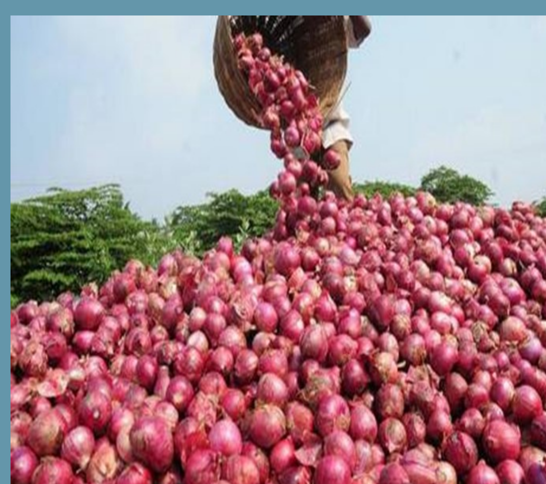
SSIGL 13

NATIONAL GUIDELINES

For Small Scale Irrigation Development in Ethiopia

Irrigation Pump Facilities Study and Design



November 2018

Addis Ababa

MINISTRY OF AGRICULTURE

National Guidelines for Small Scale Irrigation Development in Ethiopia

SSIGL 13: Irrigation Pump Facilities Study and Design

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First Edition 2018

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FORWARD

Ministry of Agriculture, based on the national strategic directions is striving to meet its commitments in which modernizing agriculture is on top of its highest priorities to sustain the rapid, broad-based and fair economic growth and development of the country. To date, major efforts have been made to remodel several important strategies and national guidelines by its major programs and projects.

While efforts have been made to create access to irrigation water and promoting sustainable irrigation development, several barriers are still hindering the implementation process and the performance of the schemes. The major technical constrains starts from poor planning and identification, study, design, construction, operation, and maintenance. One of the main reasons behind this outstanding challenge, in addition to the capacity limitations, is that SSIPs have been studied and designed using many ad-hoc procedures and technical guidelines developed by various local and international institutions.

Despite having several guidelines and manuals developed by different entities such as MoA (IDD)-1986, ESRDF-1997, MoWIE-2002 and JICA/OIDA-2014, still the irrigation professionals follow their own public sources and expertise to fill some important gaps. A number of disparities, constraints and outstanding issues in the study and design procedures, criteria and assumptions have been causing huge variations in all vital aspects of SSI study, design and implementation from region to region and among professionals within the same region and institutions due mainly to the lack of agreed standard technical guidelines. Hence, the SSI Directorate with AGP financial support, led by Generation consultant (GIRDC) and with active involvement of national and regional stakeholders and international development partners, these new and comprehensive national guidelines have been developed.

The SSID guidelines have been developed by addressing all key features in a comprehensive and participatory manner at all levels. The guidelines are believed to be responsive to the prevalent study and design contentious issues; and efforts have been made to make the guidelines simple, flexible and adaptable to almost all regional contexts including concerned partner institution interests. The outlines of the guidelines cover all aspects of irrigation development including project initiation, planning, organizations, site identification and prioritization, feasibility studies and detail designs, contract administration and management, scheme operation, maintenance and management.

Enforceability, standardization, social and environmental safeguard mechanisms are well mainstreamed in the guidelines, hence they shall be used as a guiding framework for engineers and other experts engaged in all SSI development phases. The views and actual procedures of all relevant diverse government bodies, research and higher learning institutions, private companies and development partners has been immensely and thoroughly considered to ensure that all stakeholders are aligned and can work together towards a common goal. Appropriately, the guidelines will be familiarized to the entire stakeholders working in the irrigation development. Besides, significant number of experts in the corresponding subject matter will be effectively trained nationwide; and the guidelines will be tested practically on actual new and developing projects for due consideration of possible improvement. Hence, hereinafter, all involved stakeholders including government & non-governmental organizations, development partners, enterprises, institutions, consultants and individuals in Ethiopia have to adhere to these comprehensive national guidelines in all cases and at all level whilst if any overlooked components are found, it should be documented and communicated to MOA to bring them up-to-date.

Therefore, I congratulate all parties involved in the success of this effort, and urge partners and stakeholders to show a similar level of engagement in the implementation and stick to the guidelines over the coming years.

H.E. Dr. Kaba Urgessa
State Minister, Ministry of Agriculture

SMALL SCALE IRRIGATION DEVELOPMENT VISION

Transforming agricultural production from its dependence on rain-fed practices by creating reliable irrigation system in which smallholder farmers have access to at least one option of water source to increase production and productivity as well as enhance resilience to climate change and thereby ensure food security, maintain increasing income and sustain economic growth.

ACKNOWLEDGEMENTS

The preparation of SSIGLs required extensive inputs from all stakeholders and development partners. Accordingly many professionals from government and development partners have contributed to the realization of the guidelines. To this end MOA would like to extend sincere acknowledgement to all institutions and individuals who have been involved in the review of these SSIGLs for their comprehensive participation, invaluable inputs and encouragement to the completion of the guidelines. There are just too many collaborators involved to name exhaustively and congratulate individually, as many experts from Federal, regional states and development partners have been involved in one way or another in the preparation of the guidelines. The contribution of all of them who actively involved in the development of these SSIGLs is gratefully acknowledged. The Ministry believes that their contributions will be truly appreciated by the users for many years to come.

The Ministry would like to extend its appreciation and gratitude to the following contributors:

- Agriculture Growth Program (AGP) of the MoA for financing the development and publication of the guidelines.
- The National Agriculture Water Management Platform (NAWMP) for overseeing, guidance and playing key supervisory and quality control roles in the overall preparation process and for the devotion of its members in reviewing and providing invaluable technical inputs to enrich the guidelines.
- Federal Government and Regional States organizations and their staff for their untiring effort in reviewing the guidelines and providing constructive suggestions, recommendations and comments.
- National and international development partners for their unreserved efforts in reviewing the guidelines and providing constructive comments which invaluably improved the quality of the guidelines.
- Small-scale and Micro Irrigation Support Project (SMIS) and its team for making all efforts to have quality GLs developed as envisioned by the Ministry.

The MOA would also like to extend its high gratitude and sincere thanks to AGP's multi development partners including the International Development Association (IDA)/World Bank, the Canada Department of Foreign Affairs, Trade and Development (DFATD), the United States Agency for International Development (USAID), the Netherlands, the European Commission (EC), the Spanish Agency for International Development (AECID), the Global Agriculture and Food Security Program (GAFSP), the Italy International Development Cooperation, the Food and Agriculture Organization (FAO) and the United Nations Development Program (UNDP).

Moreover, the Ministry would like to express its gratitude to Generation Integrated Rural Development Consultant (GIRDC) and its staff whose determined efforts to the development of these SSIGLs have been invaluable. GIRDC and its team drafted and finalized all the contents of the SSIGLs as per stakeholder suggestions, recommendations and concerns. The MoA recognizes the patience, diligence, tireless, extensive and selfless dedication of the GIRDC and its staff who made this assignment possible.

Finally, we owe courtesy to all national and International source materials cited and referred but unintentionally not cited.

Ministry of Agriculture

DEDICATIONS

The National Guidelines for Small Scale Irrigation Development are dedicated to Ethiopian smallholder farmers, agro-pastoralists, pastoralists, to equip them with appropriate irrigation technology as we envision them empowered and transformed.

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ACRONYMS

AGP	Agricultural Growth Program
ANSI	American National Standards Institute
API	American Petroleum Institute
ASDs	Adjustable speed rotational drives
BEP	Best Efficiency Point
BH	Brake horsepower
BOQ	Bill of quantity
CTs	Current Transformer
DCI	Ductile cast iron
DN	Nominal diameter
DOL	Direct on Line
EEPCO	Ethiopian electric power corporation
EQL	Equivalent Length
GIRDC	Generation Integrated Rural Development Consultant
GS	Galvanized steel
HD	Dynamic Head
HDL	dynamic head loss
HS	Static head
HSL	static head loss
HT	Total head
HT	Total head
ISO	International Organization for Standardization
KVA	Kilo Volt Ampere
KW	Kilowatt
LT	Low Tension
MASL	Meters above sea level
MCP	Main canals pump
MOANR	Ministry of Agriculture and Natural Resource
MOWIE	Ministry of Water, Irrigation and Electricity
NEMA	National Electrical Manufacturers Association

NPSH	Net positive suction head
NPSHA	Net positive suction head available
NPSHR	Net positive suction head required
P&ID	Piping and Instrumentation Diagram
PN	Pipe Nominal pressure
PV	Photovoltaic
PVC	polyvinyl chloride
RISE	Research Institute for Sustainable Energy
SSID	Small Scale Irrigation Development
SSIGL	Small Scale Irrigation Guideline
SSIP	small scale irrigation project
SSIS	Small Scale Irrigation Scheme
TDH	Total Dynamic Head
VTs	Voltage Transformer
WECS	Wind Energy Conversion Systems
WHP	water horsepower
XLPE	Cross-linked polyethylene

SYMBOLS

Symbol	description	Unit
A	Cross sectional area of flow	m^2
C	Coefficient of discharge or creep coefficient	unit less
D	Internal diameter of pipe	m
g	Acceleration due to gravity	m/s^2
h_f	Head loss due to friction	m
h_l	Head loss	m
h_{av}	Velocity head	m
L	Length	m
Q	Discharge or Flow	m^3/s, l/s
q	Discharge per unit length or intensity of flow	$m^3/s/m$
Q_d	Design discharge	m^3/s
v	Velocity of flow	m/s
v’	Actual velocity	m/s
V_S	velocity of suction	m/s
V_d	velocity discharge	m/s
f	Friction factor related to the roughness inside the pipe	unit less
L	Length of the pipe suction or discharge	m
d	Diameter of the pipe	m
∝	infinite resistance	ohm

PREFACE

While irrigation development is at the top of the government's priority agendas as it is key to boost production and improve food security as well as to provide inputs for industrial development. Accordingly, irrigated land in different scales has been aggressively expanding from time to time. To this end, to enhance quality delivery of small-scale irrigation development planning, implementation and management, it has been decided to develop standard SSI guidelines that must be nationally applied. In September 2017 the Ministry of Agriculture (MoA) had entrusted Generation Integrated Rural Development Consultant (GIRDC) to prepare the National Small-scale Irrigation Development Guidelines (SSIGLs).

Preparation of the SSIGLs for enhancing development of irrigated agriculture is recognized as one of the many core initiatives of the MoA to improve its delivery system and achieve the targets in irrigated agriculture and fulfill its mission for improving agricultural productivity and production. The core objective of developing SSIGLs is to summarize present thinking, knowledge and practices to enable irrigation practitioners to properly plan, implement and manage community managed SSI schemes to develop the full irrigation potential in a sustainable manner.

As the SSIGLs are prepared based on national and international knowledge, experiences and practices, and describe current and recommended practice and set out the national standard guides and procedures for SSI development, they serve as a source of information and provide guidance. Hence, it is believed that the SSIGLs will contribute to ensuring the quality and timely delivery, operation and maintenance of SSI schemes in the country. The SSIGLs attempt to explain and illustrate the important concepts, considerations and procedures in SSI planning, implementation and management; and shall be used as a guiding framework for professionals engaged in SSI development. Illustrative examples from within the country have been added to enable the users understand the contents, methodologies presented in the SSIGLs.

The intended audiences of the SSIGLs are government organizations, NGOs, CSOs and the private sector involved in SSI development. Professionally, the SSIGLs will be beneficial for experienced and junior planners, experts, contractors, consultants, suppliers, investors, operators and managers of SSI schemes. The SSIGLs will also serve as a useful reference for academia and researchers involved and interested in SSI development. The SSIGLs will guide to ensure that; planning, implementation and management of SSI projects is formalized and set procedures and processes to be followed. As the SSIGLs provide information and guides they must be always fully considered and applied by adapting them to the local specific requirements.

In cognizance with the need for quality SSIGLs, the MoA has duly considered quality assurance and control during preparation of the guidelines. Accordingly, the outlines, contents and scope of the SSIGLs were thoroughly discussed, reviewed and modified by NAWMP members (senior professionals from public, national and international stakeholder) with key stakeholders in many consultative meetings and workshops. Moreover, at each milestone of SSIGL preparation, resource persons from all stakeholders reviewed and confirmed that SSIGLs have met the demands and expectations of users.

Moreover, the Ministry has mobilized resource persons from key Federal, National Regional States level stakeholders and international development partners for review, validation and endorsement of the SSIGLs.

Several hundreds of experienced professionals (who are very qualified experts in their respective fields) from government institutions, relevant private sector and international development partners have significantly contributed to the preparation of the SSIGLs. They have been involved in all aspects of the development of SSIGLs throughout the preparation process. The preparation process included a number of consultation meetings and workshops: (i) workshop to review inception report, (ii) workshop on findings of review of existing guidelines/manuals and proposed contents of the SSIGLs, (iii) meetings to review zero draft SSI GLs, (iv) review workshop on draft SSI GLs, (v) small group review meetings on thematic areas, (vi) small group consultation meetings on its final presentation of contents and layout, (vii) consultation mini-workshops in the National States on semi-final versions of the SSIGLs, and (viii) final write-shop for the appraisal and approval of the final versions of SSIGLs.

The deliberations, concerns, suggestions and comments received from professionals have been duly considered and incorporated by the GIRD Consultant in the final SSIGLs.

There are 34 separate guidelines which are categorized into the following five parts concurrent to SSI development phases:

Part-I. Project Initiation, Planning and Organization Guideline which deals with key considerations and procedures on planning and organization of SSI development projects.

Part-II. Site Identification and Prioritization Guideline which treats physical potential identification and prioritization of investment projects. It presents SSI site selection process and prioritization criteria.

