RELIABILITY BASED DESIGN OF PUMPING STATION FOR AN URBAN WATER SUPPLY SCHEME

A DISSERTATION

Submitted in partial fulfillment of the
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(Kiswarman)

DECLARATION

Certified that the work which is being presented in the dissertation entitled " ***Reliability Based Design of Pumping Station for an Urban Water Supply Scheme*** ", in partial fulfillment of the requirements for the award of the degree of "***Master of Technology***" in Water Resources Development ***(Mechanical)***, submitted to the Water Resources Development Training Centre, Indian Institute of Technology Roorkee, India under the supervision of **DR. M. L. Kansal,** Associate Professor and **Prof. Gopal Chauhan,** professor WRDTC, Indian Institute of Technology Roorkee.

I have not submitted the matter embodied in this dissertation for the award of any other degree.

Date : June, 2004
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This is to certify that the above statement made by the candidate is correct to the best of our knowledge.

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ABSTRACT

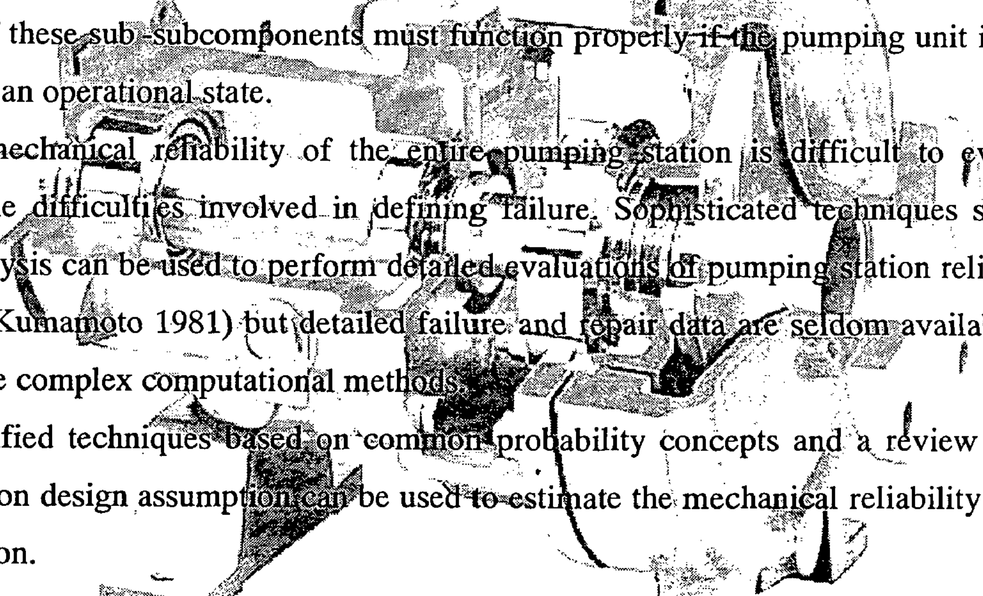
"Reliability Based Design of Pumping Station for An Urban Water Supply Scheme"

Pumping stations are one of the major components of water supply systems. A pumping station consists of one or more pumping units supported by appropriate components such as electrical, piping, control, and structural.

The pumping unit is primary subcomponent within the pumping station and includes four major sub-subcomponents: pump, driver (motor, engine, etc), power transmission, and controls. Although there are more sub-subcomponents in pumping unit, and these may be selected as the level of evaluation but the availability of failure and repair data is rather poor. Each of these sub-subcomponents must function properly if the pumping unit is to be considered in an operational state.

The mechanical reliability of the entire pumping station is difficult to evaluate because of the difficulties involved in defining failure. Sophisticated techniques such as fault tree analysis can be used to perform detailed evaluations of pumping station reliability (Henley and Kumamoto 1981) but detailed failure and repair data are seldom available for input for these complex computational methods.

Simplified techniques based on common probability concepts and a review of the pumping station design assumption can be used to estimate the mechanical reliability of the pumping station.



A common design practice is to install sufficient pumps to handle peak flows and include a spare pump of equal size to accommodate any downtime of the other pumps. Thus mechanical failure of the pumping station could be defined as the simultaneous failure of two or more pumping units while peak capacity is required.

Pipe networks seldom consist of loops only. In practice, they are of composite type consisting of loops, branches, and serial parts. However, it is usually possible to reduce a composite network to a looped network for analysis, and therefore composite networks are referred to as looped networks. Looped networks are common in densely populated areas.

They provide a greater reliability of service than branched systems, eliminate dead ends, and improve flow conditions during periods of high local demand.

EPANET can be used to plan and improve a system's hydraulic performance. Pipe, pump and valve placement and sizing, energy minimization, fire flow analysis, vulnerability studies, and operator training are just some of the activities that EPANET can assist with.

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CHAPTER

1

Introduction

CHAPTER – 1

INTRODUCTION

1.1. GENERAL

In developing countries, one of the major problems is to provide safe drinking water in adequate quantity to every citizen of the country. Existing supply systems, which were designed decades earlier, are now highly inadequate to cater for the increasing population coupled with higher per capita demand. Huge cost is thus involved in the planning, design, installation and up grading of such systems.

Because of its importance to the needs of society and industrial growth, considerable emphasis has recently been given to the condition of the nation's infrastructure. Large capital expenditure will be needed to bring the concerned systems to higher levels of serviceability and to create a vigor industrial competitiveness. One of the most vital services to industrial growth is an adequate water supply system, without which industry cannot survive.

Adequate, accessible water supply is a prerequisite for improved public health and socio-economic development. Improvements in water supply can result in a number of substantial benefits. Water systems are installed to consistently provide water in sufficient quantity to users at an acceptable pressure and quality as economically as possible. Because individual components of a system can fail, water-distribution systems are designed with a great deal of redundancy so that the system can perform adequately even some if individual components (e.g., pipes, valves, or pumps) are out of service.

The urban water distribution system is composed of three major components: pumping stations, distribution storage, and distribution piping. These components may be further divided into subcomponents, which can in turn be divided into sub-sub components. A pumping station is one of the major components of water supply systems and consists of one or more pumping units supported by four major sub-subcomponents: pump, driver (motor, engine, etc), power transmission, and controls. Each of these sub-subcomponents must function properly if the pumping unit is to be considered in an operational state.

A common design practice is to install sufficient pumps to handle peak flows and include a spare pump of equal size to accommodate any downtime of the other pumps. Thus mechanical failure of the pumping station could be defined as the simultaneous failure of two or more pumping units while peak capacity is required.

The primary function of a pump is to transfer energy from a power source to a fluid, and as a result to create flow, lift, or greater pressure on the fluid. A pump can impart three types of hydraulic energy to a fluid: lift, pressure, and velocity. In water supply, irrigation and drainage systems, pumps are commonly used to lift water from a lower elevation to a higher elevation and/or add pressure to the water.

Displacement pumps have limited capacities and are not suitable for pumping large amounts of water required. They are used mainly for very high head. Dynamic pumps (centrifugal) are commonly used for pumping of water. So in this study centrifugal pump is mainly discussed.

Piping represents the largest single component. Treated-water-storage reservoirs are the most visible components, especially elevated tanks. Pumping stations are the energy users in most water systems. Valves serve a variety of functions, from isolating system components, to regulating pressures and flows, controlling surges, controlling the direction of flow, releasing air, breaking vacuums, or relieving excess pressure.

The actual point at which a pump will operate (i.e., operating; point) is determined by combining the characteristics of the pump with the hydraulic characteristics of the system in which it is installed. For a given installation, the operating point will vary somewhat over time as water level in nearby tanks fluctuates, water consumption varies with time, valves are adjusted, and other pumps are turned on and off. The key to good pump selection is choosing a pump that will operate near its best efficiency point over the range of conditions it will experience over its life.

The characteristics of the pumping system in which the pump is installed can be represented by the system head curve which a function is giving the head required to force water into the system versus the flow rate. The intersection of the system head curve and the pump head characteristic curve is the pump operating point. There is not a single system head curve for a distribution system but rather a band of system head curves depending on the water levels in tanks and water usage rates in the system at any point in time.

Practicing Engineers are not usually involved in the detailed design of hydraulic machines. However they are frequently called upon to design systems for which these machines form an integral part. It is therefore important to have an understanding of the characteristics of pumps and their interaction with other hydraulic system components.

The objectives of the present study of Reliability Based Design of a Pumping Station are:

1. To study the components of a Water Supply System
2. To study the components of a Pumping System
3. To study the types of Rotodynamic Pumps
4. To identify the requirements of pumping units for a typical Water Supply System.

Reliability is the probability of remaining in an operating state as a function of time given that the system started in the operating state at time t = 0. The operating state (up) refers to a pump being operable, i.e., it is either in operation or can be put into operation, whereas a failed state (down) refers to a pump that has a mechanical failure and is not operable.

Availability is the probability of being found or residing in the operable state at time t given that the system started in the operating state at time t = 0. Unavailability is the probability of residing in the failed state at time t given that the system started in the operating state at time t = 0.

Simplified techniques based on common probability concepts and a review of the pumping station design assumption can be used to estimate the mechanical reliability of the pumping station.

The operator of such a water distribution system has an interest in two types of reliability. First, there is mechanical reliability, which relates to the operability of equipment i.e., obviously, it is desired to install equipment, which is not subject to frequent breakdowns. Second, there is hydraulic reliability, i.e., the utility wishes to consistently provide the specified amounts of its product at the required pressures and time and location desired

The example network used to illustrate applications of model consists of M pipelines and N demand nodes and demand elevations EL. Four loading conditions are implemented in the model by multiplying base loading condition with certain factor. The minimum head H required to satisfy the reliability are defined. A design period of 20.000 hours is used with all the operation costs, maintenance and repair cost, investment cost and depreciation with 10% salvage in any of these lifetime converted into present worth using interest rate of 8%.

With the advent of high-speed computing devices and the computer programs for solving pipe-network and pumping system problem, it has now become possible to analyze complex water distribution system and to involve people of different level of expertise in process of planning and decision. The EPANET program has received rather wider acceptance by municipalities, hence the simulation of problem has been done using program of EPANET (Version 2.0).

1.2. ORGANIZATION OF THESIS

Chapter-1 emphasizes the importance of water supply in general, followed by associated problems so as to define objectives and purposes and is concluded by scope of study presented in this dissertation.

Chapter-2 deals with components of water supply scheme and mechanical and hydraulic Reliability of pumping station.

Chapter-3 covers critical review of the literature available on pump. The classification of pumps based the principle by which energy is added to the fluid is covered and then the means by which this principle is implemented is identified, and finally specific geometries commonly used have been distinguished. The discussion on pumps in this study is restricted to centrifugal pumps.

Chapter-4 presents hydraulics and is limited to turbulent flow of water in closed conduits (pipe) flowing full. Flow in a pipe can either be steady or unsteady flow, in this study; only steady flow has been reviewed. The steady flow has been described by the continuity and energy equations and head loss. In this chapter, the type of the network i.e., serial, branching and looped has been described. Epanet program that is used for the simulation has been introduced in general at the end of this chapter.

Chapter-5 determines the optimal number of pumping units by simulation with Epanet program by combination of 2+1, 3+1, 4+1 and 5+1 units of pumps. The outage capacity for each combination has been calculated based on the simple techniques and by using common probability concepts. The result and discussion of the analysis of pumping system has been carried out integrated with the system reliability and the optimal number of the pumping unit determined based on the lowest transfer cost. The conclusion has been described in the end of this chapter.

CHAPTER

2

Components of Water Distribution System and Reliability

CHAPTER – 2

COMPONENTS OF WATER SUPPLY SCHEME AND RELIABILITY

2.1. PHYSICAL COMPONENTS

Water distribution system typically consists of a network containing large number of interconnecting pipes with occasional control valves. Reservoirs are connected at strategic points throughout the network to provide storage capacity and maintain required hydraulic pressure for consumer water demands. Rivers, lakes and tubewells are the usual sources of water supply, which is treated before being supplied to the consumers. A typical water supply scheme thus can be broadly sub-divided into two basic systems, which are:

1. Source, Treatment and Clear-water storage System.
2. Pumping, Distribution Storage and Distribution System.

A conventional water treatment system consists of raw water supply unit, mixing unit, sedimentation chamber, filtration and storage unit for clear water. Similarly, a typical urban water distribution system is composed of three major components: pumping stations, distribution storage, and distribution piping (Fig. 2.1). These components may be further divided into subcomponents, which can in turn be divided into sub-sub components. For example, the pumping station components consist of structural, electrical, piping, and pumping unit subcomponents. The pumping unit can be further divided into sub-subcomponents like pump, driver, controls, power transmission, and piping and valves. The exact definition of components, subcomponents and sub-subcomponents is somewhat fluid and depends on the level of detail of the required analysis and to a somewhat greater extent the level of detail of available data. In fact, the concept of component-subcomponent-sub-subcomponent merely defines a hierarchy of building blocks used to construct the urban water distribution system.

(a) **Sub-subcomponents**. Sub-subcomponents represent the basic building blocks of systems. They are the smallest element for which failure and repair data are available. Individual sub-subcomponents may be common to a number of subcomponents within the water distribution system. Seven sub-subcomponents can be readily identified for analysis: pipes, valves, pumps, drivers, power transmission units, controls, and storage tanks. Data requirements, including time to failure and time to repair information can be found in published sources or in those cases where existing data is insufficient, a utility specific database can be developed.

(b) **Subcomponents**. Subcomponents represent the basic building blocks for components and are composed of one or more sub-subcomponents integrated into a common operational element. For example, the pumping unit subcomponent is composed of pipes, valves, pump, driver, power transmission, and control sub-subcomponents. These subcomponents can be used to evaluate the reliability of the urban water distribution systems: pumping units, pipe links, and storage tanks.

(C) **Components**. Components represent the largest functional element comprising the urban water supply scheme. Components are composed of one or more subcomponents. For purposes of this study, only one component has been used to evaluate the reliability of the urban water distribution system i.e. pumping stations.

2.2. APPROACH TO EVALUATE RELIABILITY

The operator of a water utility has an interest in two types of reliability. First, there is mechanical reliability which relates to the operability of equipment, i.e., obviously, the utility wishes to install equipment that is not subject to frequent breakdowns. Second, there is hydraulic reliability, i.e., the utility wishes to consistently provide the specified amounts of its product (the finished water) at the required pressures, and at the time and location desired.

2.3. MECHANICAL RELIABILITY

Mechanical reliability is the ability of distribution system components to provide continuing and long-term operation without the need for frequent repairs,

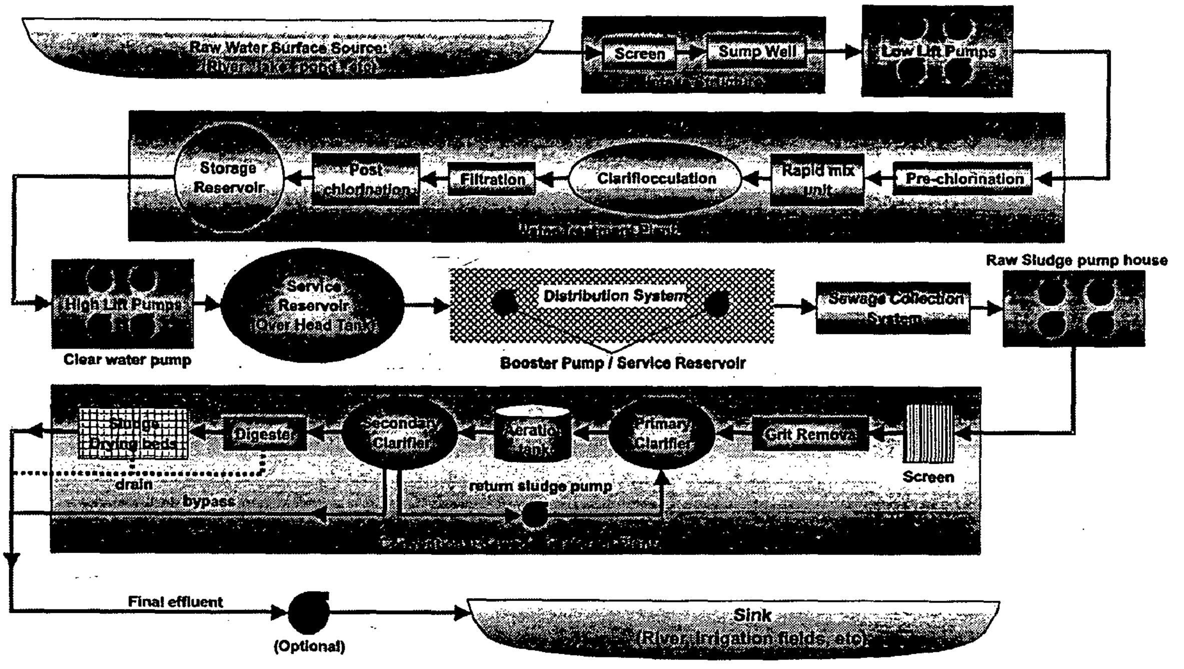


Fig. 2.1. Components of Typical Water Supply Scheme

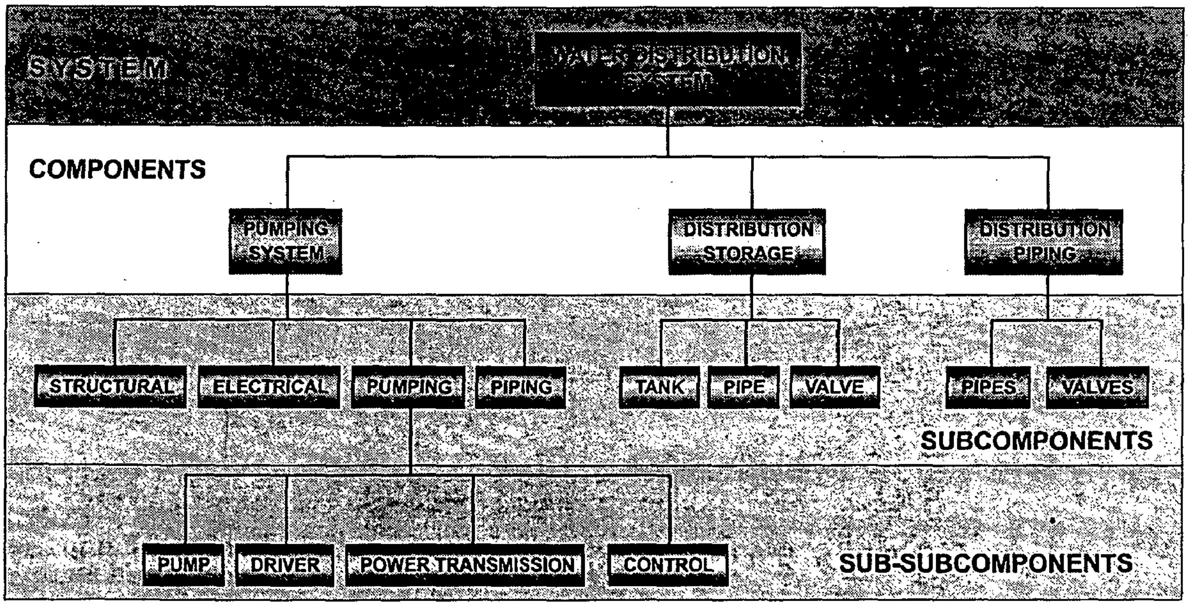


Fig. 2.2. Hierarchical Relationship of System for a Water Distribution System

modifications, or replacement of components or subcomponents. Mechanical reliability is usually defined as the probability that a component or subcomponent performs its mission within specified limits for a given period of time in a specified environment (Kaufmann et al., 1977). When quantified, mechanical reliability is merely an expression of the probability that a piece of equipment is operational at any given time. The mathematical evaluation of mechanical reliability is well developed and has been used extensively in the analysis of mechanical and electrical systems. It has proven especially useful in the analysis of aerospace systems.

As mentioned above, the water distribution system is composed of three major components:

- ✓ Pumping Stations
- ✓ Distribution Storage
- ✓ Distribution Piping

The pumping station subcomponents as mentioned earlier consist of:

- ✓ Structural
- ✓ Electrical
- ✓ Piping
- ✓ And Pumping Unit Subcomponents are;
  - o Driver
  - o Controls
  - o Power transmission
  - o Piping and valves

Mathematically, the reliability R.(t) of a component is defined as the probability that the component experiences no failures during the time interval from time zero to time t, given that it is new or repaired at time zero. Following equation used to define the reliability as

$$R(t) = \int_t^{\infty} f(t)dt$$

where f(t) is the probability density function of the time to failure of the component. The probability density function is either assumed or developed from equipment failure data, using various statistical data.

The concept of reliability is suitable for the evaluation of non-repairable components; however, for repairable components such as those most often found in water distribution systems, it is much more appropriate to use the concept of availability. Whereas the reliability is the probability that the component experiences no failures during the interval from time zero to time t, the availability of a component is the probability that the component is in operational condition at time t, given that the component was as good as new at time zero. The reliability generally differs from the availability because reliability requires the continuation of the operational state over the whole time interval. Subcomponents and sub-subcomponents contribute to component availability but not to the reliability of the component if the subcomponent or sub-subcomponent that failed before time t is repaired and is operational by time t. For example, since the driver on a pumping subcomponent is repairable, the driver may fail and be repaired a number of times between time 0 and time t yet still contribute to the availability at time t. As a result, the availability is always equal to or greater than the reliability. Figure 2.3 illustrates that the availability of a non-repairable component decreases to zero as t becomes larger, whereas the availability of a repairable component converges to a nonzero positive number.

The availability of a component can be expressed as the percentage of time that the component is in an operational state. Classically, for mechanical systems time can be divided into two categories: operational time and non-operational time. Operational time is that time in which the component performs satisfactorily. Repair or maintenance time results from component failure or scheduled maintenance requiring component shutdown. The component is non-operational during repair time. Repair time is triggered by the occurrence of a maintenance event, which is defined, as those actions required either returning or keeping the component in working order. The maintenance event has two important attributes: the time between maintenance events and the duration of the maintenance event. The time between maintenance events is referred to as the time to first failure or the time between failures and where data is lacking, is usually described by the exponential or lognormal probability distribution. The duration of the maintenance event is referred to as the repair time and is usually described by the lognormal probability distribution. These distributions are selected more for the ease with which they can be mathematically manipulated rather than their true representation of the time between failures and the length of the repair time.

There are two basic categories of maintenance event; corrective maintenance and preventive maintenance.

(a) Corrective maintenance is defined as the activity of repair after a breakdown has taken place or as unscheduled maintenance due to equipment failure.

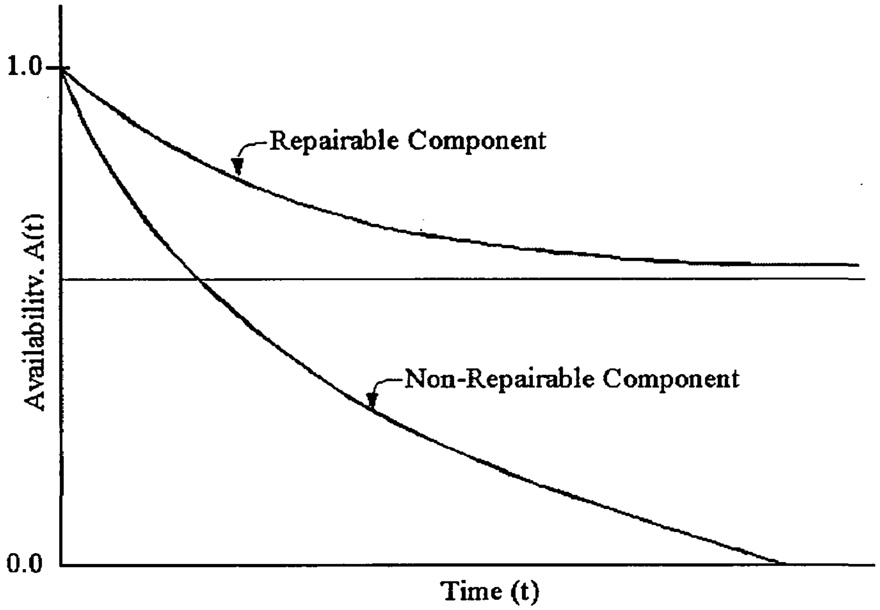


Figure 2.3. Availability for Repairable and Non-repairable Components

Corrective maintenance activities have four time periods that contribute to the unavailability of the component: time between the failure and the recognition of failure (initial response time), time awaiting repair materials, time awaiting manpower, and active repair time. The initial response time, time awaiting materials, and time awaiting manpower are utility and maintenance event specific and are often neglected in the evaluation of reliability. Active repair time is the time required to disassemble, correct the deficiency, reassemble, and return the failed equipment to an operational state. The corrective maintenance time (CMT) is the time (usually expressed in hours/ year) that a component is non-operational because of corrective maintenance activities.

(b) Preventive maintenance has a variety of potential meanings. In its most limited form, preventive maintenance is merely the inspection of equipment to prevent breakdowns before they occur. A broader definition includes activities such as repetitive servicing, and upkeep, (e.g. lubrication, painting, and cleaning). Preventive maintenance may also be called planned or routine maintenance. Since preventive maintenance is scheduled, only the active repair time contributes to the unavailability of the component. It should be noted that not all preventive maintenance activities result in component unavailability. The preventive maintenance time (PMT) is the time (usually expressed as hours/year) that a component is non-operational because of preventive maintenance activities.

Component availability can be expressed mathematically as the fraction of clock time that the component is operational, i.e., available for service. On an annual basis, this can be calculated using the following equation

$$A = \frac{(8760 - CMT - PMT)}{8760}$$

where

A = availability, fraction
CMT = corrective maintenance time, hr/yr
PMT = preventive maintenance time, hr/yr
8760 = hours in a year

Typically, mean values are used for the corrective and preventive maintenance times.

Accurate calculation of the mechanical reliability requires knowledge of the precise reliability of the basic components and the impact on mission accomplishment caused by the set of all possible failures. Thus, for a large system with many interactive components, such as a water distribution system, it is extremely difficult to analytically compute the mechanical reliability. There is no comprehensive database of failure and repair information for components and subcomponents for water distribution systems. The limited data that has been collected is summarized in Table 2.1 (in reference May L.W. [15]).

Table 2.1 Typical Failure and Repair Data for Components Other than Pipes

Subcomponent	Mean Time Between Failure	Mean Time To Repair	Preventive Maintenance	Availability
	(hrs x 10^6)	(hrs)	(hrs/yr)	
Pumps	.032066	9.6	2.	.99116
Power Transmission	.035620	2.3	7.	.99898
Motors	.066700	6.9	14.	.99816
Valves	.014440	11.6	41.	.96446
Controls	.083580	3.7	9.	.99870

2.4. MECHANICAL RELIABILITY OF PUMPING STATIONS

Pumping stations are one of the major components of a water supply system. A pumping station consists of one or more pumping units supported by appropriate electrical, piping, control, and structural components. A typical pumping station is illustrated schematically in Fig. 2.4. The evaluation of pumping station reliability is a multistep process incorporating the analysis of subcomponent and component reliability. The mechanical reliability evaluation can be conducted to any desired level of detail for which data are available. The pumping unit is the primary subcomponent within the pumping station and includes four major sub-subcomponents: pump, driver (motor, engine, etc.), power transmission, and controls. Although there are more sub-subcomponents in the pumping unit, these are selected as the level of evaluation because of the availability of failure and repair data. Each of these sub-subcomponents must function properly if the pumping unit is to be considered in an operational state.

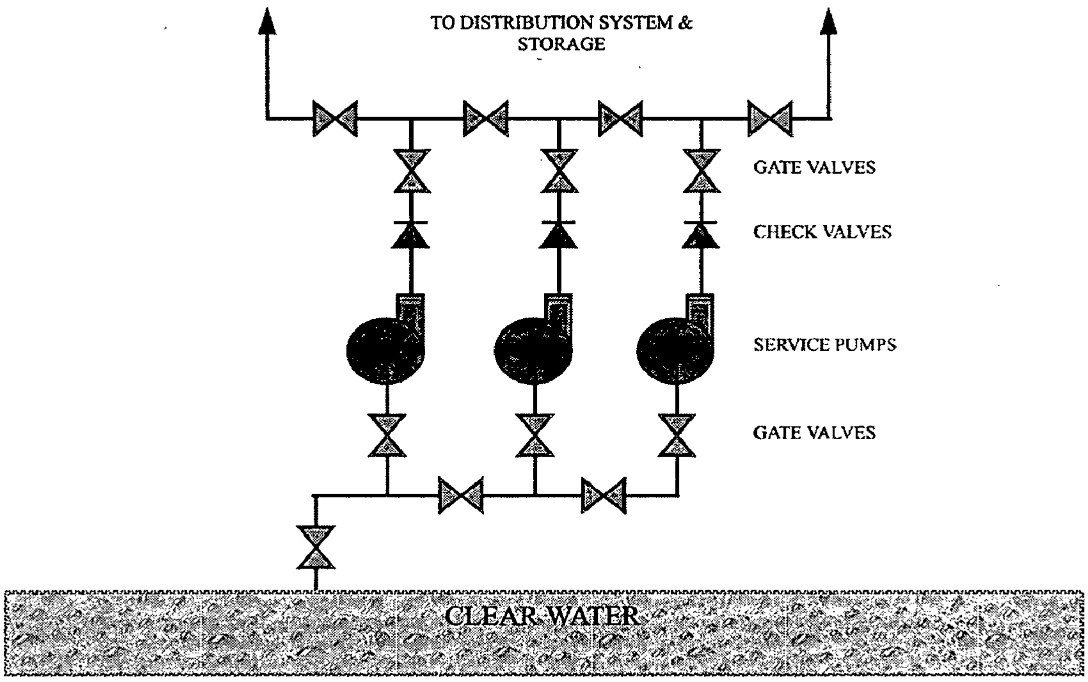


Figure 2.4 : Schematic of Typical Pumping System

A pumping station may be evaluated as consisting of four major subcomponents: pumping, electrical, piping, and structural. The major mechanical system is the pumping system consisting of one or more pumping units. For pumping units, the reliability will be measured in terms of the operational availability. Using the data for subcomponent reliability presented in table 2.1, the reliability of each pumping unit in the pump station can be easily estimated if the following assumptions are made.

(a) The pumping unit consists of the following sub-subcomponents: pump, driver, controls, and power transmission.
(b) Each sub-subcomponent must be operational for the pumping unit to be operational.
(c) The failure of each sub-subcomponent is independent of the failure of the other sub-subcomponents.
(d) The unavailability of the subsystems does not intersect, i.e., only one subsystem fails at a time.

Under the above assumptions, the pumping unit can be evaluated as a simple series problem and the availability of the individual pumping units can be calculated as follows:

$$AS = AOP \bullet AOM \bullet AOC \bullet AOPT$$

where

A S = availability of the pumping unit,
AOP = availability of the pump,
AOM = availability of the driver,
AOC = availability of the control system, and
AOPT = availability of the power transmission

Thus, the mechanical availability of the pumping unit can be calculated as follows (see Table 2.1 for availability values)

$$AS = (0.99116)\,(0.99816)\,(0.99870)\,(0.99898)$$

$$= (0.98704)$$

The mechanical reliability of the entire pumping station is difficult to evaluate because of the difficulties involved in defining failure. Does the pump station fail because of the failure of one pumping unit, etc? Probably not. Sophisticated techniques such as fault tree analysis can be used to perform detailed evaluations of pumping station reliability (Henley and Kumamoto, 1981). In addition, detailed failure and repair data are seldom available for input to these complex computational methods. Simplified techniques based on common probability concepts and a review of the pumping station design assumptions can be used to estimate the mechanical reliability of the pumping station. This simplified technique can best be illustrated through use of an example computation.

A common design practice is to install sufficient pumps to handle peak flows and include a spare pump of equal size to accommodate any downtime of the other pumps. Thus, mechanical failure of the pumping station could be defined, as the simultaneous failure of two or more pumping units while peak capacity is required. The individual pumping units have two possible operational states: available and unavailable. By treating the pumping station as a k-out-of-n reliability system, and letting AS represent the fraction of time that the pumping unit is available and US the fraction of time that the pumping unit is unavailable (note that US + AS = 1), the probability of the simultaneous failure of pumps can be calculated as follows

$$PAS=\sum_{i=0}^{n}\binom{n}{i}AS^{i}(US)^{n-i}$$

where

PAS = availability of the pumping station,

n = number of pumping units,

i = number of operational pumping units,

AS = availability of a pumping unit, and

US = unavailability (1-AS) of a pumping unit.

And

$$\binom{n}{i}=\frac{n!}{i!(n-i)!}$$

The binomial equation presented above can be expanded for range I=0 to n and rearranged as follows to provide information on the proportion of time that the pumping station will be at a specific capacity state.

$$\binom{n}{0}AS^{n}US^{0}+\binom{n}{1}AS^{n-1}US^{1}+\binom{n}{2}AS^{n-2}US^{2}+\ldots+\binom{n}{n}AS^{0}US^{n}$$

For example, in this expansion, the term

$$\binom{n}{2} AS^{n-2}US^{2}$$

yields the percentage of time that n-2 pumps are operational.

For a pumping station with three pumping units, with availability as previously calculated, the following result is obtained (table 2.2).

Table 2.2 Typical Availability Data vs. Number of Pumps

Number of Operational Pumps	Availability
3	0.993653
2	0.006333
1	0.000013
0	0.000001

Availability is the fraction of time that the respective numbers of pumps are available for service

2.5. HYDRAULIC RELIABILITY OF PUMPING STATIONS

Pumping stations may be designed to meet several demand scenarios ranging from the maximum hourly demand to the average daily demand. The hydraulic design or design flow of a specific pumping station will usually be based on the relationship between water demand and the availability of distribution storage. Once the design flow has been established, hydraulic failure of the pumping station can be defined in terms of the fraction of calendar time that the pumping station can meet this demand. The calculation of hydraulic reliability can be illustrated through use of an example computation.

Beginning with the design assumptions presented, the capacity of a pumping station can be calculated from the station pump curves. Fig 2.5 shows typical capacity-head curve for three pumps operating in parallel (for more detail explanation see Para 3.2.2.24 in Chapter 3).

For any pumping station, there are n + 1 possible capacity states, i.e., n + 1 station pumping capacities based on the number of pumping units in an operational state and these capacity states obtained from the pumping station capacity-head curve for each combination of available pumping units. Thus, for the three pumping unit example, the following capacity state-time proportion cases can be calculated as indicated in Table 2.3 below on the basis of availability in Table 2.2.

Table 2.3. Typical Availability vs. Capacity of Pumping Station

Pumping Station Capacity (discharge)	Time Proportion	Availability (hours/yr)
5200	0.993653	8704.40
4000	0.006333	55.48
2300	0.000013	0.11
0	0.000001	0.01
	1.000000	8760.00

Defining failure as the fraction of time that the pump station cannot deliver the design flow, and assuming the typical design practice of providing a spare pump, the availability of satisfying pumping capacity is 0.999986.

Note that under the assumption of equal pumping unit capacities and availability, the hydraulic and mechanical reliability of the pumping station are identical. This condition would not occur if the pumping station used pumping units of different capacities or reliabilities. It should also be noted that system elevated storage provides backup for pumping station failures.

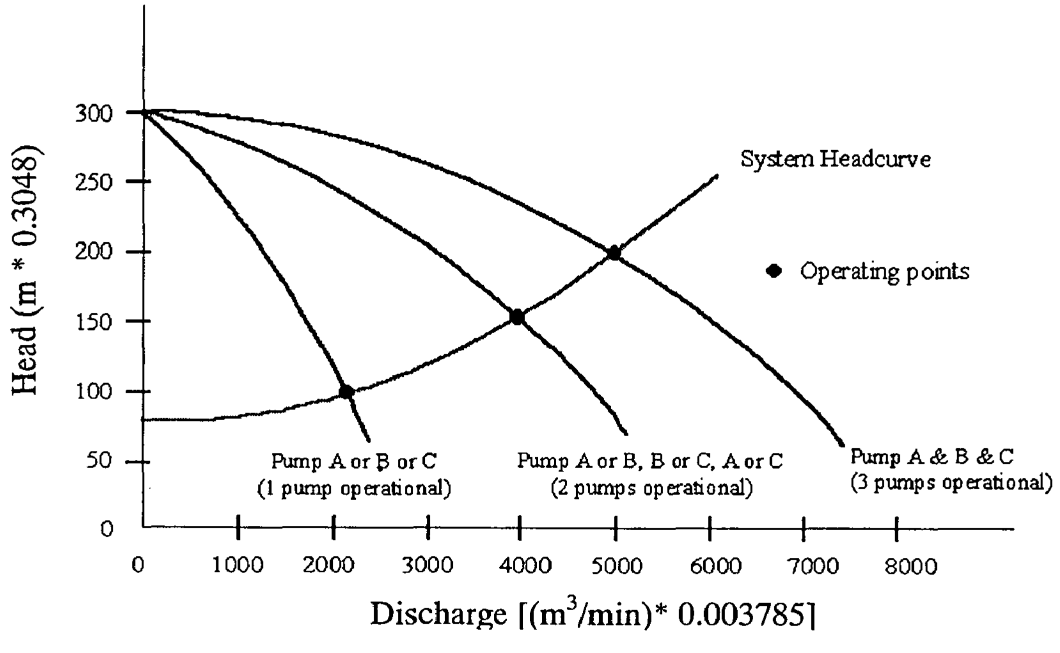


Figure 2.5. Typical Pump Station Operating Curves

CHAPTER

3

Type of Pumps and Its Characteristics

CHAPTER-3

TYPE OF PUMPS AND ITS CHARACTERISTICS

3.1. TYPES OF PUMPS

The classification of pumps as shown in fig. 3.1 first defines the principle by which energy is added to the fluid, then identifies the means by which this principle is implemented, and finally, distinguishes among specific geometries commonly used. Under this system of classification, all pumps may be divided into two major categories:

1) dynamic pumps, where continuously added energy increases velocity of the fluid and later this velocity is changed to pressure, and
2) displacement pumps where periodically added energy directly increases pressure.

Displacement pumps come in many designs and operating ranges, but they all work on the *same principle.* An increasing volume is opened to suction, filled, closed, moved to discharge, and displaced. The delivered capacity is nearly constant throughout the discharge pressure range. This constant capacity will intersect a system curve at a defined point, allowing a high degree of system control.

