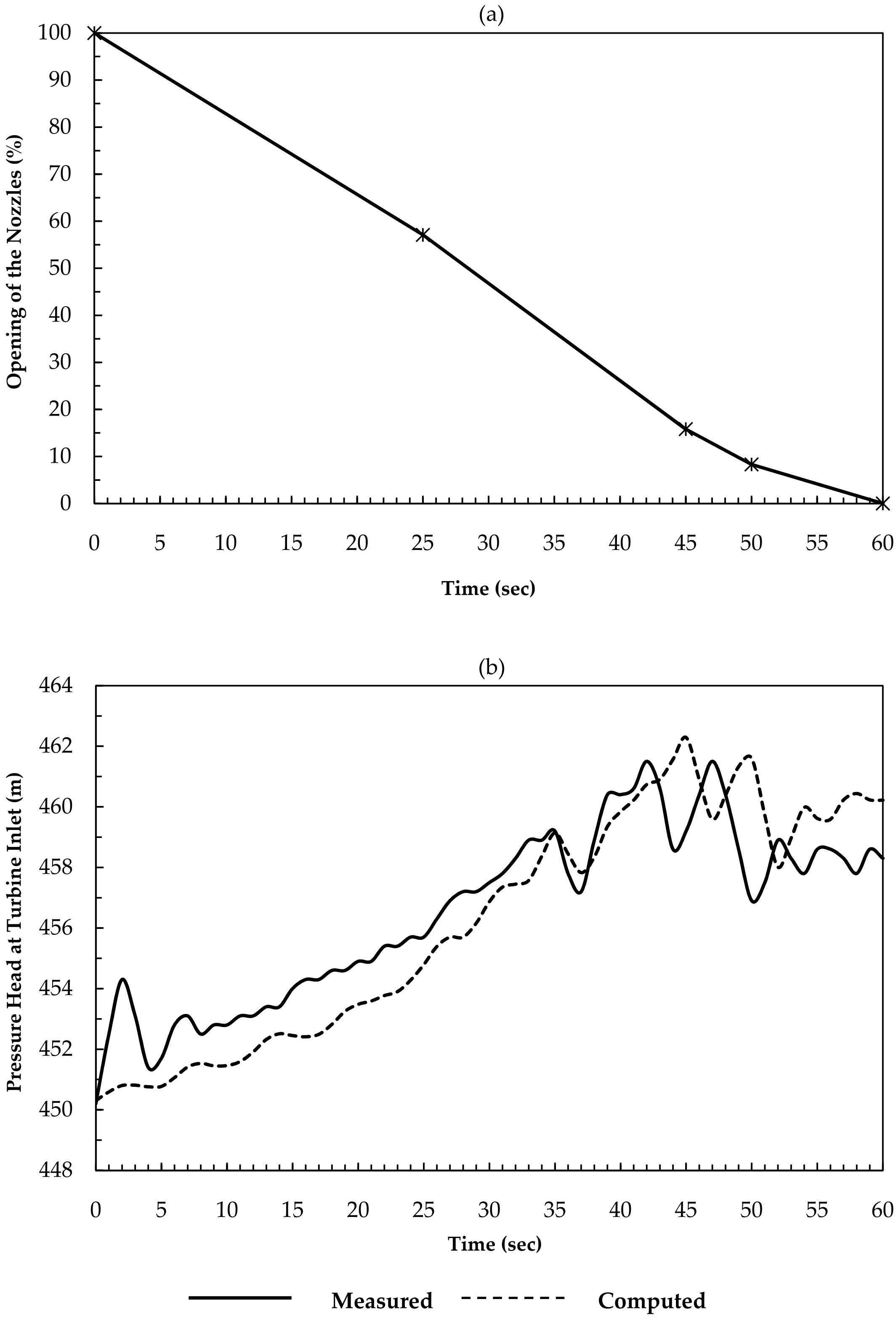


Figure 5.3 (a) Closing Law; (b) Measured and Computed Inlet Pressure of Turbine Unit 1 for Load Rejection

Just 26 minutes after the stoppage of the first turbine, the second phase of the load rejection occurred and the second turbine was stopped regularly. For this time, only second turbine was working steadily with the conditions that are given in Table 5.5.

Table 5.5 Initial Characteristics of the Pelton Turbine units for Scenario A for the Second Phase

	Turbine 1	**Turbine 2**
Discharge (m^3/s)	0	2.04
Net Head (m)	-	454.90
Turbine Output (kW)	0	8110
Closing Time of the Nozzles (sec)	Fixed	60

The graph of the measured and computed pressure histories during the closure of turbine unit 2 are given in Figure 5.4 (b). The data for the dashed line labeled “Computed” were obtained from the simulation model of the hydropower plant and given with the recorded field data on the same graph. From the figures, it can be said that the closing stroke rates of the turbines and their times are very similar. Since both turbines have the same physical properties and similar hydraulic conditions, this leads to similar pressure oscillations and head rise at the end of the penstocks. The percentage of the pressure rise over the steady state value is around 2.6 % for both phases.

According to the figure, the computed and measured transient pressures agree closely for about 42 seconds. Afterwards, relatively greater oscillations occur in measured pressure; however, they do not affect the difference between the magnitudes of maximum pressures of measured and computed results greatly.

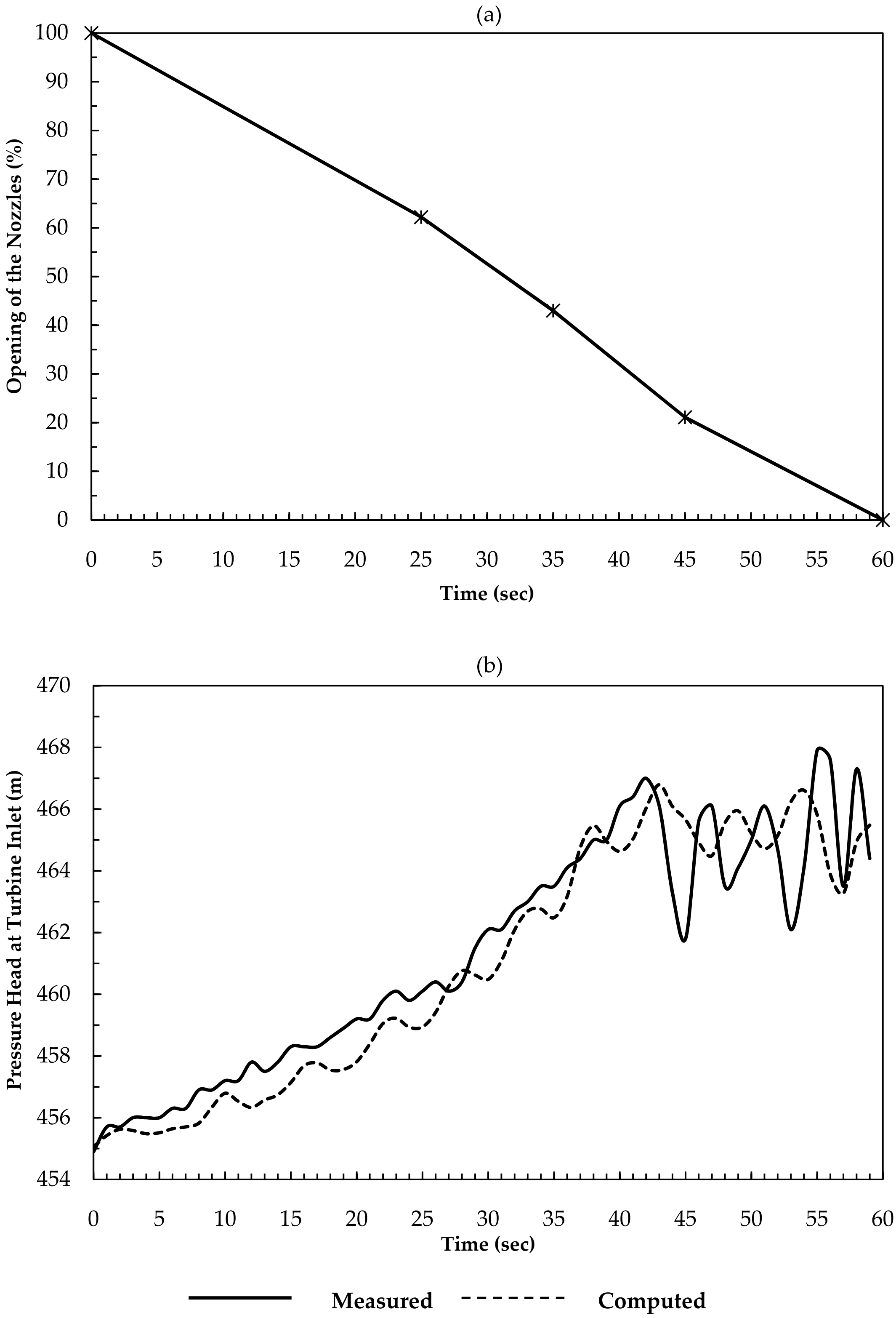


Figure 5.4 (a) Closing Law; (b) Measured and Computed Inlet Pressure of Turbine Unit 2 for Load Rejection

5.1.2.2 Scenario B: Regular Start Procedure of the Turbines during Load Acceptance

Transient state conditions in the penstock that were produced by accepting load on the turbines are simulated in this scenario. The load acceptance occurred from 0 to 15.6 MW for 168 seconds on May 23th 2010. Two turbine units were started to operate simultaneously. Data for the Pelton turbine units are given in Table 5.6.

Table 5.6 Final Characteristics of the Pelton Turbine units for the Scenario B

	Turbine x 2
Discharge (m^3/s)	1.96
Net Head (m)	450.85
Turbine Output (kW)	7800
Opening Time of the Nozzles (sec)	168

Nozzle opening percentages with respect to time and comparison of the computed transient state pressure trace with the measured on the site is illustrated in Figure 5.5 (a) and (b), respectively. While starting of the turbines from rest, nozzles are opened by 6.5 percent. At this opening, turbines begin to rotate and reach to their rated rotational speed. The opening of the nozzles is kept at this rate until the unit is synchronized to the system. For the first 60 seconds of opening, transient state pressure oscillates in an unstable manner due to the relatively small opening of the nozzles. Also in the same time range, computed and measured pressures behave very similarly. By the stroke that causes the nozzles to open by 83 %, the pressure at the end of the penstock drops barely. However, the pressure drop in the computed case is greater than the measured one and they show more rapid dissipation of transient pressure.