Part-III. Feasibility Study and Detail Design Guidelines for SSID dealing with feasibility study and design concepts, approaches, considerations, requirements and procedures in the study and design of SSI systems.

Part-IV. Contract Administration and Construction Management Guidelines for SSI development presents the considerations, requirements, and procedures involved in construction of works, construction supervision and contract administration.

Part-V. SSI Scheme Management, Operation and Maintenance Guidelines which covers SSI Scheme management and operation.

Moreover, Tools for Small Scale Irrigation development are also prepared as part of SSIGLs.

It is strongly believed and expected that; the SSIGLs will be quickly applied by all stakeholders involved in SSI development and others as appropriate following the dissemination and familiarization process of the guidelines in order to ensure efficient, productive and sustainable irrigation development.

The SSIGLs are envisioned to be updated by incorporating new technologies and experiences including research findings. Therefore, any suggestions, concerns, recommendations and comments on the SSIGLs are highly appreciated and welcome for future updates as per the attached format below. Furthermore, despite efforts in making all types of editorial works, there may still errors, which similarly shall be handled in future undated versions.

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UPDATING AND REVISIONS OF GUIDELINES

The GLs are intended as an up-to-date or a live document enabling revisions, to be updated periodically to incorporate improvements, when and where necessary; may be due to evolving demands, technological changes and changing policies, and regulatory frameworks. Planning, study and design of SSI development interventions is a dynamic process. Advancements in these aspects are necessary to cope up with the changing environment and advancing techniques. Also, based on observation feedbacks and experiences gained during application and implementation of the guidelines, there might be a need to update the requirements, provisions and procedures, as appropriate. Besides, day-by-day, water is becoming more and more valuable. Hence, for efficient water development, utilization and management will have to be designed, planned and constructed with a new set up of mind to keep pace with the changing needs of the time. It may, therefore, be necessary to take up the work of further revision of these GLs.

This current version of the GLs has particular reference to the prevailing conditions in Ethiopia and reflects the experience gained through activities within the sub-sector during subsequent years. This is the first version of the SSI development GLs. This version shall be used as a starting point for future update, revision and improvement. Future updating and revisions to the GLs are anticipated as part of the process of strengthening the standards for planning, study, design, construction, operation and management SSI development in the country.

Completion of the review and updating of the GLs shall be undertaken in close consultation with the federal and regional irrigation institutions and other stakeholders in the irrigation sub-sector including the contracting and consulting industry.

In summary, significant changes to criteria, procedures or any other relevant issues related to technological changes, new policies or revised laws should be incorporated into the GLs from their date of effectiveness. Other minor changes that will not significantly affect the whole nature of the GLs may be accumulated and made periodically. When changes are made and approved, new page(s) incorporating the revision, together with the revision date, will be issued and inserted into the relevant GL section.

All suggestions to improve the GLs should be made in accordance with the following procedures:

I. Users of the GLs must register on the MOA website: Website: www.moa.gov.et
II. Proposed changes should be outlined on the GLs Change Form and forwarded with a covering letter or email of its need and purpose to the Ministry.
III. Agreed changes will be approved by the Ministry on recommendation from the Small-scale Irrigation Directorate and/or other responsible government body.
IV. The release date of the new version will be notified to all registered users and authorities.

Users are kindly requested to present their concerns, suggestions, recommendations and comments for future updates including any omissions and/or obvious errors by completing the following revisions form and submitting it to the Ministry. The Ministry shall appraise such requests for revision and will determine if an update to the guide is justified and necessary; and when such updates will be published. Revisions may take the form of replacement or additional pages. Upon receipt, revision pages are to be incorporated in the GLs and all superseded pages removed.

Suggested Revisions Request Form (Official Letter or Email)

To: --
From: --
Date: ---
Description of suggested updates/changes: Include GL code and title, section title and # (heading/subheading #), and page #.

GL Code and Title	Date	Sections/ Heading/Subheading/ Pages/Table/Figure	Explanation	Comments (proposed change)

Note that be specific and include suggested language if possible and include additional sheets for comments, reference materials, charts or graphics.

GLs Change Action

Suggested Change	Recommended Action	Authorized by	Date

Director for SSI Directorate**:** _________________________**Date:** _________________

The following table helps to track initial issuance of the guidelines and subsequent Updates/Versions and Revisions (Registration of Amendments/Updates).

Revision Register

Version/Issue/Revision No	Reference/Revised Sections/Pages/topics	Description of revision (Comments)	Authorized by	Date

1 INTRODUCTION

Irrigation is considered as a means of renovating the agricultural economy and plays significant role in improving income and livelihoods of the community through increased agricultural production and productivity. Considering this significance of irrigated agriculture in the economy, the Government has given top priority to the irrigation sub-sector in the overall development plans of the country with the ultimate objective of enhancing agricultural production and productivity and linking it with the industrial development.

Water is thus a fundamental input to agricultural production; indeed food security cannot be achieved without supporting the rain fed agriculture with irrigated agriculture since climatic variability is challenging the agricultural sector. On the other hand, the country did not yet utilized the available surface and sub-surface water resources to the desired level.

Water can be conveyed to the available land resources either by gravity or lift/pump irrigation. Particularly due to topographic constraints it may not be always possible to irrigate suitable irrigable lands by gravity system and study and design of pumping facility must be considered for such cases. Sometimes pumping is also necessitated to apply water by pressurized irrigation systems.

Actual situations in the country indicate that pumping facilities implemented so far to benefit smallholder farmers are not efficient enough due to various problems encountered during planning, study, design, implementation and operation. As one of the limiting factors to sustain pumped irrigation schemes, proper design and selection of appropriate pump for specific area is very crucial. In this guideline study and design aspects of pumping facilities in small scale irrigation schemes will be discussed.

The study and design incorporates proper site selection for pumping facility, setting design criteria based on hydraulic and structural considerations and evaluation of different design alternatives and selection of most viable one. Design considerations also depend on the water sources for pumping like spring, stream, river, lake or underground water. The pumping facilities include the pump and power unit, the control heads, the housing, suction and delivery outlet and the pipe lines. Thus different categories of pumps for irrigation and pump driving energy sources are discussed in this guideline.

2 PURPOSE AND SCOPE OF THE GUIDELINE

2.1 PURPOSE

The purpose of the guideline is to:

- Guide engineering professionals in study and design of pumping facilities for irrigated agriculture and contribute to sustainable production and productivity.
- Standardize study and design approaches of pumping facilities in irrigated agriculture.
- Reinforce study and design skills of engineering professionals in making proper pump selection for particular site.

2.2 SCOPE

The guideline focuses on:

- Study and design of pumping facilities in the irrigated agriculture particularly small scale schemes that serve smallholder farmers.
- Addressing limiting factors related to study and design of pumping facilities in SSI schemes.
- Study and design aspects of most common types of pumping facilities in small scale irrigation development.
- Making the guideline easy to use with practical illustrations and examples in designing and selecting proper pump that suit to specific site conditions.

3 DEFINITIONS

Basic concepts in the study and design of pump Facilities:

- **Air/Vacuum Valve:** An Air/Vacuum valve is used to allow air to escape the discharge piping when pumping begins, and to prevent vacuum damage to the discharge piping when pumping stops. If the pump discharge is open to the atmosphere, an air/vacuum release valve may not be necessary. Combination air release valves are frequently used at high points in force mains to evacuate trapped air.
- **Adjustable speed rotational drives (ASDs):** Devices that allow control of a pump's speed; including mechanical devices such as hydraulic clutches and electronic devices such as eddy current clutches and variable frequency drives
- **Backpressure:** The pressure on the discharge side of the pump
- **Bearing:** A device that supports a rotating shaft, allowing it to spin, while keeping it from translating in the radial direction; a thrust bearing keeps a shaft from translating in the axial direction
- **Brake horsepower:** The amount of power (measured in units of horsepower) delivered to the shaft of a motor driven piece of equipment
- **Baseline Consumption:** Estimated pumping system annual energy consumption.
- **Best Efficiency Point (BEP):** The best efficiency point (BEP) refers to the most efficient operating point (defined by a certain rate of flow and system head) for a centrifugal pump. This is the point at which each pump should operate at optimal system design.
- **Check Valve:** A watertight fitting used in pipes to prevent back flow to the pumps and subsequent re-circulation. A check valve is sealed with a rubber seated ball type fitting.
- **Cavitation:** A phenomenon in which the local liquid pressure drops below its vapor pressure, which results in the liquid flashing to vapor. As these vapor bubbles collapse, they create vibrations and noise which can be damaging to system components, especially pump impellers.
- **Capacity** the rate of liquid flow that can be carried.
- **Centrifugal Pumps** Centrifugal pumps operate by adding kinetic energy to a liquid via a spinning impeller, where dynamic pressure is created as a result of flow resistance in the discharge passage.
- **Coupling:** Coupling refers to the connection or transmission of power between motor and pump. This includes types of coupling such as direct coupling or drive belts.
- **Coupling Efficiency:** The coupling efficiency is defined as the ratio of the energy delivered by the motor to the coupling divided by the energy delivered to the pump shaft.
- **Current**: the amount of electricity measured in amps which are flowing in a circuit.
- **Deadheading:** A condition in which all the discharge from a pump is closed off
- **Design Point:** The operating point as calculated for a pump during the system design. The actual operating point is often not at the design point.
- **Dynamic Head:** The head associated with frictional losses within the pumping system pipe network.
- **Efficiency:** In a pump, the efficiency with which the shaft power applied (not the power to the motor) is converted to flow and head; motor efficiency is the electrical equivalent of this parameter for the motor
- **Flow:** The quantity of water passing an observation point; for pumped liquids the term mass flow is often used and refers to the mass of liquid passing per unit time (however, it is more common to use volume flow; that is, the volume passing per unit time.
- **Friction losses:** Pressure losses caused purely by the resistance of the pipework and system, which must be added to the static head to obtain the total system resistance – note that friction losses vary with flow rate and that they occur in pump inlet pipework as well as outlet pipework

- **Gate Valve:** A gate valve is a simple shut-off device that is used to isolate pumps and facilitate removal. These valves should not be used to throttle flow. They should be either totally open or totally closed.
- **Head:** A measure of pressure (expressed in meters or feet) indicating the height of a column of system fluid that has an equivalent amount of potential energy. It is the sum of Dynamic and Static heads
- **Heat exchanger:** A device that transfers heat from one fluid to another
- **Impeller:** a pump component that rotates on the pump shaft and increases the pressure on a fluid by adding kinetic energy. .
- **Hydraulic Power:** The power imparted by the pump to the liquid.
- **Hydraulic horsepower (WHP):** the pump output or the liquid horsepower delivered by the pump.
- **Impeller:** The rotating component within a centrifugal pump used to increase the pressure and flow of a liquid.
- **(KVA):** Common unit for apparent power, which is the total power that appears to be flowing from a source to a load. (Kilo Volt Ampere)
- **(KW):** Common unit for real power, which is the actual net power that is flowing from a source to a load. (Kilowatt)
- **Liquid:** In the context of this document, a solution or suspension following Newtonian or Non-Newtonian laws it is water.
- **Mechanical seal:** A mechanical device for sealing the pump/shaft interface (as opposed to packing)
- **Motor:** An electric machine that uses either alternating current (A.C.) or direct current (D.C.) electricity to spin a shaft that is typically coupled to a pump. Occasionally, however, mechanisms such as a slider/crank convert this rotation to axial movement to power piston pumps.
- **Motor controller:** An electric switchbox that energizes and de-energizes an electric motor.
- **Motor input horsepower:** (EHP) - the power input to the motor expressed in horsepower.
- **Multi-stage pumps :** Pumps which contain several impellers, each feeding its output to the next stage in a serial fashion in order to generate pressures higher than a single-stage pump can achieve
- **Motor Power:** The motor power is the power consumed by the pump motor to turn the pump shaft. The motor power is the sum of the shaft power and power loss due to inefficiencies in converting electric energy into kinetic energy. Motor power may be calculated as the shaft power divided by the motor efficiency
- **Motor Efficiency:** The motor efficiency is defined as the ratio of the energy delivered to the motor divided by the energy delivered from the motor to the coupling.
- **Power:** Output from a motor is equal to the input power multiplied by the motor efficiency – it is this output which is the power absorbed by a pump, that is, the power value which features on pump characteristics.
- **Preferred operating region:** The region on a pump curve where flow remains well controlled within a range of capacities, within which hydraulic loads, vibration or flow separation will not significantly affect the service life of the pump.
- **Pressure:** Force per unit area; commonly used as an indicator of fluid energy in a pumping system (expressed in kilograms per square centimeter or pounds per square inch).
- **Prime mover:** A machine, usually an electric motor, that provides the motive force driving a pump.
- **Pump Control:** A device that activates pumps successively in response to a rising water level in the source. The control regulates the pump activity until the inflow into the wet well has ceased.
- **Pump Driver:** The device used to provide power to the pump. Alternating current electric motors are the most common type of driver.