Displacement pumps differentiate into rotary and reciprocating pumps. Rotary pumps are defined as being: *vane, piston, flexible member, lobe, gear circumferential piston, or screw* pumps. In all of the rotary designs, the chamber is created progressively through rotation of the drive shaft. There may be one or more chambers opened per revolution depending on the design. The chambers are sealed off from suction by close clearance between the rotor and the housing, or by close clearance between intermeshing rotors. Rotation of the shaft moves the chamber along the bore or housing towards discharge. The chamber is displaced to discharge by rotation. The release to discharge progresses with rotation as the volume is expelled so that the flow is typically pulsation free.

Reciprocating pumps are defined as being; *steam, power, or controlled* volume pumps. In all reciprocating pumps, there are check valves on the suction and discharge. Fluid flows through the suction valve and into the chamber as the plunger, piston, or diaphragm recedes. At the end of the stroke, the chamber is at its maximum size. The suction valve closes, the plunger moves forward into the chamber, forcing the fluid out the discharge valve. The flow from each chamber is a pulse flow. If the pump has several chambers, they are timed to have sequential pulses to minimize the overall pulsation.

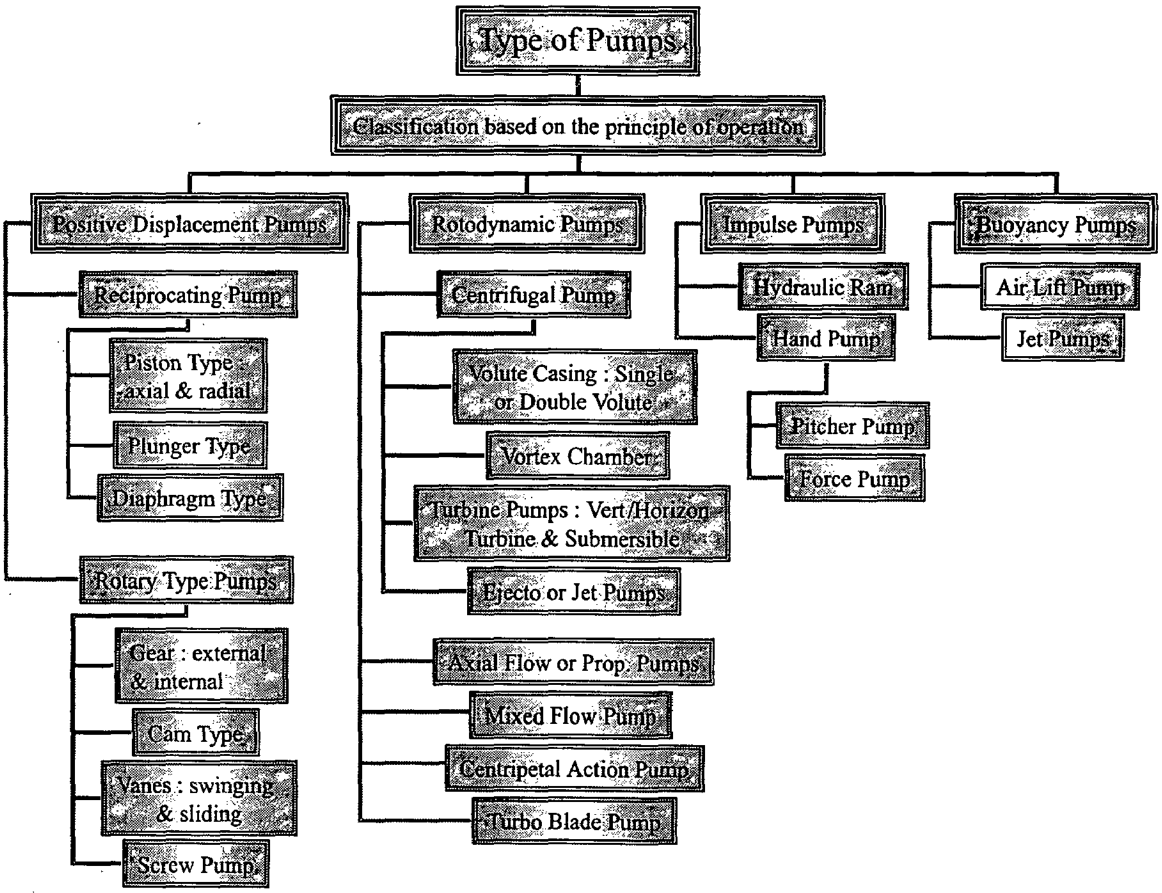


Fig. 3.1. Classification of pump based on the principle of operation

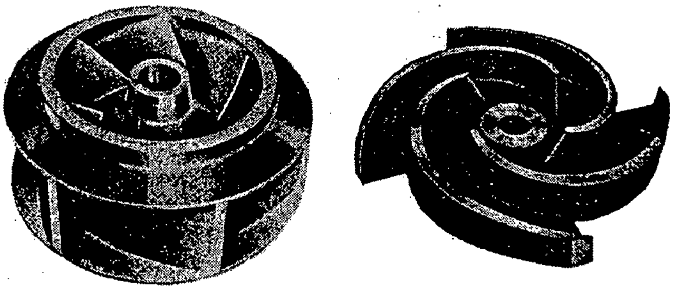
Dynamic pumps (centrifugal) are commonly used for pumping water in agricultural applications such as irrigation and drainage. Displacement pumps have limited capacities and are not suitable for pumping large amounts of water required for irrigation or drainage. They are used mainly for chemical injection in agricultural irrigation systems. So in this study centrifugal pump as mainly discussed. The centrifugal pumps are classified by whether the water leaves the impeller in the radial or axial direction with respect to the axis around which the pump rotates. In the range of heads in most water-distribution systems, radial-flow pumps are most appropriate. Axial-flow pumps find use in high-flow, low-head applications.

3.2. CENTRIFUGAL PUMPS

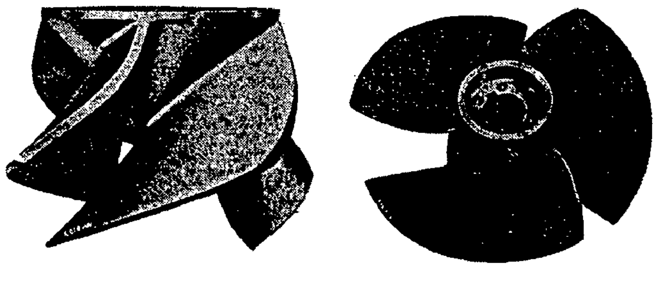
There are many different types of pumps, but in almost all cases pumps in water-distribution are *centrifugal* pumps because of their low cost, simplicity, and reliability in range of flows and head encountered. As a result, the discussion on pumps in this study is restricted to centrifugal pumps. A centrifugal pump is any pump in which fluid is energized by a rotating impeller, whether the flow is radial, axial, or a combination of both (mixed).

3.2.1. DESCRIPTION OF CENTRIFUGAL PUMPS

A centrifugal pump consists of a revolving wheel called impeller which is enclosed in an air tight casing to which suction and delivery pipes are connected. The impeller is provided with a series of backward curved blades or vanes. It is mounted on a shaft which is coupled to an external source of energy (usually an electric motor) which imparts the required energy to the impeller thereby making it to rotate.



Closed Radial   Open Radial



Mixed Flow   Propellers

Fig. 3.2 Types of Pump Impellers (isometric views)

The impellers may be classified as (1) closed or shrouded impeller; and (2) open impeller, (isometric views which are shown in fig. 3.2). A closed or shrouded impeller, is that whose vanes are provided with metal cover plates on both sides. These plates or shrouds are known as crown plate and lower or base plate. The closed impeller provides better guidance for the liquid and it is more efficient. However, this type of impeller is most suited when the liquids to be pumped are pure and comparatively free from debris. If the vanes have only the base plate and no crown plate, then the impeller is known as semi-open impeller. Such an impeller is suitable even if the liquids are charged with some debris. An open impeller is that whose vanes have neither the crown plate nor the base plate. Such impellers are useful in the pumping of liquids containing suspended solid matter, such as paper pulp, sewage and water containing sand or grit. These impellers are less liable to clog when handling liquids charged with a large quantity of debris.

At the inlet to the pump at the center of the casing a suction pipe is connected, the lower end of which dips into the liquid in the tank or sump from which the liquid is to be pumped or lifted up. The lower end of the suction pipe is fitted with a foot valve and strainer. The liquid first enters the strainer which is provided in order to keep the debris (such as leaves, wooden pieces and other rubbish) away from the pump. It then passes through the foot valve to enter the suction pipe. A foot valve is a non-return or one-way type of valve which opens only in the upward direction. As such the liquid will pass through the foot valve only upwards and it will not allow the liquid to move downwards back to the sump. At the outlet of the pump a delivery pipe is connected which delivers the liquid to the required height. Just near the outlet of the pump on the delivery pipe a delivery valve is provided. A delivery valve is a regulating valve which is provided in order to control the flow from the pump into the delivery pipe.

3.2.2. WORKING OF CENTRIFUGAL PUMP

A centrifugal pump is one of the simplest pieces of equipment in any process plant. Its purpose is to convert energy of a prime mover (a electric motor or turbine) first into velocity or kinetic energy and then into pressure energy of a fluid that is being pumped. The energy changes occur by virtue of two main parts of the pump, the impeller and the volute or diffuser. The impeller is the rotating part that converts driver energy into the kinetic energy. The volute or diffuser is the stationary part that converts the kinetic energy into pressure energy.

Note: All of the forms of energy involved in a liquid flow system are expressed in terms of meter of liquid i.e. head.

Generation of Centrifugal Force

The process liquid enters the suction nozzle and then into eye (center) of a revolving device known as an impeller. When the impeller rotates, it spins the liquid sitting in the cavities between the vanes outward and provides centrifugal acceleration. As liquid leaves the eye of the impeller a low-pressure area is created causing more liquid to flow toward the inlet. Because the impeller blades are curved, the fluid is pushed in a tangential and radial direction by the centrifugal force. This force acting inside the pump is the same one that keeps water inside a bucket that is rotating at the end of a string. Figure 3.3. below depicts a side cross-section of a centrifugal pump indicating various components through which the movement of the liquid occurs.

Conversion of Kinetic Energy to Pressure Energy

The key idea is that the energy created by the centrifugal force is *kinetic energy*. The amount of energy given to the liquid is proportional to the *velocity* at the edge or vane tip of the impeller. The faster the impeller revolves or the bigger the impeller is, then the higher will be the velocity of the liquid at the vane tip and the greater the energy imparted to the liquid.

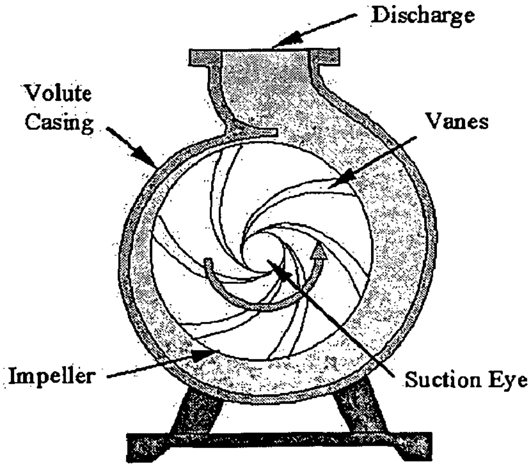


Fig. 3.3 Liquid flow path inside a centrifugal pump

This kinetic energy of a liquid coming out of an impeller is harnessed by creating a *resistance* to the flow. The first resistance is created by the pump volute (casing) that catches the liquid and slows it down. In the discharge nozzle, the liquid further decelerates and its velocity is converted to pressure according to Bernoulli's principle.

Therefore, the head (pressure in terms of height of liquid) developed is approximately equal to the velocity energy at the periphery of the impeller.

3.2.3. TYPES OF CENTRIFUGAL PUMPS

The centrifugal pumps are classified on the basis of the various characteristic features possessed by them as indicated below.

1. Relative direction of flow of liquid through impeller.
2. Types of casing.
3. Number of impellers per shaft.
4. Number of entrances to the impeller.
5. Disposition of shaft.
6. Working head.

3.2.3.1. RELATIVE DIRECTION OF FLOW OF LIQUID THROUGH IMPELLER.

On the basis of direction of flow of liquid through impeller the centrifugal pumps may be classified as radial flow pumps, mixed flow pumps and axial flow pumps.

3.2.3.1.A. RADIAL FLOW PUMP

The direction of flow for these machines is radial from the shaft of the machine (fig. 3.4). Flow enters the pump close to the shaft and the rotation of the impeller forces the flow outwards (centrifugal action) and into the volute at high velocity. The velocity is decreased and the head increased as flow expands in the volute. The large velocity change and inefficiencies in the expansion section reduces the overall efficiency of these units. For most applications an easily manufactured single stage unit will be employed. This type of unit is commonly used in water supply systems. Basically, a centrifugal radial-flow pump has two main parts: (1) a rotating element (impeller and shaft) and (2) a stationary element (casing, stuffing box and bearings). Radial flow impellers are sometimes arranged so that water can enter the *eye* of the impeller from both sides. Such an impeller is called a *double-suction impeller.* It has the same effect as *two single-suction* impellers placed back to back and results in doubling the capacity without increasing impeller diameter.

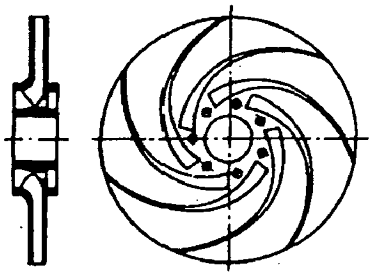
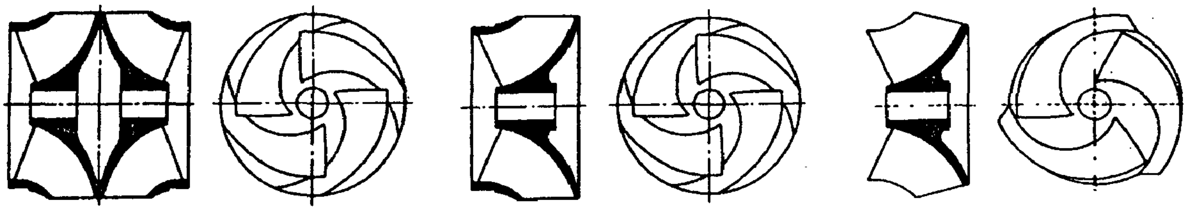


Fig. 3.4 Radial-flow Impellers

3.2.3.1.B. MIXED FLOW PUMP

Mixed-flow centrifugal pumps use both centrifugal force and some lifting action to move water (fig. 3.5). Water is discharged both radially and axially into a volute-type casing. The process is a combination of processes occurring in volute and axial-flow types of pumps. Mixed-flow impellers are often used in deep-well turbine and submersible turbine pumps. As such mixed flow pumps are generally used where a large quantity of liquid is to be discharged to low heights. Some mixed flow impellers look like screw and are known as screw impellers. Mixed flow pumps borrow characteristics from both radial flow and axial flow pumps. As liquid flows through the impeller of a mixed flow pump, the impeller blades push the liquid out away from the pump shaft and the pump suction at an angle greater than 90°.



Mixed-flow closed, double entry Mixed-flow impeller, closed Mixed-flow impeller, open

Fig. 3.5 Types of Impeller (plan and side views)

3.2.3.1.C. AXIAL FLOW PUMP

The flow of liquid through the impeller is in the axial direction only (fig 3.6). The direction of flow through these machines is parallel to the pump shaft. The action of the impeller is similar to a propeller, with relatively small changes in velocity through the pump. These machines designed to deliver very large quantities of liquid at relatively low heads and having high efficiencies. However, it is not justified to call axial flow pumps as centrifugal pumps, because there is hardly any centrifugal action in their operation.

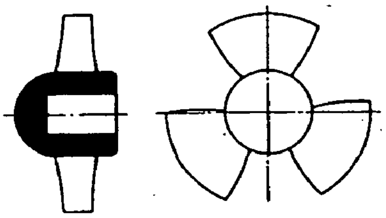


Fig. 3.6 Axial-flow Impeller

3.2.3.2. TYPES OF CASING.

According to the type of casing provided, the centrifugal pumps are classified as volute pumps, diffuser or turbine pumps and volute pumps with vortex chamber.

3.2.3.2.A. VOLUTE PUMP

In a *volute pump* the impeller is surrounded by a spiral shaped casing which is known as volute chamber. As shown in fig. 3.7. the shape of the casing is such that the area of flow section around the periphery of the impeller gradually increases from the tongue **T** towards the delivery pipe. A volute casing, the most common type, is designed to produce an equal velocity of flow around the circumference of the impeller and to reduce the velocity of the water as it enters the discharge pipe, thus creating the required pressure head.

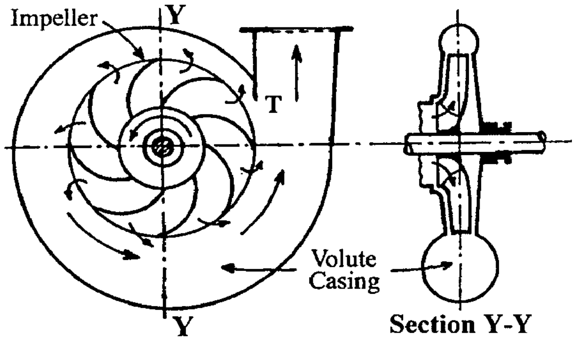


Fig. 3.7 Volute pump

3.2.3.2.B. DIFFUSER OR TURBINE PUMP

In a *diffuser* or *turbine pump,* the impeller is surrounded by a series of guide vanes mounted on a ring called diffuser ring as shown in fig. 3.8. The diffuser ring and the guide vanes are fixed in position. The adjacent guide vanes provide gradually enlarged passages for the flow of liquid. The liquid after leaving the impeller passes through these passages of increasing area, wherein the kinetic energy is converted into pressure energy, which results in a higher efficiency of the pump. After passing through the guide vanes the liquid flows into. the surrounding casing which in this case is generally circular and concentric with the impeller.

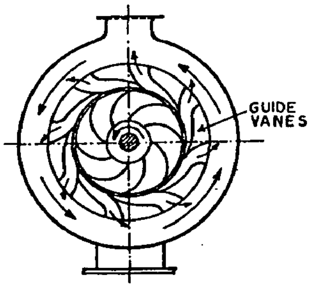


Fig. 3.8 Diffuser (or turbine) pump.

3.2.3.2.C. THE VORTEX CHAMBER

In the turbine-type pump the impeller is surrounded by stationary guide vanes which reduce the velocity of the water and convert velocity head to pressure head in the casing itself. The casing surrounding the guide vanes is usually circular and concentric with the impeller. The volute casing is so designed that considerable part of kinetic energy of the liquid leaving the impeller is converted into pressure energy, resulting in higher efficiency than is possible in a volute pump casing, because as the flow progresses from the tongue T towards the delivery pipe, more and more liquid is added to the stream from the periphery of the impeller.

However, in actual practice it has been found that in volute type of casing there is only a slight increase in the efficiency of the pump, because a considerable loss of energy takes place in eddies developed in the casing. As such a subsequent improvement over the simple volute casing was made by providing a circular chamber between the impeller and the volute chamber as shown in fig. 3.9. The circular chamber is known as *vortex* or *whirlpool chamber* and such a pump is known as *volute* pump *with vortex chamber.*

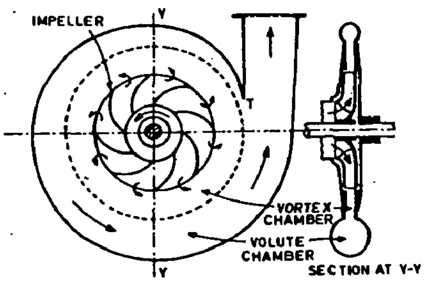


Fig. 3.9 Volute pump with vortex chamber.

3.2.3.3. NUMBER OF IMPELLERS PER SHAFT.

According to the number of impellers provided the centrifugal pumps may be classified as single stage pumps and multi-stage pumps.

MULTI STAGE PUMP

The multi-stage centrifugal pumps are used for high lifts. A *multi-stage pump* consists of two or more identical impellers mounted on the same shaft, and enclosed in the same casing. All the impellers are connected in series, so that liquid discharged with increased pressure from one impeller passes through the connecting passages to the inlet of the next impeller and so on, till the discharge from the last impeller passes into the delivery pipe. The impellers are surrounded by guide vanes which are generally provided within the connecting passages, and are meant for the recuperation of the velocity energy of the liquid leaving the impeller into pressure energy. According to the number of impellers fitted in the casing a multi-stage pump is designated as two-stage, three-stage, etc. Fig. 3.10., shows a three stage centrifugal pump.

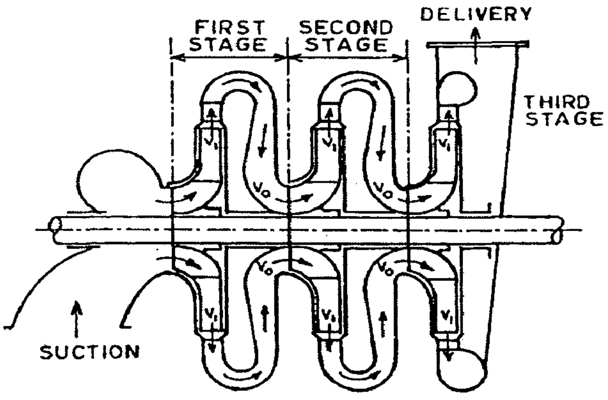
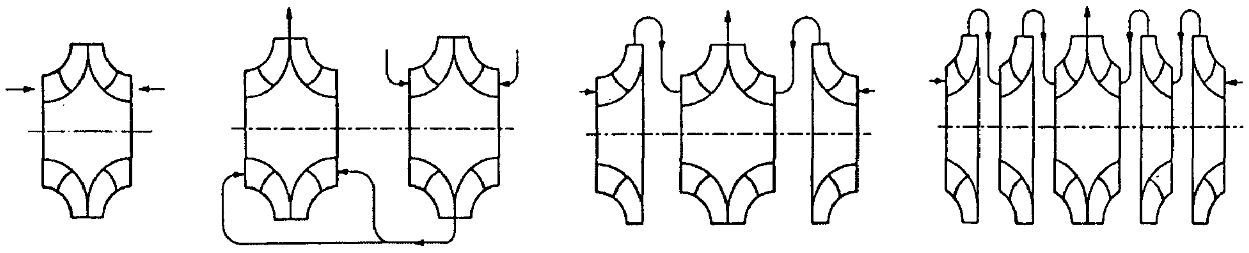


Fig. 3.10 Three-stage centrifugal pump.

3.2.3.4. NUMBER OF ENTRANCES TO THE IMPELLER.

SINGLE AND DOUBLE SUCTION

Depending on the number of entrances to the impeller the centrifugal pumps may be classified as single suction pump and double suction pump. In a *single suction (or entry) pump* liquid is admitted from a suction pipe on one side of the impeller. In a *double suction (or entry) pump* liquid enters from both sides of the impeller. A double suction pump is suitable for pumping large quantity of liquid since it provides a large inlet area.



Double-Entry, Single Stage

Double-entry, Two Stage

Double-entry, Two Stage

Double-entry, Three Stage

Fig. 3.11 Single and Multi Stage Pump Impeller Arrangement

3.2.3.5. DISPOSITION OF SHAFT

VERTICAL AND HORIZONTAL SHAFT

The centrifugal pumps may be designed with either horizontal or vertical disposition of shafts. Generally for low lifts the centrifugal pumps with horizontal shafts are adopted. However, for deep tube wells the centrifugal pumps with vertical shafts are invariably used.

3.2.3.6. WORKING HEAD.

LOW AND HIGH HEAD

According to the head developed, the centrifugal pumps may be classified as low head (or low lift) pumps, medium head (or medium lift) pumps, and high head (or high lift) pumps. *A low head (or low lift) pump is* the one which is capable of working against a total head up to 15 m. A *medium head (or medium lift) pump* is that which is capable of working against a total head more than 15 m but up to 40 m. *A high head (or high lift)* pump is the one which is capable of working against a total head above 40 m. The high head (or high lift) pumps are generally multi-stage pumps. The head produced by a centrifugal pump depends on the rim speed of the impeller. To increase the rim speed, either the rotative speed or diameter of the impeller or both must be increased. Increasing either of these has the effect of increasing the stress in the impeller material. For this reason it is usually not possible to produce very high head with one impeller. Normally a pump with a single impeller can be used to deliver the required discharge against a maximum head of about 100 m. But if the liquid is required to be delivered against a still larger head then it can be done by a multi-stage pump.

3.2.4. PERFORMANCE OF CENTRIFUGAL PUMPS

The purpose of this section is to introduce those fundamentals of centrifugal pump theory that are useful for background and sometimes necessary for selection and specifying centrifugal pumps in water and wastewater pumping applications.

CAPACITY

The capacity (flowrate, discharge or Q) of a pump is the volume of liquid pumped per unit of time, usually measured in SI units in cubic meters per second for large pump or liters per second and cubic meters per hour for small pumps.

HEAD

The term head (h or H) is the elevation of a free surface of water above (or below) a reference datum (see fig. 3.12. and 3.13). For centrifugal pump, the reference datum varies with the type of pump as shown in fig. 3.14.

In accordance with the standards of the Hydraulic Institute, distances (heads) above datum are considered positive and distances below the datum are considered negative. Each term, defined graphically in fig. 3.12. and 3.13., is expressed as the height of a water column in meters (feet) of water. H is used for total head, whereas h is used for head from the datum or for head-loss. The subscript s and d denote the pump suction and discharge, respectively. Other subscripts are defined as follows:

- ***Total static head (H_{stat}):*** The total static head is the difference in elevation in meters (feet) between the water level in the wet well and the water level at discharge (h_d - h_s).
- ***Static suction head (h_s):*** The static suction head is the difference in elevation between the wet well liquid level and the datum elevation of the pump impeller. If the wet well liquid level is below the pump datum, as in fig. 3.13., it is a static suction lift, so h_s is negative.
- ***Static discharge head (h_d):*** The static discharge head is the difference in elevation between the discharge liquid level and the pump datum elevation.

- ***Manometric suction head (h_{gs}):*** The suction gauge reading is expressed in meters (feet) measured at the suction nozzle of the pump and referenced to the pump datum elevation and atmospheric pressure.

- ***Manometric discharge head (h_{gd}):*** The discharge gauge reading is expressed in meters (feet) measured at the discharge nozzle of the pump and referenced to the centerline of the pump impeller. The gauge reading is the height that a water column would attain in a vertical pipe. It is also the distance to the hydraulic grade-line.

- ***Manometric head (H_g):*** This is the increase of pressure head, expressed in meters (feet) generated by the pump ($h_{gd} - h_{gs}$).

- ***Friction headloss (h_{fs} , h_{hd}):*** This is the head of water that must be supplied to overcome the frictional loss in the pipe. The frictional head-loss in the suction (h_{fs}) and discharge (h_{fd}) piping systems can be computed with the Hazen-Williams or Darcy-Weisbach equations (Equation, 3-9 and 3-13).

- ***Velocity head ($v^2/2g$):*** The velocity head is the kinetic energy in the liquid being pumped at any point in the system. The energy grade-line (shown solid in fig. 3.12. and 3.13.) is always above hydraulic or piezometric or manometric grade-line (dashed line) by $v^2/2g$. The velocity head in the discharge pipe, $v_d^2/2g$, is lost if the pipe discharges freely in air or if it discharges abruptly below the surface of a reservoir. Some of the velocity head can be recovered if turbulence is inhibited by a gradually expanding section, but this is ordinarily impractical with the pipe velocities normally encountered (i.e., up to 2.5 m/s or 8 ft/s).

- ***Fitting and valve losses (h_{fvs} , h_{fvd}):*** As a fluid flows through fittings and valves, energy is lost due to eddy formation and turbulence. Because the head lost in fittings and valves is small compared with the friction loss in long piping systems, the losses in fittings and valves are sometimes called "minor losses" and often ignored. But the lengths of pipe, within a pumping station are short, and the total headloss through the fittings and valves is likely to be greater than the pipe friction loss. Regardless of the shortness of pipe length, both the frictional headloss and the "minor losses" should always he computed. The loss of head through each individual fitting or valve is estimated by using Equation 3-13 *($h=Kv^2/2g$)* with K values taken from tables A1 in Appendix-A or from literature.

- ***Total dynamic head (H_t or TDH)*:** is the head against which the pump must work. It is determined by adding the static suction and discharge head (with respect to signs), the frictional headloss, the velocity heads, and the fittings and valve headlosses. The expression for determining the total dynamic head for the pumps show in Fig. 3.12. and 3.13. is given by Equations 3-1, 3-2, and 3-3.

$$H_t = h_{gd} - h_{gs} + \frac{V_d^2}{2g} - \frac{V_s^2}{2g} \tag{3-1}$$

where

$$h_{gd} = h_d + h_{fd} + \sum h_{fvd} \tag{3-2}$$

and $$h_{gs} = h_s - h_{ent} - h_{fs} - \sum h_{frs} - \frac{V_s^2}{2g} \tag{3-3}$$

Substituting Equation, 3-2 and 3-3 into Equation 3-1 and noting that $h_d - h_s = H_{stat}$

$$H_t = H_{stat} + h_{ent} + h_{fs} + h_{fd} + \sum h_{frs} + \sum h_{fvd} + \frac{V_d^2}{2g} \tag{3-4}$$

Some designers consider all headlosses except pipe friction to be "minor" losses, h_m, and rewrite Equation 3-4 as

$$H_t = H_{stat} + h_{fs} + h_{fd} + \Sigma h_m \tag{3-5}$$

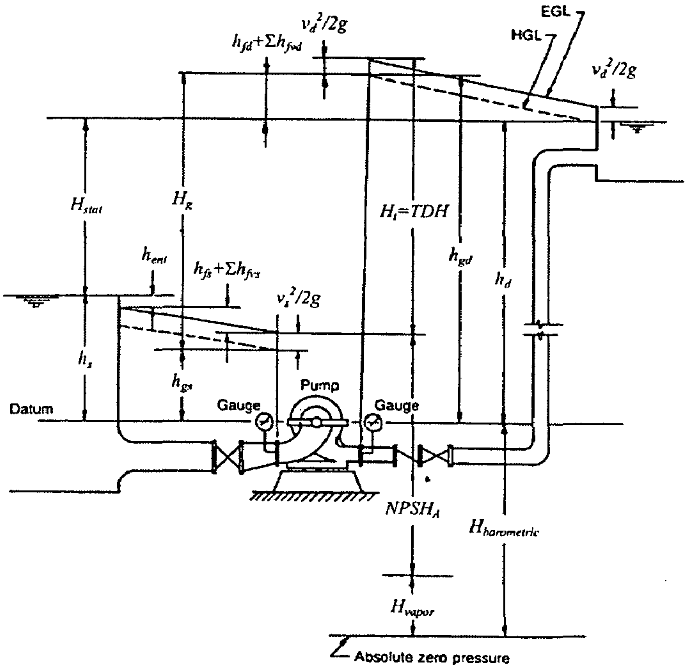


Fig. 3.12 Terminology for a pump with a positive suction head. The gauge is located to show theoretical pressures at the inlet and outlet flanges.

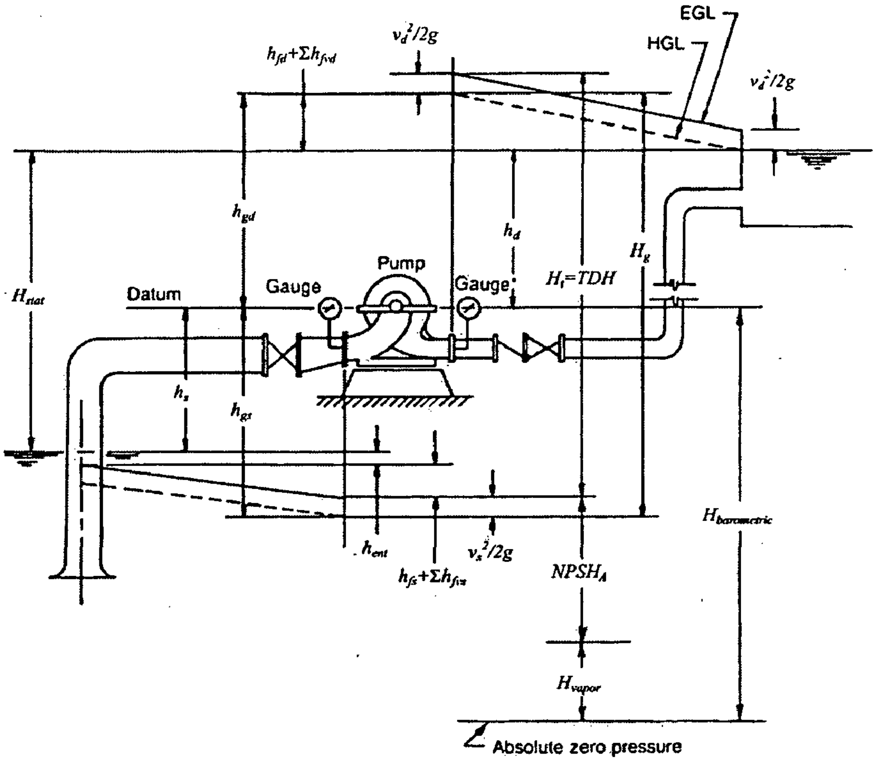


Fig. 3.13 Terminology for a pump with a negative suction head, The gauge location is to show theoretical pressures at the inlet and outlet flanges.

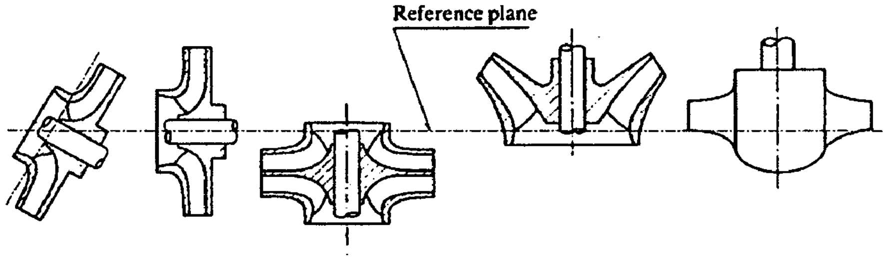


Fig. 3.14 Reference plane for NPSH value according to ISO

OUTPUT POWER

The power output of n pump is the energy delivered by the pump to the fluid. In SI units, the power output is defined as

$$Po = \rho.g.Q.H \tag{3.6a}$$

In practice it is customary to quote power in kW. It is expressed by the following equation

$$Po = \frac{\rho.Q.H}{367} \text{ in kW} \tag{3.6b}$$

where *Po* is the water power in kilowatts, ρ is the specific weight of the fluid in kg/dm^3. *Q* is the flowrate in m^3/h. H is the total dynamic head in meters,

INPUT POWER

Pump performance is measured in terms of the flowrate that a pump can discharge against a given head at a given efficiency. The pump capacity depends on the design, and design information is furnished by the pump manufacturer in a series of curves for a given pump. Pump efficiency, η, is the ratio of the useful power output (water kilowatts [wkW]) to the power input to the pump shaft. The pump power input can be expressed by the following equation in consistent units mentioned earlier

$$Po = \frac{\rho . Q . H}{367 . \eta} \tag{3-7}$$

Pump efficiencies usually range from 20 to 85% and increase with the size of the pump (see fig. 3.15.). Energy losses in a pump are volumetric, mechanical and hydraulic. Volumetric losses are those of leakage through the small clearance between wearing rings in the pump casing and the rotating element. Mechanical losses are caused by mechanical friction in the stuffing boxes and bearings, by internal disc friction, and by fluid shear. Frictional and eddy losses within the flow passages account for the hydraulic losses.

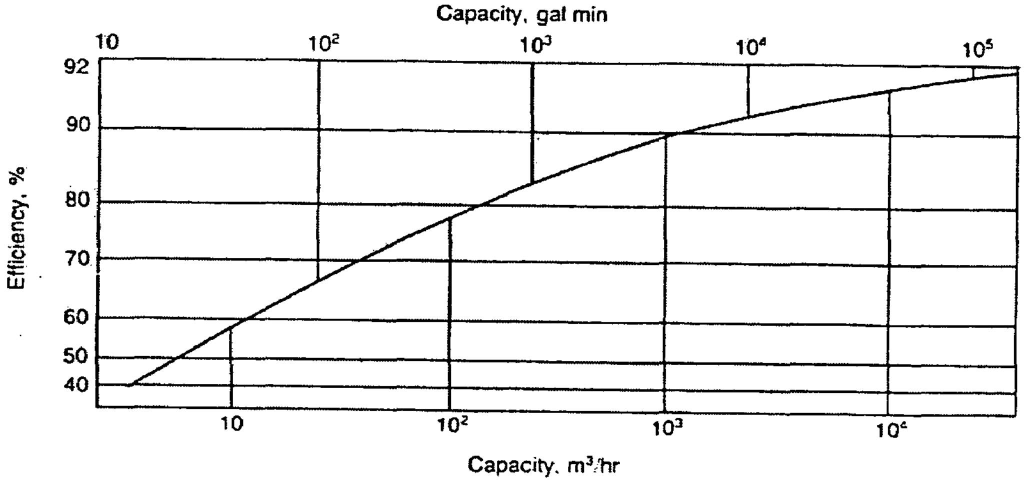


Fig. 3.15 Maximum pump efficiency attainable at the best operating point.

One distinguishes:

Nominal pump power input	P_N (power required at Q_N, H_N, and n_N)
Optimum pump power input	P_{opt} (power required at the point of optimum efficiency)
Pump power input at shut-off	P_o (power required at zero flowrate Q=0)

3.2.5. PERFORMANCE CURVES

The head that a pump can produce at various flow-rates and rotational speeds is established in pump tests conducted by the pump manufacturer. During testing, the capacity of the pump is varied by throttling a valve in the discharge pipe and the corresponding head is measured. The results of these tests and other tests with different impeller diameters are plotted to obtain a series of head-capacity (H-Q) curves for the pump at some given speed (see fig. 3.16.). Simultaneously, the power input to the pump is measured. The efficiency at various operating points is computed, and these values are also plotted in the same diagram. Together, these curves are known as "pump characteristic curves".