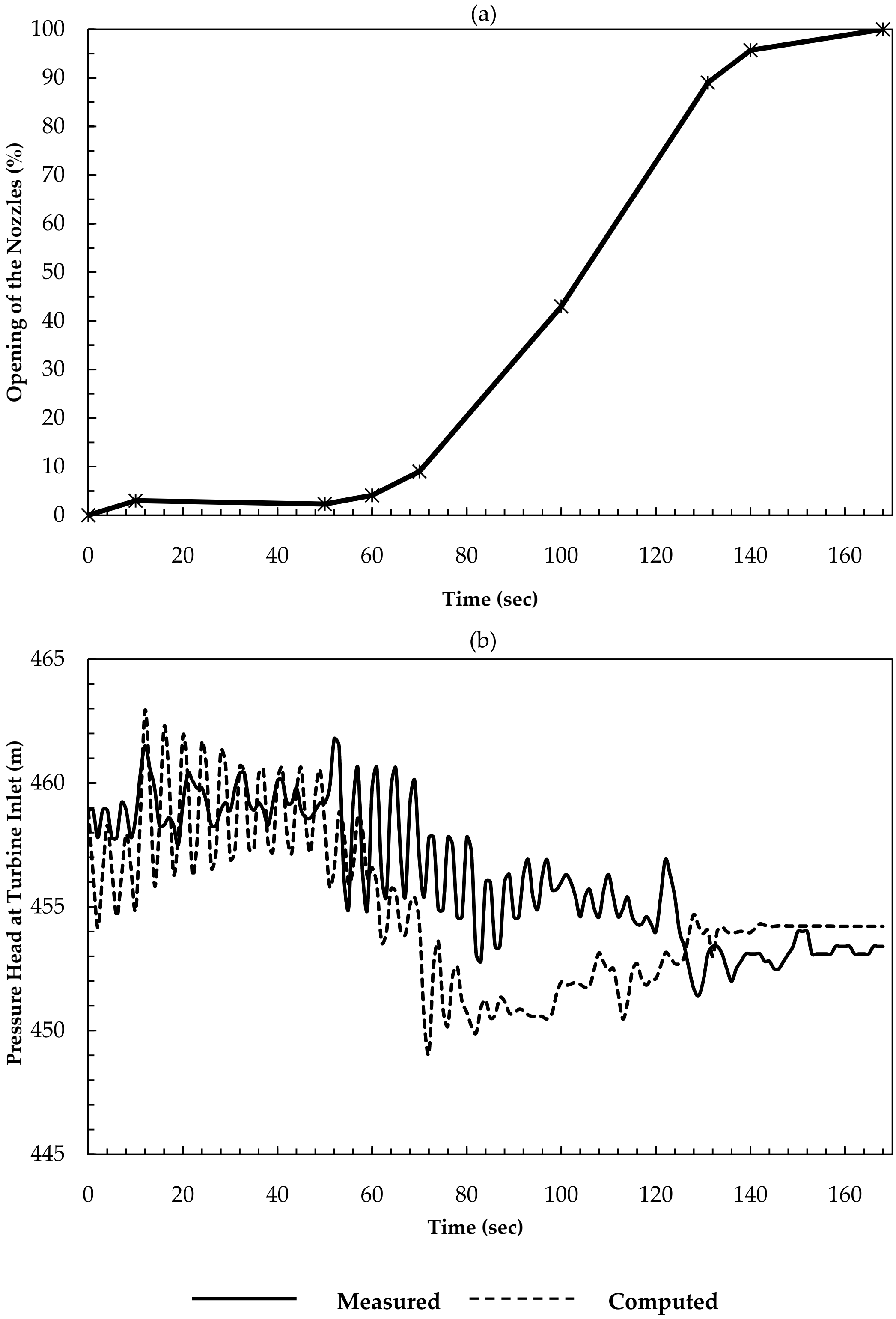


Figure 5.5 (a) Opening Law; (b) Measured and Computed Turbine Inlet Pressure of Turbines for Load Acceptance

It can be concluded from the figure that, contrary to the general view, the complete and the simultaneous start of the turbines can be handled safely if their nozzles are opened slowly.

5.1.2.3 Scenario C: Instant Stop Procedure of the Turbines during Instant Load Rejection

The load on the two turbine units were rejected instantly under governor control when they were loaded with 13.02 MW on May 23th 2010. Turbines' relative nozzle opening configuration were recorded at the site during the rejection and used in the analysis. Considering the records, after the rejection, two units were started to close simultaneously; however, the strokes and the total time of the closures were different. According to the hydraulic and machinery conditions of the turbines, the closing times were determined by the governor as 97 and 29 seconds for the first and the second turbines, respectively. After the instant load rejection, nozzles were kept totally closed. The initial characteristics of the system before the transient state are given in Table 5.7.

Table 5.7 Initial Characteristics of the Pelton Turbine units for the Scenario C

	Turbine 1	**Turbine 2**
Discharge (m^3/s)	1.51	1.73
Net Head (m)	456.02	453.49
Turbine Output (kW)	6100	6920
Closing Time of the Nozzles (sec)	97	29

The closing laws showing the strokes of each turbine are given in Figure 5.6 (a). Graphs of pressure histories of the computed and measured transient states for turbine unit 1 and 2 are given in Figure 5.6 (b) and (c), respectively.

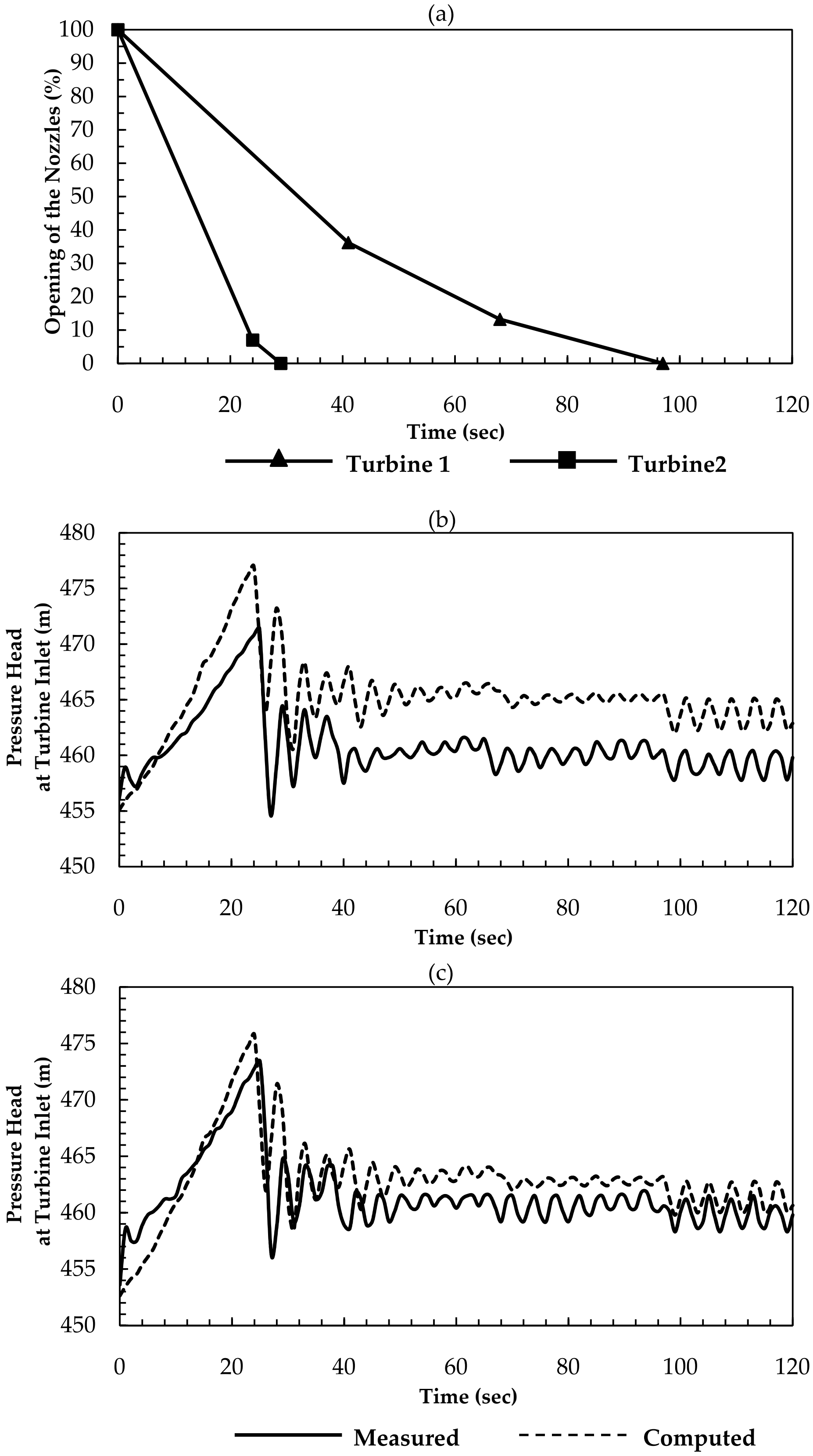


Figure 5.6 (a) Closing Laws of Turbine 1 and 2; (b) Inlet Pressure of Turbine Unit 1; (c) Inlet Pressure of Turbine Unit 2 for Instant Load Rejection

The data for the continuous line labeled “Measured” are very close for both turbine units. It can be said that the governor controls the units to keep them in similar transient state conditions. Also it is clear from the figures that the fluctuation trends of the transient pressure head at the turbine inlets are very similar for the computed and measured traces. However, there is a phase shift between them for both turbine units.

It can be concluded from transient results of three scenarios that, the validity of simulation model of the small hydropower plant is successfully established for the related operating cases.

5.2 Case Study 2: Erfelek Small Hydropower Plant

In this case study, the operation of a small hydropower plant with Francis turbines subjected to several transient states due to load rejection, load acceptance and instant load rejection is simulated. According to the operation data gathered from the operation company and a determined worst case disturbance, instant load rejection, the system is analyzed in order to investigate the power plant behavior and ensure its safety. Also three protective measures namely, flywheel, pressure relief valve and safety membrane, are considered and analyzed within the system separately to control the waterhammer pressures in the penstock.

5.2.1 Background Information about the Hydropower Plant

Erfelek Hydropower Plant is located on the Karapınar River at the Sinop province in the middle Black Sea region of Turkey (Figure 5.7). Feasibility and final design studies and construction of it were carried out by Birim Hydroelectric Production Company. Also the operation of the power plant has been conducted by the same company.It is a run of river type small hydropower plant with 6.45 MW installed capacity. It was put in operation in the beginning of April 2010.

The project consists of three diversion weirs built on individual streams. Erfelek Weir diverts water from Karapınar River by means of a lateral intake structure. The diverted water is conveyed to the forebay by the transmission line 1 which is a pipe and made of unplasticized PVC. A diversion weir with a drop type intake called Hira, located on the Hira stream and consists of an overflow spillway and a fish passage. The transmission line 2 receives water from Hira Weir and joins to the first transmission line. The third and the smallest weir is called Ebe and it is similar to the Hira with its type and dimensions. As the Ebe weir is close to the forebay, the transmission line 3 directly transmits water to the forebay. All transmission lines are buried. Flow in the first and the second transmission line is free surface. This is maintained by manholes which are open to atmosphere. However, the third transmission line has pressurized flow. The dimensions of the forebay are designed to meet the one hour demand of the turbines at peak times when the energy is most expensive. Hydraulic suction in the penstock is prevented by operating the forebay within the permissible water level ranges.

The main features of the Erfelek HP are given below.

The main penstock has a length of 1518.69 m and consists of 19 pipes with different wall thicknesses. The wall thickness of the penstock is increasing throughout the flow direction. After an optimization study, the diameter of the main penstock was determined as 1300 mm. Since there are two turbines, the main penstock branches into two pipes just upstream of the powerhouse. The diameter of the two branching pipes ensures almost the same velocity inside the branches with the velocity in the main penstock. Both branching pipes have a diameter of 900 mm. The main penstock and branches are buried from forebay to powerhouse. Therefore, in the transient simulation model of the structure, the wave speed in the pipe is calculated as it is anchored throughout. The schematic layout of the penstock and its as built properties are given in Figure 5.8 and Table 5.8, respectively.