- **Peak Load:** The peak power consumption of a site. This often determines the demand charges incurred by the site and should therefore be taken into account when considering the operating times of pumping systems.
- **Performance Curve:** Graph plotting the head required as a function of flow rate for a given pump. Also often depicted on performance curves are the shaft power, pump efficiency and suction head required. The term “pump curve” often refers to the performance curve.
- **Piping and Instrumentation Diagram (P&ID):** Schematic diagram of the pumping system including pipe layout, liquid users and associated instrumentation.
- **Positive Displacement Pumps:** Positive displacement pumps move a set volume of liquid per revolution or stroke, with pressure developed as a result of this forced discharge. Positive displacement pumps are better suited to high-viscosity applications.
- **Power input**: the electrical input to the motor expressed in kilowatts (kW). A measure of the rate at which work is done.
- **Power Factor:** Ratio of real power to apparent power.
- **Pressure; head (NPSH):** Net Positive Suction Head It usually has a subscript: NPSHR is the NPSH required by a pump at its inlet to prevent it from cavitation. NPSHA is the NPSH available from the inlet configuration in use. To avoid cavitation, NPSHA must be greater than NPSHR
- **Pump Efficiency:** The ratio of the hydraulic power (power imparted to the liquid) divided by the pump shaft input power (power delivered to the pump via motor coupling).
- **Pumping System: A** pump or group of pumps along with the other components relevant to the moving liquid. This includes the motors, coupling, piping and valves.
- **Rated duty:** The flow and head that are specified on the pump nameplate – they should be close to the values corresponding to the peak efficiency of the pump.
- **Seals:** Prevent water leaking outwards along the pump shaft – either packed glands or mechanical seals.
- **Specific gravity:** The ratio of the density of a fluid to the density of water at standard conditions.
- **Static head:** The head of water a pump must overcome before it will produce any flow; it is a result of the height difference between the suction water level and delivery water level
- **Shaft Input Power:** The power delivered to the shaft of a pump.
- **Total head :** A measure of the total energy imparted to the fluid by a pump, which includes static pressure increase and velocity head
- **Valve**: A device used to control fluid flow in a piping system – there are many types of valves with different flow control characteristics, sealing effectiveness and reliability
- **Vapor pressure:** The force per unit area that the fluid exerts in an effort to change the phase from a liquid to a vapor; it is a function of a fluid’s chemical and physical properties, and its temperature
- **Viscosity**: The resistance of a fluid to flow when subjected to shear stress

4 TYPES OF PUMPS AND AREAS OF APPLICATION

4.1 TYPES OF PUMPS

Pumps are hydraulic machines designed for transmitting fluids under pressure by transforming the mechanical energy of the driving motor into the mechanical energy of the moving fluid; pumps lift fluid up to a specified elevation, deliver it over the required distance at the horizontal plane or make it circulating in a certain closed system.

Based on their method of operation, pumps can be classified into the following two major types and common sub classifications are indicated under Figure 4-1 and Figure 4-2.

a) Dynamic Pumps

Dynamic pumps include all types of pumps using fluid velocity and the resulting momentum to pump and move the fluid through the piping system. Although dynamic pumps usually have lower efficiencies than positive displacement pumps, they require lower maintenance. They are also capable of operating at high speeds and high fluid flow rates.

b) Positive Displacement Pumps

Positive displacement pumps operate by filling and displacing liquid from a cavity. Such pumps deliver a constant flow and volume of liquid without discharge pressure or head. Positive displacement pumps are ideal for a low flow–high pressure combination.

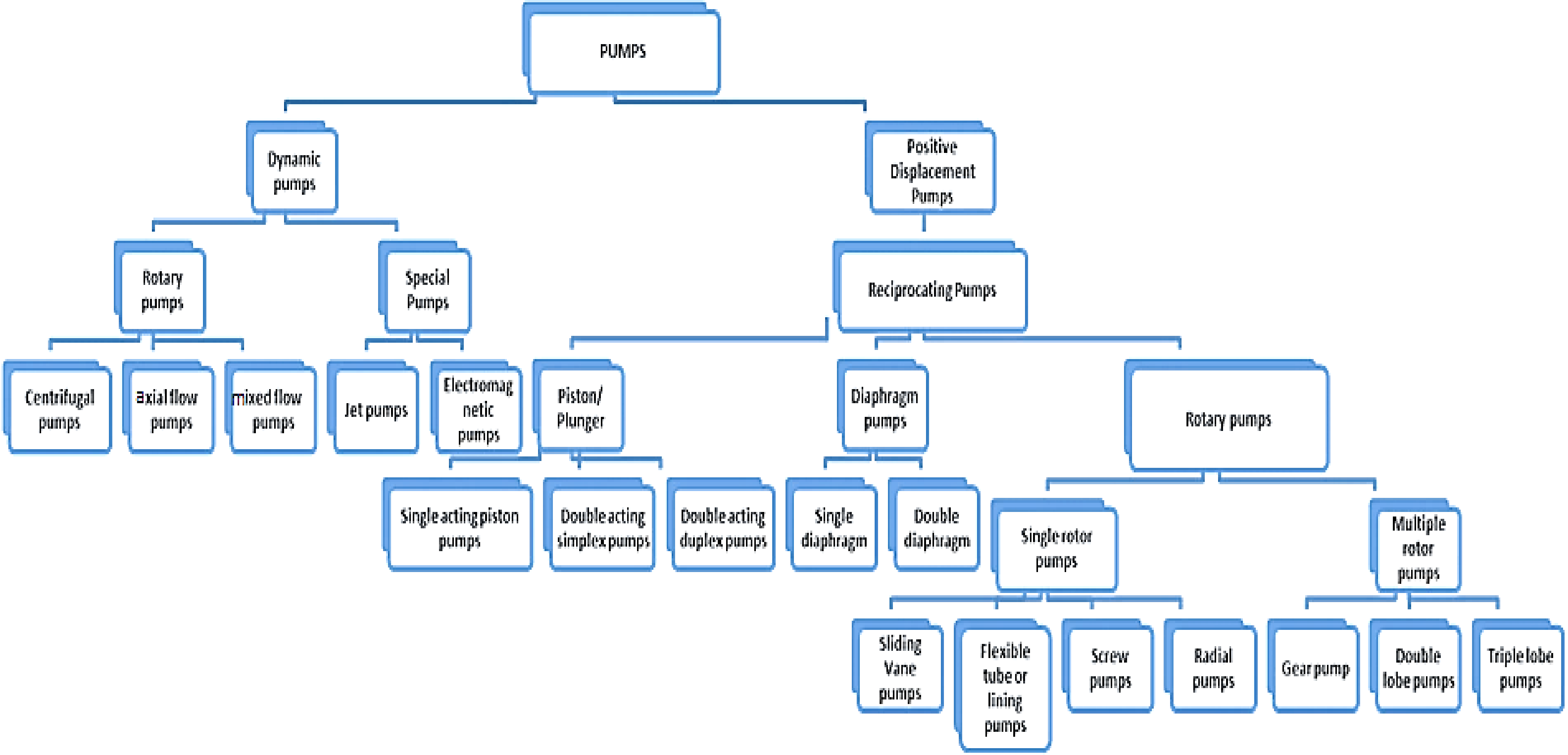


Figure 4-1: General Pump Classification-chart

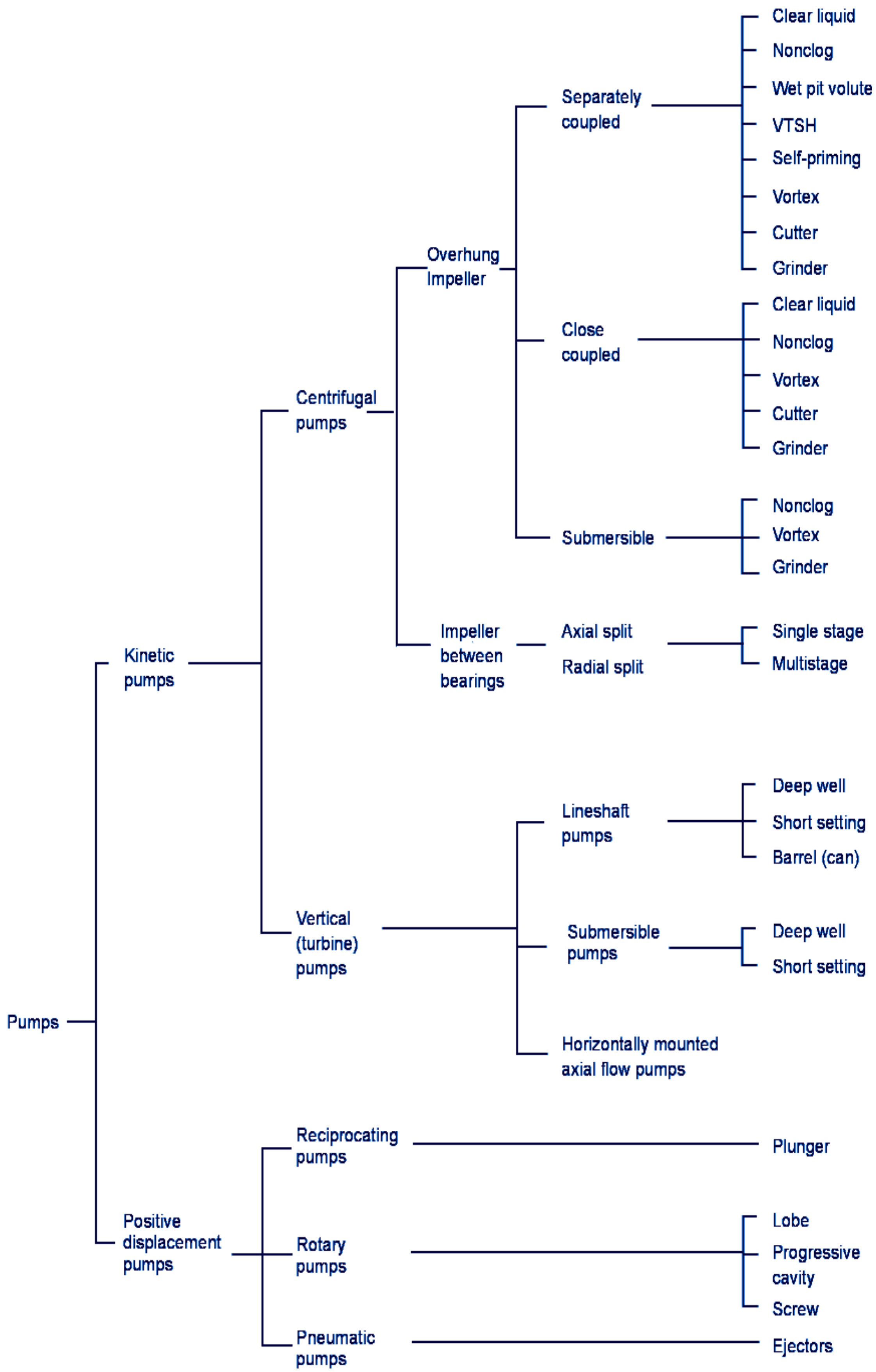


Figure 4-2 Classification of pumps by the Hydraulic Institute in 1983

4.2 AREAS OF APPLICATION

The types of pumps are given in the pump classification chart Figure 4-1 above. There are numerous ways of pump classification and the Hydraulic Institute classification is also given in Figure 4-2.

The selection of proper type and its proper installation is very important to meet desired specification and proper application. Knowledge of the variety of pumps is mandatory to propose the right pump for the right application. Commonly used pumps for irrigation are addressed in coming parts of the guideline.

a) Displacement Pumps

Displacement pumps force the water to move by displacement. Some of such pumps are piston pumps, diaphragm pumps, roller-tubes, and rotary pumps. Displacement pumps are used not only for water but for moving very thick liquids, creating very precise flow volumes, or creating very high pressures. They are also used for fertilizer injectors, spray pumps, air compressors, and hydraulic systems for machinery. With the exception of fertilizer injectors used for mixing fertilizer into irrigation water, they are not widely used for irrigation.

b) Centrifugal Pumps

Centrifugal pumps are widely used for agriculture. Centrifugal Pumps are useful since they can generally handle large quantity of fluids, provide very high flow rates (which may vary with the changes in the Total Dynamic Head (TDH) of the particular piping system) and have the ability to adjust their flow rates over a wide range. Centrifugal pumps are generally designed and suitable for liquids with relatively low viscosity that pour like water. Such pumps need **priming** to start pumping. The vertical difference between the water surface in suction sump and pump axis level is usually not more than 7-8m to avoid cavitation but mostly less than 6m.

The following information shall help to understand more about these pumps and enable to select the best kind of pump for specific purposes.

Working Conditions of Centrifugal Pumps:
Centrifugal pumps are used to induce flow or raise a liquid from a low level to a high level. These pumps work on a very simple mechanism. A centrifugal pump converts rotational energy, often from a motor, to energy in a moving fluid.

The two main parts that are responsible for the conversion of energy are the impeller and the casing. The impeller is the rotating part of the pump and the casing is the airtight passage which surrounds the impeller. In a centrifugal pump, fluid enters into the casing, falls on the impeller blades at the eye of the impeller, and is whirled tangentially and radially outward until it leaves the impeller into the diffuser part of the casing. While passing through the impeller, the fluid is gaining both velocity and pressure.

The following chief factors affect the performance of a centrifugal pump and need to be considered while choosing a centrifugal pump:

- **Working Fluid Viscosity –** can be defined as resistance to shear when energy is applied. In general, a centrifugal pump is suitable for low viscosity fluids since the pumping action generates high liquid shear.

- **Specific density and gravity of working fluid –** The density of a fluid is its mass per unit of volume. A fluid's mass per unit volume and gravity of a fluid is the ratio of a fluid's density to the density of water. It directly affects the input power required to pump a particular liquid. If you are working with a fluid other than water, it is important to consider the specific density and gravity since the weight will have a direct effect on the amount of work performed by the pump. In case of irrigation silt free water is usually assumed.
- **Operating temperature and pressure –** Pumping conditions like temperature and pressures are an important consideration for any operation. For example - High temperature pumping may require special gaskets, seals and mounting designs. Similarly, an adequately designed pressure retaining casing may be required for high-pressure conditions.
- **Net Positive Suction Head (NPSH) and Cavitation –** NPSH is a term that refers to the pressure of a fluid on the suction side of a pump to help determine if the pressure is high enough to avoid cavitation. Cavitation refers to the formation of bubbles or cavities in liquid, developed in areas of relatively low pressure around an impeller and can cause serious damage to the impeller and lead to decreased flow/pressure rates among other things. One must ensure that the system's net positive suction head available (NPSHA) is greater than the pump's net positive suction head required (NPSHR), with an appropriate safety margin.
- **Vapor pressure of the working fluid –** The vapor pressure of a fluid is the pressure, at a given temperature, at which a fluid will change to a vapor. It must be determined in order to avoid cavitation as well as bearing damage caused by dry running when the fluid has evaporated.