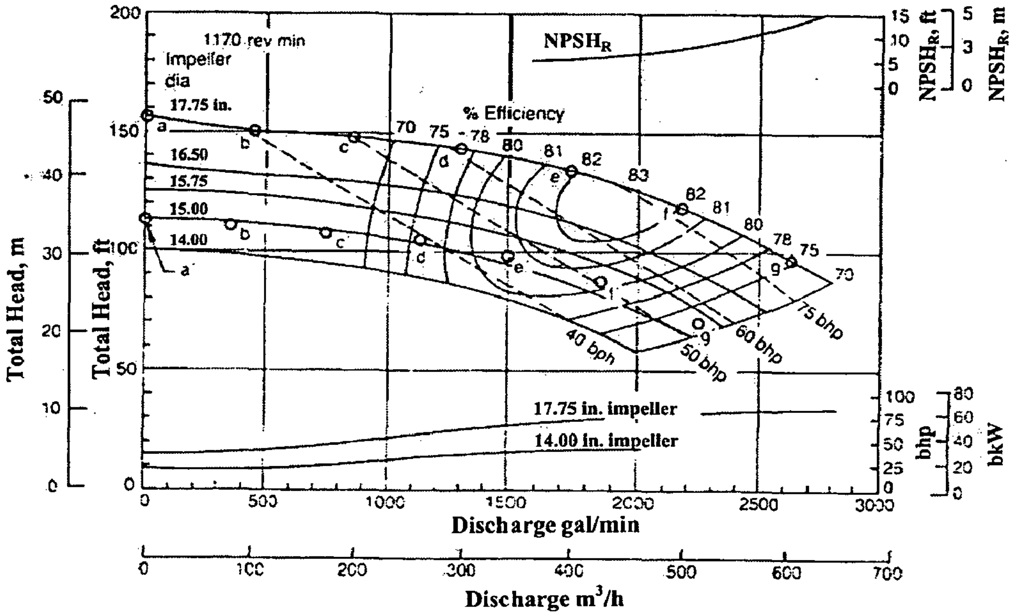


Fig. 3.16 Typical pump characteristic curves.

The operating characteristics of pumps depend on their size, speed, and design. Pumps of similar size and design are produced by many manufacturers, but they vary somewhat because of slight design modifications. The basic relationships that can be used to characterize and analyze pump performance under varying conditions include

- Energy transfer in pumps
- Flow, head, and power coefficients
- Affinity laws
- Specific speed.

3.2.6. FLOW, HEAD, AND POWER COEFFICIENTS

In centrifugal pumps, similar flow patterns occur in a series of geometrically similar pumps. By applying the principles of dimensional analysis, the following three independent dimensionless groups can be derived to describe the operation of centrifugal pumps and other rotodynamic machines. Note that the equations are the same in either SI or U.S. customary units. Only the value of the coefficient changes.

$$C_Q = \frac{Q}{nD^3} \tag{3-12}$$

$$C_H = \frac{H}{n^2D^2} \tag{3-13a}$$

or, for dimensional correctness,

$$C_H = \frac{gH}{n^2D^2} \tag{3-13b}$$

$$C_p = \frac{P}{\rho.n^3.D^5} \tag{3-14}$$

where C is a coefficient and the subscripts Q, H, and P correspond to capacity, head, and power. Q is capacity in cubic meters per second (cubic feet per second), n is speed in radians per second (revolutions per minute), D is impeller diameter in meters (feet), H is head in meters (feet), g is 9.81 m/s^2 (32.2 ft/s^2), P is the power input in kilowatts (horsepower), and ρ is the density in kilograms per cubic meter (slugs per cubic foot). Although equation 3-13b is dimensionally correct, equation 3-13a is commonly used for both SI and U.S. customary units.

Equations 3-12 through 3-14 apply only to "corresponding points", the operating points at which similar flow patterns occur. Thus, every point on a head-capacity curve for a large pump corresponds to a point on the head-capacity curve of a geometrically similar but smaller pump that operates at the same speed. This correspondence can be corrected for different speeds provided that the speeds differ by no more than about 40%.

3.2.7. AFFINITY LAWS

For a pump operating at two different speeds, the following relationships can be derived from equations 3-12 through 3-14. (Note that diameter does not change.)

$$\frac{Q_1}{Q_2} = \frac{n_1}{n_2} \tag{3-15}$$

$$\frac{H_1}{H_2} = \left(\frac{n_1}{n_2}\right)^2 \tag{3-16}$$

$$\frac{P_1}{P_2} = \left(\frac{n_1}{n_2}\right)^3 \tag{3-17}$$

where Q is flowrate, H is head. P is power, n is rotational speed, and Subscripts 1 and 2 are only for corresponding points. Note that equations 3-15 and 3-16 must be applied simultaneously to ensure that point 1 "corresponds" to point 2. Corresponding points fall upon parabolas through the origin. They do not fall upon system H-Q curves. These relationships, known collectively as "the affinity laws", are used to determine the effect of changes in speed on the capacity, head and power of a pump.

The affinity laws for discharge and head are accurate as found from actual tests for all types of centrifugal pumps including axial-flow pumps. The affinity law for power is not as accurate because efficiency increases with an increase in the size of the pump. But if the affinity law for power is used, the computed value of power can be corrected using the Moody equation. which accounts for efficiency as a function of pump size.

In applying these relationships, remember that they are based on the assumption that the efficiency remain the same when transferring from a given point on one pump curve to a homologous point on another curve. Because the hydraulic and pressure characteristics at the inlet, at the outlet, and through the pump vary with the flowrate, the errors produced by equation 3-17 may be excessive, although errors produced by Equations 3-15 and 3-16 are very small.

Approximate Relationships for Radial-Flow Pumps

To cover a wide range of flows with a minimum number of pump casings and impeller designs, manufacturers customarily offer a range of impeller diameters for each size of casing (see fig. 3.14.). In general, these radial-flow impellers are identical as cast, but the size is reduced by machining the impeller to a smaller diameter. Equations 3-12 through 3-14, written for pumps of different sizes, can be modified into equations for impellers of reduced diameter in the same pump. Dividing equation 3-13 for an impeller of diameter D_1

by the same equation for an impeller of diameter D_2 and canceling C_H and n^2 (because density, viscosity, and rotational speed are the same) gives,

$$\frac{H_1}{H_2}=\left(\frac{D_1}{D_2}\right)^2 \tag{3-18}$$

The same analysis applied to equation 3-12 gives

$$\frac{Q_1}{Q_2}=\left(\frac{D_1}{D_2}\right)^3 \tag{3-19}$$

Equation 3-19 applies only to two pumps of different size (i.e., the same scale-up in three dimensions). For a single pump, the discharge area for a radial-flow impeller is a function of impeller width and volute size and both are nearly constant even if the impeller diameter is reduced. Because area is proportional to diameter squared, equation 3-19 can be convened to an equation for a single pump by factoring out

$$\frac{Q_1}{Q_2}=\frac{D_1}{D_2} \tag{3-20}$$

Power is a function of head times discharge, so multiplying equation 3-18 and equation 3-20 gives

$$\frac{P_1}{P_2}=\left(\frac{D_1}{D_2}\right)^3 \tag{3-21}$$

These relationships, when used to predict the effect of changing the diameter of a radial-flow impeller, are somewhat less accurate than the affinity laws because the angle of the blade decreases slightly and the clearance between impeller and casing increases (which changes the geometry) as the diameter is reduced (see fig. 3.17.). Two or more impeller designs are often available (each in a range of sizes) for the same casing, but if these impellers are not geometrically similar, the affinity laws and the previous relationships are invalid.

It should be noted that the modified affinity laws are less accurate for trimming impellers of mixed and axial-flow pumps. In most designs, the pump is somewhat more effective when the pump impeller is trimmed than would be predicted using the modified affinity laws.

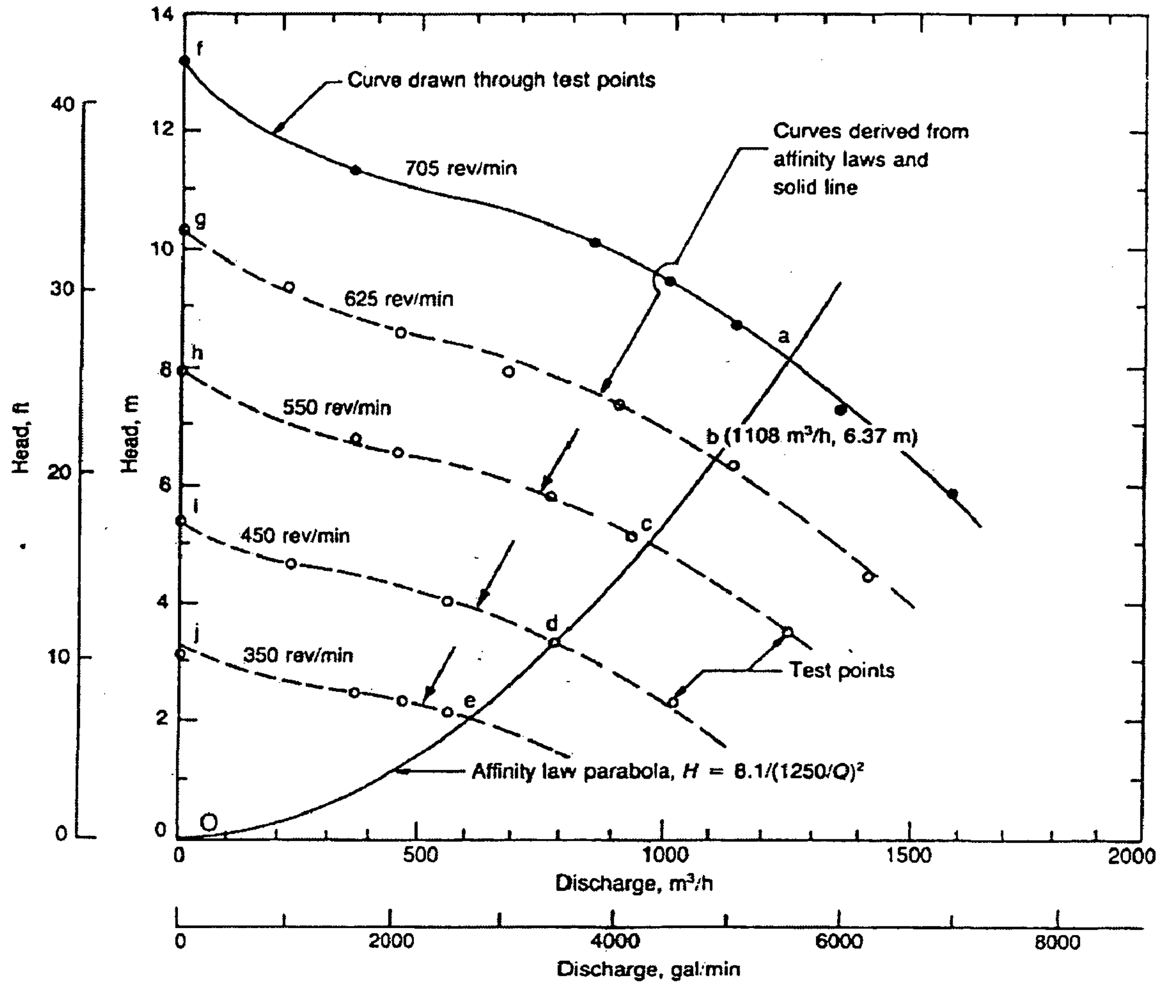


Fig. 3.17. Pump curves developed from affinity laws compared with test data.

3.2.8. SPECIFIC SPEED

For a gometrically similar series of pumps operating under similar conditions, the diameter term in equations 3-12 and 3-13a can be eliminated by dividing the square root of Equation 3-12 by the three-fourths power of Equation 3-13a:

$$n_s = \frac{C_Q^{1/2}}{C_H^{3/4}} = \frac{(Q/nD)^3}{(H_t/n^2D^2)^{3/4}} = \frac{nQ^{1/2}}{H_t^{3/4}} \qquad (3\text{-}22)$$

where n_s is the specific speed, n is in revolutions per minute (not radians per second), Q is discharge (customarily in cubic meters per second but sometimes in liters per second or cubic meters per hour), and H_t is the total dynamic head in meters. In spite of being dimensionally incorrect, it is this form of the equation that is commonly used. (A dimensionally correct expression would be derived by using equation 3-13b instead of 3-13a.

For any pump operating at any given speed, Q and H must be taken at the point of maximum efficiency. When using equation 3-22 for pumps with double suction impellers, one-half of the discharge is used unless otherwise noted. For multistage pumps, the head is the head per stage.

The variations in maximum efficiency to be expected with variations in size, capacity, and specific speed are shown in fig. 3.18. for single-stage, single-entry centrifugal pumps. The progressive changes in impeller shape as the specific speed increases are shown along the bottom of fig. 3.18. The efficiency increases with the size of the pump because of a reduction in the friction losses in the pump shrouds and a corresponding drop in the volumetric losses. The greatest efficiency is achieved in single-stage, single-entry pumps with a volute casing. In general, for a given rotational speed,

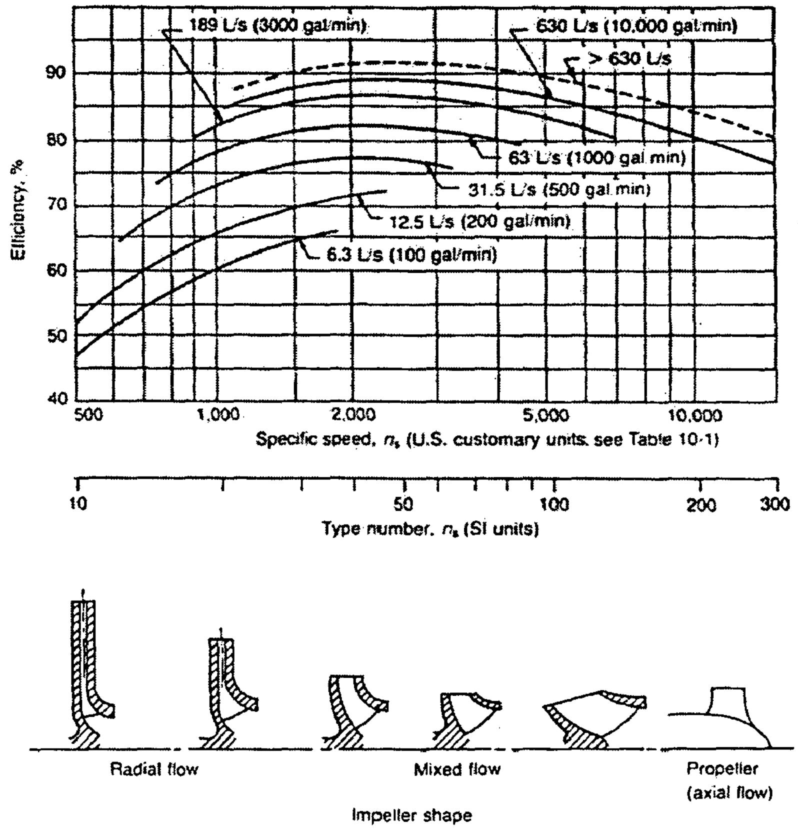


Fig. 3.18. Pump efficiency as related to specific speed and discharge.

- If the n_s value is low (specific speed <30, or in U.S. units, <1500), Q must be low and H must be high.
- If the n_s value is intermediate (specific speed of 30 to 80, or in U.S. units 1500 to 4000). Q and H must be of intermediate value.

- If the n_s value is high (specific speed >80, or in U.S. units, >4000), *Q* must be high and *H* must be low.

3.2.9. NON-DIMENSIONAL PUMP CHARACTERISTIC CURVES

Non-dimensional pump curves are obtained by expressing head, capacity, power input and efficiency as percentages of the corresponding values at the best efficiency point, and such curves are useful for (1) comparing the hydraulic properties of pumps belonging to the same type and (2) assessing the performance of pumps of different specific speeds for various applications. The general shape of these curves varies with the specific speed, the form and number of blades on the impeller, and the form of the casing used. As shown in fig. 3.19., the slope of the head-capacity curve becomes steeper as the specific speed increases. Notice the shape of the power curves in fig. 3.19a. As head decreases below the bep, the power required increases. Thus, overloading the motor is most likely to occur in pumps with low specific speeds and with flat head-capacity (H-Q) curves. Unstable pump operating curves are typically encountered in such pumps. Furthermore, because of the shape of the characteristic curve, in fig. 3.19., mixed-flow and axial-flow pumps are non-overloading in their operating range. However, depending on the impeller blade design, a dip (and consequent instability) can occur as shown in H-Q curves for pumps of high specific speeds. Prolonged operation of high-specific speed pumps in this head-capacity range should be avoided. As the specific speed increases, the effects on the power input are marked. With radial-flow pumps, where the specific speed is about 35 (about 1800 in U.S. Units), the energy input decreases with a decrease in flow. For a mixed-flow pump with a specific speed of about 85 (about 4300 in U.S. units). the power input is nearly constant. In pumps with specific speeds greater than about 125 (greater than 6400 in U.S. unit), the power absorbed by the fluid rises steeply as the discharge is decreased to zero. Thus, with mixed and axial-flow pumps, some means of unloading the pump (such as a bypass pipe with a pressure-activated valve) is needed the head rises so as to avoid overloading the pump driver. Sometimes, it is more cost effective to oversize the motor than to install an unloading device. A comparative cost analysis should be made if there is doubt.

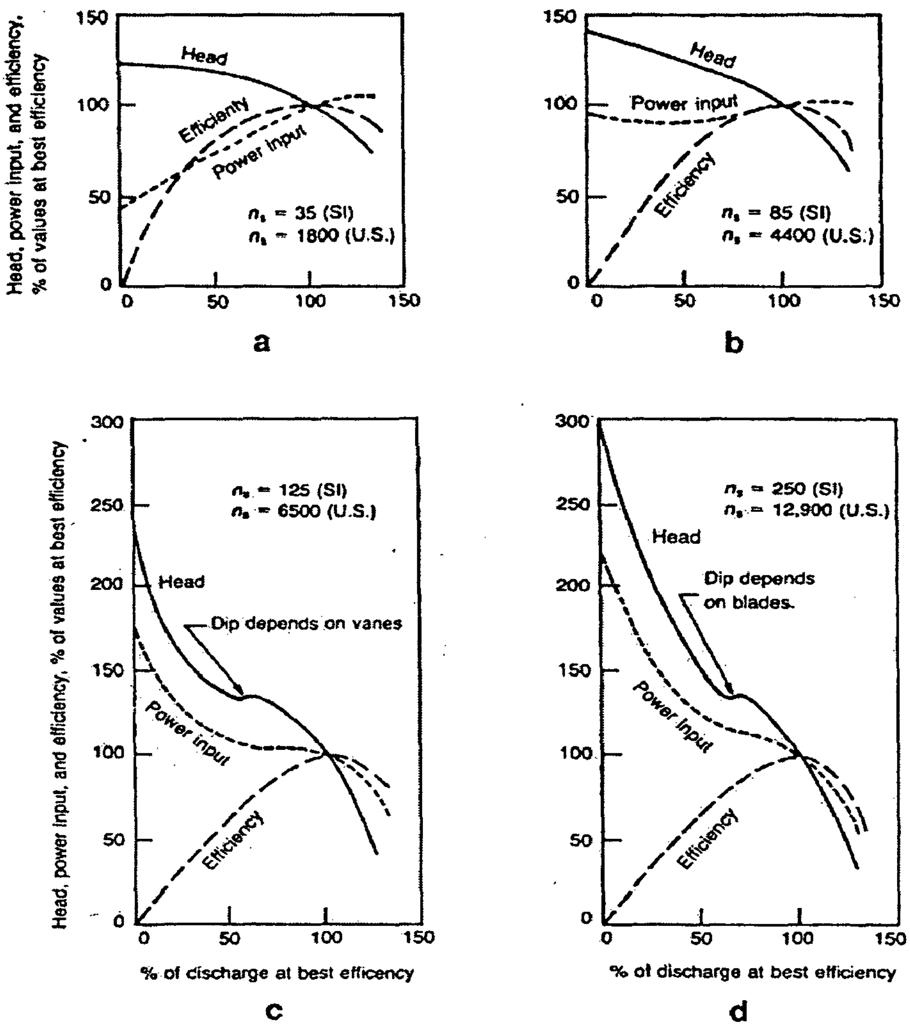
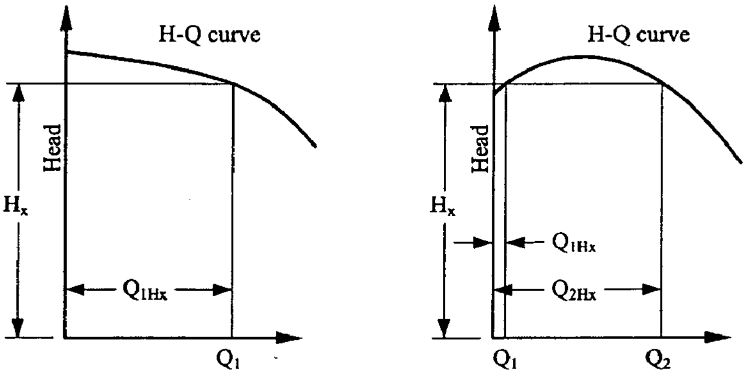


Fig. 3.19. Typical dimensionless characteristic curves for centrifugal pumps. (a) Radial flow; (b) mixed flow; (c) mixed flow; (d) axial flow.

3.2.10. STABLE AND UNSTABLE PUMP CURVES

The head-capacity curves for radial-flow centrifugal pumps can be either stable (fig. 3.20a) or unstable (fig. 3.20b), whereas the head-capacity curves for mixed-flow and axial-flow pumps are always stable (except for a small area of operation). Unstable pump curves are usually limited to specific speeds of less than 20 (<1000 in U.S. units). With stable pump curves, there is only one flowrate for each value of head, whereas two discharge values are possible for a given value of head in unstable pump curves, as can be seen from Fig. 3.20b.

The use of pumps with unstable pump curves can lead to pumping instabilities at heads greater than the shut-off head, especially where pumps are operated in parallel. It is best to avoid the use of pumps with unstable head-capacity curves entirely. In single-stage pumps, stable pump head-capacity curves can be obtained by reducing the number of impeller blades, changing the exit blade angle. and changing the configuration of the blades.



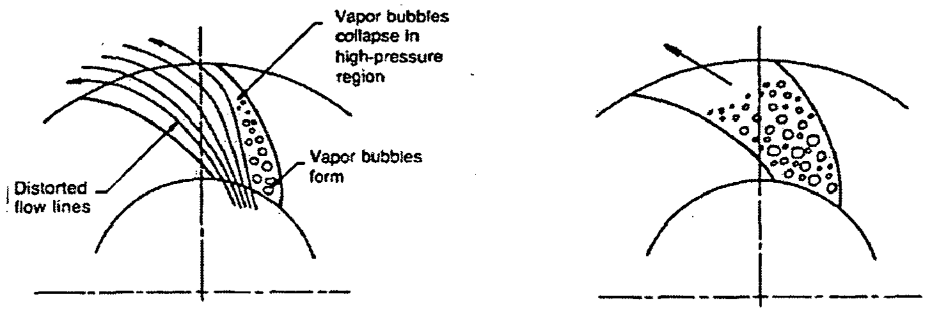
(a) Stable (b) Unstable

Fig. 3.20. Stable and unstable pump H-Q curves.

3.2.11. CAVITATION AND ITS EFFECTS

Cavitation is a potential danger, especially when pumps operate at high speeds or at a capacity much greater or much less than the best efficiency point (bep). Cavitation reduces pump capacity and efficiency and may damage the pump, sometimes very rapidly. It occurs in pumps when the absolute pressure at the inlet of the pump drops below the vapor pressure of the fluid being pumped. At first, air comes out of solution to form tiny bubbles, followed instantly by vapor as the water boils. As the vapor bubbles are transported through the impeller, they reach a zone of higher pressure where they collapse abruptly (see fig. 3.21a). If the collapse occurs on the surface of a solid, the liquid rushing in to fill the vacuous space left by the bubbles impacts tiny areas with tremendous, localized pressures and thereby pits and erodes the surface. Cavitation can also occur wherever local velocities are high, such as in vanes or nearly closed valves. The impeller eroded because the absolute pressure at the pump inlet connection was too low. The damage to the suction bell caused by water recirculated by operating the pump at 35% of its bep discharge. It has been suggested that the pitting and erosion is accelerated by simultaneous chemical attack or that the high-impact pressure causes locally high temperatures that accelerate pitting. In addition to the pitting and erosion, cavitation can also cause noise and vibration. Noise is produced by the collapse of the vapor bubbles as they enter the region of higher pressure.

The vibration is due to the imbalance and surging caused by the uneven distribution of collapsing vapor bubbles. The NPSHR curves are the basis for selecting the maximum permissible discharge. When the pump discharge rate decreases toward zero at the shut-off head, the radial load increases and the re-circulation of the pumped fluid within the impeller becomes a problem. This re-circulation can cause vibration and may also result in cavitation that can ruin the impeller. Pumps should not be operated at or near the shut-off head for extended period of time because such operation also leads to heating of the liquid and can cause severe wearing-ring rub and other problems.



(a) Partial cavitation (b) full cavitation

Fig. 3.21. Formation or vapor bubbles in a pump impeller.

3.2.12. EFFECTS OF CAVITATION ON PUMP PERFORMANCE

When cavitation occur in centrifugal pumps with type numbers less than 30 (specific speeds <1500), there is a sharp drop or cutoff in H-Q and efficiency curves, as shown in fig. 3.22. Vapor bubbles with begin to form at a lower discharge than the cutoff discharge

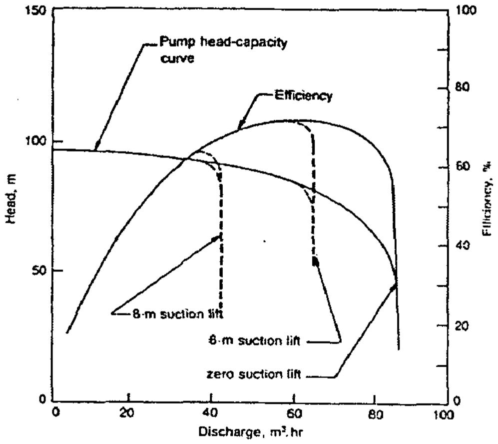


Fig. 3.22. Pump performance curves at various suction lifts with cavitation.

where cavitation is fully formed. For pumps with type numbers between 30 and 80 (specific speeds between 1500 and 4000 in U.S. units), the pump performance curves fall more gradually until the cutoff discharge is reached. For pumps with type numbers greater than 80 (specific speeds greater than 4000 in U.S. units), there is no distinct cutoff point.

3.2.13. NET POSITIVE SUCTION HEAD (NPSH)

The absolute pressure plus the velocity head at the eye of the impeller converted to absolute total dynamic head is called the "net positive suction head" and abbreviated NPSH. Pump performance declines rapidly as the NPSH becomes less than the NPSH "required" (NPSHR). The NPSHR is determined by test of geometrically similar pumps operated at constant speed and discharge but with varying suction heads. The development of cavitation is assumed to be indicated by a 3% drop in the head developed as the suction inlet is throttled, as shown in fig. 3.23. It is known, however, that the onset of cavitation occurs well before the 3% drop in head and can, indeed, develop substantially before any drop in the head can be detected. Furthermore, erosion occurs more rapidly at a 1% change in head (where vapor bubbles are small but cause a higher unit surface implosion energy) than it does at a 3% change in head (where bubbles are large with a lower unit implosion energy). Injection of air reduces the hydraulic shock imposed on the mechanical equipment. Although air injection may improve mechanical stability by acting as a shock absorber, there are distinct disadvantages and seldom does it permit long-term operation, because: (1) oxygen content increases corrosion, (2) gas increases the NPSHR, and (3) air causes noise and some vibration. In conclusion, serious erosion can occur as a result of blindly accepting data from catalogs because of the current standard (a 3% drop in head)

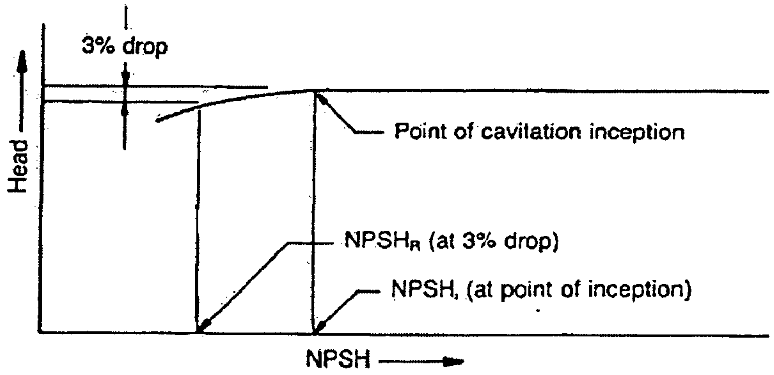


Fig. 3.23. Net positive suction head criteria as determined from pump test results.

used by most pump manufacturers. In critical installations where Continuous duty is important or where the pump is to be operated at reduced speed, the manufacturer should be required to furnish the *NPSHR* test results. Typically, *NPSHR* is plotted as a continuous curve for a pump (see fig. 3.16.). When impeller trim has a significant effect on the *NPSHR*, several curves are plotted.

The *NPSH* "available" (*NPSHA*) in the actual installation is calculated using Equation:

$$NPSHA = H_{bar} + h_s - H_{vap} - h_{fs} - \Sigma h_m - h_{vol} - FS \qquad (3\text{-}23)$$

where:

H_{bar} is the barometric pressure in m or ft of water column corrected for elevation above mean sea level. Note that storms call rêduce barometric pressure by 1.7%

h_s is the static head of the intake water surface above the eye of the impeller. If the water surface is below the eye, h, becomes minus.

H_{vap} is vapor pressure of the fluid at the maximum expected temperature.

h_{fs} is pipe friction in m or ft between the suction intake and the pump.

Σh_m is the sum of minor pipe friction losses such as entrance, bend, reducer, and valve losses. These "minor" losses are significant.

h_{vol} is the partial pressure of dissolved gases such as air in water (customarily ignored) or volatile organic matter in wastewater (customarily estimated to be about 0.6 m or 2 ft).

FS is a factor of safety used to account for uncertainty in hydraulic calculations and for the possibility of swirling or uneven velocity distribution in the intake. Also, the 3% drop in head allows some cavitation (that ought to be suppressed) to occur, and although a 3 % loss in *TDH* is typically insignificant, the shock on the equipment can reduce the life of bearings, seals, and impellers. Surprisingly large variations in NPSHR are sometime observed in tests of supposedly identical pumps. Small pumps are more sensitive to casting imperfections than large ones. Now add the fact that the NPSH.to avoid cavitation may increase substantially when the pump operates at any point but the bep, and the need for a generous factor of safety is apparent. In the past, it was customary to consider that 0.6 m or 2 ft (but no less than 20% of the NPSHR) was adequate, and some very knowledgeable engineers continue to recommend such safety factors for water and Others, however, warn that anything less than 1.5 m (5 ft) or less than 1.35 times NPSHR may be dangerous. Consult the pump manufacturer to decide which rules to follow. But be aware that to eliminate cavitation and its effect on TDH entirely, the NPSHA must depending on the operating flowrate relative to the bep, be 2 to 5 times the NPSHR. The effect of NPSH on TDH is illustrated in fig. 3.24.

No term for velocity head is seen in Equation 3-23, because velocity head is part of the absolute dynamic head.

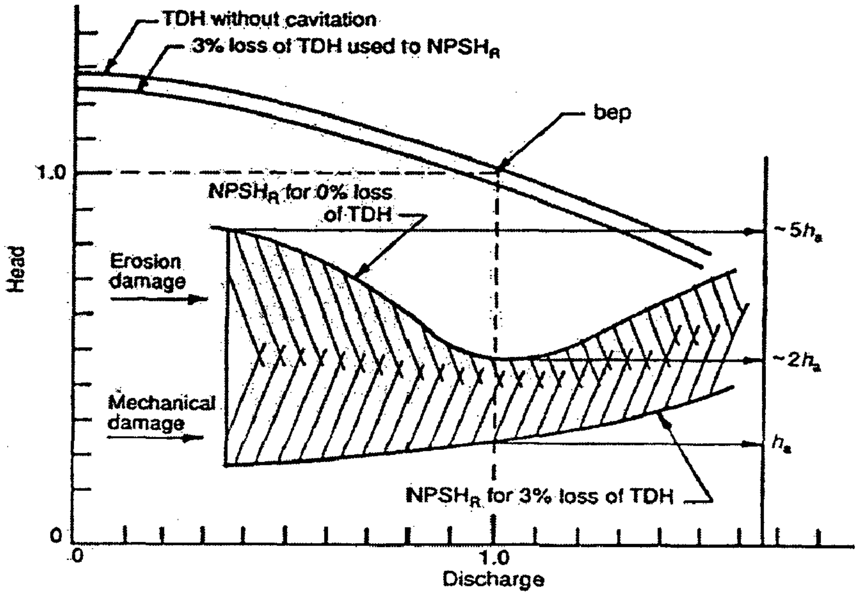


Fig. 3.24. Effect Or V PSH on TDH and pump damage.Head required at bep for 3% loss of TDH is termed ha, Coutesy of DuPont engineering wastewater pumps.

3.2.14. PREVENTION AND CONTROL OF CAVITATION

The easiest, most direct and best way to eliminate cavitation is to ensure that the internal pump pressure remains above the vapor pressure of the water. Where cavitation already exists, possible solutions might include

- Decreasing the suction lift
- Decreasing the suction losses
- Lowering the liquid temperature
- Reducing the impeller speed
- Changing the pump or the impeller
- Adding u booster pump or an inducer.

An inducer is an auxiliary axial-flow propeller attached to the pump impeller as in Fig. 3.23. It produces a small increase in the fluid pressure at the eye of the pump impeller, which reduces the risk of cavitation in the expensive impeller. Inducers are designed specifically for very low NPSHR and will not cavitate if operated at the design point. Unfortunately, the operating range that is free of cavitation is rather narrow. If the inducer codes due to cavitation, it is easier and cheaper to replace than is the pump impeller.

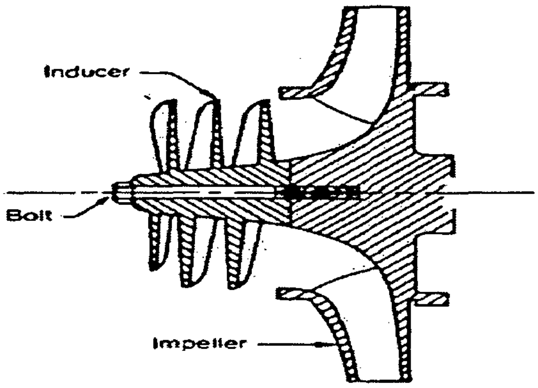


Fig. 3.25. ***Typical inducer axial-flow impeller attached to a conventional pump impeller.***

3.2.15. CAVITATION CONSTANT

The ratio of the net positive suction head at the point of cavitation inception, NPSHi (see fig. 3.23.), to the total dynamic head is known as Thoma's cavitation constant.

$$\sigma = \frac{\text{NPSHi}}{H_t} = \text{cons tan t} \qquad (3\text{-}24)$$

where H_t = total dynamic head in meters (feet). For multistage pumps, the total dynamic head is the total dynamic head per stage. In the literature, the NPSHR value for a pump is often used incorrectly in place of NPSHi. The pump NPSHR cannot be used because cavitation is already occurring at that value. Furthermore, it should be noted that there is no known relationship between the NPSHR at the 3% level and the value of NPSHi.

Because the specific speed is an indication of performance curve shape, it is possible approximately to correlate σ (and therefore NPSH_R) with the specific speed. The relationship between specific speed, Thoma's cavitation constant, and pump efficiency for single suction pumps as published by Rutschi is illustrated in fig. 3.26. For single-suction pumps

$$\sigma = \frac{K \times n_s^{4/3}}{10^6} \tag{3-25}$$

Where the values of K are given in Table 3-1. Equation, 3-24 and 3-25 are presented for background only and should not be used for making design decisions. Recommendations for NPSHR should come from the pump manufacturer for the specific operating conditions and requirements.

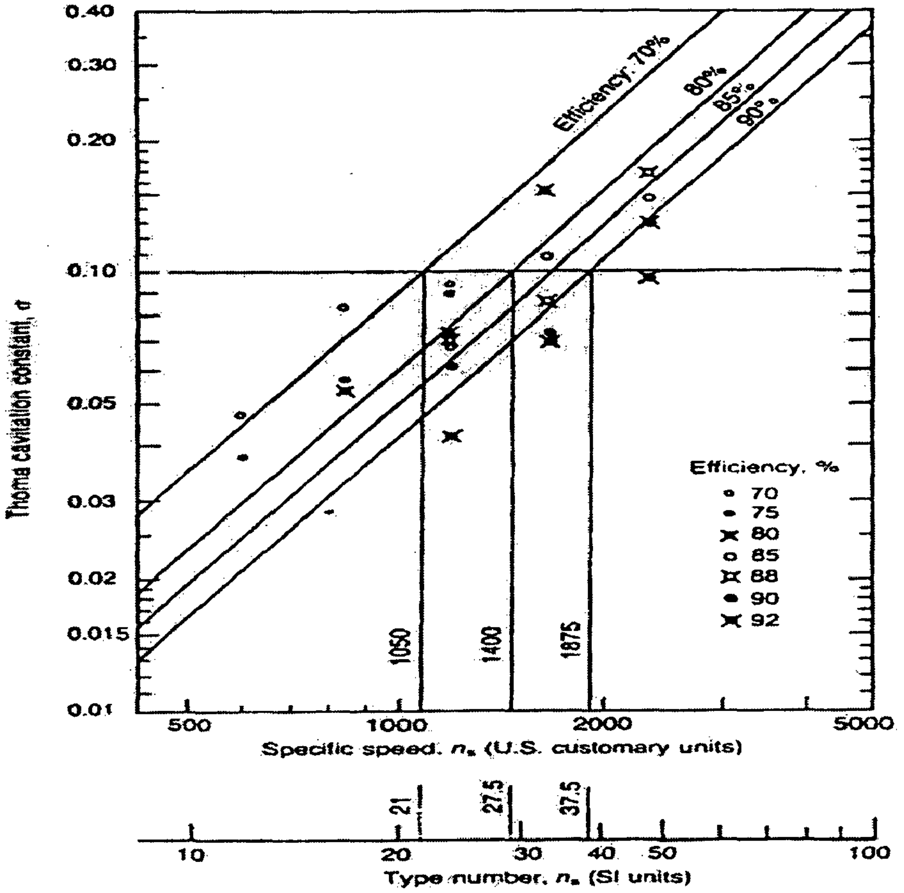


Fig. 3.26. Cavitation constant, σ, versus specific speed.