There are two Francis turbine units which have horizontal axes. Each one has 800 mm diameter inlet butterfly valves in front of them. However, the flow through the turbines is controlled by their wicket gates. Basic characteristics of the Francis turbines are presented in Table 5.9.

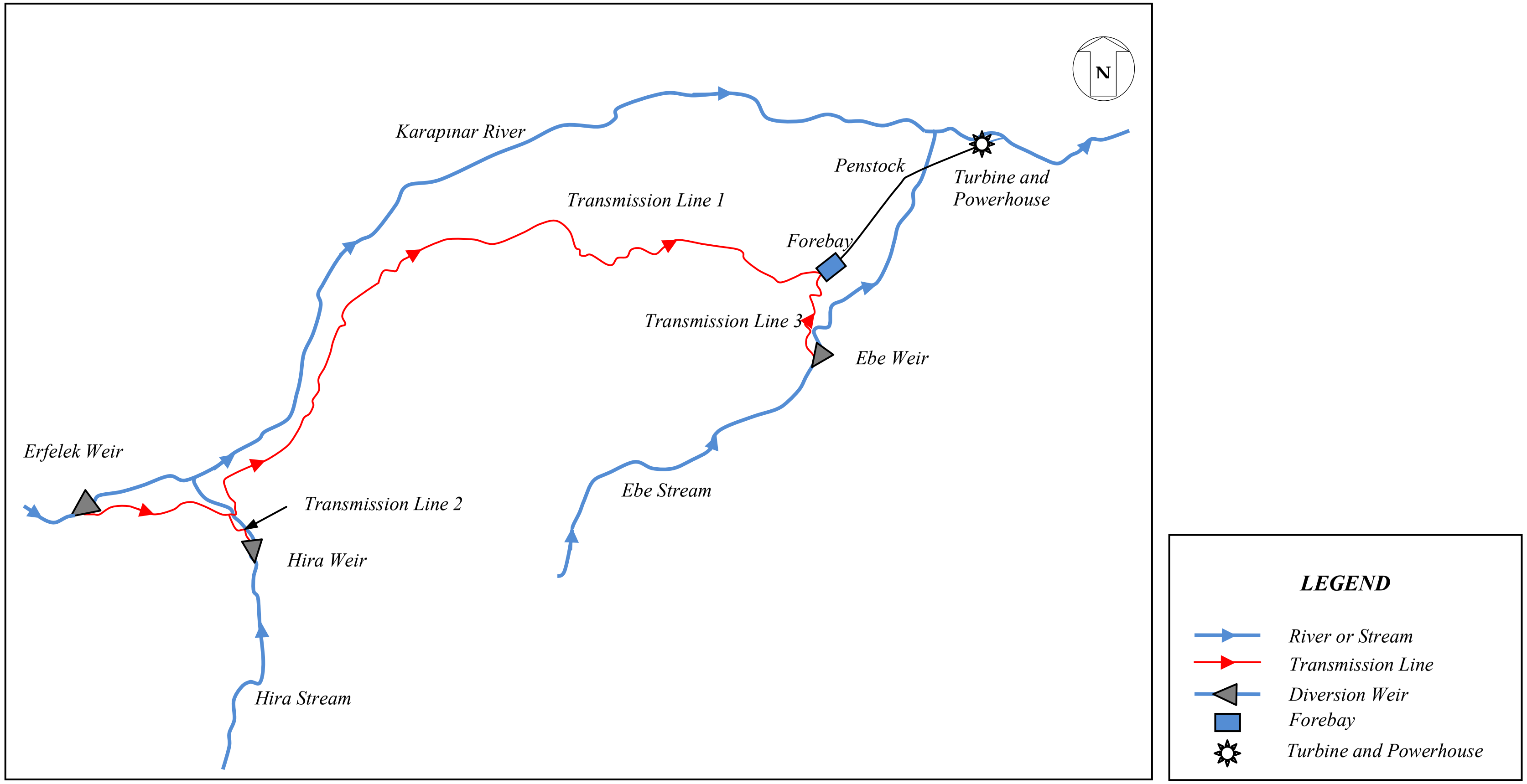


Figure 5.7 General Plan View of the Erfelek Hydropower Plant (Not Scaled)

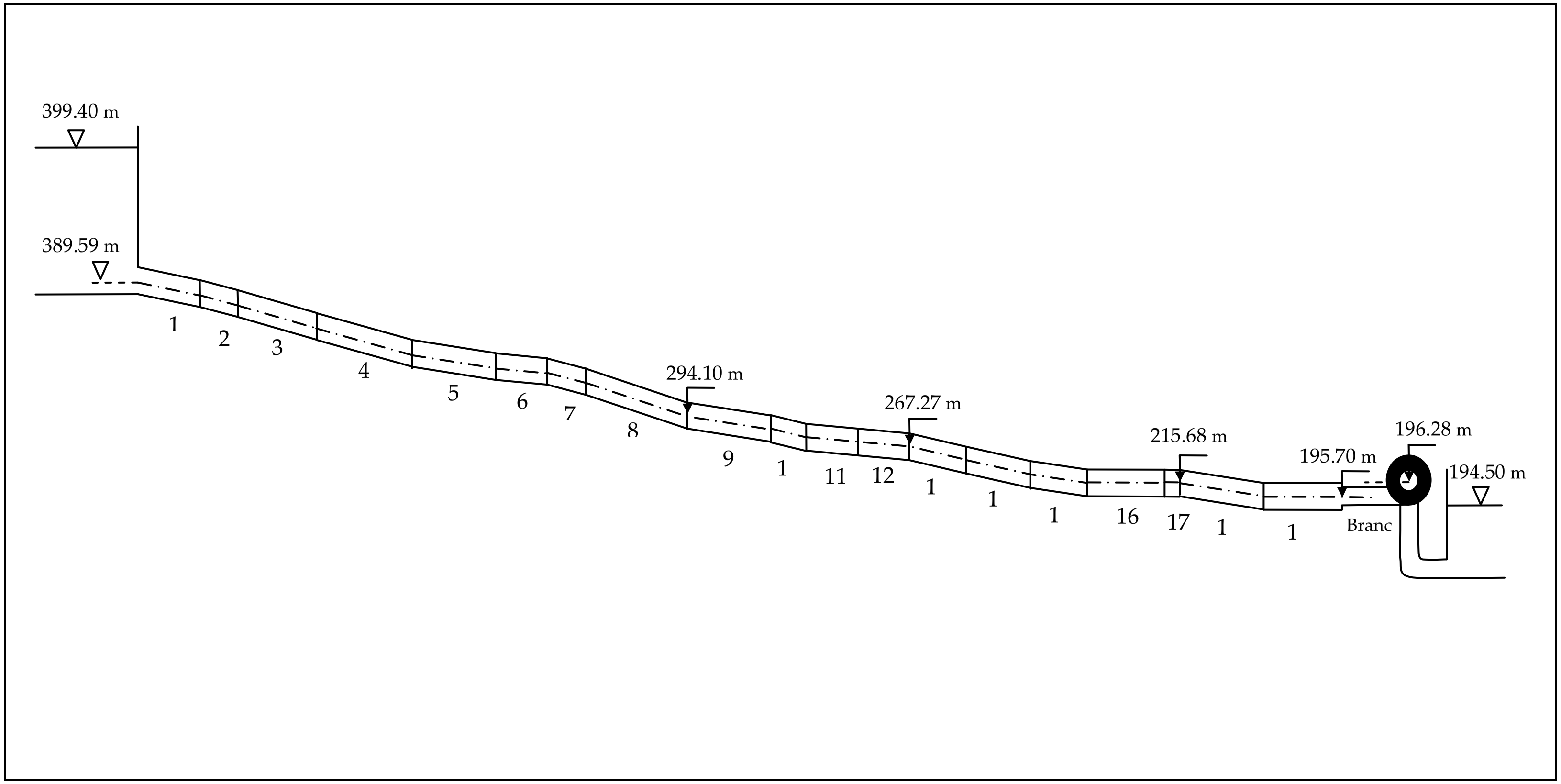


Figure 5.8 Schematic Layout of the Penstock of the Erfelek SHP

(All elevations are above the sea level.)

Table 5.8 As Built Penstock Properties of the Erfelek SHP

Pipe Number	Pipe Length	Wall Thickness (mm)	Wave Speed (m/s)	Diameter (mm)
1	34.51	8	920.36	1300
2	15.15			
3	152.99			
4	224.19			
5	136.81			
6	79.02			
7	30.13			
8	121.14			
9	57.38	10	982.63	
10	36.28			
11	31.53			
12	30.61			
13	61.36	12	1031.93	
14	74.05			
15	69.42			
16	120.10			
17	6.83			
18	172.39	14	1072.07	
19	64.80			
Branch x 2	19.78	14	1160.40	900

Table 5.9 Basic Characteristics of the Francis Turbine Units

Type	Horizontal Axis Francis
No. of Identical Turbine Units	2
Turbine Output (kW)	2 x 3225
Rated Speed (rpm)	1000
Rated discharge (m^3/s)	2 x 1.83
Nominal Gross Head (m)	204.90
Nominal Net head (m)	197.90
Moment of inertia ($kg.m^2$)	4800 (turb. + gen.)
Runner diameter (mm)	552

For the steady state computational model of the hydropower plant, following output and efficiency tabulation, provided by the manufacturer, is used. For the transient analysis and computations of the turbines, a set of differential equations are used to compute the head and flow.

Table 5.10 Turbine Discharge, Net Head and Output Table (Birim Hydroelectric Production Co., 2008)

Turbine Discharge (m^3/s)	Design Net Head (m)	Turbine Efficiency (%)	Turbine Output (kW)
1.83	197.90	90.8	3225.9
1.65	199.20	90.6	2896.9
1.46	200.36	89.9	2555.1
1.28	201.39	88.5	2200.9
1.10	202.29	86.2	1837.5
0.91	203.06	81.8	1453.1
0.73	203.69	75.7	1075.8
0.55	204.19	68.8	733.3
0.34	204.56	52.0	369.5

5.2.2 Transient Analysis of the Hydropower Plant

The transient behavior of the power plant is simulated by setting up its computational model with HAMMER. To investigate the waterhammer effects in the penstock of the hydropower plant, five different scenarios are modeled. Following table represents the scenarios with related transient cases.

Table 5.11 The Description of the Scenarios

	Operating Case	Alternative/Installation
Scenario A	Load Rejection	Different Closure Curves
Scenario B	Load Acceptance	-
Scenario C	Instant Load Rejection	Flywheel
Scenario D	Instant Load Rejection	Pressure Relief Valve
Scenario E	Instant Load Rejection	Safety Membrane

5.2.2.1 Scenario A: Regular Stop Procedure of the Turbines during Load Rejection with Different Closure Curves

In this scenario, the pressure rise at the turbine inlet and the rotational speed of the turbine runner are computed for the load rejection operation case. Regularly, when load is rejected in operation, turbines are closed in 57.61 seconds with one stroke, linearly. However, along with the regular closure, to investigate the effect of wicket gate strokes, two other closure curves are considered, and pressure rise and turbine speed rise behavior in regular closure is compared with these new closure curves. In the second and the third closure curves, the first stroke takes place during the interval of 0 s<t<19.20 s, the second stroke interval is 19.20 s<t<38.40 s and finally, similarly, third stroke interval is 38.40 s<t<57.61 s (see Figure 5.9 a). It is regarded that the turbines are in their rated conditions just before the load rejection. The initial characteristics of the Francis turbine units are given in Table 5.12.