Owing to the use in diverse range of applications, pumps come with different capacities and in various sizes. One should also consider the pressure and volume requirements of the specific operations for which you need the pump. The horsepower required is another important consideration when it comes to volume and discharge pressure.

Applications of Centrifugal Pumps:

The fact that centrifugal pumps are the most popular choice for fluid movement makes them a strong contender for many applications and as mentioned previously, they are used for agriculture, industries, boosting pressure, pumping water for domestic requirements; assisting fire protection systems, hot water circulation; sewage drainage and regulating boiler water are among the most common applications. Here **special attention is given to centrifugal pumps** as they are used for irrigation widely.

Types of centrifugal pumps:

Centrifugal pumps can be classified into several types depending on factors such as design, construction, application, service, compliance with a national or industry standard, etc. Therefore, one specific pump can belong to different groups and at times pump is known by its description itself. Some of these groups have been highlighted below:

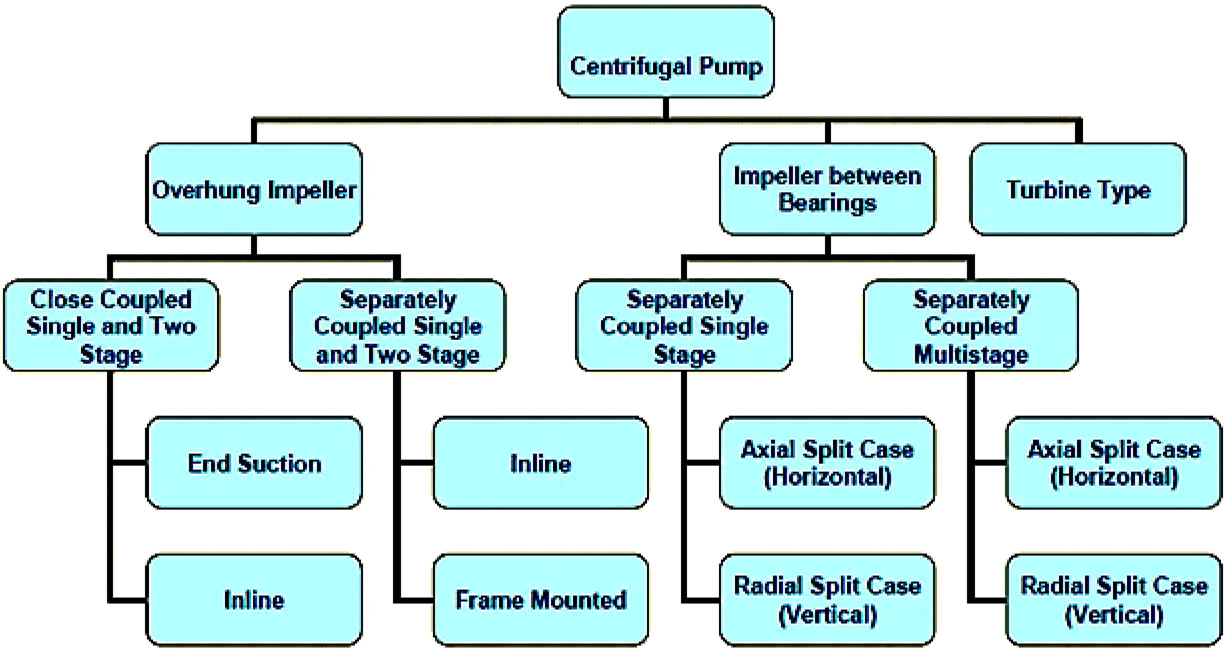


Figure 4-3 Types of centrifugal pumps

Centrifugal pumps as indicated on figure 4-2 above can be categorized based on different factors. These categorizations are discussed below.

I. Depending on number of impeller(s) in the pump

Single stage – A one impeller pump, single stage pump has a simple design and easy maintenance. They are ideal for large flow rates and low-pressure installations. They are commonly used in pumping services of high flow and low to moderate TDH (Total Dynamic Head).

Two-stage – This type of pump has two impellers operating side by side which are used for medium head applications.

Multi-stage – pump has three or more impellers in series; for high head service.

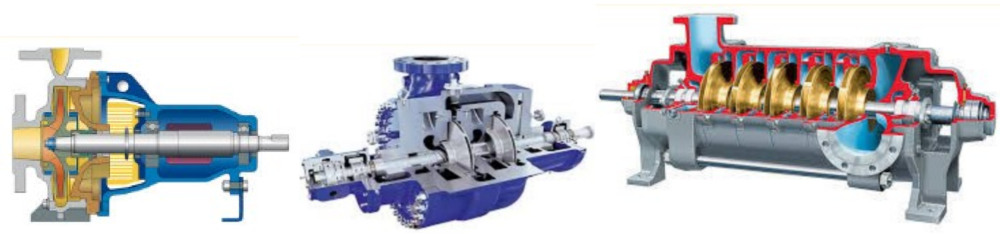


Figure 4-4 Single Stage (left), double or two stage (middle) and multi stage (right)

II. According to type of impeller

Closed impeller: Here vanes of the impeller are covered with plates on both sides. It is made of cast iron, stainless steel, cast steel or gun metal.

Semi open impeller: The vanes of the impeller are covered with plate on one side. It has less number of vanes but its height is more than the closed impeller.

Open Impeller: The vanes of the impeller are without cover plate. These are generally made of forged steel. It has less life as they are usually manufactured for rough task.

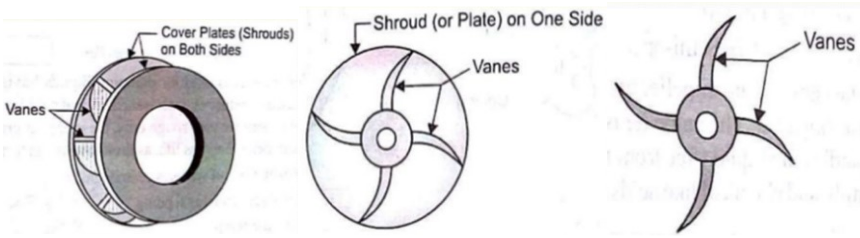


Figure 4-5 Closed impeller (left), Semi closed impeller (middle) and Open impeller (right)

III. Type of case-split

The Orientation of case-split is another factor used to categorize Centrifugal pumps:

Axial split – In these kinds of pumps, the volute casing is split axially and the split line at which the pump casing separates is at the shaft's center-line. Axial Split Pumps are typically mounted horizontally due to ease in installation and maintenance.

Radial split – Here, the pump case is split radially; the volute casing split is perpendicular to the shaft center-line.

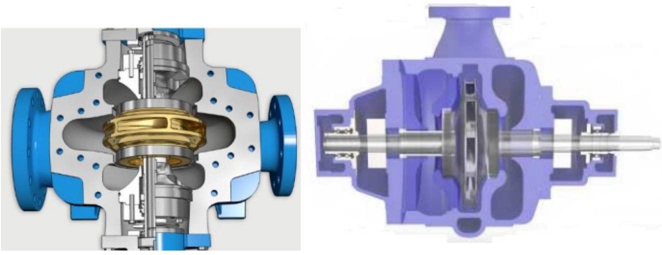


Figure 4-6 Axial split case double suction (left) and radially split case double suction (right)

IV. Categorized by type of impeller design

Single suction – This kind of pump has a single suction impeller that allows fluid to enter the blades only through one side; It has a simple design but impeller has a higher axial thrust imbalance due to flow coming in on one side of impeller only.

Double suction – This particular type of pump comes with double suction impeller that allows fluid to enter from both sides of the blades and has lower NPSHR than a single suction impeller. Split-case pumps are the most common type of pump with double suction impeller. If a pump has more than one impeller, the design of the first stage impeller will determine if the pump is of a single or double suction type.

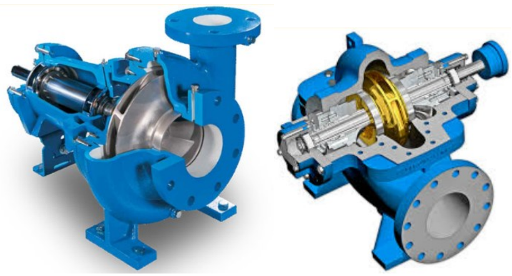


Figure 4-7 Single stage end suction pump (left) and single stage double suction pump (right)

V. Based on casing

Volute casing: In volute casing the impeller is surrounded by spiral casing. The casing is such shaped that its cross sectional area gradually increases from tongue to delivery pipe.

Vortex casing: In vortex casing annular space known as vortex or whirl pool chamber is provided between the impeller and volute casing. Liquid from the impeller flow with free vortex motion, in the vortex chamber, where its velocity is converted in to pressure energy.

Diffuser Casing: In diffuser casing the guide vanes are arranged at the outlet of the impeller. The guide vanes are shaped to provide gradually enlarged passage for flow of liquid. The kinetic energy of the liquid coming out from the impeller is converted to pressure energy during flow in guide vanes (increasing area).

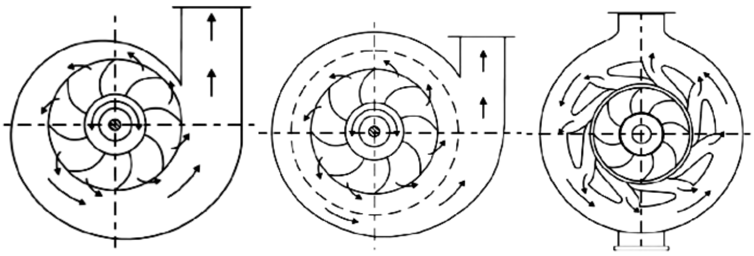


Figure 4-8 Volute casing (left), vortex casing (middle) and diffuser casing (right)

VI. Category based on type of volute

Centrifugal pumps can also be categorized based on volute namely Single volute and Double volute:

Single volute – This kind of pump Is usually used in small low capacity pumps where a double volute design is impractical due to relatively small size of volute passageway which make obtaining good quality commercial casting difficult. Pumps with single volute design have higher radial loads.

Double volute – This kind of pump volute has two partial volutes which are located 180 degrees apart resulting in balanced radial loads; most centrifugal pumps are of double volute design.

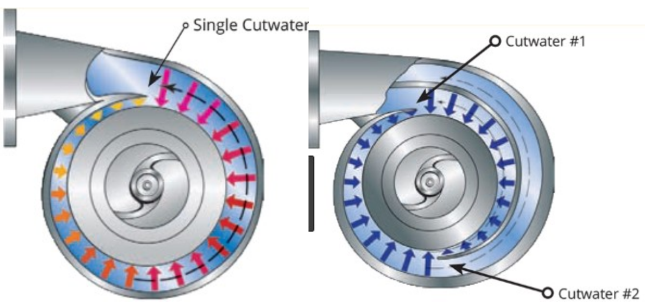


Figure 4-9 Single volute pump (left) and double volute pump (right)

VII. Depending on where the bearing support is

Bearing support is also often used to categorize Centrifugal Pump:

Overhung – where the impeller is mounted on the end of a shaft, supported by bearings on only one side. Further, the overhung pump type has a horizontal orientation of shaft or can be vertical in-line with bearing bracket.

Between-bearing – where the impeller is mounted on a shaft that has bearing support on both ends, thus impeller is located in between-bearings.

VIII. Depending on shaft orientation

Shaft orientation is another characteristic which distinguishes the type of Centrifugal pump:

Horizontal – These are pumps with shaft in horizontal plane; popular due to ease of servicing and maintenance. It is sometimes overhung or placed between bearing design.

Vertical – Vertical centrifugal pumps have their shaft in the vertical plane. They utilize a unique shaft and bearing support configuration that allows the volute to hang in the sump while the bearings are outside the sump. It is generally an overhung and of radial-split case type design.

Figure 4-10 Horizontal pump (left) and vertical pump (right)

The following are the differences between horizontal and vertical centrifugal pumps:

Horizontal centrifugal pump

1. Easy availability of its rotor and internals makes it easier to install, inspect, maintain, and service.
2. It can be coupled directly to a variety of drivers including electric motor, engine, and turbine (steam, gas or power recovery hydraulic turbine
3. It is available in either overhang design for low suction pressure service, or in between-bearing design for high suction pressure service.
4. It is available in various nozzle configurations to simplify, or match the external site piping. The nozzle configuration can be of end suction top discharge, top suction top discharge, or side suction side discharge.
5. Its low headroom requirement makes it suitable for most indoor installations.
6. It has limited applications where the NPSHR exceeds the site NPSHA; Large pumps usually require an auxiliary booster pump. (With a vertical line shaft pump, the NPSHA can be increased by lowering the setting of its impeller.
7. Bigger footprint is required for horizontal designs.

Vertical centrifugal pump

1. Most of them require large headroom for installation, servicing, and maintenance. Being of an overhang design, its hydraulic axial thrust is difficult to balance in high pressure service.