Table 3-1. Values or K for Equation 3-25[a]

Pump efficiency	K	
	SI	U.S. customary
(%)	units[b]	units[c]
70	1726	9.4
80	1210	6.3
90	796	4.3

a. For double-suction pump, use the same formula with Q equal to half of the actual value.

b. Use with *ns* value, in cubic meters per second, meters, and revolutions per minute.

c. Use with *ns* values in gallons per minute, feet, and revolutions per minute.

3.2.16. CAVITATION AT THE OPERATING POINT

If the pump Operates at low head at a flowrate considerably greater than the capacity at the best efficiency point (*bep*). Equation 3-26 is approximately correct:

$$\frac{\text{NPSHR at operating point}}{\text{NPSHR at bep}} = \left(\frac{\text{Q at operating}}{\text{Q at bep}}\right)^n \qquad (3\text{-}26)$$

where the exponent n varies from 1.25 to 3.0, depending on the design of the impeller. In most water and wastewater pumps, n lies between 1.8 and 2.8. The NPSHR at the *bep* increases with the specific speed of the pumps. For high-head pumps, it may be necessary either to limit the speed to obtain the adequate NPSH at the operating point or to lower the elevation of the pump with respect to the free water surface on the suction side.

3.2.17. PUMP OPERATING RANGES

A pump operates best at its best efficiency point. Not only is the efficiency maximum, but radial loads on the impeller and the problems of cavitation are minimized. It is good practice to limit the operating range of pump, especially radial-flow centrifugal pumps to within approximately +20% and -40% of the discharge at the bep.

3.2.18. FIXED EFFICIENCY LOSS

For a given impeller diameter, the operating range of a pump can be established by:

(1) setting a limit on the minimum acceptable efficiency and

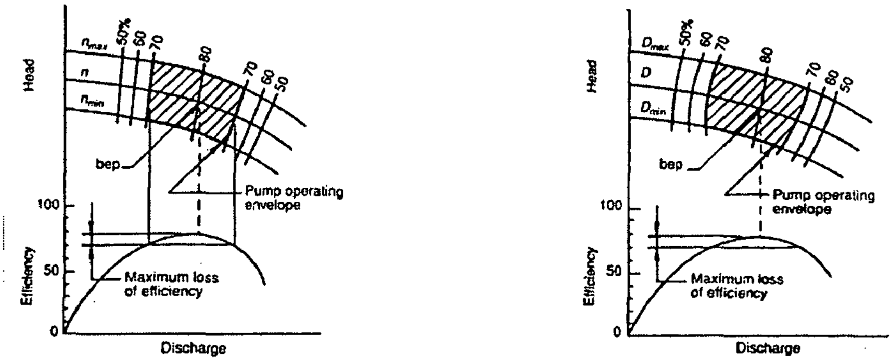
(2) setting upper and lower limits on the allowable speed changes.

Alternatively, if the diameter of the pump impeller is to be changed. the operating range of the pump can be established by:

(1) setting a limit on the minimum acceptable efficiency and

(2) setting upper and lower limits on the allowable impeller diameters.

The operating range of a pump based on these criteria is illustrated in fig. 3.27.

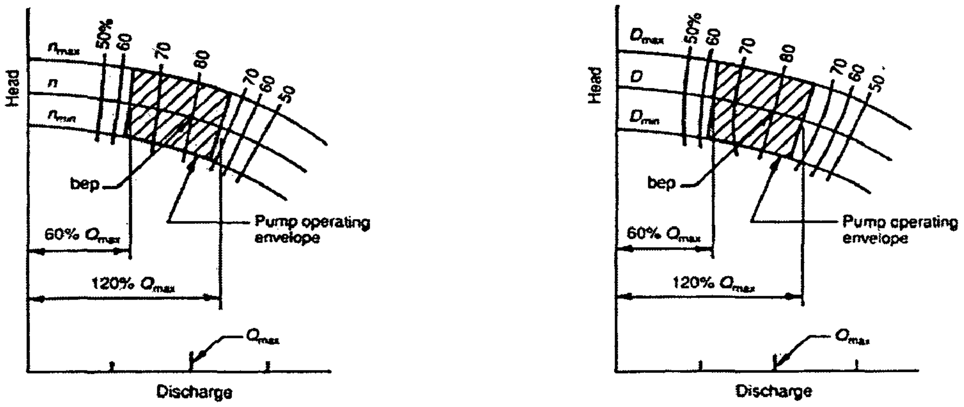


(a) Change of rotational speed (b) change of impeller diameter.

Fig. 3.27. Pump operating envelopes based on a fixed efficiency loss.

3.3.19. PERCENTAGE OF CAPACITY

An alternative approach used to establish the pump operating range is to set limits on the pump discharge as a percentage of the bep. An operating range of 60 to 120% of the bep is often recommended for pumps with specific speeds less than 40 (<2000 in U.S. units). Pump operating envelopes are defined using these values as illustrated in fig. 3.28, for changes in speed as well as for changes in the impeller diameter.



(a) Rotational speed (b) impeller diameter.

Fig. 3.28. Pump operating envelopes based on the percentage of capacity at the best efficiency point.

3.2.20. SYSTEM HEAD-CAPACITY (H-Q) CURVES

The transmission pipeline H-Q curves in fig. 3.28 are plotted over a range of flows from zero (where the only head is due to static lift) to the maximum expected (which also includes all friction, fitting, and valve losses). Because transmission pipeline or force main head-capacity curves are approximate functions of $v^2/2g$, the curve is approximately a

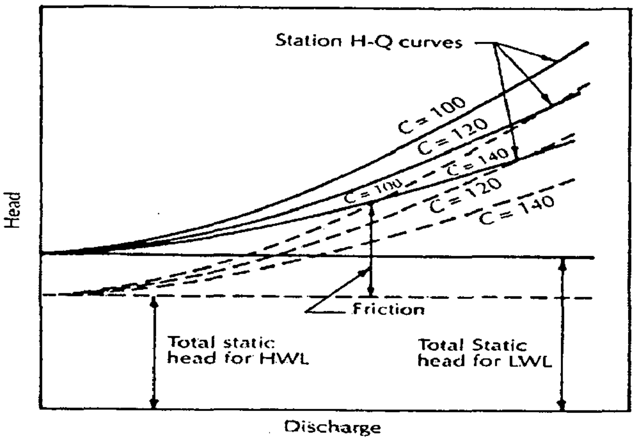


Fig. 3.29. Typical envelope of transmission pipeline head-capacity (H-Q) curves.

parabola with its apex on the zero Q line. Actually, headloss is a function of $Q^{1.85}$ not Q^2 in the Hazen-Williams equation. In the Darcy-Weisbach equation, the value of the friction factor f, changes with the Reynolds number. However, the use of a parabola is close enough for practical purposes. New pipe is expected to be very smooth for some indeterminate time. But bacterial slime, other deposit or deterioration may reduce the H-W coefficient C to as little as 120 for pipes lined with cement-mortar.

3.2.21. SINGLE-PUMP, SINGLE-SPEED OPERATION

The point on any specific system head-capacity curve at which a single-speed pump must operate is determined by superimposing the pump H-Q curve on the transmission pipeline H-Q curve as shown in fig. 3.30.

The point of intersection is the pump operating point. As the pipe ages and the pipeline H-Q curve rises, the operating point moves up and to the left along the pump H-Q curve. As pumps start and stop (or as they change speed) because of the change in water level, the path of operating points is extended still further along the pump curve. Pumps and impellers must be chosen for a range of operating conditions. The efficiency curve is usually rather broad at the top. Hence, choose an impeller that operates from slightly left to slightly right of the best efficiency point (bep). If the range in operating points is too great to fit the efficiency curve nicely, choose impeller that fits in the early years when the pipe smooth. Then a new impeller of larger diameter can be installed when conditions warrant it.

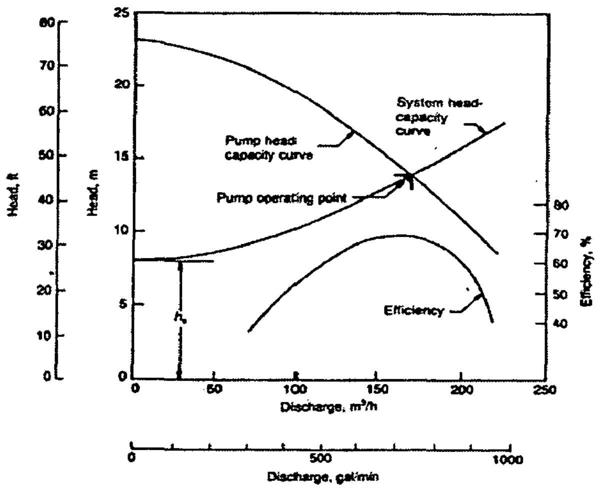


Fig. 3.30. Determining the operating point for a single-speed pump with a fixed value of h.

3.2.22. MULTIPLE PUMP OPERATION

In pumping stations where several pumps are installed. two or more pumps are usually operated in parallel. Occasionally, pumps are operated in series. The elements of the methods used to develop the combined H-Q curve for multiple pumps are outlined in the following subsections.

PARALLEL OPERATION

When two or more pump are discharging into the same header or manifold, the total flow is found by adding the individual flows at a given head. A correction (usually small) must be made, however, because the higher flow causes higher headlosses that slightly reduce the flow either pump would discharge if operated alone. One method for combining the flows is to subtract from the pump curve those station headlosses in the suction and discharge piping up to the intersection of the last pump discharge and the manifold (header). In the inset of fig. 3.31 as per fig. 3.32, the headlosses from *a* to *b* to *c* are subtracted from the H-Q curve for pump *P1*. Similarly, the losses from *a* to *d* to *c* are subtracted from the H-Q curve for pump *P2*. The modified pump curves are shown as dashed lines labeled MP1 and MP2 in the figure. Adding the abscissas (or discharges) of MP1 and MP2 produces the pump curve MP1 || MP2, where the symbol "||" means "in parallel with". Thus, $he = hg + hf$ The intersection of MP1 || MP2 with the system curve for the manifold and force main downstream from point *c* is the operating point(point *e*) for the pumps when both are running.

To find the discharge contributed by each pump, draw the horizontal line, *eh*, which intersects the modified pump curves at *f* and *g*. The discharge from *P1* is *hg*; from *P2* it is *hf*. To find the total head at which each individual pump operates, project a vertical line from *f* to *j* and from *g* to *k*. The actual operating point of *P1* is point *k* and the head on the pump is *kl*. The operating point of *P2* is point *j* and the head is *jm*. With one pump off, the other operates at point *n* (for *P1*) or point p for *P2*. The pump specifications must be written for pump *P1* operating either at point *k* or point n and for pump *P2* at point *j* or point *p*.

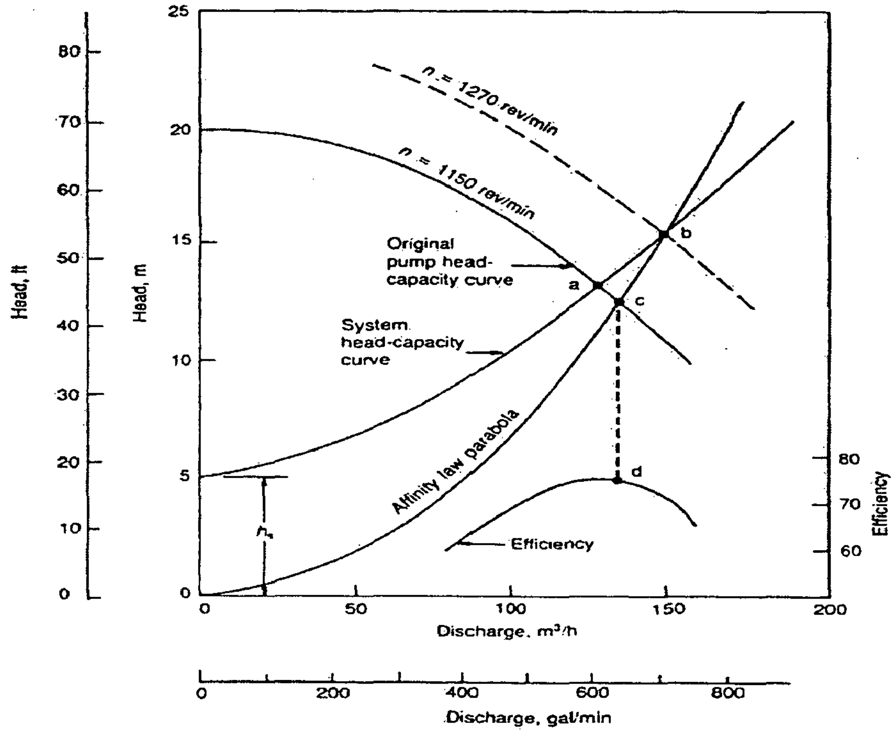


Fig. 3.31. Determining the pump operating points for a single variable-speed pump and a system curve with a fixed value of static lift (h).

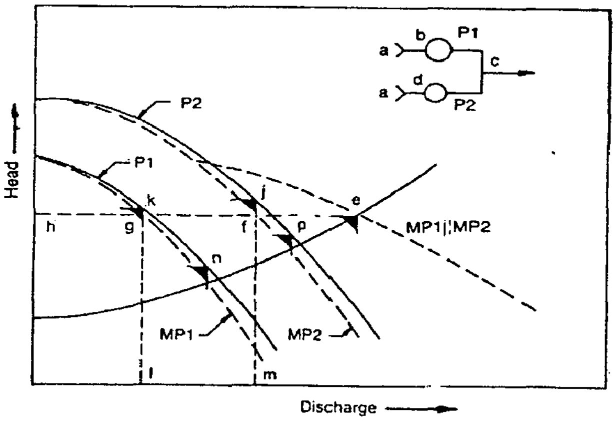
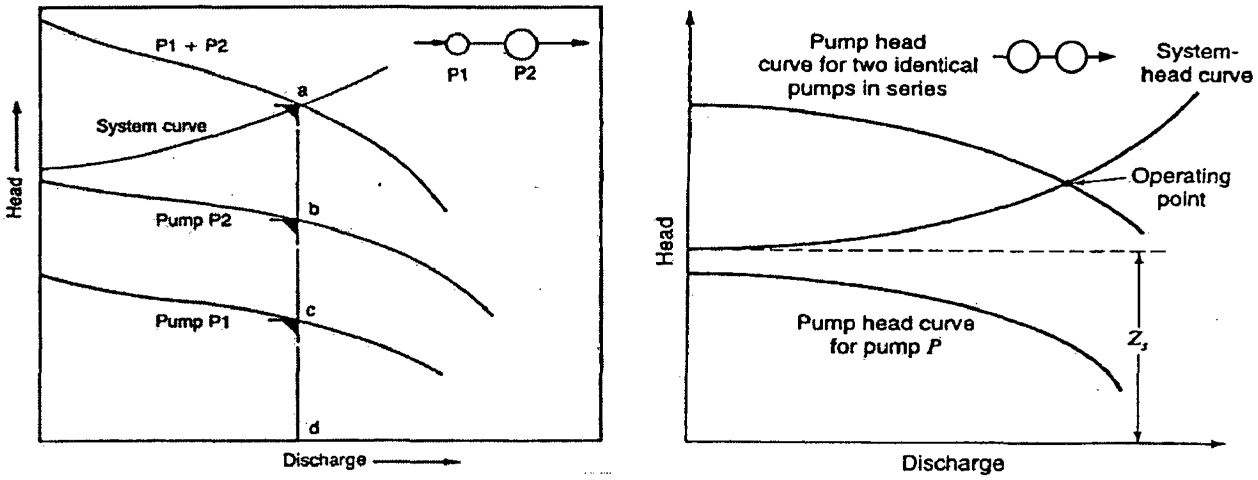


Fig. 3.32. Operation of two pumps in parallel.

SERIES OPERATION

Pump, are operated in series, as shown in fig. 3.33., when the head requirement is greater than can be obtained with a single pump. The combined head Capacity curve, *P1* + *P2*, is found by adding ordinates, that is the heads developed at the same discharge. Thus *ad* = *bd* + *cd*. The operating point is point *c* for pump *P1* and point b for pump *P2*, and the operating point for the system is point *a*.



Non-identical in series

Identical in series

Fig. 3.33. Operation of two pumps in series.

3.2.23. CONSTANT-SPEED (C/S) PUMPS IN PARALLEL

A typical schematic diagram of a pump and piping system for a pumping station with four assumed C/S pumps (one a standby) that discharge into a manifold and thence into the force main is shown in fig. 3.34. The system is divided into three basic components-force main, suction piping and discharge piping. Head-capacity (H-Q) curves are drawn for each component. The H-Q curves for the pumps are then superimposed. Collectively, the curves are called "system curves." In fig. 3.33. dynamic head losses in suction and discharge piping are combined into Curve B + C.

Curves A, A', and A'' (usually the first one, drawn) are combinations of static head plus the dynamic head losses from the manifold to the point of discharge. The curves must encompass the entire envelope of operating conditions including both HWL and LWL in the wet well and the extremes of friction losses in the force main. As shown, Curve A represents losses at the greatest static lift (LWL in the wet well) and the worst condition for pipeline resistance losses, in this example for Hazen-Williams C= 120. Curve A' represents the lowest static lift condition (HWL in the wet well) and similar pipeline losses, while curve A" represents the lowest static lift condition with the most favorable pipeline resistance losses-assumed here to be C = 145. Minor losses in the force main caused by valves and fittings can, perhaps, be ignored if the pipe is very long.

Begin the analysis by locating the operating point for the maximum head, so consider only Curve A and ignore (for the present) Curves A' and A". Determine the total required station discharge, *Q*, and draw a vertical line to intersect Curve A at Point 1. If the three duty pumps are of the same size, each must discharge *Q/3*. Draw a vertical line from *Q/3* to intersect (at Point 2) the horizontal line through Point 1. Point 2 represents the discharge and head required of the pump system that includes intake and discharge piping. The pump itself must develop enough more head, y_3, to overcome the friction and turbulence losses in the intake and discharge piping, so plot y_3 from Point 2 to Point 3. Point 3 is the required operating point or rated condition for each of the three pumps. It is unique because it represents the highest head required. Note, however, that the pumps will rarely operate at this condition.

Now choose a pump with an operating curve that includes Point 3, in other words, above and to the right of Point 3. The manufacturer's pump curve, here called Curve M, can be made to fit Point 3 exactly by changing either the speed or the diameter of the impeller (or both) according to the affinity laws as defined by Equations 3-15 through 3-18, 3-20, and 3-21. Note that these equations must be applied simultaneously. When an impeller is trimmed or the speed is changed, both head and discharge change. The curve through Point 3 is called M', the adjusted pump curve, (If the manufacturer shows the curves for a synchronous speed, the curve must also be corrected to the speed of the induction motor to be used.) Subtracting the ordinates (y_3, y_5, etc.) of Curve B + C from the adjusted pump curve, M', gives M", the corrected pump curve. Think of M" as the curve,

if suction and discharge piping were considered to be part of the pump. This curve (M") must be extended to pass through Curve A".

Another operating point, unique because it represents the maximum discharge when one pump operates alone against reduced friction and turbulence losses, is found by intersecting Curves M" and A" at Point 4 and plotting y_5 above Point 4 to locate Point 5.

All operating point, for the pumps (regardless of the wet well level, or the pipe roughness) are located along the Curve M' between Points 3 and 5. The pumps must be selected so that both points are within the manufacturer's recommendations.

It is interesting to plot Curves 2M" and 3M" to see what pump discharges and heads are developed under different conditions. For example, for HWL and minimum pipe friction, the flow and head for the pumping system is defined by the intersection of Curves 3M" and A" at Point 6. To obtain the head for the pump itself, add y_8 to Point 7 to obtain the total dynamic head at Point 8.

The conditions for both points 3 and 5 must be published in the purchase order or project specifications, because this is the only legally defensible method for communicating performance requirements to the equipment manufacturer.

Always bear in mind that pump performance is worse, terms of vibration, potential cavitation, and mechanical damage to the pump, at operating conditions different from the best efficiency point. As a general rule, off-the-shelf pumps perform best if operating conditions lie between 60 and 115% of best efficiency capacity, *bec*.

The next step (which should not be omitted) is to plot the hydraulic profile for the system curves against the profile for the pipeline under the intended modes of operation. Although extreme accuracy is not critical, it is important to assess the system as it is intended to function. Try to visualize how the system will behave under all operating conditions. Question, to be asked when performing this task are:

- ❖ Are all operating conditions properly defined by the system curves?
- ❖ What happens in the pipeline under each operating condition?
- ❖ Are there any peculiarities in the operation of the system that have not been considered in the construction of the system curves?
- ❖ Has everything that might influence pump operating conditions (and. hence, pump performance) been considered?

An example of where one can go astray is seen in fig. 3.36, in which a knee is shown in the pipeline profile. In this example, the operating point (Point 3 in fig. 3.35), for three pumps is properly defined. But when only one pump is running, the operating point (Point 5 in fig. 3.35) is wrong, because the hydraulic profile in fig. 3.36. is below the high point in the pipeline. The air-vacuum valve will open, and flow will stop until the pump increases the

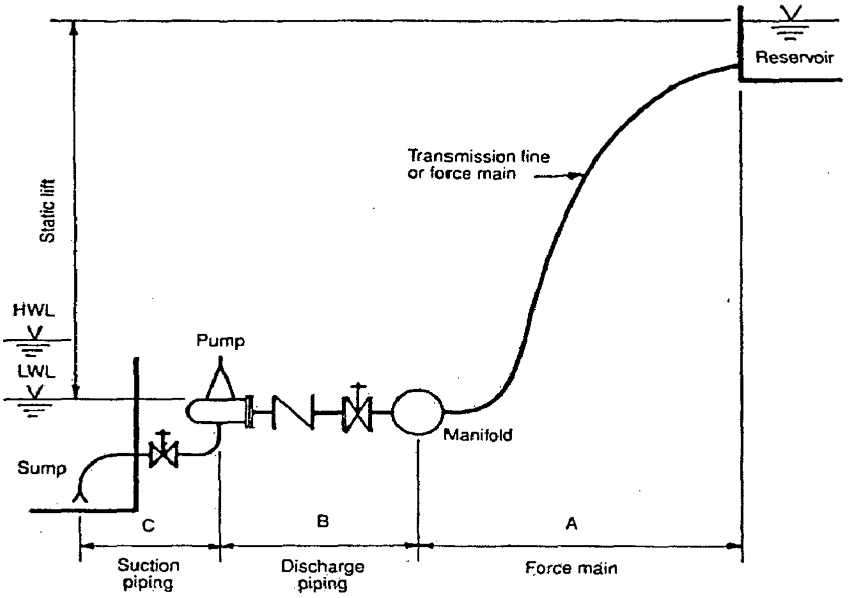


Fig. 3.34. Schematic diagram or pump and piping system. One of four (three duty) pumps is shown.

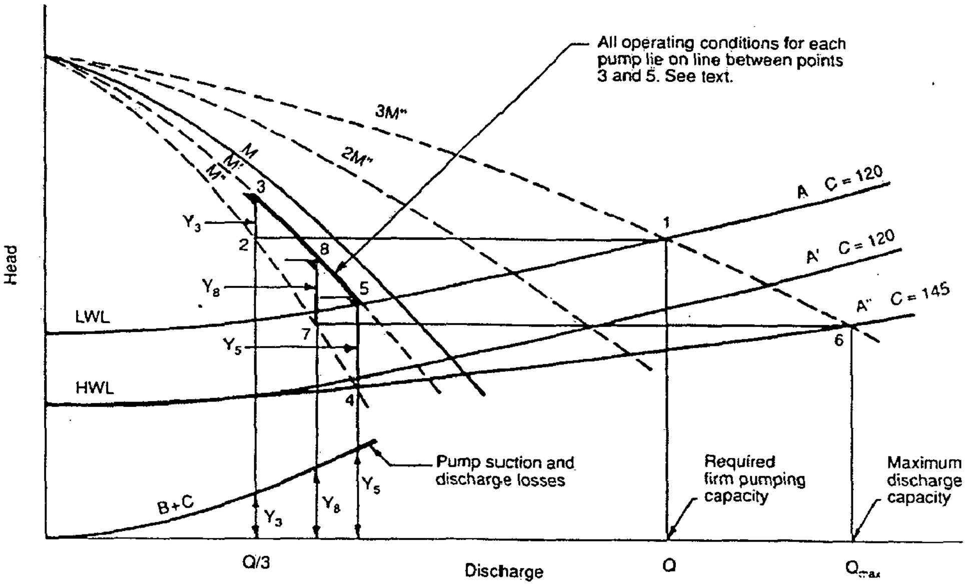


Fig. 3.35. System H Q curve analysis for three C/S duty pumps as per Fig. 3.34.

pressure enough to send water over the high point. So by plotting hydraulic profiles, it can be seen that the transmission pipeline H-Q curve must be raised, as shown in fig. 3.37, to accommodate a new static head. The true flow rate is less than the flow rate obtained from fig. 3.37, so the system will not perform as intended. Blunders of this sort are unlikely to inspire client confidence.

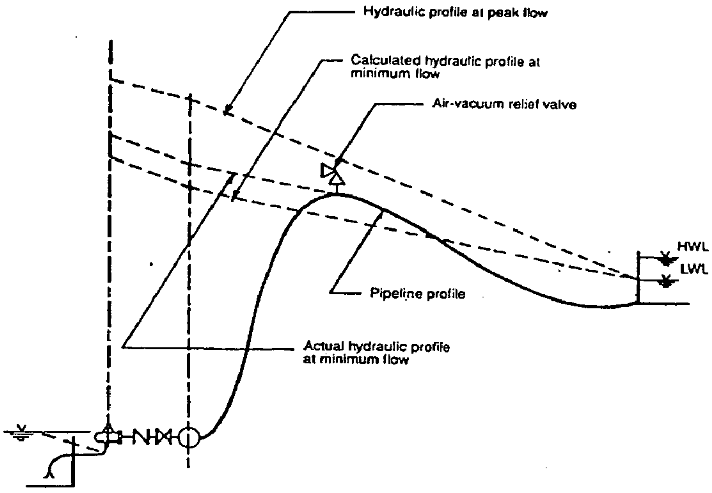


Fig. 3.36 Hydraulic profiles for Figure 3.37.

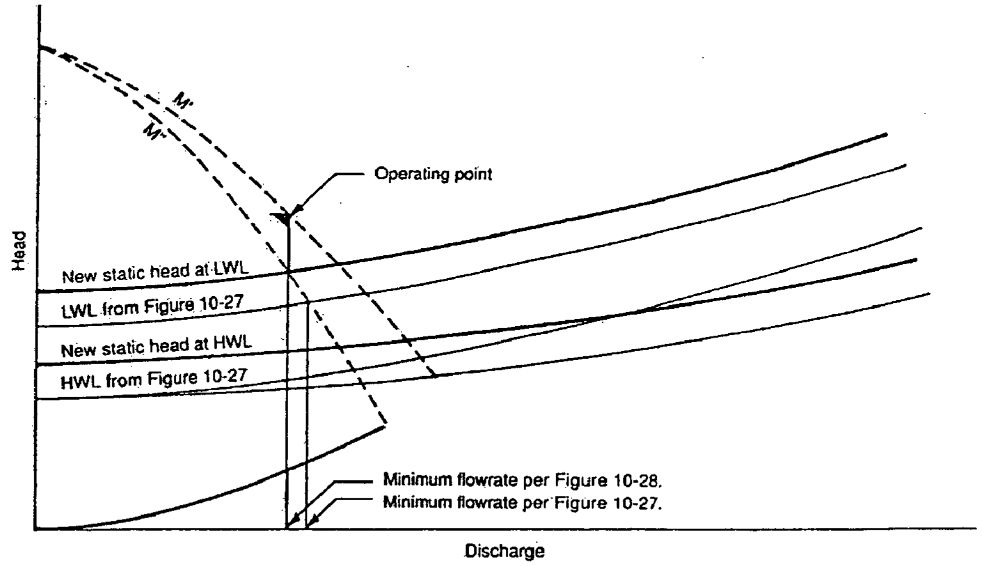


Fig. 3.37 System curve analysis for one pump operating at minimum flow as per figure 3.36 Compare with figure 3.35.

3.3. SELECTION OF PUMPS

3.3.1. INITIAL SCREENING

Before the process of selecting pumping equipment can begin, many factors relating to the application must be established. As these factors are defined, one inevitably begins to identify the type and size of equipment that will be most suitable. Initial selection factors, listed generally in order of precedence, include:

- Required design capacity (maximum flow rate)
- Operating conditions (head, maximum and minimum flowrates, submergence, and/or NPSH).

- Once these factors have been evaluated and the initial decisions have been made, the following factors must be considered:
- Mode of operation (such as in-line pumping, pumping from a well, or pumping from a clear well to a reservoir/tank)
- Type of driver (motor or engine, constant or variable speed)
- Station location, configuration, and constraints (such as the number of pumps, horizontal or vertical pumps, unit in parallel or series, wet Well or dry Well pumps, and submersible pumps).

The step, taken to complete the initial screening process are described in the following subsections. By necessity, this process is presented in a stepwise progression. In fact, it is an iterative procedure, with trade-offs from the ideal until the apparent optimum selection is found.

3.3.2. REQUIRED DESIGN CAPACITY

The required design capacity (both initially and at a future date), including the maximum, normal, and minimum flows to be pumped, must be considered when selecting size of pumping equipment.

Unless the station is intended to accommodate a wide range of flows (caused, perhaps, by substantial further growth in the service area, the following advice, derived from experience, is useful.

- Try to accommodate the peak demand with two or three duty pumps.
- Try to accommodate the normal demand with one duty pump
- Try to limit the number of pump sizes. To reduce the inventory of spare parts, one size is best. Two size are acceptable.

These considerations. Which keep the station size to a minimum, must be balanced against initial requirements. In some instances, the best solution may be to install small pumps initially that are to be replaced at some future date with larger equipment instead of more pumps. Note that the largest capital cost item is the structure itself (e.g., excavation, concrete, ventilation). In most instances, minimizing the number of pumps minimize the capital cost of the station. If smaller pump, are used initially, the suction and discharge connections should be sized properly for the future unit, use reducer as necessary for the smaller original unit. Full-size drivers and starting equipment may be preferable for the initial pumping equipment. In any event, remember to allow space for future equipment and plan how it will be installed.

Once the preliminary pump size, have been selected, the next step is to examine the complete range of operating condition, to be imposed on the equipment.

3.3.3. OPERATING CONDITIONS

The full range of operating conditions should be examined to understand the application completely. Operating conditions (minimum flow, minimum/maximum discharge heads. NPSH and/or submergence limitations, and other requirements) may dictate pump selection. A requirement for start-up and shut-down against a closed valve eliminated the pumps from consideration at a water pumping station because of the water hammer while the valve is closed suddenly. Operation against a closed valve would require larger motors, switchgear, conductors, etc. Plotting the system characteristics for all anticipated operating conditions is most helpful and is considered mandatory regardless of station size.

3.3.4. STATION LOCATION AND CONFIGURATION

Various aspects of the site selected for a pumping station may force one pump option to be favored over all others. Site considerations may include the following:

- *Size:* A small site may require the use of vertical pump, (such as vertical turbines in barrels) to reduce the size of the station floor plan.
- ***Hydraulic profile:*** The required NPSH (NPSHR) at the pump's (suction) inlet may result in a deep station. thereby discouraging the use of horizontal pumps in a dry well because of the cost associated with a large floor plan. Conversely, if a station can be shallow, horizontal pumps tray be preferred because of the prospect for reduced: (1) vibration. (2) headroom required for a crane, and (3) height of superstructure.
- ***Environment:*** Weather condition, may rule out equipment that could otherwise be mounted outdoors. Concern over noise emissions may eliminate drivers or gear boxes mounted at or above grade, and submerged motors (which are quiet) might be favored.

3.3.5. FINAL SELECTION

Final selection involves a review in more critical detail of the considerations used in the initial screening process as well as detailed discussions with the selected manufacturers and the owner. Specific actions to precede a final selection include the following:

- Perform a detailed evaluation of system hydraulics and consider the performance of candidate pumps at all possible operating condition. Furnish each candidate manufacturer with the details of the hydraulic requirements (including, the system curve), the preliminary station layout, and preliminary specifications. Ask for written comments and a proposal specific to the project with budget equipment prices.
- Show the representatives the station layout with a rough construction cost estimate and the preliminary equipment selections.
- Once comment, have been received from the owner and or pump manufacturers. the final selection of equipment, layout of the pumping station and writing of detailed pumping unit specifications call begin.

CHAPTER

4

Hydraulic of Flow In Pipe Networks

CHAPTER – 4

HYDRAULIC OF FLOW IN PIPE NETWORKS

Hydraulics presented in this chapter is limited to turbulent flow of water in closed conduits (pipes) flowing full. Pipes are connected together in various configurations (called networks) to transport water from the supply to the user. When all pipes are connected in series, the system is called a pipeline. A pipe system can include many pipes of various lengths and diameters along with valves to control the flow rate and pumps to convert between electrical energy, mechanical and hydraulic energy.

Flow in a pipe can either be steady or unsteady. For unsteady flow, the velocity is a function of time and will change within a few seconds. For steady flows the velocity is not considered a function of time, although it may change gradually. Steady flow equations can be used to analyze a water distribution system where the demands on the system change hourly, pumps turn on and off, and storage tanks fill and drain during the numerical simulation.

BASIC EQUATIONS FOR STEADY FLOW

The hydraulics of steady flow in pipe systems is described by the continuity and energy equations. Headloss is a key term in the energy equation.

4.1 CONTINUITY EQUATION

For steady flow in pipelines and pipe networks, water is considered incompressible, and the conservation of mass equation (continuity equation) reduces to the volumetric flow rate (Q)

$$Q = AV \tag{4.1}$$

where A is the cross-section area of the pipe, and V is the average velocity. The flow rate (Q) is measured in cu m per sec (cms) or cu ft per sec (cfs). Discharge rate (Q) may also be specified in liters per second (lps), gallons per minute (gpm), or million gallons per day (mgd). The continuity equation between cross-sections 1 and 2 of a pipe is

$$A_1 V_1 = A_2 V_2 \qquad (4.2)$$

Junction nodes are located where two or more pipes join together. A three pipe junction node with a constant demand (C) is shown in Fig. 4.1. The continuity equation for the junction node is

$$Q_1 - Q_2 - Q_3 - C = 0 \qquad (4.3)$$

In modeling pipe networks, all demands on the system are located at junction nodes, and the flow in pipes connecting nodes is assumed to be uniform. If a major demand is located between nodes, then an additional junction node is established at the location of the demand. Equation 4.2 serves as the continuity equation for a two-pipe junction node without a demand, where the subscripts refer to the pipe number.

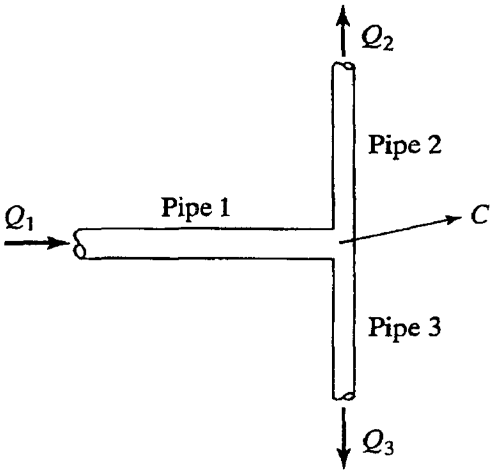


Fig. 4.1. Three-pipe junction node with a constant demand.

4.2 ENERGY EQUATION

In Fig. 4.2, water is being pumped from the lower supply reservoir through a pipeline to an upper storage reservoir. In Fig. 4.2, water is being pumped from the lower supply reservoir through a pipeline to an upper storage reservoir.

The energy grade line *(EGL)* and hydraulic grade line *(HGL)* are shown in Fig. 4.2. The *HGL* is located one velocity head ($V^2/2g$) below the *EGL*. The *EGL* and *HGL* are parallel when the pipe size is uniform. The *EGL* slopes down ward in the direction of flow because of energy loss. The vertical distance between the center of the pipe and the *HGL* is the pressure head (P/γ). If the *HGL* is above the pipe, the pressure is positive, and if the *HGL* is below the pipe, the pressure is negative. Z is the vertical distance above the datum (usually mean sea level-msl).

The energy equation written for flow in a pipeline (Fig. 4.3) is

$$\frac{V_1^2}{2g}+\frac{P_1}{\gamma}+z_1+E_P=\frac{V_2^2}{2g}+\frac{P_2}{\gamma}+z_2+H_L \qquad (4.4)$$

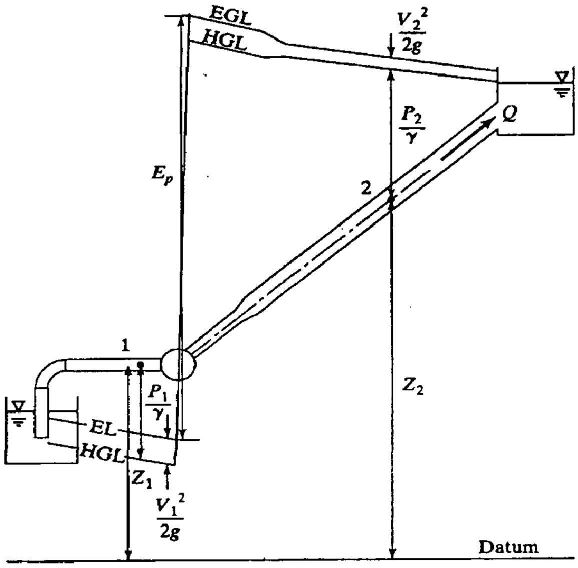


Fig. 4.2 Pump system.