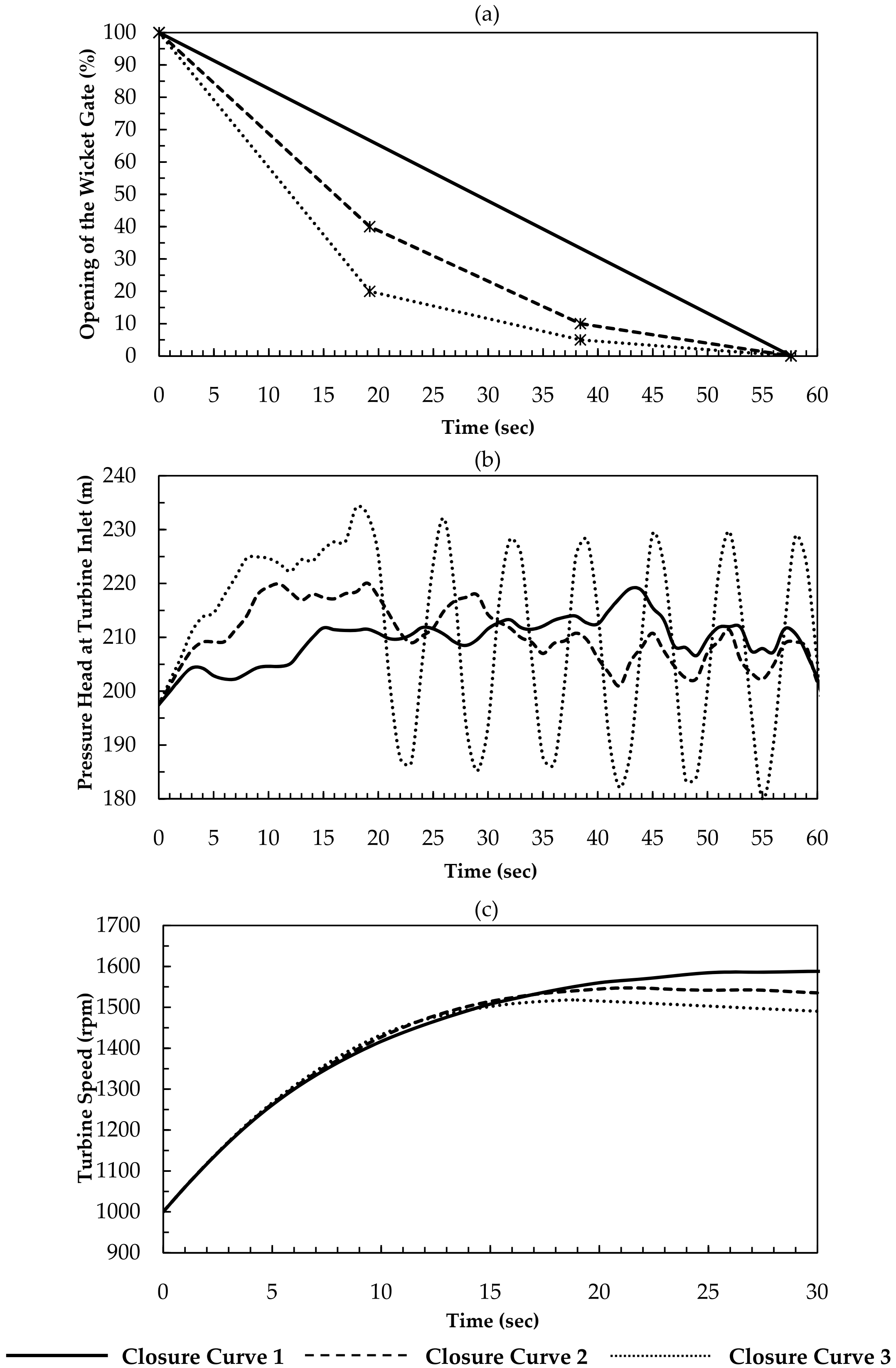


Figure 5.9 (a) Closing Laws; (b) Turbine's Inlet Pressure; (c) Turbines' Speed Rise of Three Closure Curves for Load Rejection

Table 5.12 Initial Characteristics of the Francis Turbine units for the Scenario A

	Turbine x 2
Discharge (m^3/s)	1.83
Net Head (m)	197.9
Turbine Output (kW)	3225.9
Closing Time of the Wicket Gates (sec)	57.61

The regular closing time, $T_c = 57.61s$ is greater than pressure wave return time, $T_r = 2L/a \cong 3s$, therefore, closures can be called as slow closures. For the "Closure Curve 1", by the first movement of the wicket gates, the pressure head at the turbine inlet increases relatively slightly and this increase is accompanied by the increase in the rotational turbine speed. There is 10.7 % increase in the pressure head at the turbine inlet over the steady state value, $H_n = 197.9m$, when it reaches to its maximum value. The time required for the head rise to reach its peak point, $t_{p1} = 43s$, is relatively long compared to other closure curves. After the peak point, pressure head decreases again slightly and is dampened with the frictional effects in the pipe. The point where the turbine speed starts to decrease cannot be modeled via HAMMER for this closure curve, but it is expected to happen at the instant when the peak pressure head occurs.

For the "Closure Curve 2", the pressure head at the turbine inlet reaches its maximum value at the end of the first stroke, when $t_{p2} = 19s$. The duration of the first stroke determines the peak time of the pressure head. The magnitude of the peak pressure, $H_{p2} = 220.1m$, is similar to the one obtained by the first closure curve and is nearly 11.2 % over the nominal pressure head, $H_n = 197.9m$. When the second stroke starts at t=19.20 s, the head rise slumps and the turbine speed rises and reaches its maximum value due to the increase in the pressure head. The peak value of the turbine speed is smaller than that of the first closure curve. The slight decrease in the rotational speed starts during the second stroke. By the end of the second stroke at 34.40 seconds, the slope of the closing law changes to reach the wicket gate opening to 0 %. At the beginning of the third stroke, pressure head continues to decrease slightly. Then, it oscillates with cycles in an unstable manner.

Although "Closure Curve 3" has the same stroke times with the second closure curve, it has much greater stroke percentages. In other words, the wicket gates closure rate is bigger. Similar to the second case, the peak time of the pressure head, $t_{p_3} = 18\,s$, is determined by the first stroke and the maximum head rise, $H_{p_3} = 234.2\,m$, is 18.3 % above the rated pressure head. At the end of this stroke, wicket gates dropped to 20 % opening from fully open configuration. When the second stroke starts, pressure head decreases severely. Due to the small opening of the wicket gates, the flow rate changes drastically, causing the wave speed to reflect in cycles and severe pressure rise in the penstock. The time between two successive peak points of the pressure heads is nearly $(4L/a) \cong 6\,s$. The start of the third stroke leads to more severe pressure heads at the turbine inlet. In this closure curve, the maximum value of the turbine rotational speed is $n_{p_3} = 1517.6\,rpm$, and it is the smallest compared to other closure curves. However, its magnitude is only 4.7 % smaller than that of the first closure curve. Table 5.13 shows some significant quantities in scenario A.

Table 5.13 Comparison of the Maximum Pressure Head and Turbine Speed Values for Scenario A

	Occurrence Time (s)	Maximum Pressure Head (m)	% Increase Compared to the Steady State Pressure Head	Occurrence Time (s)	Maximum Turbine Speed (rpm)	% Increase Compared to the Rated Turbine Speed
Closure Curve 1	43	219.1	10.1	30	1588.1	58.8
Closure Curve 2	19	220.1	11.2	21	1547.4	54.7
Closure Curve 3	18	234.2	18.3	19	1517.6	51.8

5.2.2.2 Scenario B: Regular Start Procedure of the Turbines during Load Acceptance

The opening of the wicket gates during load acceptance also creates waterhammer pressures in the penstock and they are investigated in this scenario. The hydraulic transient analysis is performed for the simultaneous start of the turbines. In routine operation, turbine wicket gates are opened in 360 seconds with six strokes. The final characteristics of the Francis turbine units and the order and timing of the strokes are given in Table 5.14 and 5.15, respectively. The results of the computations are presented in Figure 5.10.

Table 5.14 Final Characteristics of the Francis Turbine units for the Scenario B

	Turbine x 2
Discharge (m^3/s)	1.83
Net Head (m)	197.9
Turbine Output (kW)	3225.9
Opening Time of the Wicket Gates (sec)	360

Table 5.15 Stroke Timing and Order for Scenario B

1st Stroke	0 s<t<60 s
2nd Stroke	60 s<t<120 s
3rd Stroke	120 s<t<180 s
4th Stroke	180 s<t<240 s
5th Stroke	240 s<t<300 s
6th Stroke	300 s<t<360 s

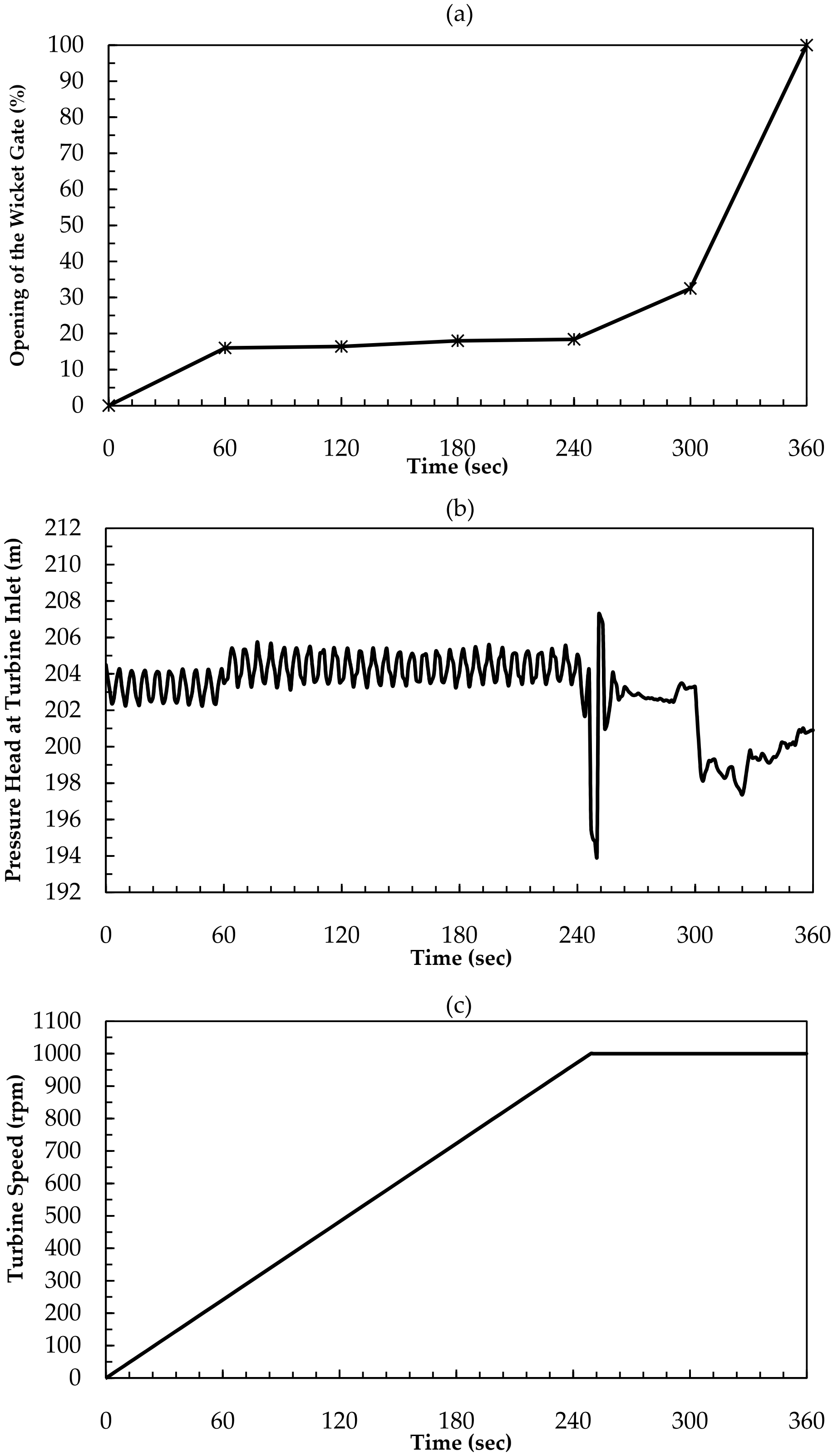


Figure 5.10 (a) Opening Law of Turbines; (b) Turbines' Inlet Pressure; (c) Turbines' Speed-up Behavior for Load Acceptance