2. Usually suitable for direct coupling to electric motor. Using engine or turbine, will require a right-angle gear drive and a universal shaft joint and a clutch.
3. It can more easily withstand higher pressure service because of its simplified bolting and confined-gasket design
4. It requires a smaller footprint and is suitable for installation where the ground surface area is limited, or is at a premium.
5. With a vertical line shaft pump the impeller setting below the ground can be lowered to increase the site NPSHA.
6. Vertical line shaft turbine pumps require large headroom for installation, servicing, and maintenance.
7. Expensive sump pit and barrel in a multistage pump is usually required.
8. There can be mechanical seal problems when pumping liquids with high dissolved or entrained gas which accumulates at the top of the stuffing box or seal chamber where venting can be difficult or less effective.

IX. On the basis of head

Pump head is pressure defined as the height to which the pump can raise the fluid to. It is important as it evaluates a pump's capacity to do its job. The most important specifications of a pump are its capabilities regarding flow and pressure. Centrifugal pumps have capacity to deliver more volume of water per unit of time with lower heads. Based on working head they are categorized as:

Low head centrifugal pumps: **<15m**
Medium head centrifugal pumps: **15m to 45m**
High head centrifugal pumps: **>45m**

X. On the basis compliance with industry standards

While choosing a centrifugal pump, the buyers should be selective based on the quality standards they have to achieve. They need to check for the following:

ANSI pump – (American National Standards Institute) **-** ANSI standards refer to dimensional standards. The pumps are also required to meet ANSI B73.1 standards, also known as ASME B73.1 – (American Society of Mechanical Engineers). The objective of this standard is to ensure interchangeability of ANSI process pumps of similar sizes. These centrifugal pumps are horizontal, end suction, single stage pumps and are comparable regardless of manufacturer.

API pump – (American Petroleum Institute) API's standard refers to the parameters of pump's construction, design, and ability to handle high temperatures and pressures. API 610 specifications and a variety of API type include API VS4, API VS7, API OH3, API OH2, API OH1, API BB1, API BB2, API BB3 etc. Centrifugal pumps must meet the requirements of the American Petroleum Institute Standard 610 for General Refinery Service.

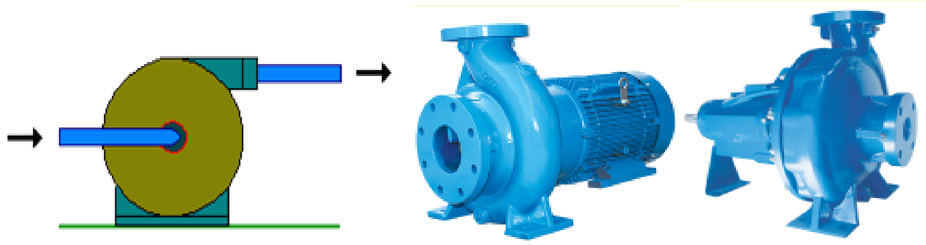
DIN pump – DIN 24256 specifications. Centrifugal pumps satisfying these standards are used in installations requiring large flow rates, abnormally high working pressures or very high temperatures. These are rarely used in mechanical building services.

ISO pump – ISO 2858, 5199 specifications, the international standard ISO 5199 specifies the requirements for class II end suction centrifugal pumps of single-stage, multistage, horizontal or vertical construction, with any drive and any installation for general application.

Nuclear pump – ASME (American Society of Mechanical Engineers) specifications.

c) End-Suction Centrifugal Pumps

End-Suction Centrifugal Pumps are the most common type of pump. An end suction pump is the most basic type of centrifugal pump typically designed with a casing. The suction is present on one end and the discharge is placed at the top. Typically the pump is "close-coupled "to the motor, that is, the pump is mounted right on the end of the motor's drive shaft and the pump case is bolted straight into the motor so that it looks like a single unit. Or the pump can be “frame-mounted” type. The water typically enters the pump through a "suction inlet "centered on one side of the pump, and exits at the end. End-suction centrifugal pumps generally need to be primed the first time they are used (including many so-called self-priming models,) after that most will not require priming unless a leak develops in the intake pipe. If the pump needs to be primed each time this means, there is a leak in the intake pipe.



a) Inlet and outlet condition b)Close-coupled type c) Frame-mounted type

Figure 4-11: End suction centrifugal pumps

End-Suction Centrifugal pumps are designed to be used as irrigation booster pumps. They are also very good for pumping water from any source where the water level is higher than the pump, where the water can flow down an intake pipe to the pump using gravity. But any time they need to actually suck the water up into the pump they perform much less efficiently. Therefore end-suction centrifugal pumps must be installed as close to the water surface level as possible. Each pump is different, so check with the manufacturer to determine the maximum height the pump can be above the water surface.

d) Submersible Pumps

Submersible pumps are installed completely underwater, including the motor. The pump consists of an electric motor and pump combined in a single unit. Typically the pump will be shaped like a long cylinder so that it can fit down inside of a well casing. Although most submersible pumps are designed to be installed in a well, many can also be laid on their side on the bottom of a lake or stream. Another common installation method for lakes and rivers is to mount the submersible pump underwater to the side of a pier pile (post). Submersible pumps don't need to be primed since they are already under water. They also tend to be more efficient because they only push the water, most submersible pumps must be installed in a special sleeve if they are not installed in a well, and sometimes they need a sleeve even when installed in a well. The sleeve forces water coming into the pump to flow over the surface of the pump motor to keep the motor cool. Without the sleeve the pump will burn up. Since the power cord runs down to the pump through the water it is very important that it shall be protected from accidental damage. Submersible pumps should not run outside water even for test purpose water is the only means for cooling.

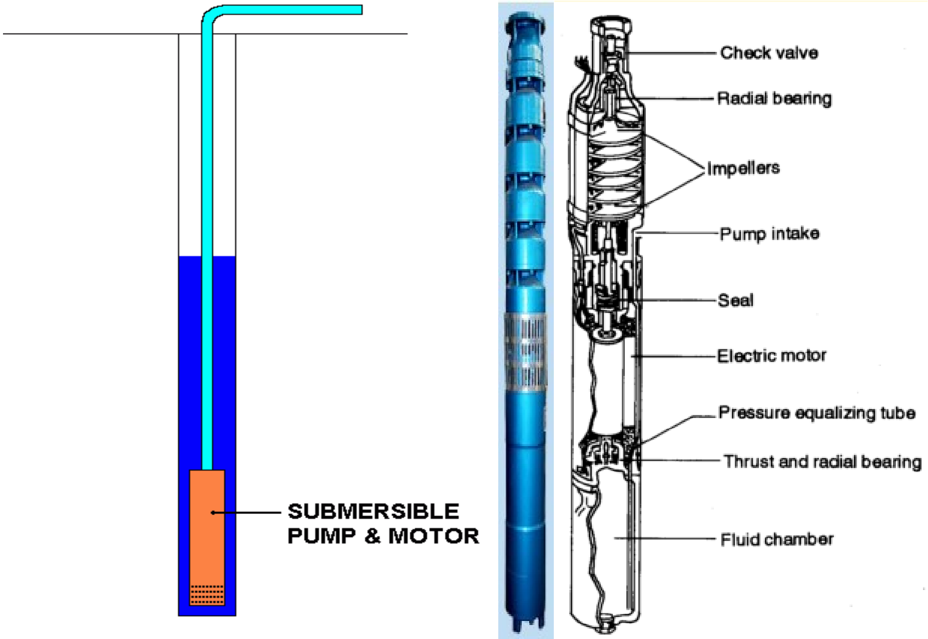


Figure 4-12: A Submersible Pump

e) Turbine pumps

A turbine pump is basically a centrifugal pump mounted underwater and attached by a shaft to a motor mounted above the water. The shaft usually extends down the center of a large pipe. The water is pumped up through the pipe and exits directly under the motor. Turbine pumps are very efficient and are used primarily for larger pump applications. Often they consist of multiple stages, each stage is essentially another pump stacked on top of the one below. It works like a train with multiple engines hitched together pulling it, each stage would be an engine. Turbine pumps are typically the type of pumps you see on farms. When you see a huge motor mounted on its end and a pipe coming out sideways below the motor that is most likely the turbine pump. The turbine pump is mounted in a large concrete vault with a pipe connecting it to the sump. The water flows by gravity into the vault where it enters the pump. The pump motors are over the vault on a frame. Usually it is common to use two or three different sized pumps side-by-side to handle different flow combinations.

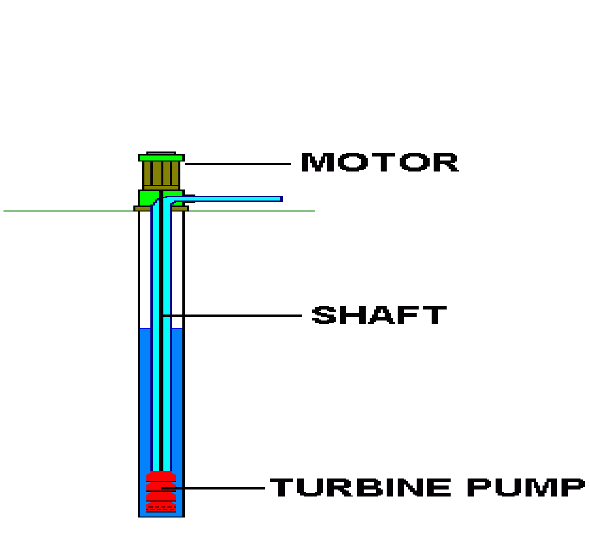


Figure 4-13: Turbine pump

4.3 SITE SELECTION CRITERIA FOR PUMPING FACILITY

General site Selection criteria for the pumping facility are:

I. The location of the pumps should be above Highest Flood Level (H.F.L.) or well protected against high floods.
II. Required quantity of water should be available as the site.
III. The pumping station should be at higher level above all the sources of contamination.
IV. The location site should be such that future growth and expansion may be possible.
V. The source of water should be permanent. In case of meandering rivers, the site should be such that the water is available at the pumping station.
VI. The site must be accessible and as close as possible to the delivery point
VII. Foundation and bank conditions must be suitable
VIII. Power source must be accounted while making site selection
IX. Availability of construction material
X. Ease of operation and maintenance is also one factor for site selection

4.4 MAJOR COMPONENTS OF PUMPING FACILITY

Major components of pumping facility are:

- The building or pump house
- The buildings are designed to accommodate the pump, motor and other couplings with door opening wide enough to allow good access and are designed for well ventilated circumstances. Ventilation mesh is usually provided. The Concrete pump seat is also provided with proper reinforcements. Usually the pump house also would have guard room and room for tools and accessories.
- The hydraulic system: the suction sump, pump, pipes, fittings, delivery unit and anchor blocks are part of the hydraulic system.
- Systematic arrangement of the hydraulic system (the pumps, pipes, fittings, suction and delivery) is very important for efficient pumping system.
- The power supply unit: For example in electrical system the power line, transformer, the cables, motor and its related components.
- The electrical and instrumentation components includes the following: a) electrical facilities configuration; b) emergency power design elements and sizing; c) fuel storage and transfer systems; d) pump station equipment selection and control system; e) pump station control room arrangement; and f) facility lighting, intrusion detection, and fire detection.
- Lifting system: The lifting system can be manual, mechanical or electrical hoisting mechanism used during installation and maintenance. This may serve to lift and re-fix the pump, motor or engine, fittings and other accessories related to the pumping system.
- The control system: pressure, flow rate, control board, control switches and fuses etc.
- Ball valve and wide body globe valves are used to control flow and pressure from a pump as well as reduce water hammer during shutdown.

5 PIPES, VALVES & FITTINGS

5.1 PIPES

The type and size of pipe plays an important role in conveyance of fluid from one position to other. The design of any pipe network involves various activities like the selection of pipe material, specification with respect to nominal pressure, wall roughness, wall thickness, nominal diameter, galvanization, etc. Different pipe materials are used for irrigation like steel, galvanized iron, aluminum, asbestos, concrete, LDPE, HDPE, PVC and cast iron pipe. The pipes can be connected with thread, flange, welds, socket and different seals or gaskets. The selection preference depends on the pressure (including water hammer), the discharge and fluid property, economy, availability in market, durability and any other issues on site like installation conditions, corrosion etc. It is always important to understand **class of the pipe as per ISO standards** to use for the intended purpose.

The following key factors apply to material selection:

- Primary consideration shall be given to materials with good market availability and documented fabrication and service performance.
- The number of different material types shall be minimized considering costs, interchangeability and availability of relevant spare parts.
- Design life.
- Operating conditions.
- Experience with materials and corrosion protection methods from conditions with similar corrosivity.
- System availability requirements.
- Philosophy applied for maintenance and degree of system redundancy.
- Weight reduction.
- Inspection and corrosion monitoring possibilities.
- Effect of external and internal environment, including compatibility of different materials.
- Evaluation of failure probabilities, failure modes, criticalities and consequences.
- Attention shall be paid to any adverse effects material selection may have on human health, environment, safety and material assets.
- Environmental issues related to corrosion inhibition and other chemical treatments.
- For main systems where materials/fabrication represent significant investments and/or operational costs.

5.2 VALVES

A valve is a mechanical device that controls the flow of fluid and pressure within a system or process. A valve controls system or process fluid flow and pressure by performing any of the following functions:

- Stopping and starting fluid flow
- Varying (throttling) the amount of fluid flow
- Controlling the direction of fluid flow
- Regulating downstream system or process pressure
- Relieving component or piping over pressure

There are many valve designs and types that satisfy one or more of the functions identified above. A multitude of valve types and designs safely accommodate a wide variety of industrial applications.