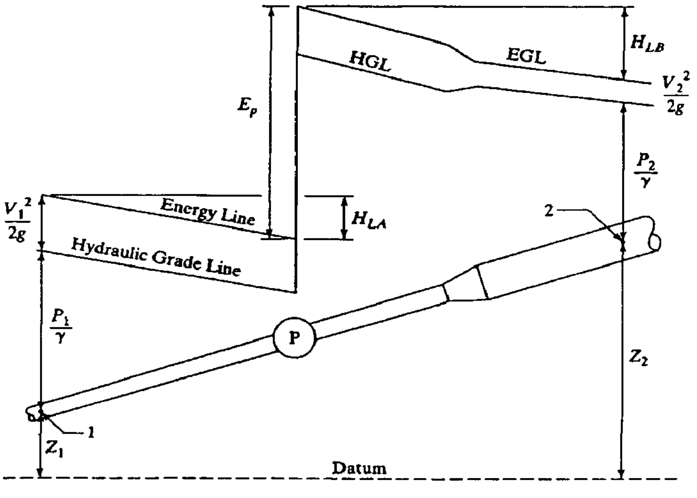


Fig. 4.3 Energy equation for pipeline flow.

where z is the elevation of the pipe, P/γ is the pressure head, $V^2/2g$ is the velocity head, E_P is the energy head added by the pump, and H_L is the total headloss between points 1 and 2. Each term in the energy equation has units of length and represents energy per unit weight of fluid (Newton-meters per Newton of fluid flowing or ft-lbs per lb).

4.3 HEADLOSS

Headlosses in pipelines are caused by pipe friction, transitions, valves, bends, and fittings. For long pipelines, pipe friction is generally the major component of headloss and the other components are often neglected. Headlosses caused by transitions, valves, bends, and fittings are referred to as minor losses and in short pipelines such as highway culverts cannot be neglected.

4.3.1 PIPE FRICTION HEADLOSS.

Headloss caused by pipe friction can be estimated using the Darcy-Weisbach equation, Hazen-Williams equation, or the Manning equation. The Darcy-Weisbach equation is

$$H_L = f\frac{L}{D}\frac{V^2}{2g} \tag{4.5}$$

where f is the friction factor, L is the length of the pipe, and D is the diameter of the pipe. The friction factor can be estimated from the Moody diagram in Fig. 4.4 and is a function of the Reynolds number ($Re = DV/\nu$) of the flow and the relative roughness of the pipe (ε/D). In most pipelines, the flow will be fully turbulent.

The hydraulic radius (R) is defined as the area (A) of the flow cross-section divided by the wetted perimeter (P). For a circular pipe flowing full

$$R = \frac{A}{P} = \frac{\frac{1}{4}\pi D^2}{\pi D} = \frac{1}{4}D \tag{4.6}$$

For noncircular pipes, the headloss can be estimated using the equations for circular pipe with 4R substituted for D. Pipe roughness values are listed in Table 4.1 for common pipe materials.

After a pipe has been in service for some time, the diameter and roughness of the pipe may change, and it may be difficult to estimate the roughness of the pipe. The Hazen-Williams equation is often used in pipe network analysis. Tables are available relating the Hazen-Williams coefficient (C_H) to the age of the pipe. The Hazen-Williams equation is

$$Q = C_w C_H A R^{0.63} S^{0.54} \tag{4.7}$$

where C_w = 0.85 for International System (SI) units [1.318 for British Gravitational (BG) units] and S is the slope of energy line. Writing the Hazen-Williams equation for headloss gives

$$H_L = S \times L = L\left(\frac{4}{D}\right)^{1.17}\left(\frac{V}{C_w C_H}\right)^{1.85} \tag{4.8}$$

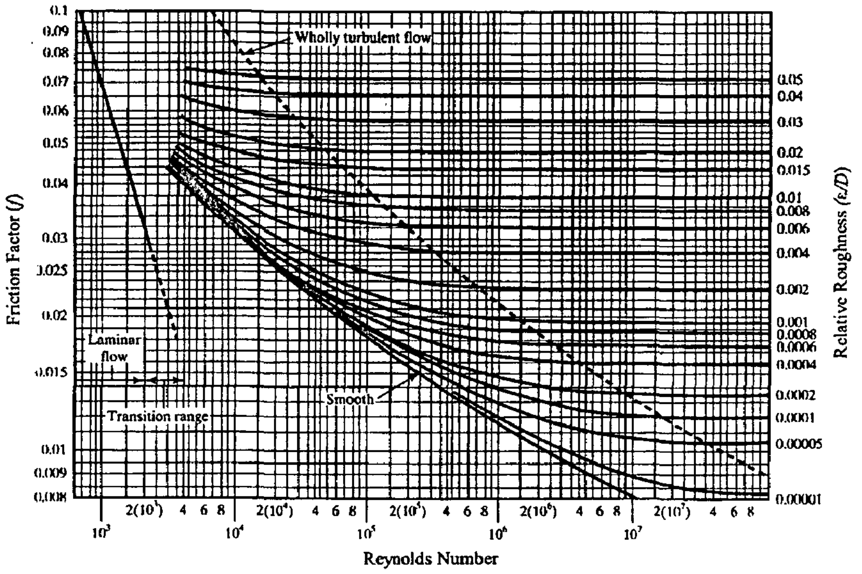


Fig. 4.4 Moody diagram of Darcy-Weisbach friction factors.

Table 4.1. Roughness Values

Material	ε mm	C_H Hazen-Williams	n Manning
Plastic, PVC	0.001	150	0.009
Asbestos cement	-	140	0.011
Welded steel	0.045	120	0.012
Riveted steel	0.9-9	110	0.015
Concrete	0.3-3	130	0.012
Asphalted iron	0.12	-	0.013
Galvanized iron	0.15	-	0.016
Cast iron (new)	0.25	130	0.013
Cast iron (old)	-	100	0.025
Corrugated metal	-	-	0.025

The Manning equation is commonly used to estimate the friction headloss in culverts and storm sewers. The Manning equation is

$$Q = \frac{C_m}{n} A R^{2/3} S^{1/2} \tag{4.9}$$

where C_m = 1.00 for SI units (1.49 for BG units) and n is the Manning roughness coefficient. Writing the Manning equation for headloss gives

$$H_L = S \times L = \frac{n^2 V^2 L}{C_m{}^2 R^{4/3}} \tag{4.10}$$

4.3.2 MINOR LOSSES.

Minor losses are caused by excessive turbulence generated by a change in flow geometry. They represent the headloss that is in excess of the normal pipe friction at transitions, bends, valves, and other fittings. The coefficient (K) is used to give the minor headloss (H_M) as a function of the velocity head

$$H_M = K \frac{V^2}{2g} \tag{4.11}$$

At transitions V is the velocity in the smaller pipe. Minor loss coefficients are listed in Appendix-6

4.4. TYPES OF NETWORKS

4.4.1. SERIAL NETWORK

A serial network is a network having no branches or loops (Fig. 4.5.). It is the simplest of all the types of pipe networks. Generally, it has one source, one sink, and one or more intermediate nodes. The first link starts at a source and the last link ends on a sink. All the intermediate nodes are connected by two links, a supply link on the upstream and a distribution link on the downstream. The direction of flow in all the links is fixed and is along the direction from the source to the sink. A path is a serial network.

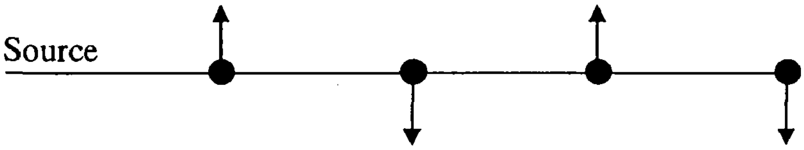


Fig. 4.5. Serial Network.

4.4.2. BRANCHING NETWORK

A branching or a *dead-end* network is a tree-like network without any loops (Fig. 4.6.). It consists of several serial networks. Usually it has one source, one or more intermediate nodes, and more than one sink. Each intermediate node is connected on the upstream by one supply link and on the downstream by one or more distribution links. The direction of flow in all the links is fixed and is along the direction from the source to several sinks.

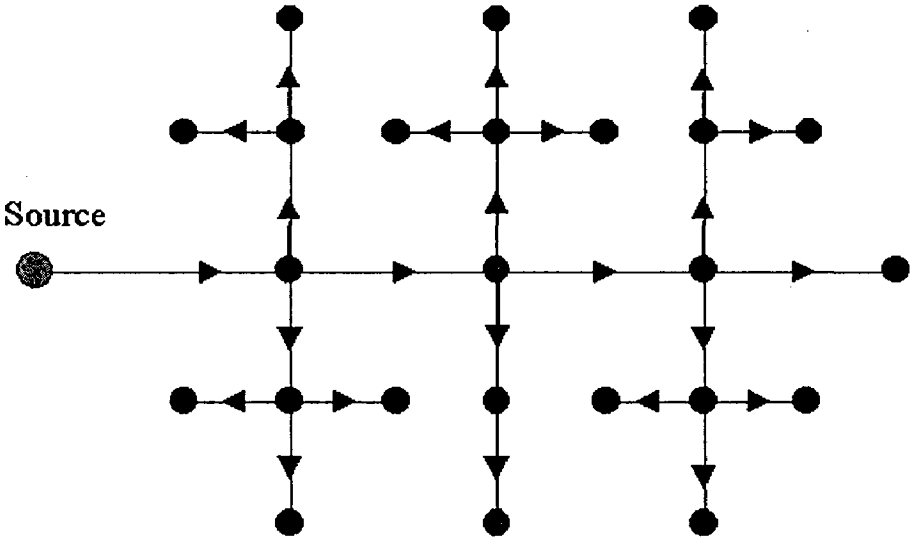


Fig. 4.6. Branching Network

Branched networks are relatively easy to design and are less costly to construct in small systems. However, they are less reliable, as any break or maintenance work in the network results in cutting off the supply to part of the system and the dead ends i mpair water quality. A network can be started with a branched system, with the ends of the branches "looped" at a later time to improve flow conditions and provide greater reliability.

4.4.3. LOOPED NETWORK

A looped network contains loops (Fig. 4.7.). It must have at least one source as in a single-source network, but may have more sources as in a multi-source network. A looped network must have at least one sink. (Why ?) The node that serves as the sink depends upon the nodal demands. For example, node F in Fig. 4.7. is a sink. However, if demand at D increases, the flow direction in links DE and DF may reverse and node D would behave as a sink. Although the direction of flow in a link is fixed for a particular demand pattern, it may change for another demand pattern. Thus, for a node, a link which behaves as a supply link for one demand pattern may behave as a distribution link for another demand pattern. Similarly, when the direction of flow changes in a link, the upstream node becomes the downstream node and vice versa.

Pipe networks seldom consist of loops only. In practice, they are of composite type consisting of loops, branches, and serial parts. However, it is usually possible to reduce a composite network to a looped network for analysis, and therefore composite networks are referred to as looped networks.

Looped networks are common in densely populated areas. They provide a greater reliability of service than branched systems, eliminate dead ends, and improve flow conditions during periods of high local demand. Looped networks may be designed based on a main loop which feeds into secondary pipes or based on a branched network with several interconnections resulting in loops. Looped networks require more valves and fittings (Fig 4.8); the omission of interconnections reduces the number of valves (Fig. 4.9) at the price of reliability.

The cost of the distribution network depends mainly on the total length of pipes installed. Therefore the layout of the network should be carefully planned and the future development of the service area should be considered.

In serial and branching networks, only one path is available for transporting water from a source to a particular node. Thus, if a link of such a network is closed for repairs or replacement, all demand nodes situated downstream of this link are completely cutoff from the source. Therefore, supply through a serial or branching network is less reliable when compared to that through a looped one where alternate paths exist. However, serial and branching networks are cheaper and usually easier to analyze than looped networks. Serial and branching networks are used for industrial water supply purposes, sprinkler irrigation projects, and for distribution of water to small communities in rural areas.

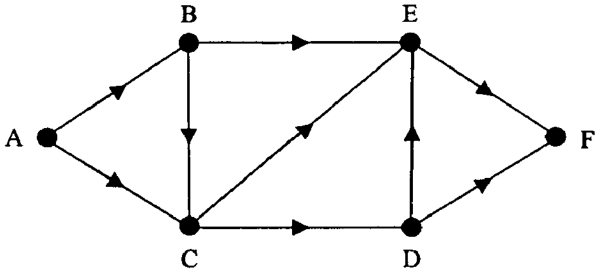


Fig. 4.7. Looped network.

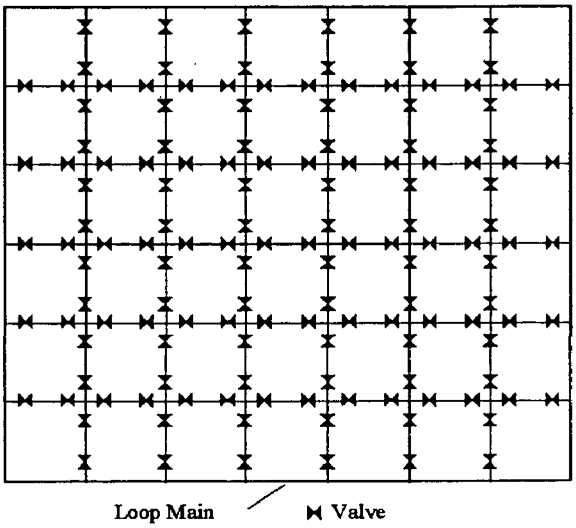


Fig. 4.8. Fully Interconnection Pipes.

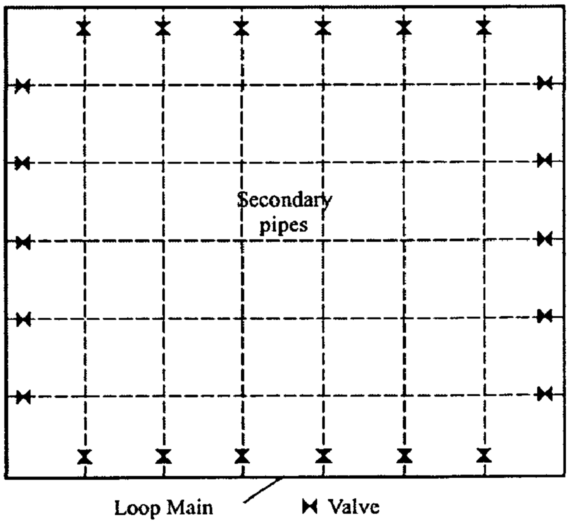


Fig. 4.9. Over Crossing Single Pipes.

Looped networks, although costlier and comparatively more difficult to analyze, arc more reliable than serial and branching networks, and therefore are used for water distribution to large communities areas.

Because serial and branching networks have similar analyses, the term branching networks will be used to include serial networks in this dissertation.

4.5. HYDRAULIC SIMULATION USING EPANET 2.0

4.5.1. DESCRIPTION

EPANET is a Windows 95/98/ME/NT program that performs extended period simulation of hydraulic and water-quality behavior within pressurized pipe networks. A network can consist of pipes, nodes (pipe junctions), pumps, valves and storage tanks or reservoirs. EPANET tracks the flow of water in each pipe, the pressure at each node, the height of water in each tank, and the concentration of a chemical species throughout the network during a simulation period comprised of multiple time steps. In addition to chemical species, water age and source tracing can also be simulated.

The Windows version of EPANET provides an integrated environment for editing network input data, running hydraulic and water quality simulations, and viewing the results in a variety of formats. These include color-coded network maps, data tables, time series graphs, and contour plots.

EPANET was developed by the Water Supply and Water Resources Division (formerly the Drinking Water Research Division) of the U.S. Environmental Protection Agency's National Risk Management Research Laboratory.

4.5.2. CAPABILITIES

EPANET provides a fully-equipped, extended period hydraulic analysis package which can:

- handle systems of any size
- compute friction head loss using the Hazen-Williams, Darcy-Weisbach, or Chezy-Manning formulas
- include minor head losses for bends, fittings, etc.

- model constant or variable speed pumps
- compute pumping energy and cost
- model various types of valves including shutoff, check, pressure regulating, and flow control valves
- allow storage tanks to have any shape (i.e., diameter can vary with height)
- consider multiple demand categories at nodes, each with its own pattern of time variation
- model pressure-dependent flow issuing from emitters (sprinkler heads)
- base system operation on simple tank level or timer controls as well as on complex rule-based controls.

In addition , EPANET's water quality analyzer can:

- model the movement of a non-reactive tracer material through the network over time
- model the movement and fate of a reactive material as it grows (e.g., a disinfection by-product) or decays (e.g., chlorine residual) with time
- model the age of water throughout a network
- track the percent of flow from a given node reaching all other nodes over time
- model reactions both in the bulk flow and at the pipe wall
- allow growth or decay reactions to proceed up to a limiting concentration
- employ global reaction rate coefficients that can be modified on a pipe-by-pipe basis
- allow for time-varying concentration or mass inputs at any location in the network
- model storage tanks as being either complete mix, plug flow, or two-compartment reactors.

EPANET's Windows user interface provides a visual network editor that simplifies the process of building piping network models and editing their properties. Various data reporting and visualization tools are used to assist in interpreting the results of a network analysis. These include graphical views (time series plots, profile plots, contour plots, etc.), tabular views, and special reports (energy usage, reaction, and calibration reports).

4.5.3. APPLICATIONS

EPANET was specifically developed to help water utilities maintain and improve the quality of water delivered to consumers through their distribution systems. It can be used to design sampling programs, study disinfectant loss and by-product formation, and conduct consumer exposure assessments. It can assist in evaluating alternative strategies for improving water quality such as altering source utilization within multi-source systems, modifing pumping and tank filling/emptying schedules to reduce water age, utilizing booster disinfection stations at key locations to maintain target residuals, and planning a cost-effective program of targeted pipe cleaning and replacement.

EPANET can also be used to plan and improve a system's hydraulic performance. Pipe, pump and valve placement and sizing, energy minimization, fire flow analysis, vulnerability studies, and operator training are just some of the activities that EPANET can assist with.

CHAPTER

5

Pumping System Analysis

CHAPTER – 5

PUMPING SYSTEM ANALYSIS

5.1. INTRODUCTION

In this chapter, hydraulic simulation using EPANET will be discussed to determine optimum number of pump, capacities, and energy consumed, etc. The example network used to illustrate applications of model consists of 17 demand nodes with demand elevations EL. Four loading condition are implemented in the model by multiplied those loading condition with certain factor. The minimum head H required to satisfy the reliability are settled. A design period (year) of the pumping system will vary depend on the number of pumps combinations used with all the operation costs, depreciation cost, investment cost with salvage value, maintenance and repair cost, in any of these life time (years) converted into present worth using interest rate of 8%. Pumping system with 2(1+1), 3(2+1), 4(3+1), 5(4+1) and 6(5+1) units of pump implemented to define optimum result of the cost and reliability.

5.2. WATER DISTRIBUTION NETWORK

A water distribution system should be able to provide, during its entire life, the required quantity of water for expected loading conditions with the desired residual pressure at all the demand nodes. In other words, hydraulic reliability, as it relates to water distribution system design, can be defined as the ability of the system to provide service with an acceptable level of interruption in spite of abnormal conditions. The evaluation of hydraulic reliability relates directly to the basic function of supplying specified quantity of water to the appropriate place at the required time under desired pressure, in this case the minimum pressure at the remotest nodes is 15 m continuously for all the 24 hours of the day.

Hydraulic failure occurs when the system cannot supply the expected amount of water to the specified location, at the specified pressure and at the specified time. The specified amount of water based on the maximum hourly demand plus needed fire flow. The specified pressure (minimum) in this study defined as 15 m water column. As the respected demand conditions may vary with space and time, there is neither a common acceptable definition of hydraulic reliability nor a consensus among researchers and practicing engineers regarding the factors that should be considered in estimation of the hydraulic reliability of a water distribution system. Network reliability factor can be defined as an average of the nodal reliability in the system.

The flows are associated with links (pipes and pumps), the demands and pressure heads at the nodes. Demands, in reality, are spread over the area, and only for modeling these are lumped and assigned to nodes. Hamberg and Shamir (1988) have suggested a schematization of discharge distribution and calculated the equivalent uniform discharge in the pipe based on the concept of equivalent head loss for several demand patterns. However, these equations can be used for reducing the size and detailing of the network model but cannot be applied in all parts of an entire system. This is mainly because of the reason that the considerable amount of work is needed to reach a substantially reduced model size this approach and some of the equivalence transformation equations depend upon the relative magnitude of flows in the network. Another drawback of this approach is lumping of demands in terms of equivalent discharge at the ends may be correct from the headloss criterion but cannot be justified from continuity point of view. Thus it is always preferred to lump the summation of individual distributed demand at the end node of the element as it will be more logical from continuity point of view and will be on the safer side from head loss criterion.

5.3. CAPACITY OF THE SYSTEM

Based upon the estimated per capita consumption and the design population, plus the demand for institutions, commercial establishments, factories and irrigation, the design average demand can be calculated. Based upon the considerations, the design year, average demand, maximum-day demand, and peak or maximum hour demand should be calculated. Prior to determining the peak demand, a decision needs to be made whether the system is to provide for fire fighting. The capacity of most of the system is based-on the maximum day demand.

Only the distribution system, including service reservoirs (overhead tank) is designed for the peak demand, which includes fire protection where this is to be provided. If an impounding reservoir is required on a stream, its capacity is based on the ultimate average demand. The design population should be about 25% greater than present population. Based on a design population of 3000, and 80 liter per connections per day on maximum day (dry season), the design capacity of raw water pumps and treatment plant should be 240 m^3/day. A design capacity of 15 m^3/hr would be satisfactory, as this would permit 16 hours operation. For other populations, the design capacity would be proportional capacities up to 25% greater would be acceptable without additional review.

Storage of 20 to 30 m^3 per 1000 population should be available in the system in a clear well or in an elevated tank. If elevated ground is available near the network, elevated storage in a ground storage tank should be provided. If the land is level, an elevated water tower is desirable or standby power should be available to provide continues water pressure, elevated water towers should be at least 3 m and not more than 5 m above the tallest structure. The demand capacity can be seen in the Table 5.1 and Figure 5.1.

5.3.1. FACTORS INFLUENCING CAPACITY

The capacity of a water supply system is based on anticipated water demand. In regions where water resources are inadequate, and where communities are without water service and without sufficient financial resources to provide a system that can meet all anticipated demands, the capacity to be provided will be governed to a large extent by the cost of providing a system. Where water of good quality is plentiful and easily developed, and where financial resources are ample, the capacity to be provided may be much greater.

Other factors that influence the capacity of the system are (a) the design period, which fixes the target date in the future for which the project must serve; (b) the population at the target date; (c) requirements for institutions, commerce, industry, and agriculture; (d) the level of service to be provided, whether full-service house connections, yard taps, or public standposts; and (e) climatological variations.

5.3.2. DESIGN PERIOD

A decision must be made as to the target year that is to be the basis for design. The optimal design period for urban water systems in developing countries is generally about 7 to 8 years. In addition, 2 to 3 years must be allotted for lead time, the time for the project to be planned, designed, constructed, and placed into operation, making a total of about 10 years from the initiation of planning. The design period may be modified by the following factors:

(a) The design period of elements of a system can be reduced if they can be easily enlarged. For example, adding another line can increase water transmission capacity, or adding another module can enlarge a treatment plant, so the first units can be designed for a shorter period. However, a dam or an intake is not easily enlarged, so they should be designed for their full useful life.

(b) The economy of scale of the system or an element of the system can affect the design period. If increasing the capacity of an element of a system 50% only increases the cost 10%, which is often the case with pipelines, where the costs of excavation, installation and backfilling are little affected by the size of the pipes, then the pipe may be designed for a capacity required further into the future.

(c) The design period might be reduced if the growth of the community is expected to be rapid. For example, if a community is expected to increase three-fold in 10 years, the cost burden to build for a 10-year period would be very heavy on the present residents of the community. A shorter design period would permit the new residents to participate financially in the system expansion. On the other hand, if the community is not expected to grow, the additional cost of providing for a longer design period would be small. Just only for the pumping systems, the period depends on manufactory recommendation.

(d) A high rate of interest for the money to be borrowed tends to reduce the design period because the higher cost of money will be a heavier burden,

(e) The useful life of the component structures and equipment can affect the design period of the elements of the system. A pump with an expected life of 20,000 years should be designed for that period of useful life. If it were designed with a smaller capacity, it would need to be replaced before it is worn out. If it were designed with a larger capacity, it would be worn out before its full capacity would be needed.

Accordingly, design periods for major elements of the project should be selected at the very start of the planning, so that the population to be served at the end of the design period, upon which the size of the project is to be based, can be decided.

In general, the useful life of individual system components has been used in this study in term of working hour; the life of the pumps is taken as 20,000 hrs after reference Mahesh Varma [12].

5.3. PHYSICAL COMPONENTS IN WATER DISTRIBUTION SYSTEM

In general, a WDS is modeled as a collection of links connected to nodes in some specified branched or looped configuration. The links represent pipes, pumps, and control valves. The nodes represent junctions, tanks, and reservoirs. The example network used to illustrate applications is shown in Fig. 5.1. This network consists of 17 demand nodes, with the demands and node elevations listed in Table 5.1. A pump used as device to transport water from source (reservoir) to the Distribution Network and that raises the hydraulic head and flow of water to fulfill demand requirement.

Table 5.1. Network Characteristics

Network Table - Nodes			Network Table - Links				
Node ID	Elevation m	Base Demand CMD	Link ID	Length m	Diameter mm	Roughness	Status
Junc. 1	180	-14300	Pipe 1	1400	350	100	open
Junc. 2	178	600	Pipe 2	1700	300	100	open
Junc. 3	179	1000	Pipe 3	1000	300	100	open
Junc. 4	180	900	Pipe 4	900	250	100	open
Junc. 5	181	1200	Pipe 5	1350	200	100	open
Junc. 6	183	900	Pipe 6	900	300	100	open
Junc. 7	182	800	Pipe 7	1100	250	100	open
Junc. 8	181	800	Pipe 8	1400	200	100	open
Junc. 9	180	1200	Pipe 9	900	200	100	open
Junc. 10	182	1200	Pipe 10	1000	200	100	open
Junc. 11	181	600	Pipe 11	1200	150	100	open
Junc. 12	181	800	Pipe 12	1100	300	100	open
Junc. 13	183	1200	Pipe 13	800	200	100	open
Junc. 14	184	800	Pipe 14	1400	200	100	open
Junc. 15	179	800	Pipe 15	800	250	100	open
Junc. 16	180	600	Pipe 16	1100	200	100	open
Junc. 17	181	900	Pipe 17	1200	250	100	open
-	-	-	Pipe 18	800	200	100	open
-	-	-	Pipe 19	900	150	100	open
-	-	-	Pipe 20	1400	200	100	open
-	-	-	Pipe 21	1200	150	100	open

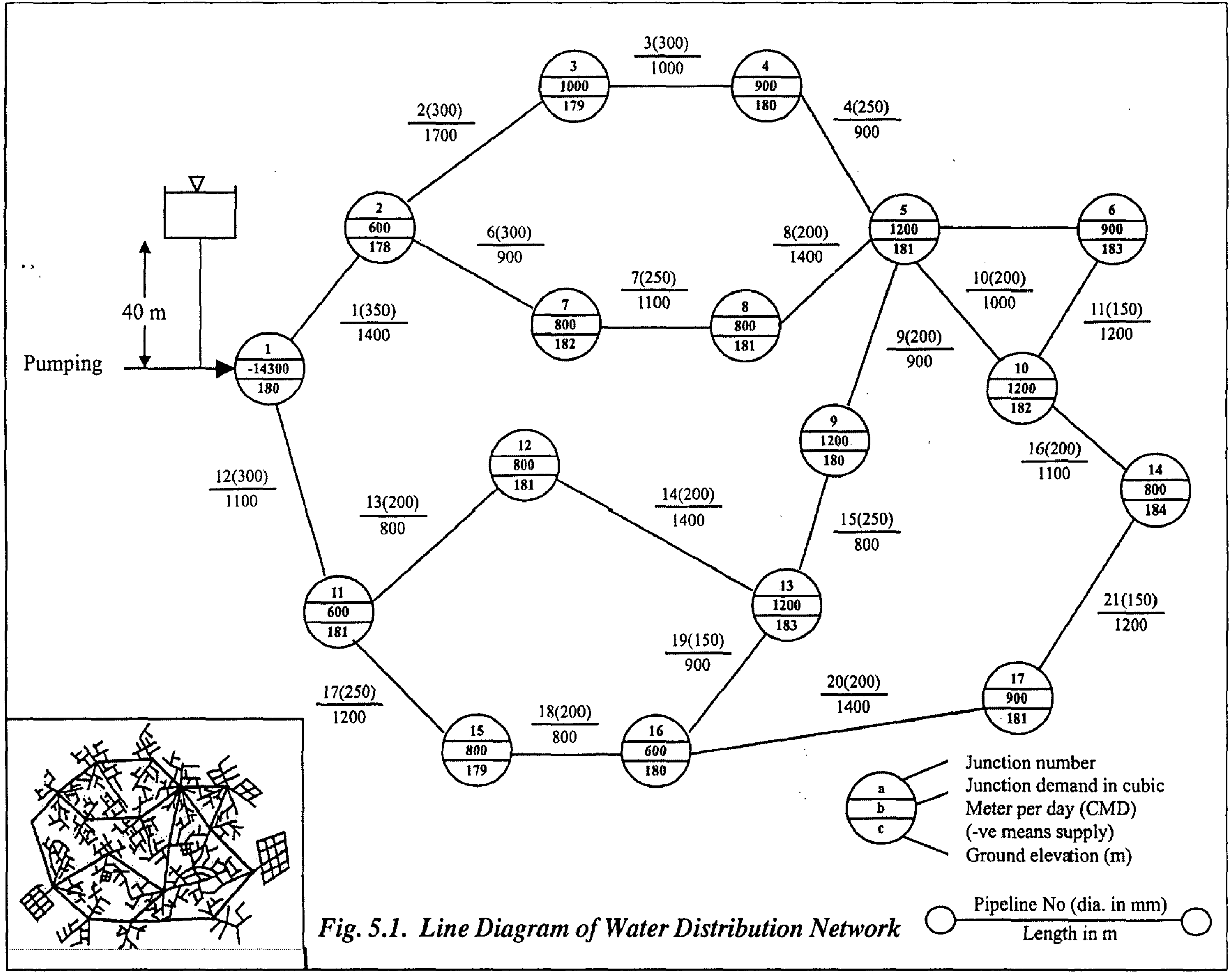


Fig. 5.1. Line Diagram of Water Distribution Network

5.~~4~~3.1. PUMPS

Pumps selected based on the manufacturers published design data and must able to create flow and head as required by the demand in this case the pump head specified by 50 m while flow selected as according to the number of pump chosen and the pumps must be sized for peak hour loading condition. The type of pump to be used is *centrifugal* pump (single-stage) because of their low cost, simplicity, and reliability in range of flows and head encountered.

MULTIPLE OPERATIONS

Pumps are links that impart energy to a fluid thereby raising its hydraulic head. Parallel combinations are commonly used for lifting water that varies in quantity (from minimum to maximum demand) and head (source at high level and service reservoir empty to vice versa). In this study, the pumping stations with at least two pumps must be available; each of adequate capacity operating alone, so that one is available for standby. They should be operated alternatively. In the simulation one or combination of two, three, four or five number of identical pumps is operated in parallel are implemented to develop flow and head required with pressure at the remotest of nodes not less than 15 m. When two or more pumps are discharging into the same header or manifold, the total flow is found by adding the individual flow at given head. A more extensive explanation for multiple pumps is given in chapter 3. In lieu of a pump curve, the pump could be represented as a constant energy device, one that supplies a constant amount of energy (horsepower or kilowatts) to the fluid for all combinations of flow and head. Flow through a pump is unidirectional and EPANET will not allow a pump to operate outside the range of its pump curve.

Pumps can be turned on and off at preset times or when certain conditions exist in the network. A pump's operation can also be described by assigning it a time pattern of relative speed settings. Each pump can be assigned an efficiency curve. Pumping for less than 24 hours requires larger pumps, larger storage tanks and greater power for pumping. This is partially offset by reduced operator time.

5.3.2. JUNCTIONS

Junctions are the end-points of links and are used to represent supply points such as reservoirs or tanks, points where pipes changes size, intersection of links and points of demand. In this case there are 17 nodes with demand requirement as shown in Table 5.1. The basic input data required for junctions are:

- elevation above some reference (usually mean sea level)
- water demand (rate of withdrawal from the network)
- initial water quality. (blank ; no water quality analysis is being made)

The minimum head required are 15m for all nodes for all loading conditions.

5.3.3. RESERVOIR

Reservoir is node that represents an external source or sink of water to the network. It is used to model such things as lake, river, groundwater aquifer, and tie-in to other systems. The primary input properties for a reservoir are its hydraulic head (equal to the water surface elevation if the reservoir is not under pressure and measured from sea level). Because a reservoir is a boundary point to a network, its head cannot be affected by what happens within the network. Therefore it has no computed output properties. In this study only one reservoir available with head of 175 m measured from sea level).

5.3.4. OVERHEAD TANK

Tank is node with storage capacity, where the volume of stored water can vary with time during a simulation. The primary input properties for tank (see Fig. 5.2.) are :

- bottom elevation (where water level is zero)
- diameter
- initial, minimum and maximum water levels
- initial water quality. (blank ; no water quality analysis is being made)

Tank is required to operate within their minimum and maximum levels. Pumps will start operation if a tank is reaching at its minimum level (3 m) and stops when it is at its maximum level (8 m), the level measured from bottom of the tank. So different head is 5 m with volume under control 2,454 m^3 or 17% of total demand of 14,300 CMD or adequate for serving 4 hours at full loading condition and the remaining one 2 m level (about 1000 m^3) for anticipate fire fighting so total volume of tank about 25% of the maximum day demand capacity for operation 24-hour would be satisfactory. Tank Pumping for less than 24 hours requires larger pumps, larger storage tanks and greater power for pumping. This is partially offset by reduced operator time.

Tank controlled so that always in a state of full level at midnight, so that the combination of the number and capacity of the pump and scheduling also have to be accommodated to fulfill this rule.

LOCATION AND ELEVATION

Tank location and elevation has affect pipe size and pump requirements. The best choice for service storage location if available, is a ground tank at an elevated location near the community to be served. If the land is flat, the storage tank must be elevated on the tower. It is often convenient to place the tank either at the site of the treatment plant or pumping station (to facilitate and maintenance), or at a central location in the distribution network. If the area to be served stretches over a long distance, with the one end, the reservoir might be built at the other end of town. This allows the tank to be lower and the distribution pipes to be smaller, because water would be fed to consumers from two directions. Such a location provides added reliability in event of breaks in the transmission main.

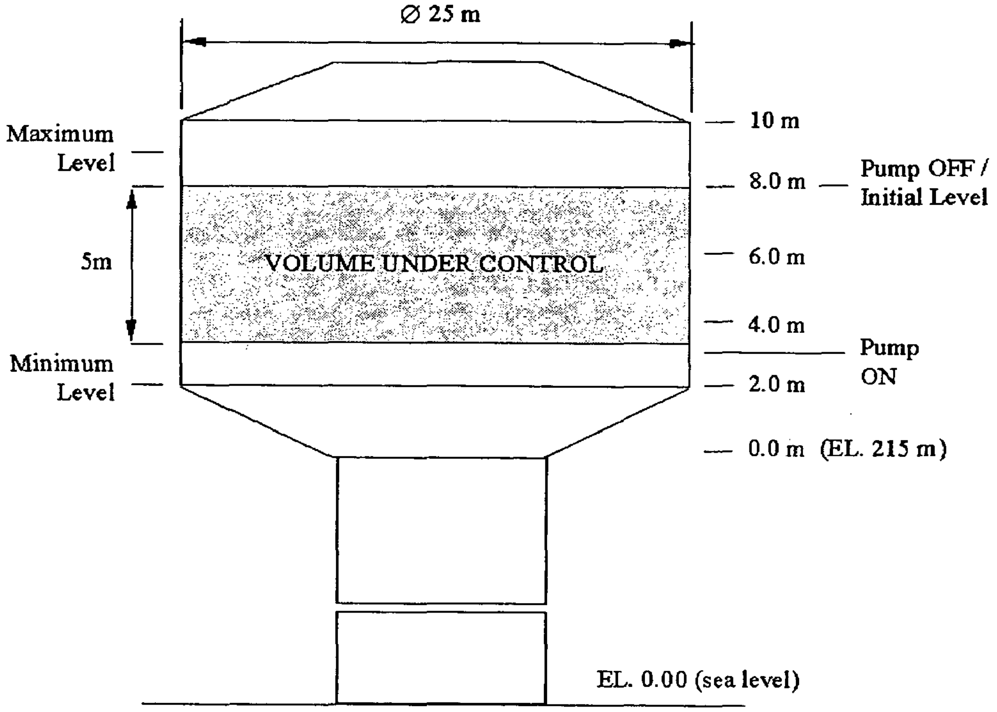


Fig. 5.2. Level of Volume Control for Overhead Tank

The required elevation depends upon the local situation such as the length of pipelines to the furthest service from the tank and the height of buildings to be served. For a village with one-floor houses, the elevations would range from 3 to 10 m. Where there are taller buildings, the tank may need to be higher, although individual tall buildings may be more economically served with their own booster pumps and elevated tanks.

If there is a difference in elevation of 60 meters or more between the water level in the tank and lower points in the system, savings are possible by locating part of the storage in a tank at a lower level, providing a zone of lower pressure. Excessive pressure increases the frequency of breaks and increases leakage significantly and increases power costs as well.

In this study, the overhead tank placed near the pumping station and the bottom elevation of the tank defined 215 m measured from sea level.

5.3.5. PIPES

The sizing of pipes in a distribution network is based on:

(1) the peak demand of the consumers.