As a result of the opening of the wicket gates during the first stroke, pressure head drops instantaneously. However, this drop is relatively small because of the small opening of the wicket gates. Owing to the same reason, pressure head oscillates with the period of wave reflection time in cycles. Also in this stage, turbine rotational speed starts to rise linearly. The second stroke has very small effect in the opening, and it has an adverse effect on the pressure head. The increase rate of the turbined flow is suddenly dropping with the second stroke and this causes the turbine inlet pressure to increase. The third and fourth strokes have relatively smaller effects in the opening; therefore, the wave speed reflection goes on and the pressure head oscillates throughout these movements. The turbine rotational speed reaches to its rated value and at the end of the fourth stroke. Then, to synchronize the unit to the grid with the fifth and final strokes, wicket gates are opened further.

The critical pressure head is determined by the fifth stroke. Because of the high increase rate in the gate opening, just after the action of it, pressure decreases severely and minimum head drop occurs. The wicket gate opening is large enough to relief the pressure wave, therefore after one reflection of the wave, pressure head fluctuation dampens. By the movement of the wicket gates during the final stroke, the pressure head falls again; however, the pressure head drop is smaller for this time, because the increase rate of the wicket gate opening is greater than the previous stroke. Then, with small fluctuations, the pressure head dampens and after the fully opened position of the gates, it reaches to its steady state value.

5.2.2.3 Scenario C: Instant Stop Procedure of the Turbines during Instant Load Rejection with the Protective Measure "Flywheel"

In the operation, when the load is instantly rejected or in case of a mechanical failure, the turbine has to be stopped in few seconds contrary to the regular operating cases. As a worst case scenario, the sudden stoppage of the turbine in 11 seconds with one stroke is also analyzed herein. For the scenario, both turbines stopped at the same time when they are working in their rated conditions. As a protective measure, two flywheels having different moment of inertia values are added to the generator and turbine couple and their effects on waterhammer pressures during the transient state are investigated. The as built moment of inertia of the rotating parts of the hydropower plant is 4800 kg.m^2. The system is analyzed for the as built conditions first, and then, a reasonable and applicable flywheel, which increases the total rotating mass of inertia 1200 kg.m^2, is considered. Finally a fictitious, much larger GD_g^2 value is regarded for the analysis. Here, G is the weight of rotating parts and D_g is the radius of gyration of rotating mass. The initial hydraulic characteristics of the turbines are given in Table 5.16.

Table 5.16 Initial Characteristics of the Francis Turbine units for the Scenario C, D & E

	Turbine x 2
Discharge (m^3/s)	1.83
Net Head (m)	197.9
Turbine Output (kW)	3225.9
Closing Time of the Wicket Gates (sec)	11

Results of transient simulations of the scenario are presented in Figure 5.11.

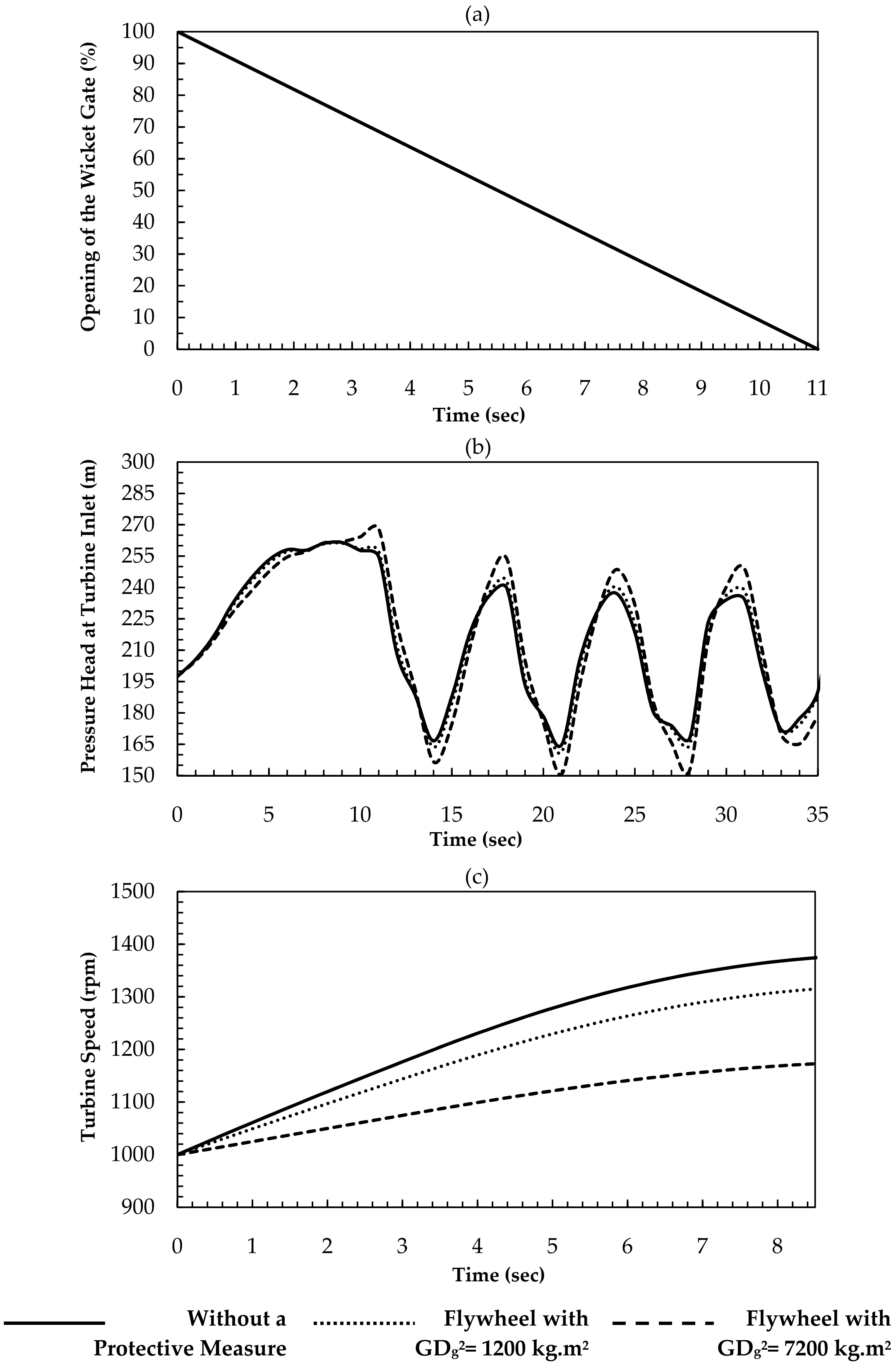


Figure 5.11 (a) Closing Law; (b) Turbines' Inlet Pressure; (c) Turbines' Speed Rise for Instant Load Rejection with Flywheel Effects

It can be seen from Figure 5.11 (b) that the maximum pressure rise at the turbine inlet during a sudden closure is about 32.1 % over the steady state pressure, $H_n = 197.9\,m$. For the simulation that has the biggest moment of inertia, this pressure rise ratio is about 35.5 %. The maximum and minimum transient pressure head values are close to each other for different moment of inertias. Also it is clear that as the value of GD_g^2 rises, the maximum pressure head increases and the minimum pressure head drops in the penstock. Because, when the magnitude of GD_g^2 increases, the rate of turbine speed change reduces, and this causes a reduction in the rate of change of turbined flow. The greater the flow velocity change in the turbine, the greater the pressure changes in the penstock.

For all simulations, when the closure starts, pressure rises sharply and drops severely. This pressure rise is accommodated by the turbine speed rise. After the fully closed position of the gates, the pressure fluctuates and turbine speed dampens. However, the explicit behavior of the rotational speed cannot be computed with HAMMER after the closure takes place for this scenario and closing law.

By use of flywheels, increasing the moment of inertia of rotating parts reduce the maximum rotational speed of the turbine significantly. From Figure 5.11 (c), it is clear that flywheels are very effective for preventing the turbine excessive overspeeding. Significant values of the results of scenario C are represented in Table 5.17.

Table 5.17 Comparison of the Maximum Pressure Head and Turbine Speed Values for Scenario C

	Occurrence Time (s)	Maximum Pressure Head (m)	% Increase Compared to the Steady State Pressure Head	Maximum Turbine Speed (rpm)	% Increase Compared to the Rated Turbine Speed
Without a Protective Measure	9	261.5	32.1	1376.4	37.6
Flywheel with GD_g^2=1200 kg.m^2	9	261.1	32.0	1316.9	31.7
Flywheel with GD_g^2=7200 kg.m^2	11	268.1	35.5	1174.3	17.4

5.2.2.4 Scenario D: Instant Stop Procedure of the Turbines during Instant Load Rejection with the Protective Measure "Pressure Relief Valve"

According to the simulation results of the previous scenario, flywheels are not effective in diminishing the maximum surges in the penstock. Although transient pressures resulting from the instant load rejection are not above the maximum allowable pressure of the system, for regulation of the pressure in the penstock, in scenario D, the system is analyzed with a pressure relief valve (PRV) for instant load rejection case. PRVs are commonly used in small hydroelectric power plants. A surge tank or an air chamber require relatively large amount of construction time and work and are not economic solutions for SHPs. Therefore a PRV, loaded by a spring, is placed 20 m away from the branch junction with a set pressure of 220 m on the main penstock. The schematic layout of the powerhouse and the pressure relief valve is given in Figure 5.12. The initial hydraulic characteristics of the system and the closing law of turbines are the same as those with the previous scenario's and given in Table 5.16 and Figure 5.11 (a), respectively.