5.2.1 Basic parts of a valves

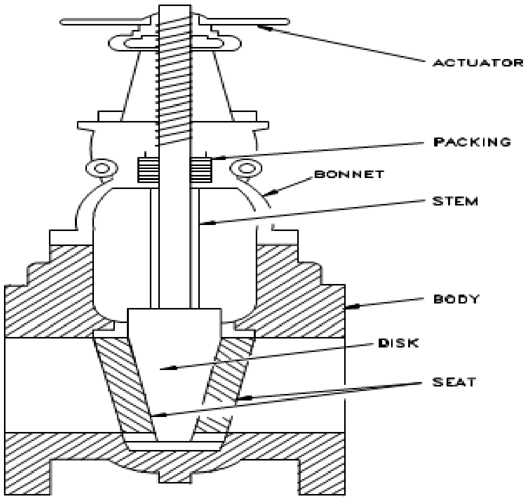


Figure 5-1: Valve parts

Valve body

The body, sometimes called the shell, is the primary pressure boundary of a valve. It serves as the principal element of a valve assembly because it is the framework that holds everything together. The body, the first pressure boundary of a valve, resists fluid pressure loads from connecting piping. It receives inlet and outlet piping through threaded, bolted, or welded joints.

The basic form of a valve body typically is not spherical, but ranges from simple block shapes to highly complex shapes in which the bonnet, a removable piece to make assembly possible, forms part of the pressure resisting body.

Valve bonnet

The cover for the opening in the valve body is the *bonnet*. In some designs, the body itself is split into two sections that bolt together. Like valve bodies, bonnets vary in design. Some bonnets function simply as valve covers, while others support valve internals and accessories such as the stem, disk, and actuator.

The bonnet is the second principal pressure boundary of a valve. It is cast or forged of the same material as the body and is connected to the body by a threaded, bolted, or welded joint. In all cases, the attachment of the bonnet to the body is considered a pressure boundary. This means that the weld joint or bolts that connect the bonnet to the body are pressure-retaining parts.

Valve trim

The internal elements of a valve are collectively referred to as a valve's *trim*. The trim typically includes a *disk, seat, stem,* and *sleeves needed* to guide the stem. A valve's performance is determined by the disk and seat interface and the relation of the disk position to the seat.

Because of the trim, basic motions and flow control are possible. In rotational motion trim designs, the disk slides closely past the seat to produce a change in flow opening.

Disk and seat

For a valve having a bonnet, the disk is the third primary principal pressure boundary. The disk provides the capability for permitting and prohibiting fluid flow. With the disk closed, full system pressure is applied across the disk if the outlet side is depressurized. For this reason, the disk is a pressure-retaining part. Disks are typically forged and, in some designs, hard-surfaced to provide good wear characteristics. A fine surface finish of the seating area of a disk is necessary for good sealing when the valve is closed. Most valves are named, in part, according to the design of their disks.

Stem

The *stem*, which connects the actuator and disk, is responsible for positioning the disk. Stems are typically forged and connected to the disk by threaded or welded joints. For valve designs requiring stem packing or sealing to prevent leakage, a fine surface finish of the stem in the area of the seal is necessary. Typically, a stem is not considered a pressure boundary part.

Connection of the disk to the stem can allow some rocking or rotation to ease the positioning of the disk on the seat. Alternately, the stem may be flexible enough to let the disk position itself against the seat.

Valve actuator

The *actuator* operates the stem and disk assembly. An actuator may be a manually operated hand wheel, manual lever, motor operator, solenoid operator, pneumatic operator, or hydraulic ram. In some designs, the actuator is supported by the bonnet. In other designs, a yoke mounted to the bonnet supports the actuator.

Except for certain hydraulically controlled valves, actuators are outside of the pressure boundary. Yokes, when used, are always outside of the pressure boundary.

Valve Packing

Most valves use some form of packing to prevent leakage from the space between the stem and the bonnet. *Packing* is commonly a fibrous material (such as flax) or another compound (such as Teflon) that forms a seal between the internal parts of a valve and the outside where the stem extends through the body.

Valve packing must be properly compressed to prevent fluid loss and damage to the valve's stem. If a valve's packing is too loose, the valve will leak, which is a safety hazard. If the packing is too tight, it will impair the movement and possibly damage the stem.

5.2.2 Types of valves

Because of the diversity of the types of systems, fluids, and environments in which valves must operate, a vast array of valve types have been developed. Examples of the common types are the globe valve, gate valve, ball valve, plug valve, butterfly valve, diaphragm valve, check valve, pinch valve, and safety valve. Each type of valve has been designed to meet specific needs. Some valves are capable of throttling flow, other valve types can only stop flow, others work well in corrosive systems, and others handle high pressure fluids. Each valve type has certain inherent advantages and disadvantages. Understanding these differences and how they affect the valve's application or operation is necessary for the successful operation of a facility.

Although all valves have the same basic components and function to control flow in some fashion, the method of controlling the flow can vary dramatically. In general, there are four methods of controlling flow through a valve.

- Move a disc, or plug into or against an orifice (for example, globe or needle type valve).
- Slide a flat, cylindrical, or spherical surface across an orifice (for example, gate and plug valves).
- Rotate a disc or ellipse about a shaft extending across the diameter of an orifice (for example, a butterfly or ball valve).
- Move a flexible material into the flow passage (for example, diaphragm and pinch valves).

Each method of controlling flow has characteristics that make it the best choice for a given application of function.

5.2.2.1 Gate Valves

A *gate* valve is a linear motion valve used to start or stop fluid flow; however, it does not regulate or throttle flow. The name gate is derived from the appearance of the disk in the flow stream. Figure below illustrates a gate valve.

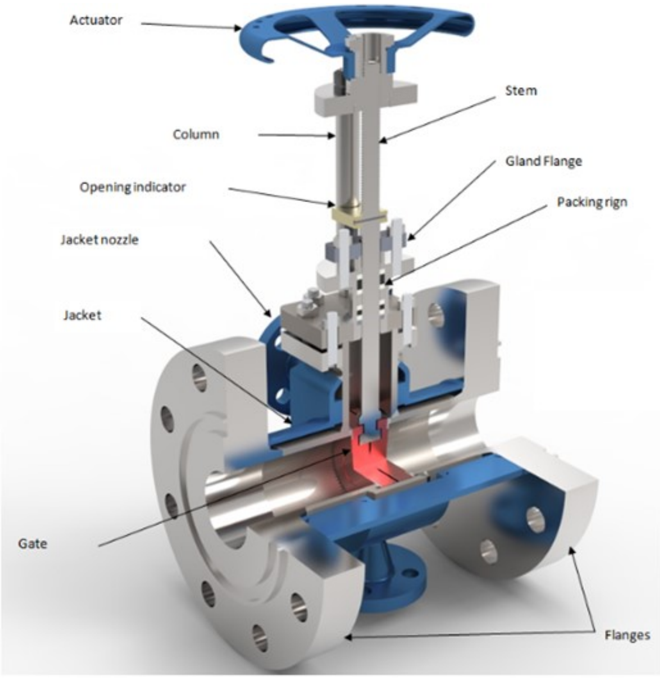


Figure 5-2: Gate valve

5.2.2.2 Globe valves

A globe valve is a linear motion valve used to stop, start, and regulate fluid flow. A Z-body globe valve is illustrated in Figure below.

The globe valve disk can be totally removed from the flow path or it can completely close the flow path. The essential principle of globe valve operation is the perpendicular movement of the disk away from the seat. This causes the annular space between the disk and seat ring to gradually close as the valve is closed. This characteristic gives the globe valve good throttling ability, which permits its use in regulating flow. Therefore, the globe valve may be used for both stopping and starting fluid flow and for regulating flow.

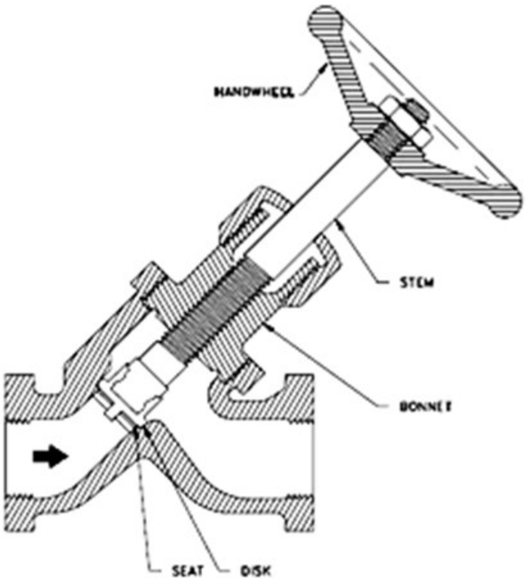


Figure 5-3: Globe valve

5.2.2.3 Ball valves

A ball valve is a rotational motion valve that uses a ball-shaped disk to stop or start fluid flow. The ball, shown in below, performs the same function as the disk in the globe valve. When the valve handle is turned to open the valve, the ball rotates to a point where the hole through the ball is in line with the valve body inlet and outlet. When the valve is shut, the ball is rotated so that the hole is perpendicular to the flow openings of the valve body and the flow is stopped.

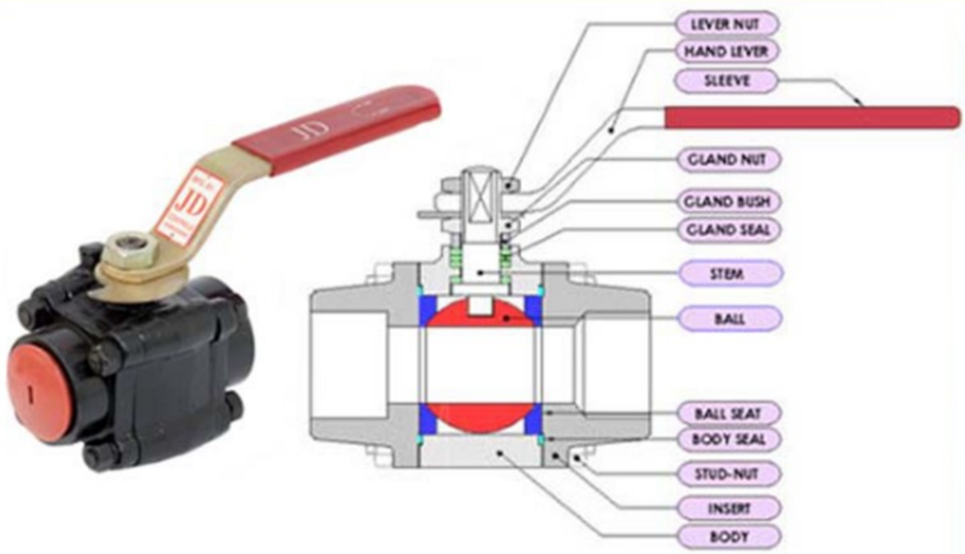


Figure 5-4: Ball valves

5.2.2.4 Plug valves

A plug valve is a rotational motion valve used to stop or start fluid flow. The name is derived from the shape of the disk, which resembles a plug. A plug valve is shown below. The simplest form of a plug valve is the petcock. The body of a plug valve is machined to receive the tapered or cylindrical plug. The disk is a solid plug with a bored passage at a right angle to the longitudinal axis of the plug.

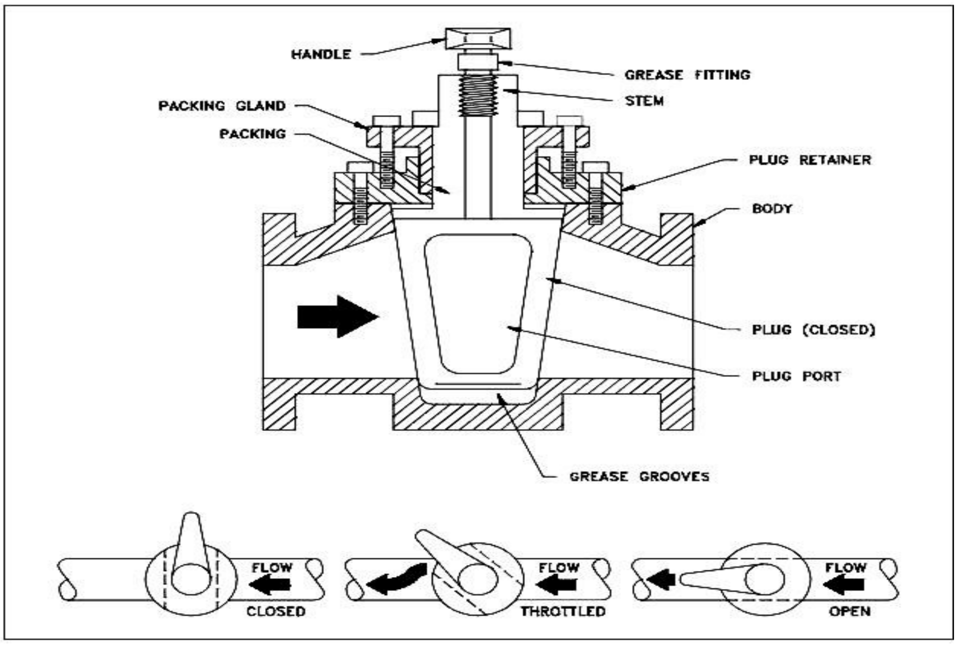


Figure 5-5: Plug valves

In the open position, the passage in the plug lines up with the inlet and outlet ports of the valve body. When the plug is turned 90 degree from the open position, the solid part of the plug blocks the ports and stops fluid flow.

Plug valves are available in either a lubricated or non-lubricated design and with a variety of styles of port openings through the plug as well as a number of plug designs.

5.2.2.5 Diaphragm valves

A diaphragm valve is a linear motion valve that is used to start, regulate, and stop fluid flow. The name is derived from its flexible disk, which mates with a seat located in the open area at the top of the valve body to form a seal. A diaphragm valve is illustrated in Figure below.