(2) the minimum required pressure head to be maintained anywhere in the network, to protect the network from contamination by backflow of wastewater and infiltration of contaminated groundwater; and to supply services. If future extension of the network is planned, the head losses to these sections should be considered.

(3) the available static head provided by the source feeding into the supply system. This source may either be an elevated tank, a ground reservoir at a higher elevation, or a pressure tank fed by a pump.

In principle the pipe sizing is simple: the pressure losses at peak flow should not exceed the difference between the available head and the minimum residual head, however, it is more difficult because the available head may itself be varied by pump selection or height selection of service tanks. Larger pumps and higher tanks permit the use of smaller pipes. In small systems without special conditions, pipe sizing can be based on a velocity in the pipes of about 1 m/s at the design flow.

For a larger system or a system with special conditions (for instance, a supply area spread over a long distance or with large consumers in a certain area), the evaluation of several options with the help of computer-aided design may be considered such as EPANET.

Several programs for microcomputers have been developed to calculate pipe sizes and costs of many possible options in a short time.

Pipes are links that convey water from one point in the network to another. Assumes that all pipes are full at all times and the suitable value of Hazen-William coefficient for each of the pipeline and if not prescribed then a default value of 100 is taken. Flow direction is from the end at higher hydraulic head (internal energy per weight of water) to that at lower head. The number of pipes in the network is 17 lines. The principal hydraulic input parameters for pipes are (the numerical data, see table 5.1.):

- start and end nodes
- diameter
- length
- roughness coefficient (for determining headloss)
- status (open, closed, or contains a check valve).
- The status parameter allows pipes to implicitly contain shutoff (gate) valves and check (non-return) valves (which allow flow in only one direction).

The hydraulic head lost by water flowing in a pipe due to friction with the pipe walls can be computed using one of three different formulas:

- Hazen-Williams formula
- Darcy-Weisbach formula
- Chezy-Manning formula

With the Darcy-Weisbach formula different methods to compute the friction factor f depending on the flow regime:

- The Hagen–Poiseuille formula is used for laminar flow (Re < 2,000).
- The Swamee and Jain approximation to the Colebrook-White equation is used for fully turbulent flow (Re > 4,000).
- A cubic interpolation from the Moody Diagram is used for transitional flow (2,000 < Re < 4,000).

Pipes can be set open or closed at preset times or when specific conditions exist, such as when tank levels fall below or above certain set points, or when nodal pressures fall below or above certain values.

5.3.6. VALVES

Fittings and valves are integral components of a pumping system and influence pump selection and network characteristics. Since fittings and valves serve the control the flow of fluids, they influence NPSH available to the pump, pressure developed in the pump station, and hydraulic transients in the pipeline which may develop. Most valves in a pumping system are for isolation service and, as such, are either open or closed. Check valves respond to flow direction and opened and closed automatically.

Valves are links that limit the pressure or flow at a specific point in the network. Their principal input parameters include:

- start and end nodes
- diameter
- setting
- status.

The computed outputs for a valve are flow rate and headloss. The different types of valves included in EPANET are:

- Pressure Reducing Valve (PRV)
- Pressure Sustaining Valve (PSV)
- Pressure Breaker Valve (PBV)
- Flow Control Valve (FCV)
- Throttle Control Valve (TCV)
- General Purpose Valve (GPV).

PRVs limit the pressure at a point in the pipe network. EPANET computes in which of three different states a PRV can be in:

- partially opened (i.e., active) to achieve its pressure setting on its downstream side when the upstream pressure is above the setting
- fully open if the upstream pressure is below the setting
- closed if the pressure on the downstream side exceeds that on the upstream side (i.e., reverse flow is not allowed).PSVs maintain a set pressure at a specific point in the pipe network. EPANET computes in which of three different states a PSV can be in:

- partially opened (i.e., active) to maintain its pressure setting on its upstream side when the downstream pressure is below this value
- fully open if the downstream pressure is above the setting closed if the pressure on the downstream side exceeds that on the upstream side (i.e., reverse flow is not allowed).

PBVs force a specified pressure loss to occur across the valve. Flow through the valve can be in either direction. PBV's are not true physical devices but can be used to model situations where a particular pressure drop is known to exist.

FCVs limit the flow to a specified amount. The program produces a warning message if this flow cannot be maintained without having to add additional head at the valve (i.e., the flow cannot be maintained even with the valve fully open).

TCVs simulate a partially closed valve by adjusting the minor head loss coefficient of the valve. A relationship between the degree to which a valve is closed and the resulting head loss coefficient is usually available from the valve manufacturer.

GPVs are used to represent a link where the user supplies a special flow - head loss relationship instead of following one of the standard hydraulic formulas. They can be used to model turbines, well draw-down or reduced-flow backflow prevention valves. Shutoff (gate) valves and check (non-return) valves, which completely open or close pipes, are not considered as separate valve links but are instead included as a property of the pipe in which they are placed.

Each type of valve has a different type of setting parameter that describes its operating point (pressure for PRVs, PSVs, and PBVs; flow for FCVs; loss coefficient for TCVs, and head loss curve for GPVs).

Valves can have their control status overridden by specifying they be either completely open or completely closed. A valve's status and its setting can be changed during the simulation by using control statements. Because of the ways in which valves are modeled the following rules apply when adding valves to a network:

- a PRV, PSV or FCV cannot be directly connected to a reservoir or tank (use a length of pipe to separate the two)
- PRVs cannot share the same downstream node or be linked in series

- two PSVs cannot share the same upstream node or be linked in series
- a PSV cannot be connected to the downstream node of a PRV.

Note : Only FCV type has been used in the pumping system for this study.

5.4. NON-PHYSICAL COMPONENTS

In addition to physical components, EPANET employs three types of informational objects – curves, patterns, and controls - that describe the behavior and operational aspects of a distribution system.

5.4.1. PUMP CURVE

A Pump Curve represents the relationship between the head and flow rate that a pump can deliver at its nominal speed setting. Head is the head gain imparted to the water by the pump and is plotted on the vertical (Y) axis of the curve in feet (meters).

Flow rate is plotted on the horizontal (X) axis in flow units. A valid pump curve must have decreasing head with increasing flow. A different shape of pump curve will be used depending on the number of points supplied.

Single-Point Curve - A single-point pump curve is defined by a single head-flow combination that represents a pump's desired operating point. EPANET adds two more points to the curve by assuming a shutoff head at zero flow equal to 133% of the design head and a maximum flow at zero head equal to twice the design flow. It then treats the curve as a three-point curve.

Three-Point Curve - A three-point pump curve is defined by three operating points: a Low Flow point (flow and head at low or zero flow condition), a Design Flow point (flow and head at desired operating point), and a Maximum Flow point (flow and head at maximum flow).

Multi-Point Curve – A multi-point pump curve is defined by providing either a pair of head-flow points or four or more such points. EPANET creates a complete curve by connecting the points with straight-line segments. For variable speed pumps, the pump curve shifts as the speed changes

Note: In this study, only Single-Point Curve has been used.

5.4.2. EFFICIENCY CURVE

An Efficiency Curve determines pump efficiency (Y in percent) as a function of pump flow rate (X in flow units). Efficiency should represent wire-to-water efficiency that takes into account mechanical losses in the pump itself as well as electrical losses in the pump's motor. The curve is used only for energy calculations. A 65% pump efficiency has been used in this study.

5.4.3. TIME PATTERNS

A Time Pattern is a collection of multipliers that can be applied to a quantity to allow it to vary over time. Nodal demands, reservoir heads, pump schedules, and water quality source inputs can all have time patterns associated with them. The time interval used in all patterns is a fixed value. Within this interval a quantity remains at a constant level, equal to the product of its nominal value (as shown in table 5.1) and the pattern's multiplier for that time period. Four loading conditions are considered with the demands as shown in Fig. 5.3. The examples assume that the tank becomes full at the mid-night. Although all time patterns must utilize the same time interval, each can have a different number of periods. When the simulation clock exceeds the number of periods in a pattern, the pattern wraps around to its first period again.

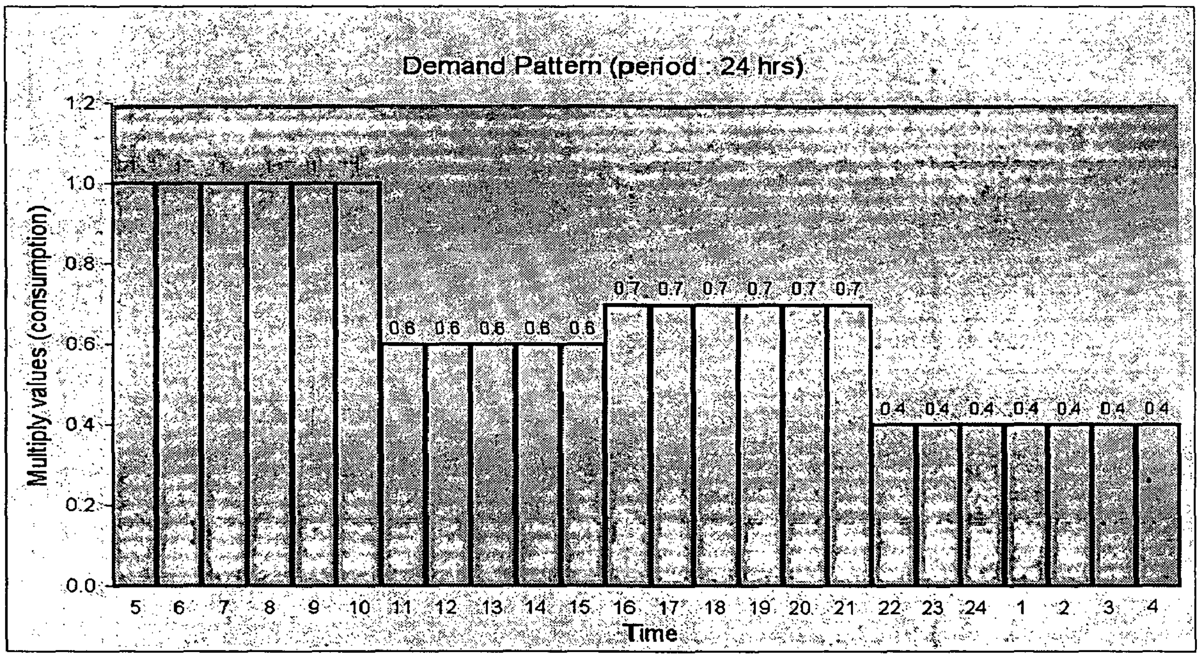


Fig. 5.3. Demand Time Pattern

In this study, time patterns work consider a junction node with an average demand of 14,300 CMD. The time pattern interval has been set out vary in (6, 5, 6 and 7 hours) and a pattern with the following multipliers; 1, 0.6, 0.7 and 0.4, respectively. These four loading conditions occur from 5 am to 10 am, 11am to 15 pm, 15 pm to 21 pm and 21 pm to 4 am.

5.4.4. CONTROLS

Controls are statements that determine how the network is operated over time. They specify the status of selected links as a function of time, tank water levels, and pressures at select points within the network. There are two categories of controls that can be used:

- Simple Controls
- Rule-Based Controls

Simple controls depend on only a single condition in the network (e.g., a water level in a certain tank) while rule-based controls depend on a number of conditions occurring simultaneously.

Simple Controls to allow a link to be controlled based on the time of day. Simple Controls change the status or setting of a link based on

- the water level in a tank,
- the pressure at a junction,
- the time into the simulation,
- or the time of day.

Rule-Based Controls allows the control of links to be based on a combination of conditions that might exist in the network over an extended period simulation.

In this study Rule-Based Control has been applied, and the rule as shown below:

Based-Control for 1 unit pump (see also Fig. 5.2.)

```
RULE 1
IF TANK LEVEL BELOW 3
THEN PUMP STATUS IS OPEN
RULE 2
IF TANK LEVEL ABOVE 8
THEN PUMP STATUS IS CLOSED
```

5.5. HYDRAULIC SIMULATION BY EPANET PROGRAM

EPANET's hydraulic simulation model computes hydraulic heads at junctions and flow rates through links for a fixed set of reservoir levels, tank levels, and water demands over a succession of points in time. From one time step to the next reservoir levels and junction demands are updated according to their prescribed time patterns while tank levels are updated using the current flows solution. The solution for heads and flows at a particular point in time involves solving simultaneously the conservation of flow equation for each junction and the headloss relationship across each link in the network. This process, known as hydraulically balancing the network, requires using an iterative technique to solve the nonlinear equations involved. EPANET employs the Gradient Algorithm for this purpose.

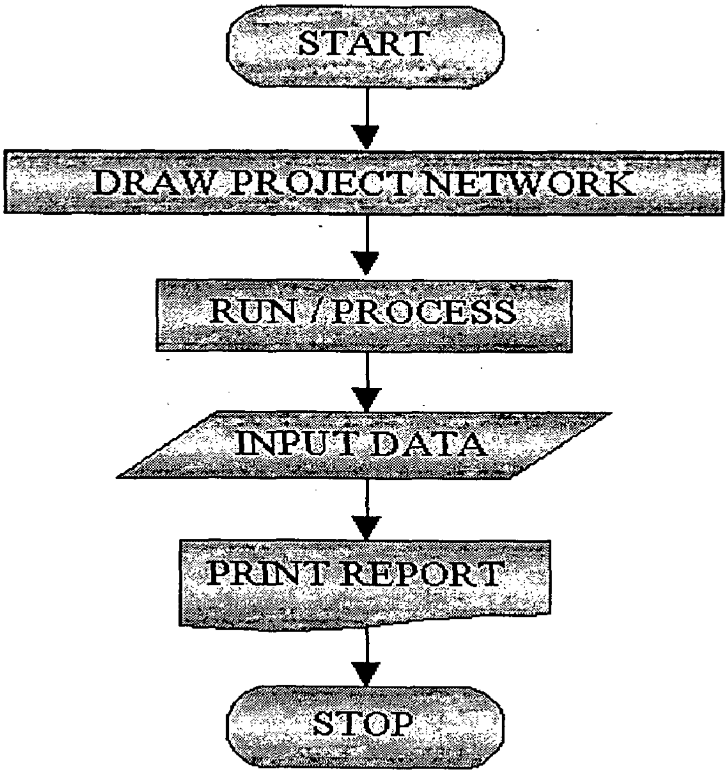


Fig. 5.4. Flow Chart for Execution EPANET Simulation

5.5.1. DATA ENTRY PROPERTY FOR EPANET SIMULATION

Data Entry For EPANET Simulation used in this study

1. JUNCTIONS Property
   - Elevation :
   - Base Demand :
   - Demand Pattern :
2. RESERVOIRS Property
   - Total Head :
3. TANKS Property
   - Elevation :
   - Initial Level :
   - Minimum Level :
   - Maximum Level :
   - Diameter :
4. PUMPS Property
   - Pump Curve :
   - Initial Status : Closed
5. PIPES Property
   - Length :
   - Diameter :
   - Roughness :
   - Loss Coefficient :
   - Initial Status : Open
6. CURVES (for Pump)
   - Curve ID :
   - Curve Type : Pump
   - Flow :
   - Head :
7. PATTERN
   - Time Period :
   - Multiplier :
8. ENERGY OPTION
   - Pump Efficiency :
   - Energy Price :
9. CONTROLS
   - Rule Based (Chosen)

5.5.2. OUTPUT PROPERTY OF EPANET SIMULATION

GRAPH

- ✓ Time Series for Nodes
- ✓ Time Series for Links
- ✓ Profile Plot
- ✓ Contour Plot
- ✓ Frequency Plot for Nodes
- ✓ Frequency Plot for Links
- ✓ System Flow

TABLE

1. *Network Nodes*
   - Elevation
   - Base Demand
   - Demand
   - Head
   - Pressure
2. *Network Links*
   - Length
   - Diameter
   - Roughness
   - Flow
   - Velocity
3. *Energy*
   - ✓ Usage Factor
   - ✓ Average Efficiency
   - ✓ kW-hr/m^3
   - ✓ Average kW
   - ✓ Peak kW
   - ✓ Cost per Day

4. *Full Report*
   - ✓ Link-Node Table
     - Link
     - Start Node
     - End Node
     - Length
     - Diameter
   - ✓ Energy Usage
   - ✓ Node Results at Time: T hrs
     - Demand
     - Head
     - Pressure
   - ✓ Link Results at Time: T hrs
     - Flow
     - Velocity
     - Unit Headloss
     - Status

5.5.3. SIZE OF SUCTION AND DISCHARGE PIPE FOR CENTRIFUGAL PUMP

Size of suction and discharge pipe has been defined by EPANET Simulation with the parameter mentioned below:

Input of Simulation:

Number of unit : 1

Loading Condition : See Fig. 5.3. and 5.4. and Table 5.1.

Curve type : Single-Point Curve

- ✓ Continuous Mode Operation : $H = 66.67 - 1.593E - 7Q^2$
- ✓ Intermittent Mode Operation : $H = 66.67 - 6.344E - 8Q^2$

Head : 50 m

Design Flow

- ✓ Continuous Mode Operation : 10,230 CMD
- ✓ Intermittent Mode Operation : 16,210 CMD

Control : Rule-Based Controls (see Chapter 5.4.4)

Suction Pipe : Length = 5 m ; Diameter = 400 mm

Discharge Pipe : Length = 12.5 m ; Diameter = 350 mm

Pump Elevation : 177 m (measured from sea level)

Reservoir Head : 175 m (measured from sea level)

Minor Coefficients : see Appendix 5 & 6 (in this simulation, loss coefficient applied for pipeline in pumping station only)

Physical Construction of the model for pumping station used in this study; see Appendix 4

Output of Simulation

Max. Discharge

- ✓ Continuous Mode Operation : 12,400 CMD
- ✓ Intermittent Mode Operation : 18,500 CMD

Based on the result of the maximum discharge, the sizes of the suction and discharge pipes for the further simulation are mentioned in the table 5.2.

Table 5.2.
Size of Suction and Discharge Pipe
(After Simulation and as per Specification of Kirloskar Pump see Appendix- 2 & 3)

No. of Units	Continuous Mode Operation			Specifications		
	Flow (MCD)	Flow (m^3/h)	Flow (l/s)	Type	Suc. Dia (mm)	Dis. Dia. (mm).
5	2,480	103	29	100-40	125	100
4	3,100	129	36	100-40	125	100
3	4,133	172	48	125-40	150	125
2	6,200	258	72	125-41	150	125
1	12,400	517	144	N/A	N/A	N/A

No. of Units	Intermittent Mode Operation			Specifications		
	Flow (MCD)	Flow (m^3/h)	Flow (l/s)	Type	Suc. Dia (mm)	Dis. Dia. (mm).
5	3,700	154	43	100-40	125	100
4	4,625	193	54	125-40	150	125
3	6,167	257	71	125-41	150	125
2	9,250	385	107	150-40	200	150
1	18,500	771	214	N/A	N/A	N/A

Physical Construction of the model for pumping station used in this study; see Appendix 4

5.6. RESULTS AND ANALYSIS

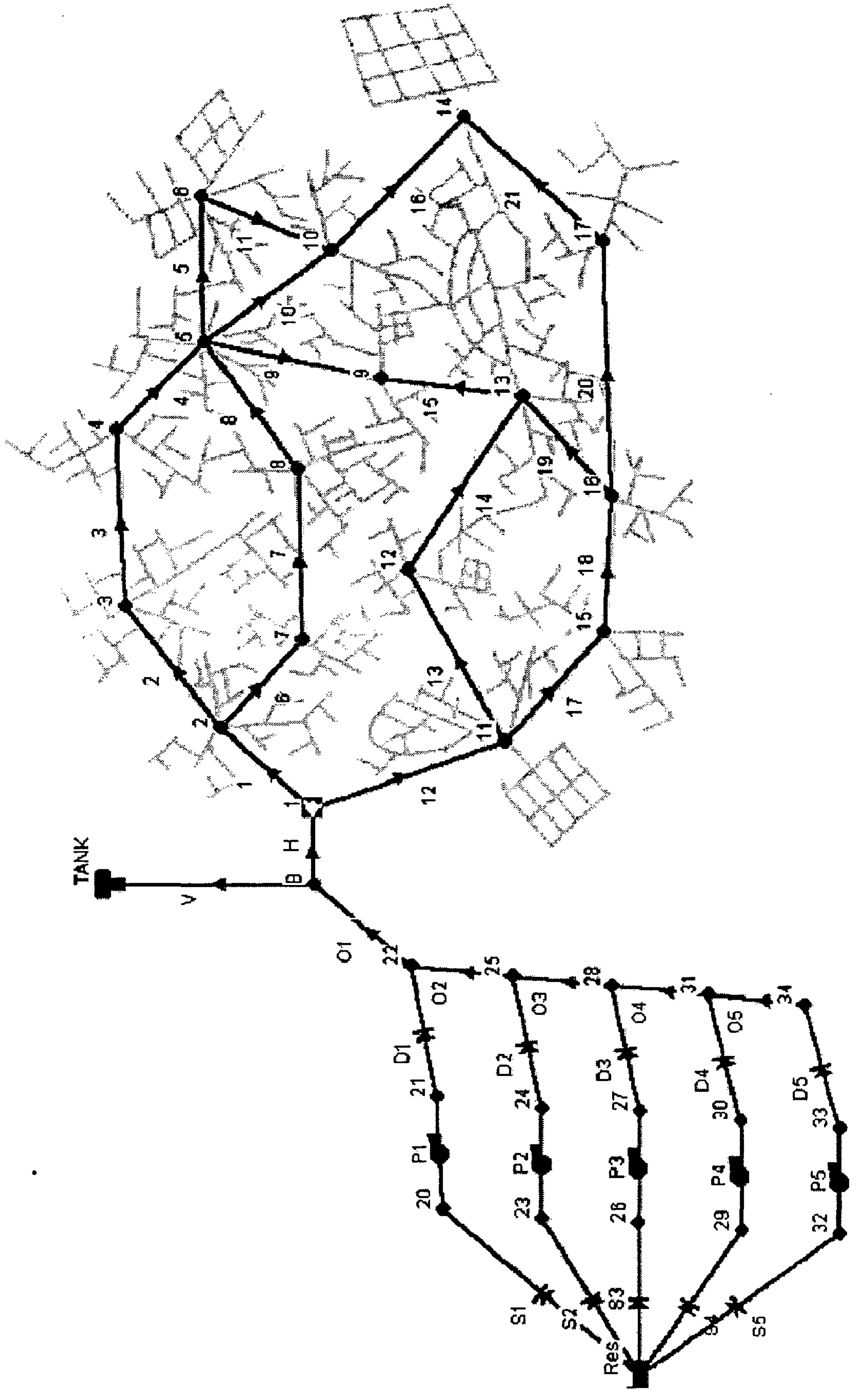


Fig. 5.5. Pumping Units and Distribution Network Representation for Simulation (5 Units)

Fig. 5.6
System Flow Balance & Water Level
(Result of Simulation – Continuous Mode Operation)

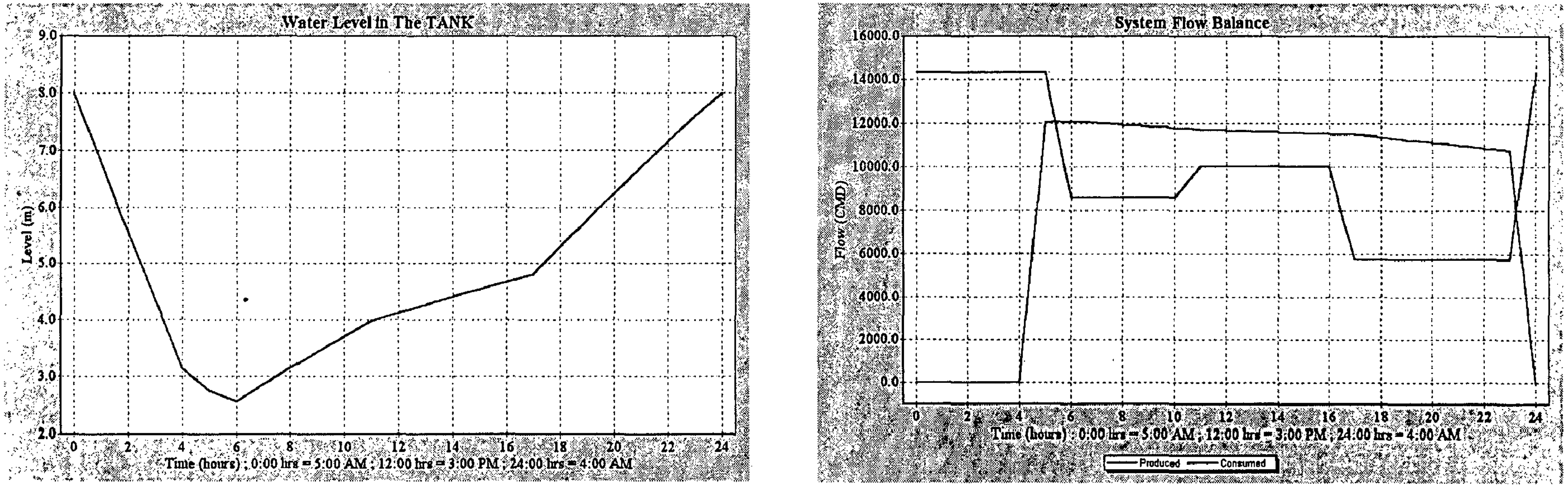


Table 5.3
Simulation Data for Continuous Mode Operation

Description	Number of Combination			
	2+1 (3 units)	3+1 (4 units)	4+1 (5 units)	5+1 (6 units)
Mode of Operation	: Continuous	: Continuous	: Continuous	: Continuous
Number of Pump (Ops.)	: 2 Units	: 3 Units	: 4 Units	: 5 Units
Pump Curve Type Curve	: Single-Point	: Single-Point	: Single-Point	: Single-Point
	: $H = 66.67 - 3.12E - 7Q^2$	: $H = 66.67 - 1.131E - 6Q^2$	: $H = 66.67 - 1.786E - 6Q^2$	: $H = 66.67 - 3.249E - 6Q^2$
Design Flow	: 7,310 CMD	: 3,840 CMD	: 3,055 CMD	: 2,265 CMD
Design Head	: 50 m	: 50 m	: 50 m	: 50 m
Pumping Control	: Rule-Based Control	: Rule-Based Control	: Rule-Based Control	: Rule-Based Control
Demand Pattern	: See Chapter 5.4.3.	: See Chapter 5.4.3.	: See Chapter 5.4.3.	: See Chapter 5.4.3.

Fig. 5.7
System Flow Balance & Water Level
(Result of Simulation – Intermittent Mode Operation)

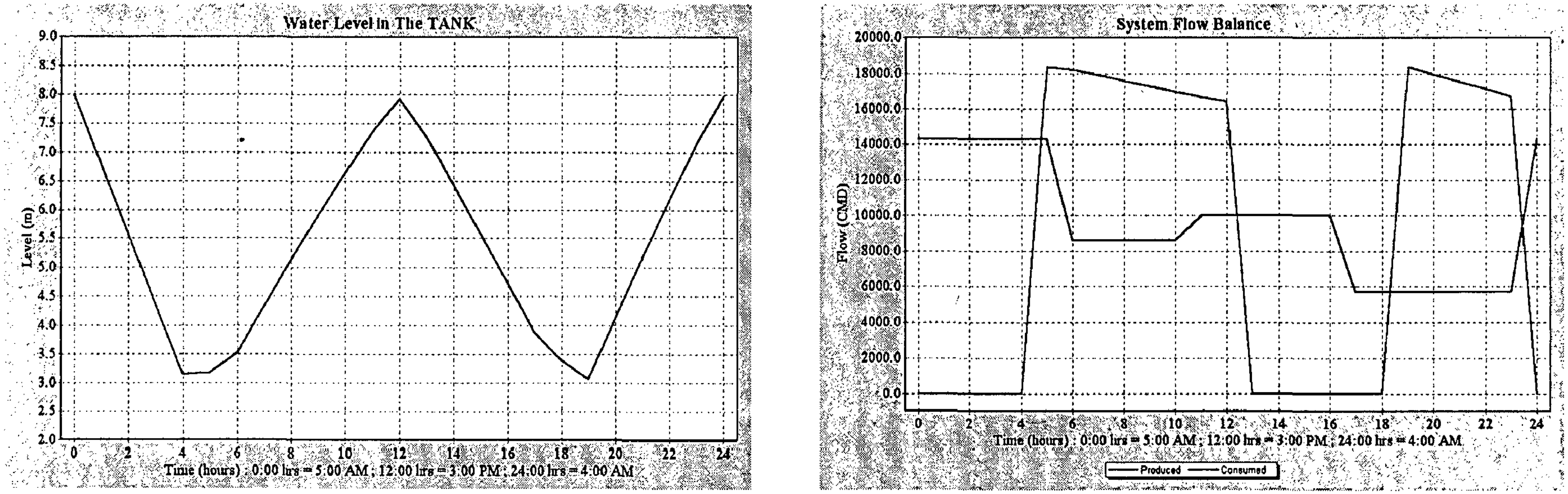


Table 5.4
Simulation Data for Intermittent Mode Operation

Description	Number of Combination			
	2+1 (3 units)	3+1 (4 units)	4+1 (5 units)	5+1 (6 units)
Mode of Operation	: Intermittent	: Intermittent	: Intermittent	: Intermittent
Number of Pump (Ops.)	: 2 Units	: 3 Units	: 4 Units	: 5 Units
Pump Curve Type Curve	: Single-Point	: Single-Point	: Single-Point	: Single-Point
	: $H = 66.67 - 1.279E - 8Q^2$	: $H = 66.67 - 2.747E - 7Q^2$	: $H = 66.67 - 7.563E - 7Q^2$	: $H = 66.67 - 8.769E - 7Q^2$
Design Flow	: 11,415 CMD	: 7,790 CMD	: 4,695 CMD	: 4,360 CMD
Design Head	: 50 m	: 50 m	: 50 m	: 50 m
Pumping Control	: Rule-Based Control	: Rule-Based Control	: Rule-Based Control	: Rule-Based Control
Demand Pattern	: See Chapter 5.4.3.	: See Chapter 5.4.3.	: See Chapter 5.4.3.	: See Chapter 5.4.3.

Table 5.5 Summary of Flow of The Pumps For Continuous Mode Operation (19 hrs.)

5 Units Continuous Mode Operation + 1 Unit Standby

Sheet 1 of 4

Row No.	Descriptions	Units	C1	C2	C3	C4	C5	C6	C7	C8	C9	C10	C11	C12
			5:00 AM	6:00 AM	7:00 AM	8:00 AM	9:00 AM	10:00 AM	11:00 AM	12:00 PM	1:00 PM	2:00 PM	3:00 PM	4:00 PM
R1	Pump No.1	(m^3/day)	OFF	OFF	OFF	OFF	2,410.95	2,419.50	2,404.51	2,389.70	2,375.12	2,360.76	2,347.01	2,339.43
R2	Pump No.2	(m^3/day)	OFF	OFF	OFF	OFF	2,410.49	2,419.04	2,404.05	2,389.24	2,374.66	2,360.31	2,346.56	2,338.99
R3	Pump No.3	(m^3/day)	OFF	OFF	OFF	OFF	2,410.22	2,418.77	2,403.78	2,388.98	2,374.40	2,360.05	2,346.30	2,338.72
R4	Pump No.4	(m^3/day)	OFF	OFF	OFF	OFF	2,410.09	2,418.64	2,403.66	2,388.85	2,374.27	2,359.92	2,346.18	2,338.60
R5	Pump No.5	(m^3/day)	OFF	OFF	OFF	OFF	2,410.06	2,418.61	2,403.62	2,388.81	2,374.24	2,359.89	2,346.14	2,338.57
R6	Sum of Flow for 5 Units	(m^3/day)	---	---	---	---	12,051.81	12,094.56	12,019.62	11,945.58	11,872.69	11,800.93	11,732.19	11,694.31
R7	Ave. Flow each Unit	(m^3/day)	---	---	---	---	2,410.36	2,418.91	2,403.92	2,389.12	2,374.54	2,360.19	2,346.44	2,338.86
R8	Max Flow	(m^3/day)	---	---	---	---	2,410.95	2,419.50	2,404.51	2,389.70	2,375.12	2,360.76	2,347.01	2,339.43

Row No.	Descriptions	Units	C13	C14	C15	C16	C17	C18	C19	C20	C21	C22	C23	C24
			5:00 PM	6:00 PM	7:00 PM	8:00 PM	9:00 PM	10:00 PM	11:00 PM	12:00 AM	1:00 AM	2:00 AM	3:00 AM	4:00 AM
R1	Pump No.1	(m^3/day)	2,331.99	2,324.68	2,317.51	2,310.48	2,301.80	2,275.89	2,250.13	2,224.66	2,199.50	2,174.65	2,150.10	OFF
R2	Pump No.2	(m^3/day)	2,331.54	2,324.23	2,317.07	2,310.04	2,301.36	2,275.45	2,249.70	2,224.23	2,199.08	2,174.22	2,149.68	OFF
R3	Pump No.3	(m^3/day)	2,331.28	2,323.97	2,316.81	2,309.78	2,301.10	2,275.20	2,249.44	2,223.98	2,198.83	2,173.98	2,149.44	OFF
R4	Pump No.4	(m^3/day)	2,331.15	2,323.85	2,316.68	2,309.66	2,300.97	2,275.07	2,249.32	2,223.86	2,198.71	2,173.86	2,149.32	OFF
R5	Pump No.5	(m^3/day)	2,331.12	2,323.82	2,316.65	2,309.62	2,300.94	2,275.04	2,249.29	2,223.83	2,198.68	2,173.83	2,149.29	OFF
R6	Sum of Flow for 5 Units	(m^3/day)	11,657.08	11,620.55	11,584.72	11,549.58	11,506.17	11,376.65	11,247.88	11,120.56	10,994.80	10,870.54	10,747.83	---
R7	Ave. Flow each Unit	(m^3/day)	2,331.42	2,324.11	2,316.94	2,309.92	2,301.23	2,275.33	2,249.58	2,224.11	2,198.96	2,174.11	2,149.57	---
R8	Max Flow	(m^3/day)	2,331.99	2,324.68	2,317.51	2,310.48	2,301.80	2,275.89	2,250.13	2,224.66	2,199.50	2,174.65	2,150.10	---
	Overall Ave. Flow 5 Units Opr. (R6 ; C5 - C23)	(m^3/day)	11,541.79											
	Overall Ave. Flow each Unit (R7 ; C5 - C23)	(m^3/day)	2,308.36											
	Overall Ave. Max Flow (R8 ; C5 - C23)	(m^3/day)	2,419.50											
		l/s	28.00											

4 Units Continuous Mode Operation + 1 Unit Standby

Sheet 2 of 4

Row No.	Descriptions	Units	C1	C2	C3	C4	C5	C6	C7	C8	C9	C10	C11	C12
			5:00 AM	6:00 AM	7:00 AM	8:00 AM	9:00 AM	10:00 AM	11:00 AM	12:00 PM	1:00 PM	2:00 PM	3:00 PM	4:00 PM
R1	Pump No.1	(m^3/day)	OFF	OFF	OFF	OFF	3,008.95	3,019.74	3,001.10	2,982.69	2,964.56	2,946.71	2,929.62	2,920.24
R2	Pump No.2	(m^3/day)	OFF	OFF	OFF	OFF	3,008.44	3,019.23	3,000.59	2,982.18	2,964.06	2,946.21	2,929.12	2,919.74
R3	Pump No.3	(m^3/day)	OFF	OFF	OFF	OFF	3,008.20	3,018.99	3,000.35	2,981.94	2,963.82	2,945.98	2,928.89	2,919.51
R4	Pump No.4	(m^3/day)	OFF	OFF	OFF	OFF	3,008.14	3,018.93	3,000.29	2,981.88	2,963.75	2,945.91	2,928.82	2,919.44
R5	Sum of Flow for 4 Units	(m^3/day)	---	---	---	---	12,033.73	12,076.89	12,002.33	11,928.69	11,856.19	11,784.81	11,716.45	11,678.93
R6	Ave. Flow each Unit	(m^3/day)	---	---	---	---	3,008.43	3,019.22	3,000.58	2,982.17	2,964.05	2,946.20	2,929.11	2,919.73
R7	Max Flow	(m^3/day)	---	---	---	---	3,008.95	3,019.74	3,001.10	2,982.69	2,964.56	2,946.71	2,929.62	2,920.24

Row No.	Descriptions	Units	C13	C14	C15	C16	C17	C18	C19	C20	C21	C22	C23	C24
			5:00 PM	6:00 PM	7:00 PM	8:00 PM	9:00 PM	10:00 PM	11:00 PM	12:00 AM	1:00 AM	2:00 AM	3:00 AM	4:00 AM
R1	Pump No.1	(m^3/day)	2,911.02	2,901.97	2,893.10	2,884.40	2,873.63	2,841.35	2,809.24	2,777.51	2,746.16	2,715.19	2,684.60	OFF
R2	Pump No.2	(m^3/day)	2,910.52	2,901.48	2,892.61	2,883.91	2,873.15	2,840.86	2,808.77	2,777.04	2,745.69	2,714.72	2,684.14	OFF
R3	Pump No.3	(m^3/day)	2,910.29	2,901.25	2,892.38	2,883.68	2,872.92	2,840.64	2,808.54	2,776.82	2,745.47	2,714.50	2,683.93	OFF
R4	Pump No.4	(m^3/day)	2,910.22	2,901.18	2,892.31	2,883.61	2,872.85	2,840.57	2,808.48	2,776.75	2,745.41	2,714.44	2,683.87	OFF
R5	Sum of Flow for 4 Units	(m^3/day)	11,642.05	11,605.88	11,570.40	11,535.60	11,492.55	11,363.42	11,235.03	11,108.12	10,982.73	10,858.85	10,736.54	---
R6	Ave. Flow each Unit	(m^3/day)	2,910.51	2,901.47	2,892.60	2,883.90	2,873.14	2,840.86	2,808.76	2,777.03	2,745.68	2,714.71	2,684.14	---
R7	Max Flow	(m^3/day)	2,911.02	2,901.97	2,893.10	2,884.40	2,873.63	2,841.35	2,809.24	2,777.51	2,746.16	2,715.19	2,684.60	---
	Overall Ave. Flow 4 Units Ops. (R5 ; C5 - C23)	(m^3/day)	11,527.13											
	Overall Ave. Flow each Unit (R6 ; C5 - C23)	(m^3/day)	2,881.78											
	Overall Ave. Max Flow (R7 ; C5 - C23)	(m^3/day)	3,019.74											
		l/s	34.95											