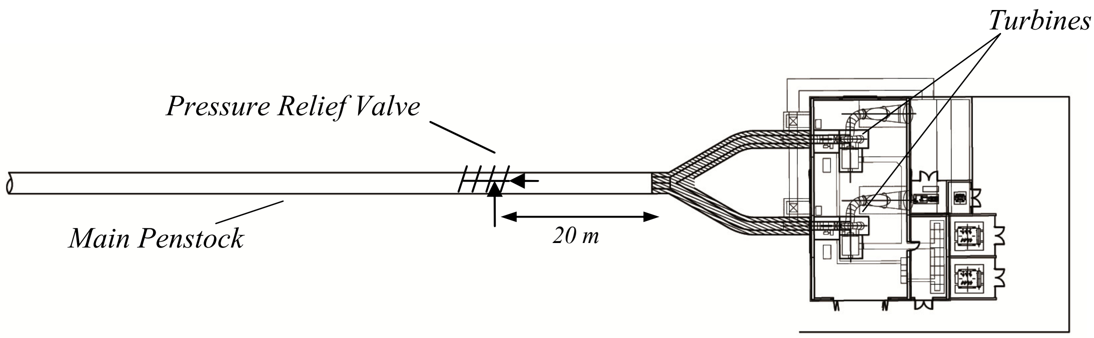


Figure 5.12 Plan view of the Powerhouse and the Location of Pressure Relief Valve

Pressure-time response of the system with and without pressure relief valve protection is given in Figure 5.13.

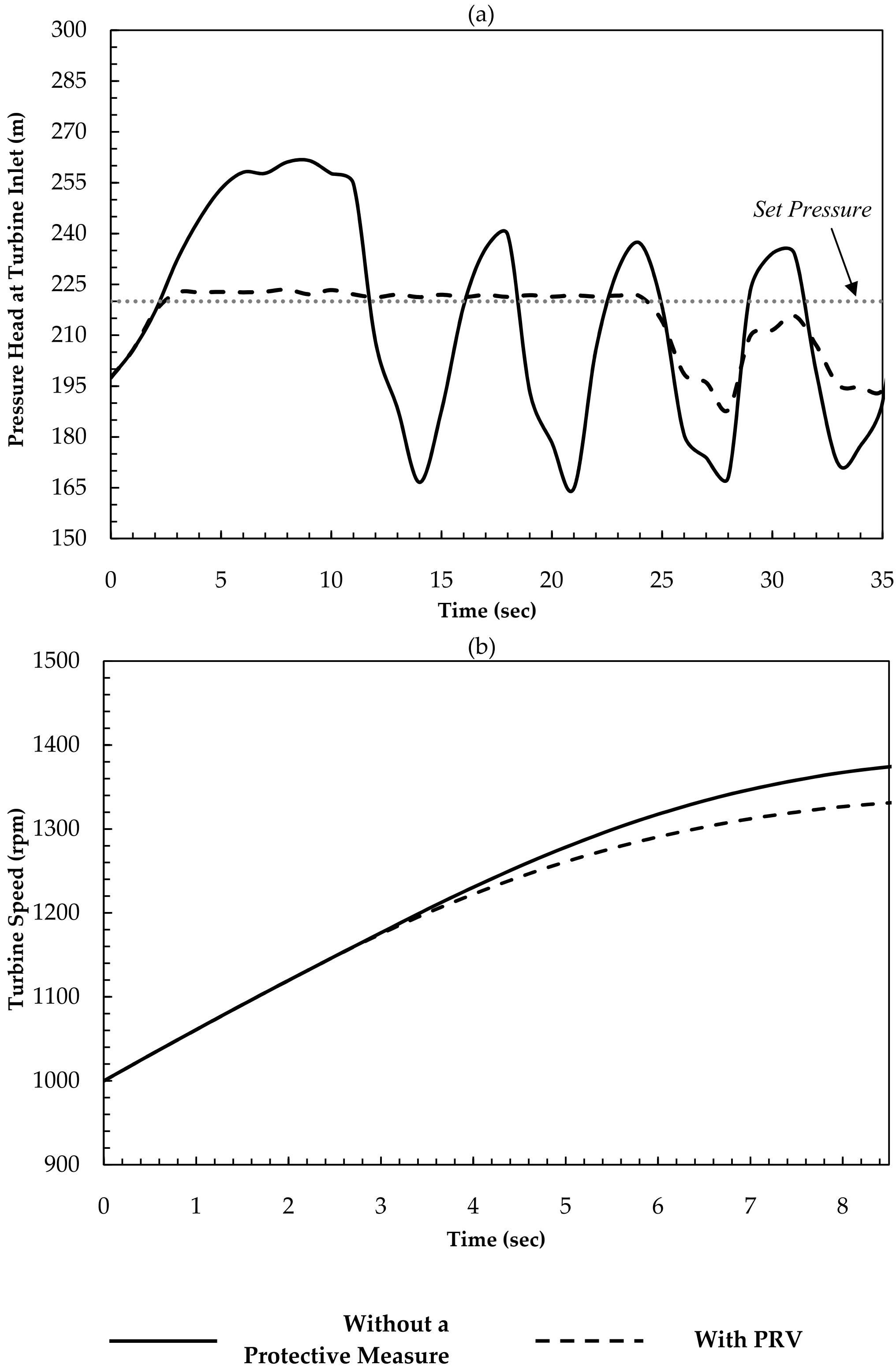


Figure 5.13 (a) Turbines' Inlet Pressure; (b) Turbines' Speed Rise with PRV Effect for Instant Load Rejection

Just before the transient state, the surge relief valve is closed. When the load on the unit is instantly rejected, by the closure of the wicket gates, pressure surge develops in the penstock and it propagates through it. Then, this pressure wave reaches the PRV. As the pressure increase is greater than the threshold pressure head, H_T=220 m, of the relief valve, it causes the PRV to open at t=3 seconds. One of the basic characteristic of the PRV is that it can open very quickly to decrease the surge pressures. Therefore, just after the pressure exceeds the threshold point, it acts rapidly. The head rise on the penstock is diminished by releasing some quantity of water in 22 seconds through the PRV (see Figure 5.13 a). As it is loaded by a spring, the release of that water is regulated and this provides controlled pressure waves in the penstock. As a consequence of this relief, pressure head at the turbine inlet is kept at nearly 222 m during the opening of the PRV. When the transient state pressure decreases to the set pressure point, valve is closed at t=24 seconds. The closing time is regarded as long enough not to cause secondary waterhammer pressures in the penstock. One of the advantages of the PRV is the closure of the relief opening after releasing the maximum pressure surge. This full closure causes the pressure not to drop instantaneously following the relief. Afterwards, the relieved pressure drops mildly and fluctuates until it dampens with the friction. It is clear that, system will stabilize faster than the unprotected one. Also the minimum pressure occurred in the penstock is kept relatively small with pressure relief valve. Moreover, it can be seen from Figure 5.13 (b) that, the maximum turbine rotational speed is decreased from 1376.4 rpm to 1332.7 rpm with a decrease rate of 4.4 % over its synchronous speed, 1000 rpm.

One disadvantage of the PRV may be the regular maintenance necessity of it; however reliability and efficiency of it in preventing large pressure rises in the penstock makes it a standalone protective measure, as can be seen from results, and it may easily replace a surge tank in small hydropower plants.

5.2.2.5 Scenario E: Instant Stop Procedure of the Turbines during Instant Load Rejection with the Protective Measure "Safety Membranes"

As an alternative to the PRV, another transient control device that acts directly due to pressure rise in the penstock, called safety membrane, is used in the system and its effects are investigated for instant load rejection case in this scenario. Despite being an effective protective measure, relief valves are more expensive and may require more maintenance work than safety membranes. In the present scenario, like a design study, safety membranes are designed to work alone as a protection device and protect the unit and penstock efficiently. Safety membranes are placed on the penstock, near the turbines for safer operation. They consist of three membranes with 10 m of intervals and a diameter of 300 mm. Schematic illustration of installed safety membranes are presented in Figure 5.14.

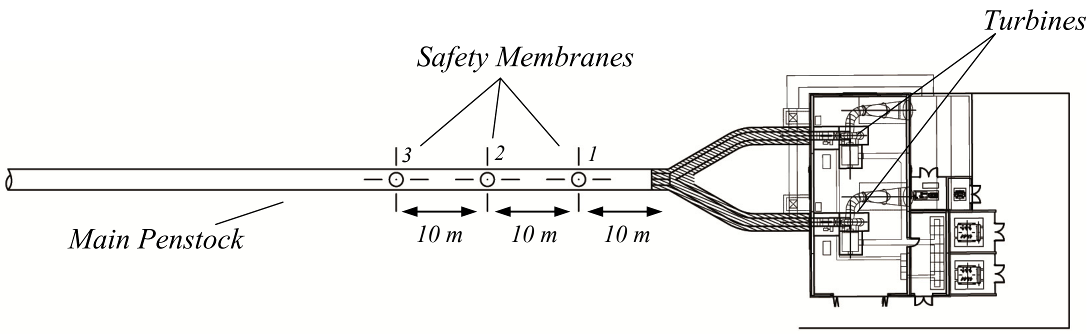


Figure 5.14 Plan view of the Powerhouse and the Location of Safety Membranes

These controlled weak points are designed to rupture in sequence when the pressure on the membranes rises above their set point. First one, nearest to the turbines, is designed to rupture first. When the discharge through it is inadequate to relief the pressure rise, the second one and similarly the third one will rupture successively. Figure 5.15 (a) and (b) shows transient pressure heads at turbines' inlets and turbine speed rise, respectively.

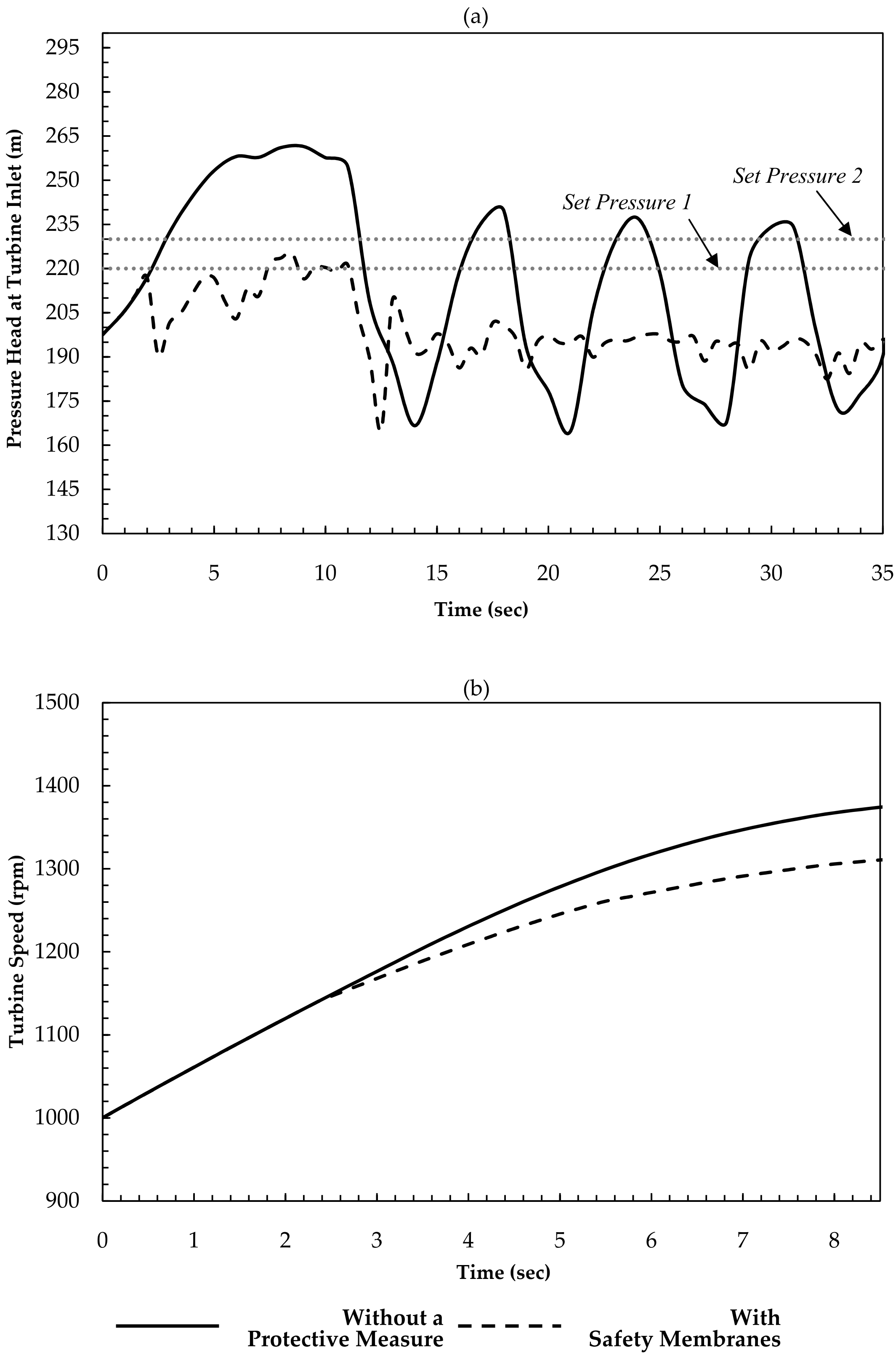


Figure 5.15 (a) Turbines' Inlet Pressure; (b) Turbines' Speed Rise with Safety Membranes Effect for Instant Load Rejection