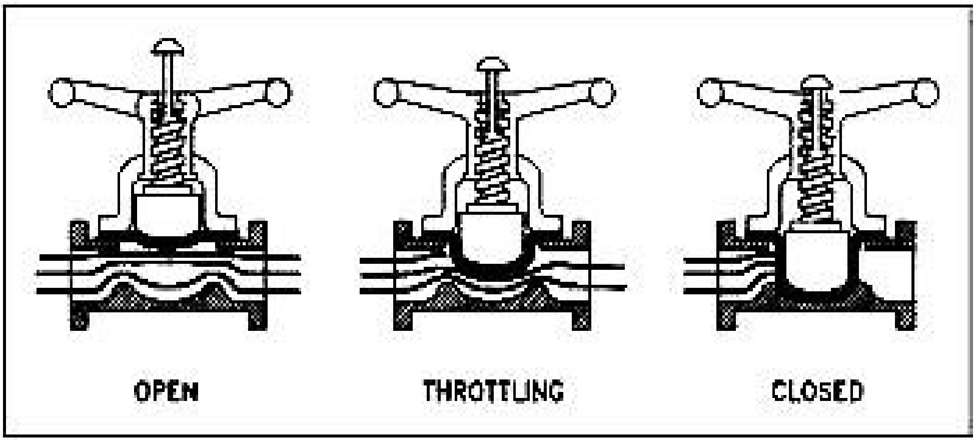


Figure 5-6: Straight through diaphragm valve

Diaphragm valves are, in effect, simple "pinch clamp" valves. A resilient, flexible diaphragm is connected to a compressor by a stud molded into the diaphragm. The compressor is moved up and down by the valve stem. Hence, the diaphragm lifts when the compressor is raised. As the compressor is lowered, the diaphragm is pressed against the contoured bottom in the straight through.

5.2.2.6 Reducing valves

Reducing valves automatically reduce supply pressure to a preselected pressure as long as the supply pressure is at least as high as the selected pressure. As illustrated below, the principal parts of the reducing valve are the main valve; an upward-seating valve that has a piston on top of its valve stem, an upward-seating auxiliary (or controlling) valve, a controlling diaphragm, and an adjusting spring and screw.

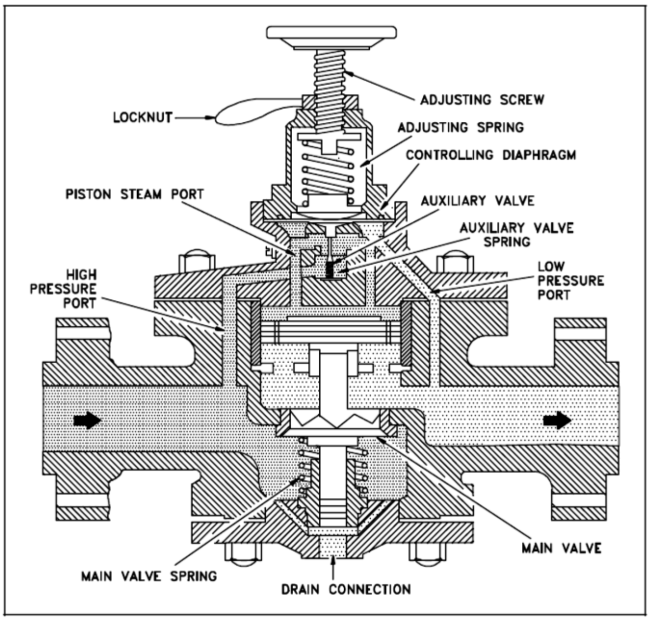


Figure 5-7: Variable reducing valve

Reducing valve operation is controlled by high pressure at the valve inlet and the adjusting screw on top of the valve assembly. The pressure entering the main valve assists the main valve spring in keeping the reducing valve closed by pushing upward on the main valve disk. However, some of the high pressure is bled to an auxiliary valve on top of the main valve. The auxiliary valve controls the admission of high pressure to the piston on top of the main valve. The piston has a larger surface area than the main valve disk, resulting in a net downward force to open the main valve. The auxiliary valve is controlled by a controlling diaphragm located directly over the auxiliary valve.

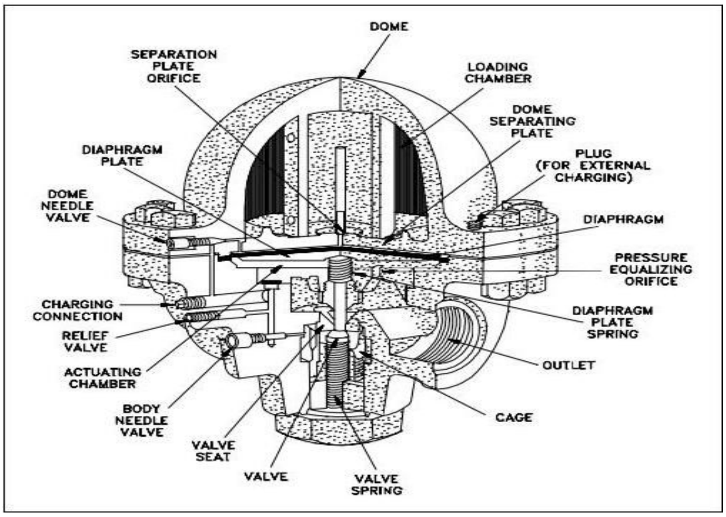


Figure 5-8: Non-variable reducing valves

Non-variable reducing valves eliminate the need for the intermediate auxiliary valve found in variable reducing valves by having the opposing forces react directly on the diaphragm. Therefore, non-variable reducing valves are more responsive to large pressure variations and are less susceptible to failure than are variable reducing valves.

5.2.2.7 Butter fly valves

A butterfly valve, illustrated in Figure below, is a rotary motion valve that is used to stop, regulate, and start fluid flow. Butterfly valves are easily and quickly operated because a 90^{o} rotation of the handle moves the disk from a fully closed to fully opened position. Larger butterfly valves are actuated by hand wheels connected to the stem through gears that provide mechanical advantage at the expense of speed.

Butterfly valves possess many advantages over gate, globe, plug, and ball valves, especially for large valve applications. Savings in weight, space, and cost are the most obvious advantages. The maintenance costs are usually low because there are a minimal number of moving parts and there are no pockets to trap fluids.

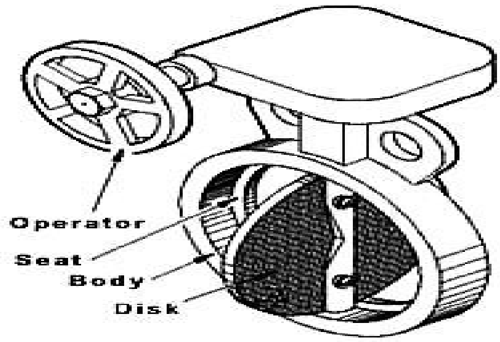


Figure 5-9: Butterfly valve

5.2.2.8 Check valves

Check valves are designed to prevent the reversal of flow in a piping system. These valves are activated by the flowing material in the pipeline. The pressure of the fluid passing through the system opens the valve, while any reversal of flow will close the valve. Closure is accomplished by the weight of the check mechanism, by back pressure, by a spring, or by a combination of these means. The general types of check valves are swing, tilting-disk, piston, butterfly, and stop.

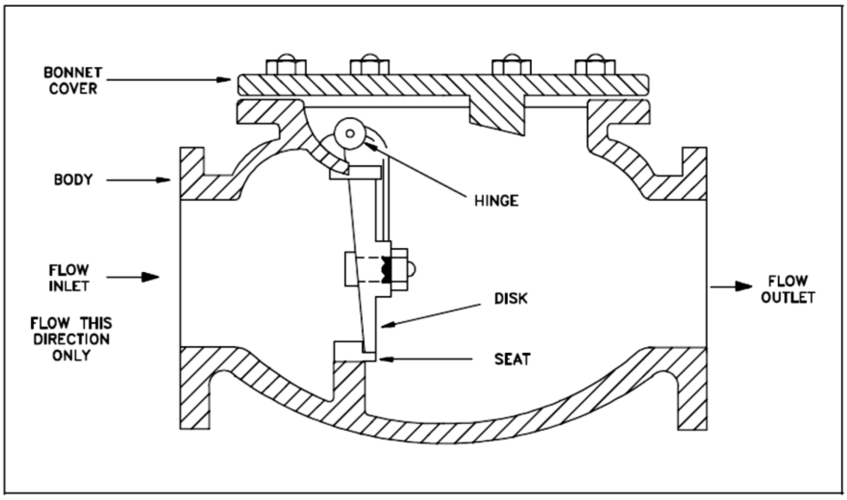


Figure 5-10: Swing check valve

A lift check valve, illustrated in Figure below, is commonly used in piping systems in which globe valves are being used as a flow control valve. They have similar seating arrangements as globe valves.

Lift check valves are suitable for installation in horizontal or vertical lines with upward flow. They are recommended for use with steam, air, gas, water, and on vapor lines with high flow velocities. These valves are available in three body patterns: horizontal, angle, and vertical.

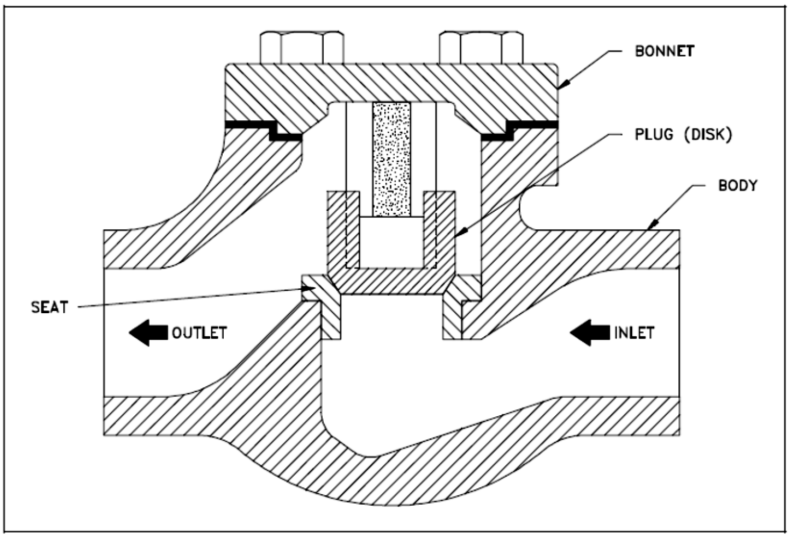


Figure 5-11: Lift check valve

5.2.2.9 Pressure release valve

Obviously as the name suggests pressure relief valves are a safety device designed to open when system pressure (i.e. in a vessel or pipework) becomes too great and may damage equipment or endanger personnel if not relieved.

The most common type is the spring operated valve. A valve feather under spring pressure is seated in the valve body and exposed to system pressure. When the system pressure overcomes the spring pressure the valve feather will move in the seat creating an exit to atmosphere allowing the system gas or liquid to escape.

5.2.2.10 Air/vacuum valve

The Air/Vacuum Valve has been designed with stainless steel trim to give years of trouble free operation. The Air/vacuum valve is typically mounted on a pipeline at the high points or large changes in grade. The valve will exhaust large quantities of air during system start-up and allow air to re-enter the line upon system shut down or after a power failure. The valves are needed to maintain pipeline efficiency while providing protection from adverse pressure condition. The Size, Maximum Working Pressure, and Model No. are stamped on the nameplate for reference.

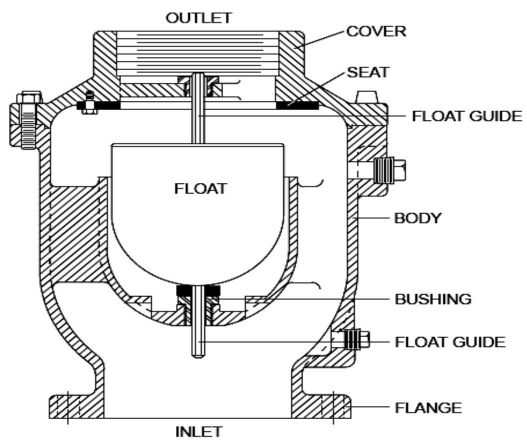


Figure 5-12: Air/vacuum Valve

Operation of the valve:

The Air/Vacuum Valve is designed to exhaust large quantities of air during system startup and allow air to reenter the line upon system shutdown or line break. As water enters the valve during startup, the float will rise and close the outlet port. The valve will remain closed until system pressure drops to near zero pressure. It will open during shutdown to perform a dual purpose. First, it eliminates the possibility of a vacuum forming and a potential pipeline collapse. Second, it allows rapid drainage of the line when system maintenance is required. The only moving parts in the valve are the float and the float guide. The float guide assures that the float enters the seat at the optimum angle and prevents float contact with any surface other than the resilient seat thus assuring a long, maintenance free life. An optional Anti-Slam Valve may be bolted to the bottom of the air/vacuum valve to provide regulated closure.

Installation of the valve:

The installation of the valve is important for its proper operation. The valves must be installed at the system high points in the vertical position with the inlet down. For pipeline service, a vault with freeze protection, adequate screened venting, and drainage should be provided. During closure, some fluid discharge will occur so vent lines should extend to an open drain for in-plant installations. A shutoff valve should be installed below the valve in the event servicing is required. A spool piece is required when mating to a wafer butterfly valve.

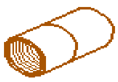
Valve operation rules:

Do not close a valve before opening another!
Do not close a valve before stopping a pump!