3 Units Continuous Mode Operation + 1 Unit Standby

Sheet 3 of 4

Row No.	Descriptions	Units	C1	C2	C3	C4	C5	C6	C7	C8	C9	C10	C11	C12
			5:00 AM	6:00 AM	7:00 AM	8:00 AM	9:00 AM	10:00 AM	11:00 AM	12:00 PM	1:00 PM	2:00 PM	3:00 PM	4:00 PM
R1	Pump No.1	(m^3/day)	OFF	OFF	OFF	OFF	4,013.53	4,027.86	4,002.99	3,978.43	3,954.24	3,930.44	3,907.63	3,895.09
R2	Pump No.2	(m^3/day)	OFF	OFF	OFF	OFF	4,012.98	4,027.31	4,002.45	3,977.89	3,953.71	3,929.90	3,907.10	3,894.56
R3	Pump No.3	(m^3/day)	OFF	OFF	OFF	OFF	4,012.83	4,027.16	4,002.30	3,977.74	3,953.56	3,929.76	3,906.95	3,894.42
R4	Sum of Flow for 3 Units	(m^3/day)	---	---	---	---	12,039.34	12,082.33	12,007.74	11,934.06	11,861.51	11,790.10	11,721.68	11,684.07
R5	Ave. Flow each Unit	(m^3/day)	---	---	---	---	4,013.11	4,027.44	4,002.58	3,978.02	3,953.84	3,930.03	3,907.23	3,894.69
R6	Max Flow	(m^3/day)	---	---	---	---	4,013.53	4,027.86	4,002.99	3,978.43	3,954.24	3,930.44	3,907.63	3,895.09

Row No.	Descriptions	Units	C13	C14	C15	C16	C17	C18	C19	C20	C21	C22	C23	C24
			5:00 PM	6:00 PM	7:00 PM	8:00 PM	9:00 PM	10:00 PM	11:00 PM	12:00 AM	1:00 AM	2:00 AM	3:00 AM	4:00 AM
R1	Pump No.1	(m^3/day)	3,882.78	3,870.69	3,858.83	3,847.21	3,832.83	3,789.79	3,746.98	3,704.68	3,662.87	3,621.57	3,580.78	OFF
R2	Pump No.2	(m^3/day)	3,882.25	3,870.16	3,858.31	3,846.68	3,832.31	3,789.27	3,746.47	3,704.17	3,662.36	3,621.07	3,580.29	OFF
R3	Pump No.3	(m^3/day)	3,882.10	3,870.02	3,858.16	3,846.54	3,832.16	3,789.12	3,746.33	3,704.03	3,662.23	3,620.93	3,580.15	OFF
R4	Sum of Flow for 3 Units	(m^3/day)	11,647.13	11,610.87	11,575.30	11,540.43	11,497.30	11,368.18	11,239.78	11,112.88	10,987.46	10,863.57	10,741.22	---
R5	Ave. Flow each Unit	(m^3/day)	3,882.38	3,870.29	3,858.43	3,846.81	3,832.43	3,789.39	3,746.59	3,704.29	3,662.49	3,621.19	3,580.41	---
R6	Max Flow	(m^3/day)	3,882.78	3,870.69	3,858.83	3,847.21	3,832.83	3,789.79	3,746.98	3,704.68	3,662.87	3,621.57	3,580.78	---
	Overall Ave. Flow 3 Units Ops. (R4 ; C5 - C23)	(m^3/day)	11,532.17											
	Overall Ave. Flow each Unit (R5 ; C5 - C23)	(m^3/day)	3,844.06											
	Overall Max Flow (R6 ; C5 - C23)	(m^3/day)	4,027.86											
		l/s	46.62											

2 Units Continuous Mode Operation + 1 Unit Standby

ConSheet 4 of 4

Row No.	Descriptions	Units	C1	C2	C3	C4	C5	C6	C7	C8	C9	C10	C11	C12
			5:00 AM	6:00 AM	7:00 AM	8:00 AM	9:00 AM	10:00 AM	11:00 AM	12:00 PM	1:00 PM	2:00 PM	3:00 PM	4:00 PM
R1	Pump No.1	(m^3/day)	OFF	OFF	OFF	OFF	6,016.97	6,038.58	6,001.26	5,964.39	5,928.10	5,892.37	5,858.15	5,839.37
R2	Pump No.2	(m^3/day)	OFF	OFF	OFF	OFF	6,016.50	6,038.10	6,000.78	5,963.92	5,927.62	5,891.90	5,857.68	5,838.90
R3	Sum of Flow for 2 Units	(m^3/day)	---	---	---	---	12,033.47	12,076.68	12,002.04	11,928.31	11,855.72	11,784.27	11,715.83	11,678.27
R4	Ave. Flow each Unit	(m^3/day)	---	---	---	---	6,016.74	6,038.34	6,001.02	5,964.16	5,927.86	5,892.14	5,857.92	5,839.14
R5	Max Flow	(m^3/day)	---	---	---	---	6,016.97	6,038.58	6,001.26	5,964.39	5,928.10	5,892.37	5,858.15	5,839.37

Row No.	Descriptions	Units	C13	C14	C15	C16	C17	C18	C19	C20	C21	C22	C23	C24
			5:00 PM	6:00 PM	7:00 PM	8:00 PM	9:00 PM	10:00 PM	11:00 PM	12:00 AM	1:00 AM	2:00 AM	3:00 AM	4:00 AM
R1	Pump No.1	(m^3/day)	5,820.92	5,802.82	5,785.06	5,767.65	5,746.10	5,681.47	5,617.21	5,553.69	5,490.93	5,428.94	5,367.73	OFF
R2	Pump No.2	(m^3/day)	5,820.45	5,802.35	5,784.60	5,767.18	5,745.64	5,681.01	5,616.75	5,553.24	5,490.49	5,428.50	5,367.29	OFF
R3	Sum of Flow for 2 Units	(m^3/day)	11,641.37	11,605.17	11,569.66	11,534.83	11,491.74	11,362.48	11,233.96	11,106.93	10,981.42	10,857.44	10,735.02	---
R4	Ave. Flow each Unit	(m^3/day)	5,820.69	5,802.59	5,784.83	5,767.42	5,745.87	5,681.24	5,616.98	5,553.47	5,490.71	5,428.72	5,367.51	---
R5	Max Flow	(m^3/day)	5,820.92	5,802.82	5,785.06	5,767.65	5,746.10	5,681.47	5,617.21	5,553.69	5,490.93	5,428.94	5,367.73	---
	Overall Ave. Flow 2 Units (R3 ; C5 - C23)	(m^3/day)	11,526.36											
	Overall Ave. Flow each Unit (R4 ; C5 - C23)	(m^3/day)	5,763.18											
	Overall Max Flow (R5 ; C5 - C23)	(m^3/day)	6,038.58											
		l/s	69.89											

Table 5.6 Summary of Flow of The Pumps For Intermittent Mode Operation (8+5=13 hrs.)

5 Units Intermittent Mode Operation + 1 Unit Standby

Sheet 1 of 4

Row No.	Descriptions	Units	C1	C2	C3	C4	C5	C6	C7	C8	C9	C10	C11	C12
			5:00 AM	6:00 AM	7:00 AM	8:00 AM	9:00 AM	10:00 AM	11:00 AM	12:00 PM	1:00 PM	2:00 PM	3:00 PM	4:00 PM
R1	Pump No.1	(m^3/day)	OFF	OFF	OFF	OFF	3,678.04	3,644.57	3,579.35	3,515.20	3,452.14	3,390.19	3,331.15	3,282.05
R2	Pump No.2	(m^3/day)	OFF	OFF	OFF	OFF	3,676.47	3,643.00	3,577.81	3,513.69	3,450.65	3,388.72	3,329.71	3,280.63
R3	Pump No.3	(m^3/day)	OFF	OFF	OFF	OFF	3,675.54	3,642.09	3,576.91	3,512.80	3,449.78	3,387.86	3,328.86	3,279.79
R4	Pump No.4	(m^3/day)	OFF	OFF	OFF	OFF	3,675.11	3,641.65	3,576.48	3,512.38	3,449.36	3,387.46	3,328.46	3,279.39
R5	Pump No.5	(m^3/day)	OFF	OFF	OFF	OFF	3,674.99	3,641.53	3,576.37	3,512.26	3,449.25	3,387.34	3,328.34	3,279.28
R6	Sum of Flow for 5 Units	(m^3/day)	---	---	---	---	18,380.15	18,212.84	17,886.92	17,566.33	17,251.18	16,941.57	16,646.52	16,401.14
R7	Ave. Flow each Unit	(m^3/day)	---	---	---	---	3,676.03	3,642.57	3,577.38	3,513.27	3,450.24	3,388.31	3,329.30	3,280.23
R8	Max Flow	(m^3/day)	---	---	---	---	3,678.04	3,644.57	3,579.35	3,515.20	3,452.14	3,390.19	3,331.15	3,282.05

Row No.	Descriptions	Units	C13	C14	C15	C16	C17	C18	C19	C20	C21	C22	C23	C24
			5:00 PM	6:00 PM	7:00 PM	8:00 PM	9:00 PM	10:00 PM	11:00 PM	12:00 AM	1:00 AM	2:00 AM	3:00 AM	4:00 AM
R1	Pump No.1	(m^3/day)	OFF	OFF	OFF	OFF	OFF	OFF	3,676.05	3,591.05	3,506.98	3,423.87	3,341.74	OFF
R2	Pump No.2	(m^3/day)	OFF	OFF	OFF	OFF	OFF	OFF	3,674.47	3,589.51	3,505.47	3,422.39	3,340.29	OFF
R3	Pump No.3	(m^3/day)	OFF	OFF	OFF	OFF	OFF	OFF	3,673.55	3,588.60	3,504.59	3,421.52	3,339.43	OFF
R4	Pump No.4	(m^3/day)	OFF	OFF	OFF	OFF	OFF	OFF	3,673.11	3,588.18	3,504.17	3,421.11	3,339.03	OFF
R5	Pump No.5	(m^3/day)	OFF	OFF	OFF	OFF	OFF	OFF	3,672.99	3,588.06	3,504.05	3,420.99	3,338.92	OFF
R6	Sum of Flow for 5 Units	(m^3/day)	---	---	---	---	---	---	18,370.17	17,945.40	17,525.26	17,109.88	16,699.41	---
R7	Ave. Flow each Unit	(m^3/day)	---	---	---	---	---	---	3,674.03	3,589.08	3,505.05	3,421.98	3,339.88	---
R8	Max Flow	(m^3/day)	---	---	---	---	---	---	3,676.05	3,591.05	3,506.98	3,423.87	3,341.74	---
	Overall Ave. Flow 5 Units Opr. (R6 ; C5 - C23)	(m^3/day)	17,456.67											
	Overall Ave. Flow each Unit (R7 ; C5 - C23)	(m^3/day)	3,495.41											
	Overall Ave. Max Flow (R8 ; C5 - C23)	(m^3/day)	3,678.04											
		l/s	42.57											

4 Units Intermittent Mode Operation + 1 Unit Standby

Sheet 2 of 4

Row No.	Descriptions	Units	C1	C2	C3	C4	C5	C6	C7	C8	C9	C10	C11	C12
			5:00 AM	6:00 AM	7:00 AM	8:00 AM	9:00 AM	10:00 AM	11:00 AM	12:00 PM	1:00 PM	2:00 PM	3:00 PM	4:00 PM
R1	Pump No.1	(m^3/day)	OFF	OFF	OFF	OFF	4,589.41	4,548.03	4,467.12	4,467.12	4,309.25	4,232.35	4,159.06	4,098.12
R2	Pump No.2	(m^3/day)	OFF	OFF	OFF	OFF	4,587.68	4,546.31	4,465.42	4,465.42	4,307.61	4,230.74	4,157.46	4,096.55
R3	Pump No.3	(m^3/day)	OFF	OFF	OFF	OFF	4,586.86	4,545.50	4,464.62	4,464.62	4,306.83	4,229.97	4,156.71	4,095.81
R4	Pump No.4	(m^3/day)	OFF	OFF	OFF	OFF	4,586.63	4,545.27	4,464.40	4,464.40	4,306.62	4,229.76	4,156.50	4,095.60
R5	Sum of Flow for 4 Units	(m^3/day)	---	---	---	---	18,350.58	18,185.11	17,861.56	17,861.56	17,230.31	16,922.82	16,629.73	16,386.08
R6	Ave. Flow each Unit	(m^3/day)	---	---	---	---	4,587.65	4,546.28	4,465.39	4,465.39	4,307.58	4,230.71	4,157.43	4,096.52
R7	Max Flow	(m^3/day)	---	---	---	---	4,589.41	4,548.03	4,467.12	4,467.12	4,309.25	4,232.35	4,159.06	4,098.12

Row No.	Descriptions	Units	C13	C14	C15	C16	C17	C18	C19	C20	C21	C22	C23	C24
			5:00 PM	6:00 PM	7:00 PM	8:00 PM	9:00 PM	10:00 PM	11:00 PM	12:00 AM	1:00 AM	2:00 AM	3:00 AM	4:00 AM
R1	Pump No.1	(m^3/day)	OFF	OFF	OFF	OFF	OFF	OFF	4,584.34	4,478.85	4,374.49	4,271.30	4,169.32	OFF
R2	Pump No.2	(m^3/day)	OFF	OFF	OFF	OFF	OFF	OFF	4,582.61	4,477.15	4,372.83	4,269.67	4,167.72	OFF
R3	Pump No.3	(m^3/day)	OFF	OFF	OFF	OFF	OFF	OFF	4,581.79	4,476.35	4,372.04	4,268.90	4,166.97	OFF
R4	Pump No.4	(m^3/day)	OFF	OFF	OFF	OFF	OFF	OFF	4,581.56	4,476.13	4,371.83	4,268.69	4,166.76	OFF
R5	Sum of Flow for 4 Units	(m^3/day)	---	---	---	---	---	---	18,330.30	17,908.48	17,491.19	17,078.56	16,670.77	---
R6	Ave. Flow each Unit	(m^3/day)	---	---	---	---	---	---	4,582.58	4,477.12	4,372.80	4,269.64	4,167.69	---
R7	Max Flow	(m^3/day)	---	---	---	---	---	---	4,584.34	4,478.85	4,374.49	4,271.30	4,169.32	---
	Overall Ave. Flow 4 Units Ops. (R5 ; C5 - C23)	(m^3/day)	17,454.39											
	Overall Ave. Flow each Unit (R6 ; C5 - C23)	(m^3/day)	4,366.48											
	Overall Ave. Max Flow (R7 ; C5 - C23)	(m^3/day)	4,589.41											
		l/s	53.12											

3 Units Intermittent Mode Operation + 1 Unit Standby

Sheet 3 of 4

Row No.	Descriptions	Units	C1	C2	C3	C4	C5	C6	C7	C8	C9	C10	C11	C12
			5:00 AM	6:00 AM	7:00 AM	8:00 AM	9:00 AM	10:00 AM	11:00 AM	12:00 PM	1:00 PM	2:00 PM	3:00 PM	4:00 PM
R1	Pump No.1	(m^3/day)	OFF	OFF	OFF	OFF	6,120.52	6,065.12	5,956.90	5,850.44	5,745.78	5,642.96	5,544.97	5,463.51
R2	Pump No.2	(m^3/day)	OFF	OFF	OFF	OFF	6,118.66	6,063.27	5,955.07	5,848.64	5,744.01	5,641.22	5,543.25	5,461.82
R3	Pump No.3	(m^3/day)	OFF	OFF	OFF	OFF	6,118.14	6,062.76	5,954.57	5,848.15	5,743.53	5,640.74	5,542.78	5,461.35
R4	Sum of Flow for 3 Units	(m^3/day)	---	---	---	---	18,357.32	18,191.15	17,866.54	17,547.23	17,233.32	16,924.92	16,631.00	16,386.68
R5	Ave. Flow each Unit	(m^3/day)	---	---	---	---	6,119.11	6,063.72	5,955.51	5,849.08	5,744.44	5,641.64	5,543.67	5,462.23
R6	Max Flow	(m^3/day)	---	---	---	---	6,120.52	6,065.12	5,956.90	5,850.44	5,745.78	5,642.96	5,544.97	5,463.51

Row No.	Descriptions	Units	C13	C14	C15	C16	C17	C18	C19	C20	C21	C22	C23	C24
			5:00 PM	6:00 PM	7:00 PM	8:00 PM	9:00 PM	10:00 PM	11:00 PM	12:00 AM	1:00 AM	2:00 AM	3:00 AM	4:00 AM
R1	Pump No.1	(m^3/day)	OFF	OFF	OFF	OFF	OFF	OFF	6,114.37	5,973.28	5,833.72	5,695.74	5,559.39	OFF
R2	Pump No.2	(m^3/day)	OFF	OFF	OFF	OFF	OFF	OFF	6,112.51	5,971.45	5,831.93	5,693.99	5,557.68	OFF
R3	Pump No.3	(m^3/day)	OFF	OFF	OFF	OFF	OFF	OFF	6,111.99	5,970.94	5,831.44	5,693.50	5,557.20	OFF
R4	Sum of Flow for 3 Units	(m^3/day)	---	---	---	---	---	---	18,338.87	17,915.67	17,497.09	17,083.23	16,674.27	---
R5	Ave. Flow each Unit	(m^3/day)	---	---	---	---	---	---	6,112.96	5,971.89	5,832.36	5,694.41	5,558.09	---
R6	Max Flow	(m^3/day)	---	---	---	---	---	---	6,114.37	5,973.28	5,833.72	5,695.74	5,559.39	---
	Overall Ave. Flow 3 Units Ops. (R4 ; C5 - C23)	(m^3/day)	17,434.41											
	Overall Ave. Flow each Unit (R5 ; C5 - C23)	(m^3/day)	5,811.47											
	Overall Max Flow (R6 ; C5 - C23)	(m^3/day)	6,120.52											
		l/s	70.84											

2 Units Intermittent Mode Operation + 1 Unit Standby

Sheet 4 of 4

Row No.	Descriptions	Units	C1	C2	C3	C4	C5	C6	C7	C8	C9	C10	C11	C12
			5:00 AM	6:00 AM	7:00 AM	8:00 AM	9:00 AM	10:00 AM	11:00 AM	12:00 PM	1:00 PM	2:00 PM	3:00 PM	4:00 PM
R1	Pump No.1	(m^3/day)	OFF	OFF	OFF	OFF	9,195.15	9,111.47	8,948.59	8,788.36	8,630.83	8,476.06	8,328.54	8,205.83
R2	Pump No.2	(m^3/day)	OFF	OFF	OFF	OFF	9,193.51	9,109.83	8,946.98	8,786.78	8,629.27	8,474.52	8,327.03	8,204.34
R3	Sum of Flow for 2 Units	(m^3/day)	---	---	---	---	18,388.66	18,221.30	17,895.57	17,575.14	17,260.10	16,950.58	16,655.57	16,410.17
R4	Ave. Flow each Unit	(m^3/day)	---	---	---	---	9,194.33	9,110.65	8,947.79	8,787.57	8,630.05	8,475.29	8,327.79	8,205.09
R5	Max Flow	(m^3/day)	---	---	---	---	9,195.15	9,111.47	8,948.59	8,788.36	8,630.83	8,476.06	8,328.54	8,205.83

Row No.	Descriptions	Units	C13	C14	C15	C16	C17	C18	C19	C20	C21	C22	C23	C24
			5:00 PM	6:00 PM	7:00 PM	8:00 PM	9:00 PM	10:00 PM	11:00 PM	12:00 AM	1:00 AM	2:00 AM	3:00 AM	4:00 AM
R1	Pump No.1	(m^3/day)	OFF	OFF	OFF	OFF	OFF	OFF	9,192.04	8,979.79	8,769.83	8,562.23	8,357.06	OFF
R2	Pump No.2	(m^3/day)	OFF	OFF	OFF	OFF	OFF	OFF	9,190.40	8,978.18	8,768.25	8,560.68	8,355.54	OFF
R3	Sum of Flow for 2 Units	(m^3/day)	---	---	---	---	---	---	18,382.44	17,957.97	17,538.08	17,122.91	16,712.60	---
R4	Ave. Flow each Unit	(m^3/day)	---	---	---	---	---	---	9,191.22	8,978.99	8,769.04	8,561.46	8,356.30	---
R5	Max Flow	(m^3/day)	---	---	---	---	---	---	9,192.04	8,979.79	8,769.83	8,562.23	8,357.06	---
	Overall Ave. Flow 2 Units (R3 ; C5 - C23)	(m^3/day)	17,467.01											
	Overall Ave. Flow each Unit (R4 ; C5 - C23)	(m^3/day)	8,777.54											
	Overall Max Flow (R5 ; C5 - C23)	(m^3/day)	9,195.15											
		l/s	106.43											

Table 5.7 Summary of Electrical Energy Consumed (Output of EPANET Simulation)

Continuous Mode Operation

Pump	Usage Factor	Avg. Effic.	Kw-hr /m3	Avg. kW	Peak kW	Cost/day
P1	79.17	65	0.23	56.82	58.28	3,238.46
P2	79.17	65	0.23	56.81	58.28	3,238.30
					Total Cost: Rs.	6,476.77
P1	79.17	65	0.21	33.51	33.98	1910.21
P2	79.17	65	0.21	33.51	33.98	1910.13
P3	79.17	65	0.21	33.51	33.98	1910.1
					Total Cost: Rs.	5730.43
P1	79.17	65	0.22	26.09	26.56	1,487.12
P2	79.17	65	0.22	26.09	26.55	1,487.02
P3	79.17	65	0.22	26.09	26.55	1,486.97
P4	79.17	65	0.22	26.09	26.55	1,486.96
					Total Cost: Rs.	5,948.07
P1	79.17	65	0.21	19.87	20.12	1,132.80
P2	79.17	65	0.21	19.87	20.12	1,132.74
P3	79.17	65	0.21	19.87	20.12	1,132.70
P4	79.17	65	0.21	19.87	20.12	1,132.68
P5	79.17	65	0.21	19.87	20.12	1,132.68
					Total Cost: Rs.	5,663.59

Intermittent Mode Operation

Pump	Usage Factor	Avg. Effic.	Kw-hr /m3	Avg. kW	Peak kW	Cost/day
P1	54.17	65	0.23	87.35	89.97	3,406.50
P2	54.17	65	0.23	87.34	89.96	3,406.14
					Total Cost: Rs.	6,812.65
P1	54.17	65	0.23	58.72	60.46	2290.08
P2	54.17	65	0.23	58.71	60.45	2289.66
P3	54.17	65	0.23	58.71	60.44	2289.54
					Total Cost: Rs.	6869.27
P1	54.17	65	0.25	48.72	61.09	1,900.17
P2	54.17	65	0.25	48.71	61.09	1,899.62
P3	54.17	65	0.25	48.7	61.09	1,899.35
P4	54.17	65	0.25	48.7	61.09	1,899.28
					Total Cost: Rs.	7,598.42
P1	54.17	65	0.23	34.26	35.3	1,335.96
P2	54.17	65	0.23	34.25	35.3	1,335.64
P3	54.17	65	0.23	34.24	35.29	1,335.46
P4	54.17	65	0.23	34.24	35.29	1,335.37
P5	54.17	65	0.23	34.24	35.29	1,335.34
					Total Cost: Rs.	6,677.77

Note:
Cost of Energy per kWH = Rs. 3

Table 5.8 Unit Outage and Capacity Outage Probability table For Continuous mode Operation

Units of Pumps		Max Flow (each unit)	
In Duty	Stand by		
2	1	6,039 CMD	70 l/sec
3	1	4,028 CMD	47 l/sec
4	1	3,020 CMD	35 l/sec
5	1	2,332 CMD	27 l/sec

Unit Outage Probability Table (2 + 1) Units

Unit out of service 3 x 70 l/sec	Capacity out of service l/sec	Individual Probability (R = 0.96, Q = 0.04)		Expected Outage (l/sec)
0	0	0.96^3	= 0.884736	0
1	70	$3 \times 0.96^2 \times 0.04$	= 0.110592	7.73
2	140	$3 \times 0.96 \times 0.04^2$	= 0.004608	0.64
3	210	0.04^3	= 0.000064	0.01
		SUM	= 1.000000	8.39

Unit Outage Probability Table (3 + 1) Units

Unit out of service 4 x 47 l/sec	Capacity out of service l/sec	Individual Probability (R = 0.94, Q = 0.06)		Expected Outage l/sec
0	0	0.94^4	= 0.780749	0
1	47	$4 \times 0.94^3 \times 0.06$	= 0.199340	9.29
2	94	$6 \times 0.94^2 \times 0.06^2$	= 0.019086	1.79
3	140	$4 \times 0.94 \times 0.06^3$	= 0.000812	0.11
4	186	0.06^4	= 0.000013	0.00
		SUM	= 1.000000	11.20

Unit Outage Probability Table (4 + 1) Units

Unit out of service 5 x 35 l/sec	Capacity out of service l/sec	Individual Probability (R = 0.92, Q = 0.08)		Expected Outage l/sec
0	0	0.92^5	= 0.659082	0
1	35	$5 \times 0.92^4 \times 0.08$	= 0.286557	10.02
2	70	$10 \times 0.92^3 \times 0.08^2$	= 0.049836	3.48
3	105	$10 \times 0.92^2 \times 0.08^3$	= 0.004334	0.45
4	140	$5 \times 0.92 \times 0.08^4$	= 0.000188	0.03
5	175	0.08^5	= 0.000003	0.00
		SUM	= 1.000000	13.98

Unit Outage Probability Table (5 + 1) Units

Unit out of service 6 x 27 l/sec	Capacity out of service l/sec	Individual Probability (R = 0.90, Q = 0.10)		Expected Outage l/sec
0	0	0.90^6	= 0.531441	0
1	27	$6 \times 0.90^5 \times 0.10$	= 0.354294	9.56
2	54	$15 \times 0.90^4 \times 0.10^2$	= 0.098415	5.31
3	81	$20 \times 0.90^3 \times 0.10^3$	= 0.014580	1.18
4	108	$15 \times 0.90^2 \times 0.10^4$	= 0.001215	0.13
5	135	$6 \times 0.10^5 \times 0.90$	= 0.000054	0.01
6	162	0.10^6	= 0.000001	0.00
		SUM	= 1.000000	16.19

Table 5.9 Unit Outage and Capacity Outage Probability table For Intermittent mode Operation

Units of Pumps		Max Flow (each)	
In Duty	Stand by		
2	1	9,195 CMD	106 l/sec
3	1	6,121 CMD	71 l/sec
4	1	4,589 CMD	53 l/sec
5	1	3,678 CMD	43 l/sec

Unit Outage Probability Table (2 + 1) Units

Unit out of service 3 x 106 l/sec	Capacity out of service l/sec	Individual Probability (R = 0.96, Q = 0.04)	Expected Outage l/sec
0	0	$0.96^3 = 0.884736$	0
1	106	$3 \times 0.96^2 \times 0.04 = 0.110592$	11.77
2	213	$3 \times 0.96 \times 0.04^2 = 0.004608$	0.98
3	319	$0.04^3 = 0.000064$	0.02
		SUM = 1.000000	12.77

Unit Outage Probability Table (3 + 1) Units

Unit out of service 4 x 71 l/sec	Capacity out of service l/sec	Individual Probability (R = 0.94, Q = 0.06)	Expected Outage l/sec
0	0	$0.94^4 = 0.780749$	0
1	71	$4 \times 0.94^3 \times 0.06 = 0.199340$	14.12
2	142	$6 \times 0.94^2 \times 0.06^2 = 0.019086$	2.70
3	213	$4 \times 0.94 \times 0.06^3 = 0.000812$	0.17
4	283	$0.06^4 = 0.000013$	0.00
		SUM = 1.000000	17.00

Unit Outage Probability Table (4 + 1) Units

Unit out of service 5 x 53 l/sec	Capacity out of service l/sec	Individual Probability (R = 0.92, Q = 0.08)	Expected Outage l/sec
0	0	$0.92^5 = 0.659082$	0
1	53	$5 \times 0.92^4 \times 0.08 = 0.286557$	15.22
2	106	$10 \times 0.92^3 \times 0.08^2 = 0.049836$	5.29
3	159	$10 \times 0.92^2 \times 0.08^3 = 0.004334$	0.69
4	212	$5 \times 0.92 \times 0.08^4 = 0.000188$	0.04
5	266	$0.08^5 = 0.000003$	0.00
		SUM = 1.000000	21.25

Unit Outage Probability Table (5 + 1) Units

Unit out of service 6 x 43 l/sec	Capacity out of service l/sec	Individual Probability (R = 0.90, Q = 0.10)	Expected Outage l/sec
0	0	$0.90^6 = 0.531441$	0
1	43	$6 \times 0.90^5 \times 0.10 = 0.354294$	15.08
2	85	$15 \times 0.90^4 \times 0.10^2 = 0.098415$	8.38
3	128	$20 \times 0.90^3 \times 0.10^3 = 0.014580$	1.86
4	170	$15 \times 0.90^2 \times 0.10^4 = 0.001215$	0.21
5	213	$6 \times 0.10^5 \times 0.90 = 0.000054$	0.01
6	255	$0.10^6 = 0.000001$	0.00
		SUM = 1.000000	25.54

Table 5.10. Summary of The Pumping Units Analysis Continuous Mode Operation

	Description	Units	Unit of Pumps Installed (+1 unit stantdby)			
			2+1	3+1	4+1	5+1
RESULT OF EPANET SIMULATION	Filling Duration	hrs.	19	19	19	19
	Average Flow of all Pumps (except 1 standby)	m^3/day	11,526	11,532	11,527	11,542
	Max Flow of Pump (P1)	m^3/day	6,039	4,028	3,020	2,332
	Flow Design of Pumps	m^3/day	7,310	3,840	3,055	2,265
	Head Design of Pumps	m	50	50	50	50
	Usage Factor	%	79.17	79.17	79.17	79.17
	Average Efficiency	%	65.00	65.00	65.00	65.00
	kw-hr / m^3	kw-hr/m^3	0.23	0.21	0.22	0.21
	Average Kw	kW	56.82	33.51	26.09	19.87
	Peak Kw	kW	58.28	33.98	26.56	20.12
	Energy Cost per Day (per unit)	Rs.	3,238	1,910	1,487	1,133
	Energy Cost per Day (except 1 unit standby)	Rs.	6,477	5,730	5,948	5,664
EVALUATION	HP (Peak)	HP	79.26	46.21	36.12	27.36
	Capacity Available in The Common Market	HP	80	46	36	27
	Pump Price per Unit	Rs.	72,500.00	48,000.00	41,000.00	35,200.00
	Pump Cost	Rs.	217,500.00	192,000.00	205,000.00	211,200.00
	Life Time (Years) * after rounded		4.326	3.845	3.605	3.461
	Life Time (Years) * after rounded to nearest digit	Years	5	4	4	4
	Depreciation per year = (0.9 x Pump Cost)/Lifetime	Rs.	39,150.00	43,200.00	46,125.00	47,520.00
	Energy Cost (electrical) per Year (@ Rs.3 per kwh)	Rs.	2,364,021.05	2,091,606.95	2,171,045.55	2,067,210.35
	Maintenance and Repair Cost = 0.7 x Depr. Cost	Rs.	27,405.00	30,240.00	32,287.50	33,264.00
	Investment Cost $P_{average}$ (Pav)	Rs.	139,200.00	127,200.00	135,812.50	139,920.00
	Annual Cost of the Pumping Units	Rs.	2,569,776.05	2,292,246.95	2,385,270.55	2,287,914.35
	Present Worth (i = 8%)	Rs.	10,260,370.63	7,592,212.65	7,900,318.61	7,577,862.53
	Expected Outage	l/sec	8.39	11.20	13.98	16.19
	Calculation of Unit Pumping Transfer Cost per m^3 of Water Based on Annual Cost					
	Production Capacity per Year	m^3	3,330,636.93	3,332,317.64	3,330,861.12	3,335,096.23
	Losses due to Expected Outage per Year	m^3	209,387.76	279,695.42	349,031.62	404,308.77
	Usage Production per Year	m^3	3,121,249.16	3,052,622.22	2,981,829.51	2,930,787.47
	Cost per Unit m^3	Rs.	0.8233	**0.7509**	0.7999	0.7806
	Ranking (Lowest Cost) Based on Annual Cost		4	**1**	3	2
	Calculation of Unit Pumping Transfer Cost per m^3 of Water Based on Over Lifetime					
	Production Capacity	m^3	16,653,184.63	13,329,270.57	13,323,444.49	13,340,384.93
	Losses due to Expected Outage	m^3	1,046,938.81	1,118,781.67	1,396,126.46	1,617,235.07
	Usage Production	m^3	15,606,245.82	12,210,488.90	11,927,318.03	11,723,149.86
	Cost per Unit m^3 Based on Over Lifetime	Rs.	0.6575	**0.6218**	0.6624	0.6464
	Ranking (Lowest Cost) Based on Over Lifetime		3	1	4	2

Note :

Unit Price of Pumps; see Appendix-1
Salvage value = 10%
Life Time (in year) calculated based on hourly of 20,000 hrs lifetimes
Depreciation calculated based on the Straight Line Method
Annual maintenance and repair cost is 70% of Depreciation
Investment Cost = $Pav - \frac{P(n+1) + S(n-1)}{2n}$
n = Life in years ; P = Total Initial Cost ; Pav = Average Value ; S = Salvage Value

Table 5.11. Summary of The Pumping Units Analysis
Intermittent Mode Operation

	Description	Units	Unit of Pumps Installed (+1 unit standby)			
			2+1	3+1	4+1	5+1
RESULT OF EPANET SIMULATION	Filling Duration	hrs.	8 + 5 = 13	8 + 5 = 13	8 + 5 = 13	8 + 5 = 13
	Average Flow of all Pumps (except 1 standby)	m^3/day	17,467	17,434	17,454	17,457
	Max Flow of Pump per Unit	m^3/day	9,195	6,121	4,589	3,678
	Flow Design of Pumps	m^3/day	11,415	7,790	4,695	4,360
	Head Design of Pumps	m	50	50	50	50
	Usage Factor	%	54.17	54.17	54.17	54.17
	Average Efficiency	%	65.00	65.00	65.00	65.00
	kw-hr / m^3	kw-hr/m^3	0.23	0.23	0.21	0.23
	Average Kw	kW	87.35	58.72	39.60	34.26
	Peak Kw	kW	89.97	60.46	40.74	35.30
	Energy Cost per Day (per unit)	Rs.	3,407	2,290	1,545	1,336
	Energy Cost per Day (except 1 unit standby)	Rs.	6,813	6,869	6,177	6,678
EVALUATION	HP (Peak)	HP	122.36	82.23	55.41	48.01
	Capacity Available in The Common Market	HP	120	80	55	50
	Pump Price per Unit	Rs.	108,900.00	72,500.00	52,000.00	49,900.00
	Pump Cost	Rs.	326,700.00	290,000.00	260,000.00	299,400.00
	Life Time (Years) * after rounded		6.32	5.62	5.27	5.06
	Life Time (Years) * after rounded to nearest integer	Years	6	6	5	5
	Depreciation per year = (0.9 x Pump Cost)/Lifetime	Rs.	49,005.00	43,500.00	46,800.00	53,892.00
	Energy Cost (electrical) per Year (@ Rs.3 per kwh)	Rs.	2,486,617.25	2,507,283.55	2,254,743.70	2,437,386.05
	Maintenance and Repair Cost = 0.7 x Depr. Cost	Rs.	34,303.50	30,450.00	32,760.00	37,724.40
	Investment Cost $P_{average}$ (Pav)	Rs.	204,187.50	181,250.00	166,400.00	191,616.00
	Annual Cost of the Pumping Units	Rs.	2,774,113.25	2,762,483.55	2,500,703.70	2,720,618.45
	Present Worth (i = 8%)	Rs.	12,824,391.73	12,770,629.03	9,984,584.76	10,862,640.59
	Expected Outage	l/sec	12.77	17.00	21.25	25.54
	Calculation of Production Cost per m^3					
	Production Capacity per Year	m^3	3,453,372.83	3,446,927.54	3,450,878.05	3,451,330.04
	Losses due to Expected Outage per Year	m^3	218,154.93	290,418.67	362,945.84	436,307.50
	Usage Production per Year	m^3	3,235,217.89	3,156,508.86	3,087,932.21	3,015,022.55
	Cost per Unit m^3	Rs.	0.8575	0.8752	**0.8098**	0.9024
	Ranking (Lowest Cost) Based on Annual Cost		2	3	1	4
	Calculation of Unit Pumping Transfer Cost per m^3 of Water Based on Over Lifetime					
	Production Capacity	m^3	20,720,236.96	20,681,565.21	17,254,390.26	17,256,650.22
	Losses due to Expected Outage	m^3	1,308,929.60	1,742,512.04	1,814,729.20	2,181,537.48
	Usage Production	m^3	19,411,307.36	18,939,053.17	15,439,661.06	15,075,112.74
	Cost per Unit m^3 Based on Over Lifetime	Rs.	0.6607	0.6743	**0.6467**	0.7206
	Ranking (Lowest Cost) Based on Over Lifetime		2	3	1	4

Note :

Unit Price of Pumps; see Appendix-1
Salvage value = 10%
Life Time (in year) calculated based on hourly of 20,000 hrs lifetimes
Depreciation calculated based on the Straight Line Method
Annual maintenance and repair cost is 70% of Depreciation
Investment Cost = $P_{av} = \frac{P(n+1)+S(n-1)}{2n}$
n = Life in years ; P = Total Initial Cost ; Pav = Average Value ; S = Salvage Value

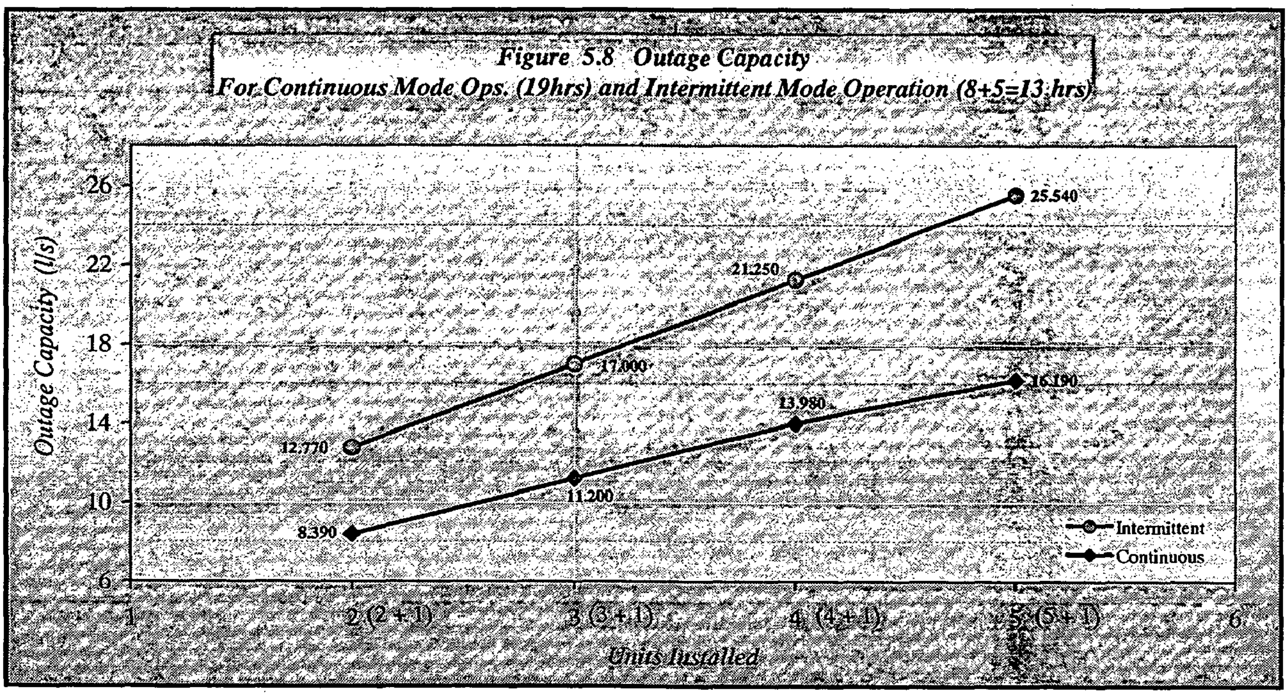


Figure 5.8 Outage Capacity
For Continuous Mode Ops. (19hrs) and Intermittent Mode Operation (8+5=13 hrs)

5.7. DISCUSSION OF RESULTS

From the outcome of the simulation under continuous mode operation (Table 5.10) for all of unit pump combinations at design head of 50m, and average efficiency of 65%, it can be noticed that the duration of operation either for less or more number unit of pump combinations are same i.e., 19 hours (first row of the table 5.10) with the average flows of all pumps (except 1 unit as standby) being nearly same i.e. about 11,530 CMD (2nd row of the table 5.10) and usage factor of pump for one day is 79.17% (6th row of the table 5.10).