To keep the pressure rise under a certain level in the penstock, the first and second membranes rupture pressure is set up to 220 m, which is labeled as “Set Pressure 1”, and the third one’s set pressure is selected as 230 m and labeled as “Set Pressure 2” in Figure 5.15 (a). The initial hydraulic characteristics of the system and the closing law of turbines are the same as those with the scenario C and given in Table 5.16 and Figure 5.11 (a), respectively.

By the instant rejection of the load and the initiation of the wicket gate closure, waterhammer pressure in the penstock rises to the set pressure of the membranes in 2 seconds and causes the first safety membrane to rupture. Releasing some quantity of water, it provides a drop in pressure; however, after the reflection of the wave speed from the forebay and further closing of the wicket gates, the pressure rises again to 220 m and causes the second membrane to explode. Similarly, released quantity of water from the second membrane is insufficient to suppress the pressure rise and third one ruptures. After this final explode, the closure of the wicket gates is fully completed. As safety membranes are free and uncontrolled openings, after every relief of the transient pressure, it drops instantaneously. After the fully closure takes place, the pressure oscillates in an unstable manner. It is possible to observe that considerable amount of pressure rise at the end of the penstock is dampened out thanks to the safety membranes. Moreover, their existence on the system decreases the maximum turbine rotational speed from 1376.4 rpm to 1312.8 rpm and this means a 6.4 % decrease over the synchronous speed, 1000 rpm.

Despite the replacement of a safety membrane may be troublesome; they are reliable and can be simply operated. It can be concluded from the results that safety membranes can also work alone and protect the penstock from large waterhammer pressures effectively.

Comparison of significant values of scenario D and E is given in Table 5.18.

Table 5.18 Comparison of the Maximum Pressure Head and Turbine Speed Values of Scenario D & E

	Occurrence Time (s)	Maximum Pressure Head (m)	% Increase Compared to the Steady State Pressure Head	Maximum Turbine Speed (rpm)	% Increase Compared to the Rated Turbine Speed
Without a Protective Measure	9	261.5	32.1	1376.4	37.6
With Pressure Relief Valve	8	223.5	12.9	1332.7	33.3
With Safety Membranes	8.5	225.2	13.8	1312.8	31.3

5.3 Discussions on the Case Studies

In this chapter, various transient states of two small hydropower plants were modeled by the computer software. For both systems, by using real operating data, the time dependent variable, namely pressure at the end of the penstocks were computed for the transient states. Turbine rotational speed during waterhammer was also modeled for Francis units. By comparing the computed results of "Case Study 1" with its measured ones, the validation of the software is satisfied. Following inferences and discussions are made for this case study by considering its compared results.

There is a good agreement between measured and computed results in regular stop procedure. Especially maximum values of transient pressures are very close. However, there are relatively little differences between the results. The lack of agreement may be the consequence of the improper transient behavior description of the simulation model. In the model, Pelton turbine nozzles were characterized by needle valve. The rate of change of the flow area of needle valve may be different from nozzle's flow area change rate. Phase and small amplitude shifts in transient pressures might be caused by this reason.

In load acceptance case, the agreement between results is well established for 60 seconds. In this time range, nozzles are opened and kept at the lowest rate at which the turbine rotates at its rated speed. It can be concluded that, this start up procedure prevents low transient state pressures in the penstock and hence the formation of vapor cavities. When the unit was synchronizing with the grid system, simultaneously, nozzle opening was increasing. This action caused a divergence between measured and computed results. In the model, unsteady friction factors were assumed to be the same with the steady state's friction factors. During relatively low pressures, the variation of the wave speed throughout the penstock might have affected the transient frictional losses, and might have caused disagreements between computed and measured results.

The governor controlled instant load rejection case illustrates that the mechanically identical turbine units may be started to close at the same time with different closure times. The closing law and order of the turbines are determined according to their hydraulic conditions. As the main penstock of the power plant branches just upstream of the powerhouse, the transient state pressures at the end of the branching penstocks are also balanced with this operating procedure and the possibility of existence of big pressures and forces on the branch is eliminated. For both turbines, measured and computed transient pressures have similar oscillation trends. In other words there is a good agreement between the shapes of pressure curves; but, there are shifts between them. These shifts may be caused by the existence of air in the water before the rejection or entrainment of it during the rejection.

"Case Study 2" is a hydropower plant that has been operated only for six months. During the design of this power plant, transient analysis was done by considering its operating cases. The aim of this case study is to examine other possible operating applications and alternatives for further practices and/or problems of the operation. Following discussions and interpretations are made regarding the simulation results.

The effect of different wicket gate closing laws on the transient state pressures at the penstock and turbine rotational speed is investigated with the load rejection case of the hydropower plant. To observe the effects of valve closing law and its strokes, three different closing laws which have the same closure times but different strokes are studied. According to the results, if the wicket gate closing rate is greater during the first stroke, bigger head rises occurs in the penstock. This is because when the gates are closed faster, discharge passing through the turbine decreases. At the same time, the maximum value of turbine rotational speed reduces. Similarly, slower the gates are closed at the first stroke, greater the turbine rotational speed takes place. Results showed that rapid closing would result in an increase in transient pressures in the penstock and decrease in the maximum turbine rotational speed rise. Simultaneously reducing the maximum pressure rise and turbine rotational speed with closing law can only be achieved with an optimization study.

As mentioned in the first case study, like in Pelton turbines, complete starting of the two Francis turbines simultaneously does not cause very low pressures in the penstock if the wicket gates are opened slowly and the unit is synchronized with the grid system after the turbines reach their rated speed with the speed no load gate position.

Additionally, the instantaneous disconnection of the generator from the grid system is investigated in scenario C. Following this action, the rotational speed of the turbine and hence the frequency of the system increases rapidly. The governor must act immediately to keep the turbine and generator couple stable. The stable operation of a hydropower plant can be handled by keeping the turbine speed rise within the permissible limits. It can be effectively controlled by the sufficient moment of inertia of the system. Turbine inertia is very small relative to the generator. Therefore, to increase the GD_g^2 value, generator's inertia may be increased or a flywheel can be added to the system as a more economical solution. It is clear from the computations done for two flywheel effects that, when the system inertia increases, the turbine rotational speed substantially reduces. However, oscillation trend of the transient state pressures at the end of the penstock does not change greatly. Moreover, the maximum and minimum pressure head values are close to each other for three different moment of inertia values. It can be concluded that, as a protective measure, a flywheel is very effective for decreasing turbine rotational speed; however, it is insufficient in reducing the maximum transient state pressures in the penstock.

Since the flywheels do not have satisfactory effects on reducing the rising pressure in the penstock, two different and simple protection measures acting as a result of pressure rise are used within the system and their effects are simulated on both turbine rotational speed and pressure head rise in the penstock. These devices are pressure relief valve and safety membranes. The instant load rejection case that is considered for the simulations does not result in great pressures that exceed the maximum allowable pressure of the penstock; however, regulated transient pressures in the penstock and turbine rotational speed ensure safer operation, minimize maintenance cost and may extend lifetime of the mechanical equipment. Threshold pressures of both PRV and safety membranes are set to yield the same maximum pressure in the penstock with the regular stop, in case of the instant stop of the turbines. In both scenario E and D, protection measures reduce the maximum pressure surge approximately 19 % over steady state pressure. Without protection, generated pressure wave propagates along the penstock and creates cycling effect. The existence of each protective device separately minimizes this effect. Their behavior shows that they are effective, reliable and can be used as the only measure of safety in a small hydropower plant to protect its penstock. Also, the effect of both protective measures on pressure rise and turbine speed proves that there is no need for complicated and expensive protection devices in SHPs.

CHAPTER 6

CONCLUSIONS AND FINAL REMARKS

Growing energy consumption of Turkey has led the government to promote the private sector in investing clean and renewable energy sources; especially, small hydropower plants. Nowadays design studies of many small hydropower plants are being conducted by several companies. In design of a hydropower plant it is inevitable to analyze the system for unsteady flow conditions. However, a small hydropower plant can not be treated as a large scale HP in transient analysis. Their transient behavior is different since small hydropower plants have considerably long penstocks and small inertia turbines. In this thesis, waterhammer phenomenon in SHPs is investigated. Based on the discussions of this study and results of the simulations, following conclusions can be drawn:

- Dynamic simulation of any small hydropower plant by the help of computer software is inevitable because they save time and help the designer in complicated studies of waterhammer analysis. A system with its possible components and protective devices can be modeled with various transient scenarios and alternatives to develop a cost effective hydropower plant easily. However, to have a proper description of transient behavior in the model system should be defined properly.
- For the operational safety of a small hydropower plant, holding a detailed transient analysis that considers both usual and extreme operating conditions is the key factor. Every possible excitation that starts transient

state in the penstock should be considered. An ignored worst case disturbance may cause undesirable consequences in unprotected systems.

- A Pelton turbine unit can be modeled as a needle valve in a computational model. Compared results proved that their transient behavior in a system is very similar to the needle valves.

- Transient states of Pelton turbines are more controllable compared to Francis turbines. Even if, load on a Pelton unit instantly rejected, needle valves can be closed slowly compared to Francis turbines wicket gates.

- During load acceptance procedure of both turbine types, speed no load gate position of the wicket gates and needle valves play very important role. It prevents the occurrence of excessively low pressures and hence the formation of vapor cavities in the penstock.

- The closing law of both wicket gates and needle valves has a vital effect on waterhammer pressures in the penstock and turbine rotational speed rise. In order to cope with transient situations by dampening the waterhammer effects and reduce the possible risk of damage to the system, an optimum closing law can be chosen instead of installing auxiliary protective devices on the system.

- Flywheels can easily reduce the speed rise of reaction turbines during transient states. Rather than decreasing the pressure rise in the penstock, they are very effective in preventing the turbine from reaching its runaway speed.

- Speed rise of impulse turbines in small hydropower schemes is not considerably large and they may not require any protective device that prevents turbine from overspeeding.

- An installed pressure relief valve or safety membrane can be effectively used in reducing pressure peaks of waterhammer, protecting the penstock and preventing turbines from runaway speed in small hydropower plants. They might be preferred as standalone protective measures instead of surge tanks or air chambers in these types of hydropower plants.

For prospective researches, a study that acquires small hydropower plant operation data would be very helpful for investigation of waterhammer problems. The experiences gathered from these investigations may also help developing a computer program that only deals with transient simulation and operation of small hydropower plants and even determination of protection devices. Moreover a built-in simulation tool that analyses the economy of the protection devices with an optimization study would be beneficial.

In addition to the above stated recommendations, investigation of the damping effect of the forebay, as a surge tank, on pressure rise in the penstock and speed rise of the turbine in small hydropower schemes may be the aspect of future studies.

REFERENCES

Adamkowski, A. (2001). Case Study: Lapino Powerplant Penstock Failure. *ASCE Journal of Hydraulic Engineering* , 541-555.

Bentley HAMMER. (2010). *HAMMER-Water hammer and transient analysis software-from Bentley*. Retrieved April 12, 2010, from http://www.bentley.com/en-US/Products/HAMMER/Product-Overview.htm

Bentley HAMMER V8i Edition User's Guide. (n.d.).

Bergant, A., Simpson, A.R., Tijsseling A.S. (2004). *Water Hammer With Column Separation: A Review of Research in the Twentieth Century.* Centre for Analysis, Scientific computing and Applications.

Birim Hydroelectric Production Co. (2008). *Final Design Report of Erfelek HP.* Ankara.

Chaudhry, M. H. (1987). *Applied Hydraulic Transients.* New York: Van Nostrand Reinhold Company Limited.

Creager, P. W. and Justin, D. J. (1950). *Hydroelectric Handbook.* New York: John Wiley & Sons.

DSI. (2006). *Water Use Right Agreements of Hydroelectric PPs.* Retrieved February 03, 2010, from General Directorate of State Hydraulic Works: http://www.dsi.gov.tr/skatablo/Tablo1.htm

Gagnon, L., Bélanger, C., Uchiyama, Y. (2002). Life-cycle assessment of electricity generation options: The status of research in year 2001. *Energy Policy* , 1267-1278.

GAMA Energy Inc. (2008). *Final Design Report of Çakırlar HP.* Ankara.

Gulliver, S. J. and Arndt, E. R. (1991). *Hydropower Engineering Handbook.* McGraw-Hill.

Hagihara, S., Yokota, H., Goda, K., Isobe, K. (1979). Stability of a Hydraulic Turbine Generating Unit Controlled by P.I.D. Governor. *IEEE Transactions on Power Apparatus and Systems* , 2294-2298.

Hovey, L. M. (1962). Optimum Adjustment of Hydro Governors on Mantoba Hydro System. *American Institute of Electrical Engineers* , 581-586.

Jiandong, T., Naibo, Z., Xianhuan, W., Jing, H., Huishen D. (1997). *Mini Hydropower.* Hangzhou: John Wiley & Sons.

Jimenez, O. F. and Chaudhry, M. H. (1987). Stability Limits of Hydroelectric Power Plants. *ASCE Journal of Energy Engineering* , 50-60.

Karadzic, U., Bergant, A., Vukoslavcevic, P. (2009). A Novel Pelton Turbine Model for Water Hammer Analysis. *Strojniski vestnik - Journal of Mechanical Engineering* , 369-380.

Küçükali, S. and Barış, K. (2009). Assessment of small hydropower (SHP) development in Turkey: Laws, regulations and EU policy perspective. *Energy Policy*, 3872-3879.

Laguna, M., Upadhyay, D., Taylor, S. (2006). Flowing to the East: Small Hydro in Developing Countries. *Renewable Energy World* , 126-131.

Linsley, R. K., Franzini, J. B., Freyberg, L. D., Tchobanoglous, G. (1992). *Water-Resources Engineering.* McGraw-Hill.

Lüdecke, H. J. and Kothe, B. *Water Hammer.* KSB.

McStraw, B. (1996). *Inspection of Steel Penstocks and Pressure Conduits.* Colorado: United States Departments of the Interior Bureau of Reclamation.

Ni, F., Hu, P., Wang, Q. (1996). Numerical Simulation of Hydraulic Transients in Hydropower Plant Using Safety Membranes. *ASCE Journal of Hydraulic Engineering* , 298-300.

Paish, O. (2002). Small Hydro Power: Technology and Current Status. *Renewable and Sustainable Energy Reviews* , 537–556.

Parmakian, J. (1963). *Waterhammer Analysis.* New York: Dover Publications.

Peicheng, H., Pusheng, Z., Elkouh, A. F. (1989). Relief Valve and Safety Membrane Arrangement in Lieu of Surge Tank. *ASCE Journal of Energy Engineering* , 78-83.

Penche, C. (2004). *Guide on How to Develop a Small Hydropower Plant.* Belgium: European Small Hydropower Association.

Ramos, H. and Almeida, B. (2002). Parametric Analysis of Water-Hammer Effects in Small Hydropower Schemes. *ASCE Journal of Hydraulic Engineering* , 689-696.

Ramos, H., Almeida, A. B., Portela, M., Almeida H. P. (2000). *Guidelines for Design of Small Jydropower Plants.* Belfast, North Ireland: WREAN and DED.

Selek, B., Kırkgöz, M. S., Selek, Z. (2004). Comparison of computed water hammer pressures with test results for the Çatalan power plant in Turkey. *Canadian Journal of Civil Engineering* , 78-85.

Souza, O. H., Barbieri, N., Santos, A. H. M. (1999). Study of hydraulic transients in hydropower plants through simulation of nonlinear model of penstock and hydraulic turbine model. *IEEE Transactions on Power Systems* , 1269-1272.

Stephenson, D. (2002). Simple Guide for Design of Air Vessels fo Water Hammer Protection of Pumping Lines. *ASCE Journal of Hydraulic Engineering* , 792-797.

Vakil, A. and Firoozabadi, B. (2009). Investigation of Valve-Closing Law on the Maximum Head Rise of a Hydropower Plant. *Scientia Iranica Mechanical Engineering*, 222-228.

Warnick, C. C. (1984). *Hydropower Engineering.* New Jersey: Prentice-Hall.

WEC. (2007). *2007 Survey of Energy Resources.* London: World Energy Council.

Wylie, E. B. (1984). Fundamental Equations of Waterhammer. *Journal of Hydraulic Engineering, ASCE* , 539-542.

Wylie, E. B., Streeter, V. L., Suo, L. (1993). *Fluid Transients in Systems.* New Jersey: Prentice-Hall.

Yanmaz, M. (2006). *Applied Water Resources Engineering.* Ankara: METU Press.

Yüksek, Ö., Kankal, M., Kömürcü, M. İ., Önsoy, H., Akpınar, A. (2007). The Importance of Hydropower Plants in Turkey's Energy Planning. *International Congress on River Basin Management* (pp. 720-732). Antalya: General Directorate of State Hydraulic Works.

APPENDIX

In this thesis, two case studies under various operation conditions such as load rejection, load acceptance and instant load rejection are studied. A software, HAMMER, that utilizes method of characteristics for solving nonlinear differential equations of transient flow is used in the study. In determination of the approximate maximum and minimum pressure head values in the penstock during waterhammer, rigid water column theory (RWCT) can be used if the transient state is caused by uniform movement of the gate. In this theory, water is considered to be incompressible, and pipe walls do not stretch regardless of the pressure inside the pipe. For uniform gate operations, pressure head change can be calculated by (Parmakian, 1963)

$$\Delta H = H_g\left(\frac{K}{2} \pm \sqrt{K + \frac{K^2}{4}}\right) \tag{A.1}$$

and

$$K = \left(\frac{LV_0}{gH_gT}\right)^2 \tag{A.2}$$

where;

ΔH :	Change in the pressure head, [m]	V_0 :	Initial velocity, [m/s]
H_g :	Gross head, [m]	g :	Gravitational acceleration, [m/s^2]
K :	Dimensionless parameter	T :	Time of closure or opening, [sec]
L :	Length of the penstock, [m]		

Following tables show the comparison of maximum and minimum transient pressure head values of two case studies with related scenarios, according to different solution methods.

Table A.1 Computation Table for Rigid Water Column Theory Pressure Peaks for Çakırlar SHP

Scenario	L (m)	V_0 (m/s)	H_g (m)	T (sec)	K	ΔH_{max} (m)	ΔH_{min} (m)	H_{max} (m)	H_{min} (m)
A (Turbine unit 1)	1116.67	3.45	460.56	60	2.01×10^{-4}	6.58	-6.49	467.14	454.07
A (Turbine unit 2)	1116.67	1.96	463.43	60	6.45×10^{-5}	3.74	-3.71	467.17	459.72
B	1116.67	3.77	461.82	108*	7.41×10^{-5}	3.99	-3.96	465.81	457.86
C (Turbine unit 1)	1116.67	3.12	459.63	97	6.33×10^{-5}	3.67	-3.64	463.30	455.99
C (Turbine unit 2)	1116.67	3.12	459.63	29	7.10×10^{-4}	12.40	-12.08	472.03	447.55

* Opening of the nozzles is assumed to be uniform after 60 seconds.

Table A.2 Computation Table for Rigid Water Column Theory Pressure Peaks for Erfelek SHP

Scenario	L (m)	V_0 (m/s)	H_g (m)	T (sec)	K	ΔH_{max} (m)	ΔH_{min} (m)	H_{max} (m)	H_{min} (m)
A	1538.47	2.76	204.9	57.61	1.34×10^{-3}	7.64	-7.37	212.54	197.53
B	1538.47	2.76	204.9	120*	3.09×10^{-4}	3.63	-3.57	208.53	201.33
C & D & E	1538.47	2.76	204.9	11	3.68×10^{-2}	43.25	-35.71	248.15	169.19

* Opening of the wicket gate is assumed to be uniform after 240 seconds.

Table A.3 Comparison of Maximum and Minimum Transient Pressure Head Values of Çakırlar SHP with Related Scenarios According to Different Solution Methods

Scenario	Measured	HAMMER Solution	RWCT Solution
	Maximum Pressure Head (m)	Maximum Pressure Head (m)	Maximum Pressure Head (m)
A (Turbine unit 1)	461.50	462.29	467.14
A (Turbine unit 2)	467.90	466.79	467.17
C (Turbine unit 1)	471.40	476.98	463.30
C (Turbine unit 2)	473.40	475.76	472.03
		HAMMER Solution	**RWCT Solution**
		Minimum Pressure Head (m)	Minimum Pressure Head (m)
B	451.70	449.06	457.86

Table A.4 Comparison of Maximum and Minimum Transient Pressure Head Values of Erfelek SHP with Related Scenarios According to Different Solution Methods

Scenario	HAMMER Solution	RWCT Solution
	Maximum Pressure Head (m)	Maximum Pressure Head (m)
A	219.08	212.54
C&D&E	261.50	248.15
	HAMMER Solution	**RWCT Solution**
	Minimum Pressure Head (m)	Minimum Pressure Head (m)
B	193.97	201.33

It should be noted that RWCT provides a simple way for determining waterhammer effects for slow valve (gate) operations. For rapid valve operations, the elastic waterhammer theory must be used.