For Intermittent mode operation (Table 5.11) with same design parameter same as above, it can be determined that the duration of operation either for less or more number unit of pumps combination are also same i.e., 8+5 = 13 hours (2nd row of the table 5.11) with average flow of all pumps working (except 1 unit as standby) being nearly same i.e. about 17,500 CMD (2nd row of table 5.11) and usage factor of pump at 54.17% (6th row of the table 5.11).

Considering energy cost @ Rs.3 per kWh, peak energy consumed each unit for continuous mode operation vary from 20.12 – 58.28 kW (9th row of the table 5.10) for units number of pump combination 5+1 to 2+1 respectively, which have influence on energy cost per unit per day from Rs. 1,133 for unit with number of pump combinations of 5+1 to Rs. 3,328 for pump combinations of 2+1. For intermittent mode operation from 35.3 – 89.97 kW for the units at pump combinations of 5+1 to 2+1 respectively, which have influence to energy cost per unit per day from Rs. 1,336 for units with pump combinations of 5+1 to Rs. 3,407 for units with pump combinations of 2+1.

Based on 20,000 hrs lifetime, lifetime of pumps (in year) will vary depending upon duration of operation of each pump. Since the parameter in calculation formula either for depreciation, present worth, so on in term of year, it is not simply to analyze the system with having fraction number of lifetime. For simplicity in calculating purpose, therefore the fraction number has to be rounded up to next integer number.

The significant cash flows in this study have to determined by combined costs associated with several costs i.e., purchase expanse, energy cost based on Rs.3 per kWh, salvage value (10% of initial cost of the pumps), depreciation (by using straight line method), investment cost with salvage value 10% and maintenance and repair cost (70% of depreciation cost).

Transfer cost defined by considering losses due to system failure (Outage capacity), depends upon reliability of the system and the capacity of the pumps used (see table 5.6 and 5.7).

In Continuous mode operation (table 5.10), the lowest transfer cost in the system with combination of 3+1, whether calculated on the annual cost or present worth. When the calculation is based on annual cost, the lowest transfer cost will be Rs. 0.7509 per m^3, and when the calculation is based on the present worth, the transfer cost will be Rs. 0.6218 per m^3.

In intermittent mode operation (table 5.11), the lowest transfer cost are also in the same combination whether the calculation is based on the annual cost or based on present worth, i.e. in the units with combination of 3+1. When the calculation is based on annual cost, the transfer cost will be Rs. 0.8098 per m^3, and when the calculation is based on the present worth, the transfer cost will be Rs. 0.6467 per m^3.

Hence the system under Continuous Mode Operation gives the lowest value i.e. Rs. 0.7695 per m^3 (calculation is based on the annual cost) and Rs. 0.6218 when calculation is based on present worth and the combination of the pumps in the system with combination of 3+1.

Based on the result of simulation with three pump (3+1) indicated that every pump have to work during 19 hours per day hence the duration of pumps operation on the system with combination of 3+1 (1 pump standby while the three other are working) is 14.75 hours per day per unit. The operation time schedule will be as shown below.

Operation Schedule for Continuous Mode Operation (19 hrs)
With system combination of 3+1

Pumps	Operation (hours)				
	4.75 hrs 9:00AM – 1:45PM	4.75 hrs 1:45PM – 6:30PM	4.75 hrs 6:30PM – 11:15PM	4.75 hrs 11:15PM – 3:00AM	19.00 hrs
P1	Standby	▬	▬	▬	14.25 hrs
P2	▬	Standby	▬	▬	14.25 hrs
P3	▬	▬	Standby	▬	14.25 hrs
P4	▬	▬	▬	Standby	14.25 hrs

From the reliability point of view, the expected outage of the system can be seen at figure 5.8. Losses due to failure of the system show tendency to increase with the number of pump combinations, but the losses expressed in terms of per unit of pump in fact follow at decreasing trend with increase in the number of pump combinations. At continuous mode operation, the lowest amount of losses is 8.390 l/s till 16.190 l/s., while on the systems with intermittent mode operation, the lowest amount of losses (expected outage) is 12.770 l/s. which may occur in the system with pump combinations of 2+1 and highest level of 25.540 l/s at intermittent mode operation with pumps combination of 5+1.

In the intermittent mode operation, the level of the losses due to system failure is higher when compared to the systems with continuous mode operation. This is because of the following reasons:

(a) Decrease in the total working time to fulfill demand capacity because pump capacities on the system with particular number of pump combinations is higher which also means the increased water pumped individually. Since the loss due to outage of the units is function of the flow, so more flow will produce more losses for individual unit.

(b) Degree of reliability for one system with same capacity will be lower at systems having more units compared to systems having a few units. This is because of the MTTF factor of the smaller pumps are smaller. This is due to the fact that certain level of clearance accuracy for small parts are more difficult to be achieved as compared to bigger part, and it will give more chance to affect either the decreasing performance capability or increasing noise, chatter, etc. which may cause more failure.

Based on the schedule above and noting that there is four times changing over, which means that there is four times load fluctuation occur in the system. When the pump discharge rate decreases toward zero at the shut-off head, the radial load increases and the re-circulation of the pumped fluid within the impeller becomes a problem. This re-circulation can cause vibration and may also result in cavitation that can ruin the impeller. Pumps should not be operated at or near the shut-off head for extended period of time because such operation also leads to heating of the liquid and can cause severe wearing-ring rub and other problems, which can decrease the life time of the pumping unit. For more detail explanation see Chapter 3.

However, the rescheduling the operation hours will also result in the similar situation and thus for shutting down and or running up of the centrifugal pump the following characteristics should be considered:

(i) Head v/s Discharge Curve
  - ➢ Corresponding to zero or no discharge (i.e., when the pump is started but delivery valve is closed) the head developed is maximum which is known as the shut off head of the pumps)
  - ➢ With the increase in the discharge the head produced by the pump decreases. The maximum discharge being limited by a certain minimum head, below which the pump will not work.

(ii) Input Power v/s Discharge Curve
  - ➢ Corresponding to zero discharge the power input will be minimum
  - ➢ With the increase in the discharge the input power increases. Under normal or non-overloading condition this increase in the input power continues up to a certain discharge (usually the designed discharge).

(iii) Water hammer is result of an event, which is associated with a rapid velocity (or pressure) change, the result of an accident or an abnormal operation on the pumping and or pipeline system.

5.8. CONCLUSION

Component of water distribution system are:

1. Sources of supply
2. Pumping system
3. Water Treatment
4. Service Reservoir
5. Distribution system

Pumping stations are one of the major components of a water supply system. A pumping station consists of one or more pumping units supported by appropriate electrical, piping, control, and structural components.

The pumping unit is the primary subcomponent within the pumping station and includes four major sub-subcomponents: pump, driver (motor, engine, etc.), power transmission, and controls.

Number of Pumps defined by calculation of lowest transfer cost, this is by combined costs associated with several costs i.e., purchase expanse, energy cost, salvage value, depreciation cost, investment cost and maintenance and repair cost.

The mechanical reliability of the entire pumping station is difficult to evaluate because of the difficulties involved in defining failure. Does the pump station fail because of the failure of one pumping unit, etc? Probably not. Sophisticated techniques such as fault tree analysis can be used to perform detailed evaluations of pumping station reliability (Henley and Kumamoto, 1981). In addition, detailed failure and repair data are seldom available for input to these complex computational methods. Simplified techniques based on common probability concepts and a review of the pumping station design assumptions can be used to estimate the mechanical reliability of the pumping station.

Accurate calculation of the mechanical reliability requires knowledge of the precise reliability of the basic components and the impact on mission accomplishment caused by the set of all possible failures. Thus, for a large system with many interactive components, such as a water distribution system, it is extremely difficult to analytically compute the mechanical reliability. There is no comprehensive database of failure and repair information for components and subcomponents for water distribution systems.

Closed Conduit. Several factors, which need to be considered when designing such, closed conduit systems. The primary functions of water distribution systems are

- ✓ To meet the water demands of users while maintaining acceptable pressures in the system
- ✓ Supply water for fire protection at specific locations within the system, while maintaining acceptable pressures for normal service

- ✓ Provide sufficient level of redundancy to support minimum level of service during emergency conditions (ie. power loss or water main failure)

Centrifugal Pumps. There are many different types of pumps, but in almost all cases pumps in water-distribution are *centrifugal* pumps because of their low cost, simplicity, and reliability in range of flows and head encountered. A centrifugal pump is any pump in which fluid is energized by a rotating impeller, whether the flow is radial, axial, or a combination of both (mixed).

Demand Requirement. A water distribution system should be able to provide, during its entire life, the required quantity of water for expected loading conditions with the desired residual pressure at all the demand nodes. In other words, hydraulic reliability, as it relates to water distribution system design, can be defined as the ability of the system to provide service with an acceptable level of interruption in spite of abnormal conditions.
Hydraulic failure occurs when the system cannot supply the expected amount of water to the specified location, at the specified pressure and at the specified time. As the respected demand conditions may vary with space and time

Demand Capacity. Based upon the estimated per capita consumption and the design population, plus the demand for institutions, commercial establishments, factories and irrigation, the design average demand can be calculated. Based upon the considerations, the design year, average demand, maximum-day demand, and peak or maximum hour demand should be calculated. Prior to determining the peak demand, a decision needs to be made whether the system is to provide for fire fighting. The capacity of most of the system is based-on the maximum day demand.

Peak Demand. Only the distribution system, including service reservoirs (overhead tank) is designed for the peak demand, which includes fire protection where this is to be provided. If an impounding reservoir is required on a stream, its capacity is based on the ultimate average demand. The design population should be about 25% greater than present population. Based on a design population of 3000, and 80 liter per connections per day on maximum day (dry season), the design capacity of raw water pumps and treatment plant should be 240 m^3/day. A design capacity of 15 m^3/hr would be satisfactory, as this would permit 16 hours operation.

For other populations, the design capacity would be proportional. Capacities up to 25% greater would be acceptable without additional review.

Back up Power. Most of water system pumps are electrically driven, and power outages do occur. All but the smallest utilities should have some standby pumping capacity with an alternative source of power. If power supply is considered very reliable, it may be adequate to simply provide two or more connections to electrical utilities, preferably at widely separate points in the electrical grid.

Emergency Storage Capacity. In addition to fire storage and equalization storage provided in the tank, utilities should provide for some emergency storage. For smaller utilities, the combined storage should be sufficient to meet demands for approximately 24 hours while the number of hours of storage will be smaller for larger utilities. The amount of storage required is inversely related to the reliability of the water source and treatment. For example, if the utility purchases its water from another system through a single connection, it should have considerable storage. If the utility has several reliable surface and groundwater sources, only fire and equalization storage may be required.

Maintaining High Tank Water Levels. If an emergency should occur, the length of time before the utility is without water will depend heavily on the amount of water in storage at the time of the emergency. Therefore, from a reliability standpoint, the utility should keep its tanks at all times. However, this directly conflicts with the need for the tanks t, "float" in response to temporal variations in water use which in turn red peak energy usage. The operating policies of the utility must balance these conflicting goals. In general, the water level should remain in the top of the tank.

A regular schedule for operating standby equipment should be followed.

- ✓ Water distribution systems are traditionally designed on the basis of "established" design criteria or "rules of thumb" that attempt to provide water service that is 100 percent reliable. This is generally accomplished by providing standby equipment or, as in the case of piping systems, constructing a looped system providing water to a particular point in the system from two or more directions. Reliability engineering concepts can be applied to the evaluation of a water distribution system to provide a quantitative measure of system reliability.

- ✓ Water distribution system operators are concerned about both mechanical reliability and hydraulic reliability. Mechanical reliability is a measure of equipment maintainability whereas hydraulic reliability measures the ability of the system to provide water in the desired amounts and at appropriate pressures. Analytical techniques are available to quantify the mechanical reliability of water distribution system components. Since nearly all components of a water distribution system are repairable, most of these techniques are based on the concept of availability. The evaluation of hydraulic reliability requires the use of simulation modeling to evaluate system performance.

Overhead Tank (Service Reservoir). Tank location and elevation effect pipe size and pump requirements. The best choice for service storage, if available, is a ground tank at an elevated location near the community to be served. If the land is flat, the storage tank must be elevated on a tower; it is convenient to place the tank either at the site of the treatment plant or pumping station (to facilitate operation and maintenance), or at a central location in the distribution network. If the area to be served stretches over a long distance, with the source at one end, the reservoir might be built at the other end of town. This allows the tank to be lower and the distribution pipes to be smaller, because water would be fed to consumers from two directions. Such a location provides added reliability in event of a break in transmission main.

Distribution Network. Distribution systems are classified as branched network and looped networks. Branched networks are relatively easy to design and are less costly to construct in small systems. However, they are less reliable, as any break or maintenance work in the network results in the cutting off the supply to part of the system and the dead ends impair water quality. A network can be started with branched system, with the ends of the branches "looped" at a later time to improve flow conditions and provide greater reliability.

EPANET is a Windows 95/98/NT program that performs extended period simulation of hydraulic and water-quality behavior within pressurized pipe networks. A network can consist of pipes, nodes (pipe junctions), pumps, valves and storage tanks or reservoirs. EPANET tracks the flow of water in each pipe, the pressure at each node, the height of water in each tank, and the concentration of a chemical species throughout the network during a simulation period comprised of multiple time steps. In addition to chemical species, water age and source tracing can also be simulated.

EPANET was specifically developed to help water utilities maintain and improve the quality of water delivered to consumers through their distribution systems. It can be used to design sampling programs, study disinfectant loss and by-product formation, and conduct consumer exposure assessments. It can assist in evaluating alternative strategies for improving water quality such as altering source utilization within multi-source systems, modifying pumping and tank filling/emptying schedules to reduce water age, utilizing booster disinfections stations at key locations to maintain target residuals, and planning a cost-effective program of targeted pipe cleaning and replacement.

EPANET can also be used to plan and improve a system's hydraulic performance. Pipe, pump and valve placement and sizing, energy minimization, fire flow analysis, vulnerability studies, and operator training are just some of the activities that EPANET can assist with.

5.9. SCOPE FOR FURTHER STUDY

Further work can be carried out involving reliability of a water distribution system. Since very less data are available for various components of water distribution system for reliability analysis purpose, it is suggested that a data base on component failures and repairs should be generated, so that realistic reliability can be estimated for the future work.

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APPENDICES

Appendix 1

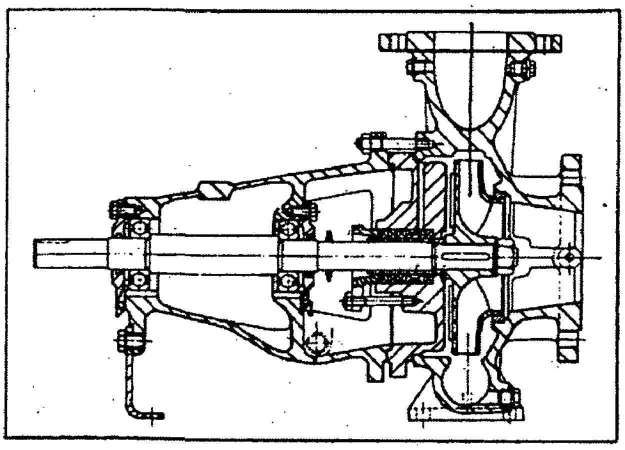
Prices List of Centrifugal pumps

Power HP	Price Rs.	Power HP	Price Rs.	Power HP	Price Rs.
1	8000	21	31000	60	56000
2	8500	22	32000	65	61000
3	8900	23	32200	70	66000
4	9300	24	32700	75	69500
5	11000	25	33500	80	72500
6	14000	26	35000	85	78000
7	14200	27	35200	90	82300
8	14400	28	35700	95	84000
9	14600	29	36000	100	91000
10	15400	30	36300	110	99500
11	17000	31	36500	120	108900
12	18000	32	37000	130	120500
13	19000	33	37500	140	127300
14	20000	34	39000	150	139500
15	21000	35	39500	160	147800
16	22000	36	41000	200	169500
17	24000	40	43000	205	195000
18	26000	45	48000	210	215300
19	28000	50	49900		
20	30000	55	52000		

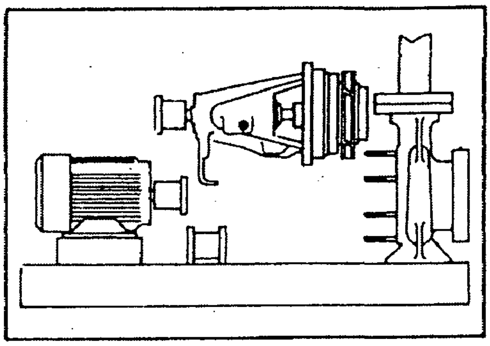
Appendix 2

Cross Sectional View and Performance Characteristics Of Centrifugal Pump

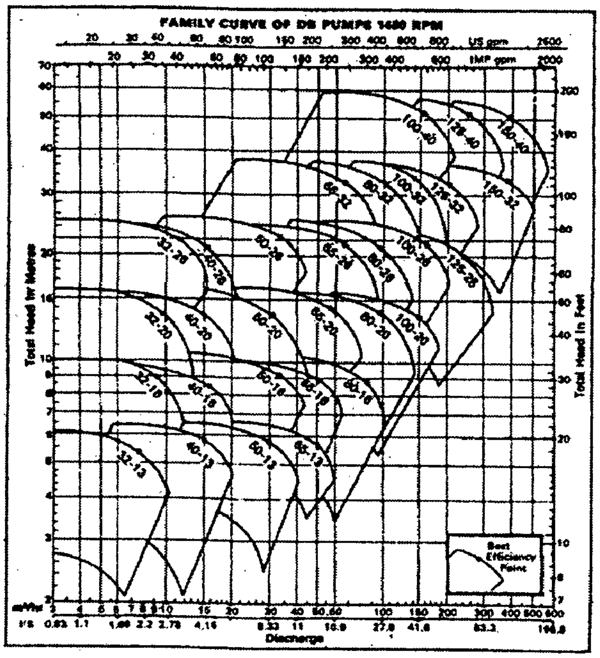
CROSS SECTIONAL VIEW

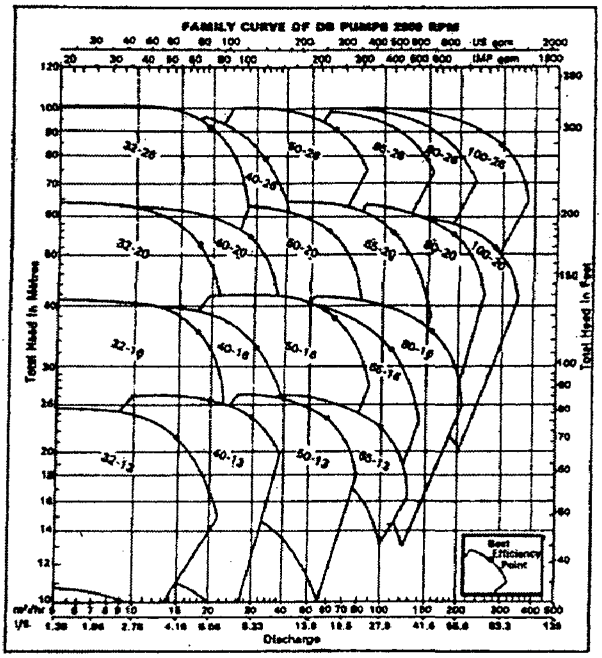


BACK PULL OUT ARRANGEMENT



PERFORMANCE CHARACTERISTICS





Specifications are subject to change without notice.

KIRLOSKAR BROTHERS LIMITED

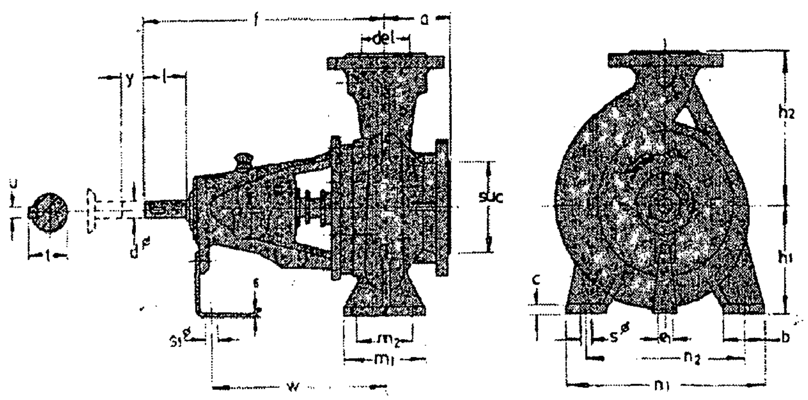
REGD. & SALES HEAD OFFICE: UDYOG BHAVAN, TILAK ROAD, PUNE-411 002 (INDIA)

Regional Offices: Ahmedabad, Bangalore, Bhubaneswar, Calcutta, Indore, Jaipur, Madras, New Delhi, Pune and Secunderabad

Appendix 3

General Dimension of Centrifugal pumps (Kirloskar)

General Dimension



WORKING PRESSURES AND FLANGE DRILLING

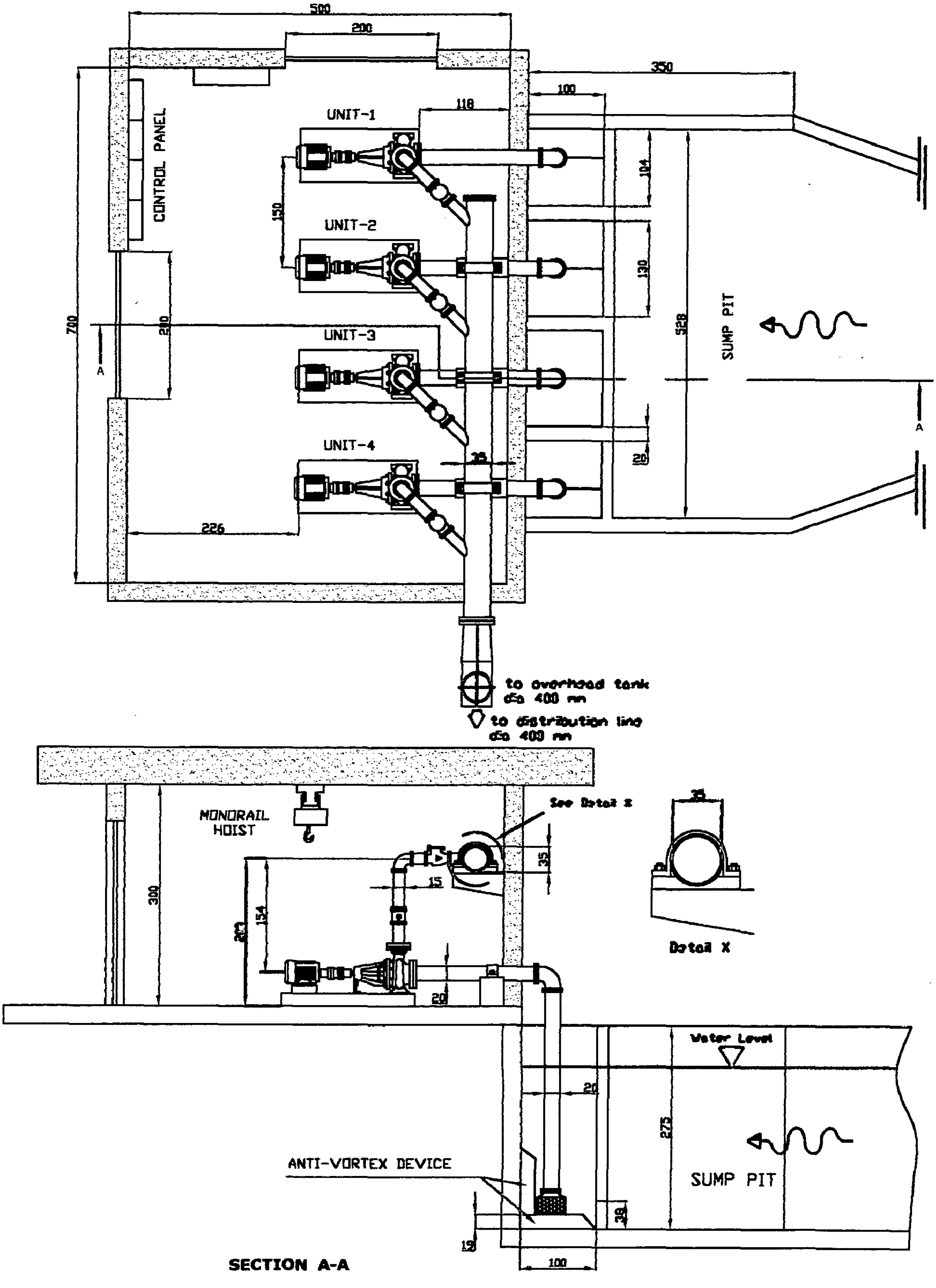
PUMP MODELS	Pressure in Kg/cm²		Flange Standard & Drilling
	Working upto 95°C	Hydraulic Test	
32/13, 32/16, 32/20, 32/26 40/13, 40/16, 40/20, 40/26 50/16, 50/20, 50/26 100/20, 100/26, 100/32 125/26, 125/32.	16.0	21.0	All Companion flanges are drilled to DIN2533 ND 16/BS 4504 NP-16
50/13, 100/40, 125/40	13.0	17.0	
65/13, 65/16, 65/20, 65/26, 65/32 80/16, 80/20, 80/26, 80/32 150/32, 150/50	10.0	16.0	

1. Drillings of flanges as per BS-10 Table D & E can be given only for pump sizes DB 100-20 and above at extra prices.
2. Flange drillings can be given as per Standard ASA Class 150 or BS 4504, NP 10 at extra prices.

Pump							Mounting Dimensions										Shaft End				Dist.	
Model	Del	Suc	a	f	h1	h2	b	c	m1	m2	n1	n2	s	e1	s1	w	d	l	t	u	y	net weight
65-20	65	80	100	360	180	225	65	14	125	95	320	250	14	110	14	267	24	50	27	8	100	52
80-20	80	100	125	470	180	250	65	14	125	95	345	280	14	110	14	342	32	80	35	10	140	70
100-20	100	125	125	470	200	280	80	16	160	120	360	280	18	110	14	342	32	80	35	10	140	85
100-40	100	125	140	530	280	355	100	18	200	150	500	400	23	110	14	370	42	110	45	12	140	177
125-40	125	150	140	530	315	400	100	18	200	150	500	400	23	110	14	370	42	110	45	12	140	188
150-40	150	200	160	530	315	450	100	18	200	150	550	450	23	110	14	370	42	110	45	12	140	205

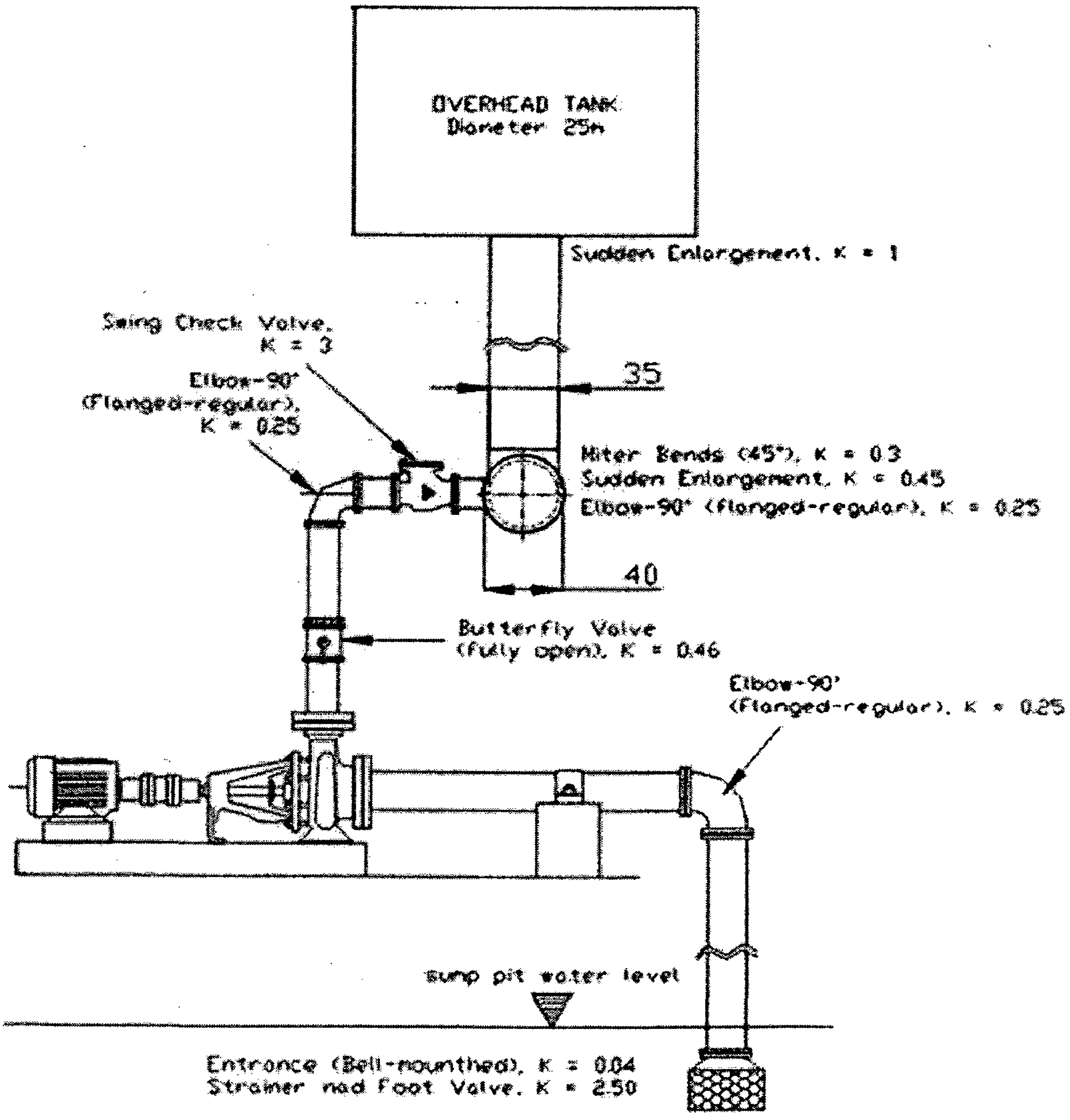
Appendix 4

Pumping System Arrangement



Appendix 5

Minor Loss Coefficients (K) applied on the Simulation



Appendix 6

Minor Loss Coefficients (K) various Components of Pipeline

Appurtenance 1. Sudden Enlargement	Head Loss Coefficient[a]
$D_1 : D_2$	Range of V_1 : 0.6 - 13 m/s
1 : 1.2	0.11 - 0.08
1 : 1.4	0.26 - 0.20
1 : 1.6	0.40 - 0.32
1 : 1.8	0.51 - 0.40
1 : 2.0	0.60 - 0.47
1 : 2.5	0.74 - 0.58
1 : 3.0	0.83 - 0.65
1 : 4.0	0.92 - 0.72
1 : 5.0	0.96 - 0.75

2 Gradual Enlargement							
θ	2°	4°	6°	8°	10°	12°	15°
Kge	0.033	0.039	0.046	0.055	0.078	0.10	0.16
θ	20°	30°	40°	50°	60°	75°	90°
Kge	0.31	0.49	0.60	0.67	0.72	0.72	0.67

Appurtenance	Head Loss Coefficients
3 Exit	
Bell-mouthed	0.1
Sharp-cornered	1
Pipe into still water or air (free discharge)	1 ($V_2 = 0$)

4 Sudden Contraction	
$D_1 : D_2$	Range of V_2: 0 6-13 m/s
1.2 : 1	0.07 - 0.11
1.4 : 1	0.17 - 0.20
1.6 : 1	0.26 - 0.24
1.8 : 1	0.34 - 0.27
2.0 : 1	0.38 - 0.29
2.5 : 1	0.42 - 0.31
3.0 : 1	0.44 - 0.33
4.0 : 1	0.47 - 0.34
5.0 : 1	0.48 - 0.35

Appurtenance	Head Loss Coefficients
5. Gradual Contraction	
Ordinary	0.25
Bell-mouthed	0.10
Streamlined	0.04
6. Entrance	
Bell-mouthed	0.04
Slightly rounded	0.23
Sharp edged	0.50
Re-entrant (pipe projecting into	1.00
Strainer and foot valve	2.50
7. Bends and Elbows	
a. Elbow-45°	
Flanged-regular	0.20 – 0.30
Flanged-long radius	0.18 – 0.20
Screwed-regular	0.30 – 0.42
b. Elbows-90°	
Flanged--regular	0.21 – 0.30
Flanged-long radius	0.18 – 0.20
Intersection of two cylinders (welded pipe-not rounded)	1.25 - 1.8
Screwed-short radius	0.90
Screwed-medium radius	0.75
Screwed-long radius	0.60
c. Miter Bends	
Angle of bend, E3	
5 °	0.016 – 0.024
10 °	0.034 – 0.044
15 °	0.042 – 0.062
22.5 °	0.066 – 0.154
30 °	0.130 – 0.165
45 °	0.236 – 0.320
60 °	0.471 – 0.684
90 °	1.129 – 1.265
d. Return Bend (2 Nos. 90°)	
Flanged-regular	0.38
Flanged-long radius	0.25
Screwed	2.20
8. Tees	
Standard--bifurcating	1.5 – 1.8
Standard-90° turn	1.8
Standard-run o1 tee	0.6
Reducing-run of tee (in terms of velocity at smaller end)	
2:01	0.90
4:01	0.75

Appurtenance	Head Loss Coefficients
9 Obstructions I	
Pipe-area to flow-area-at obstruction ratio	
1.1	0.21
1.4	1.15
1.6	2.40
2.0	5.55
3.0	15.00
40	27.30
5.0	42.00
6.0	57.00
7.0	72.50
10.0	121.00

10. Flow Meters I		
a. Venturi meters		
Throat to Inlet Diameter Ratio	Long Tube Type	Short Tube Type
1 : 3	1.00 - 1.20	2.43
1 : 2	0.44 - 0.52	0.72
2 : 3	0.25 - 0.30	0.32
3 : 4	0.20 - 0.23	0.24

b. Orifice Meters		
Orifice to pipe diameter ratio		
1 : 4	4.80	
1 : 3	2.50	
1 : 2	1.00	
2 : 3	0.40	
3 : 4	0.24	

Appurtenance	Head Loss Coefficients
11 Valves	
a) Gate Valves	
fully open	0.19
three-fourths open	1.15
one-half open	5.60
one-quarter open	24.00
b) Butterfly Valves	
Closure angle, q	
0° (fully open)	0.30
10°	0.46
20°	1.38
30°	3.6
40°	10
50°	31
60°	94

r) Diaphragm Valves	
fully open	2
three-fourths open	3
one-half open	4
one-quarter open	21
d Plug Globe or Stop Valve	
fully open	4
three-fourths open	4.6
one-half open	6
one-quarter open	780
e) Check (Retlux) Valves	
Swing check (fully open)	3
Ball type (fully open)	2.5 – 3.5
Horizontal lift type	8 - 12
f) Fool Valve with Strainer	2.5
g) Pressure Reducing Valve	10