5.3 FITTINGS

Fittings are bends, junctions, reducers, tapers, joint adaptors and similar items which are not joints or flow control equipment. Fittings are attached to pipes to provide various functions, ranging from length extension to direction changes and so on. Here are some of the most common pipe fittings and their functions:

Table 5-1: Types of Fittings

 Bell Reducer	**Bell Reducer.** A bell reducer has female threads on both ends. Bell reducers are generally not available in PVC (so, despite what the caption says above, this drawing is not of a PVC fitting).
 Cap	**Cap.** A cap may have a solvent weld socket end or a female threaded end. The other end is closed off. If a solvent weld cap is used to provide for a future connection point, be sure to leave several inches of pipe before the cap! When the cap is cut off for the future connection there will need to be enough pipe present to glue a new fitting onto! I can't begin to tell you how many times I've seen a solvent weld cap butted right up against another fitting, making it impossible to ever use the capped connection again!
 Coupling	**Coupling.** A coupling connects two sections of pipe together. Couplings may have solvent weld socket ends or female threaded ends.
 Cross	**Cross.** A cross connects four pipe sections together. Crosses may have solvent weld socket ends or female threaded ends (no female threads available for PVC). Crosses are special order parts at many suppliers. Crosses create a great deal of stress on the pipe because they have four connection points. In theory this is the same principle that makes a 3 leg stool (a "tee") more steady than a 4 leg stool (a "cross"). I recommend that you avoid using crosses in most situations. Use two tees.
 Female Adapter	**Female Adapter.** Female adapters are used to add a female threaded pipe connection on a solvent welded pipe. Never use female adapters when converting to a metallic pipe. The metal male pipe threads tend to split the PVC fittings. Place a metal coupling on the metallic pipe then use a PVC male adapter. Metal male threads should never be inserted into any female threaded PVC fitting!
 Male Adapter	**Male Adapter.** Male adapters are used to add a male threaded pipe connection to a solvent weld pipe section.
 Plug	**Plug.** Used to plug an unused fitting outlet. May have female threads or a solvent weld spigot. In most cases a threaded plug is used to provide a connection point for future use. If solvent welded in place the plug is never going to be removed!
 Side Outlet Ell	**Side Outlet Ell.** Side outlet ells are an ell with a side outlet. They most commonly have two 3/4″ or 1″ solvent weld sockets, with a 1/2″ side outlet having female threads. Side outlet ells are common in residential sprinkler systems, but are seldom used in commercial installations. The side outlet is listed last when stating the side outlet ell size. Example: 1x1x1/2 SO ELL SST has a 1/2″ threaded side outlet.

 Tee	**Tee.** The most common fitting! Available with all female thread sockets, all solvent weld sockets, or with opposed solvent weld sockets and a side outlet with female threads. Many configurations of “reducer tees” are available, meaning that one or more of the sockets is smaller than the others. Tees are always labeled as TEE with the side outlet as the last size. The larger of the other two sockets is always listed first.

6 HEAD IN PUMPING SYSTEM

Figure 6-1 illustrates the general situation of pumping system between the suction sump and the delivery tank. The pump axis level is reference for suction and delivery sides.

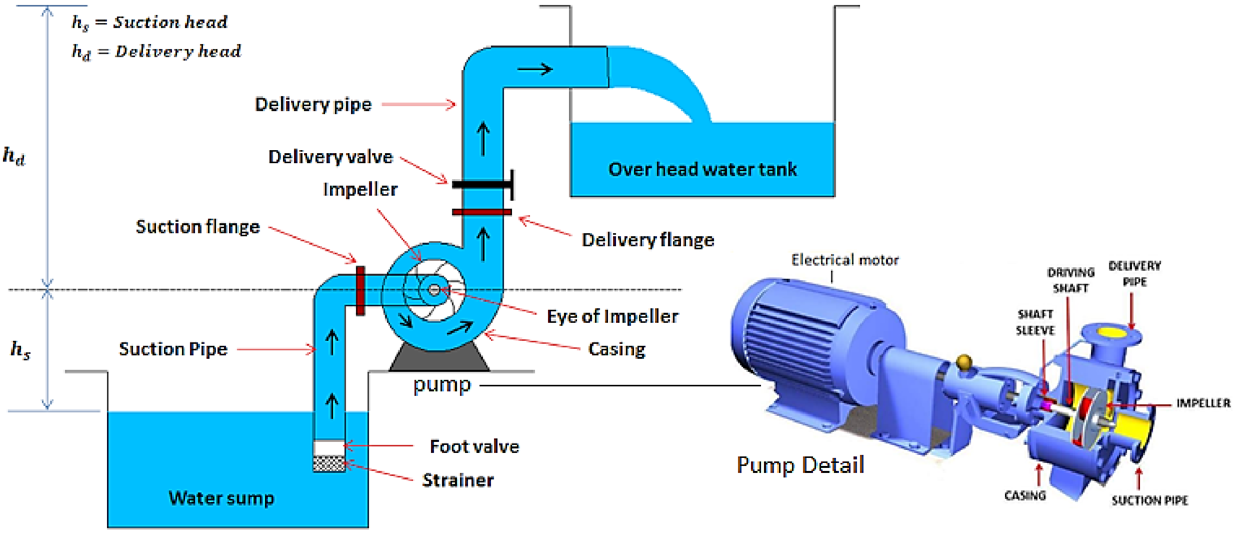


Figure 6-1 General situation of irrigation pumping system

6.1 TOTAL DYNAMIC HEAD OR TOTAL PUMPING HEAD

Head is the expression of the potential energy imparted to a liquid to move it from one level to another. Total dynamic head or total pumping head is the head that the pump is required to impart to a fluid in order to meet the head requirement of a particular system, whether this is a town water supply system or an irrigation system. The total dynamic head is made up of static suction lift or static suction head, static discharge head, total static head, required pressure head, friction head and velocity head. Figure 6-2 shows the various components making up the total dynamic head. Source: FAO irrigation Manual Module-5, 2001.

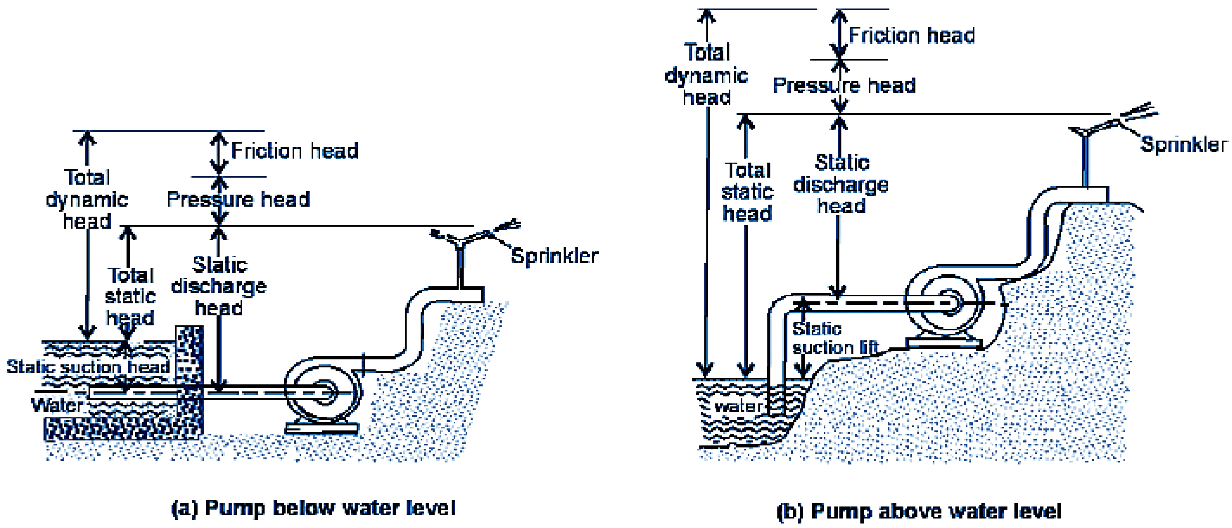


Figure 6-2 Components of total dynamic head

6.2 STATIC SUCTION HEAD OR STATIC SUCTION LIFT

When a pump is installed such that the level of the water source is above the eye of the impeller (flooded suction), then the system is said to have a positive suction head at the eye of the impeller. However, when the pump is installed above the water source, the vertical distance from the surface of the water to the eye of the impeller is called the static suction lift.

6.3 STATIC DISCHARGE HEAD

This is the vertical distance or difference in elevation between the point at which water leaves the impeller and the point at which water leaves the system, for example the outlet of the highest sprinkler in an overhead irrigation system.

6.4 TOTAL STATIC HEAD

When no water is flowing (static conditions), the head required to move a drop of water from water source to the highest sprinkler or outlet point is equal to the total static head. This is simply the difference in elevation between where we want the water and where it is now. For systems with the water level is above the pump, the total static head is the difference between the elevations of the water and the sprinkler (Figure 6-2a).

Total Static Head = Static Discharge Head – Static Suction Head
For systems where the water level is below the pump, the total static head is the static discharge head plus the static suction lift (Figure 6-2b).
Total Static Head = Static Discharge Head + Static Suction Lift

6.5 FRICTION HEAD

The pressure decreases in pipe flow because of the friction against the walls of the pipe over its length. Therefore, the pump needs to provide the necessary energy to the water to overcome the friction losses. The losses must be considered both for the suction part and the discharge part of the pump. The magnitude of the friction head can be calculated using either hydraulic formulae or tables and graphs.

6.6 PRESSURE HEAD

Except for the cases where water is discharged to a reservoir, or a canal, a certain head to operate an irrigation system is required. For example, in order for a sprinkler system to operate, a certain head is required.

6.7 VELOCITY HEAD

This energy component is not shown in Figure 6-2. It is very small and is normally not included in practical pressure calculations. Most of the energy that a pump adds to flowing water is converted to pressure in the water. Some of the energy is added to the water to give the velocity it requires to move through the pipeline. The faster the water is moving the larger the velocity head. The amount of energy that is needed to move water with a certain velocity is given by the formula:

Velocity Head = $V^2/2g$ **Equation 1**

Where:

V = the velocity of the water (m/s)

g = the gravitational force which is equal to 9.81 (m/s2)

Keller and Bliesner (1990) recommend that for centrifugal pumps the diameter of the suction pipe should be selected such that the water velocity V< 3.3 m/s in order to assure good pump performance. However, according to MoWR Design Manual on Pumping Facilities, 2001 the suction velocity is taken about 1.5 m/s while for delivery line is about 2m/s.

6.8 DRAWDOWN

Usually, the level of the water in a well or even a reservoir behind a dam does not remain constant. In the case of a well, after pumping starts with a certain discharge, the water level lowers. This lowering of the water level is called drawdown. In the case of a dam or reservoir, fluctuation of the water level is common and depends on water inflow, evaporation and water withdrawal. The water level increases during the rainy season, followed by a decrease during the dry season because of evaporation and withdrawal of the stored water. This variation in water level will affect the static suction lift or the static suction head and, correspondingly, the total static head.

7 LOSSES IN FLOW

A pipeline is a circular conduit used to convey fluid from one location in the system to another. A pipeline consists of a circular pipe full of fluid, the valves and fittings used to direct the flow of fluid through the pipe in the operation. Each of these items affects the head loss in the pipeline.

Flow of fluid through a pipe is resisted by viscous shear stresses within the fluid and the turbulence that occurs along the internal pipe wall, which is dependent on the roughness of the pipe material. This resistance is termed pipe friction and is usually measured in **meters head** of the fluid, which is why it is also referred to as the head loss due to pipe friction.

I. Friction loss

When fluid flows inside a pipeline, friction occurs between the moving fluid and the stationary pipe wall. This friction converts some of the fluid's hydraulic energy to thermal energy. This thermal energy cannot be converted back to hydraulic energy, so the fluid experiences a drop in pressure. This conversion and loss of energy is known as head loss. The head loss in a pipeline with Newtonian fluids can be determined using the Darcy equation.

Pipe Friction Loss Calculations:
A large amount of research has been carried out over many years to establish various formulae that can calculate head loss in a pipe. Most of this work has been developed based on experimental data.

Overall head loss in a pipe is affected by a number of factors which include the viscosity of the fluid, the size of the internal pipe diameter, the internal roughness of the inner surface of the pipe, the change in elevation between the ends of the pipe and the length of the pipe along which the fluid travels.

Valves and fittings on a pipe also contribute to the overall head loss that occurs, however these must be calculated separately using fitting loss factors.

a. Darcy Weisbach Formula

The Darcy formula or the Darcy-Weisbach equation as it tends to be referred to, is now accepted as the most accurate pipe friction loss formula, and although more difficult to calculate and use than other friction loss formula, with the introduction of computers, it has now become the standard equation for hydraulic engineers.

Darcy-Weisbach equation:

$$\boldsymbol{h_f = f * \frac{L}{D} * \frac{V^2}{2g}} \qquad \text{Equation 2}$$

Where;
hf = the head loss due to friction (SI units: m); Note: This is also proportional to the piezometric head along the pipe;
L = is the length of the pipe (m);
D = is the hydraulic diameter of the pipe (for a pipe of circular section, this equals the internal diameter of the pipe) (m);
V = the average flow velocity, experimentally measured as the volumetric flow rate per unit cross-sectional wetted area (m/s);

g = the local acceleration due to gravity (m/s^2);

f = a dimensionless parameter called the Darcy friction factor, resistance coefficient or simply friction factor. f can be found from a Moody diagram or calculated;

Note: This friction factor must not be confused with the fanning friction factor with is one-fourth of the Darcy friction factor.

For Laminar flow: $f = \frac{64}{\mathbf{Re}}$ Equation 3

Re is the Reynolds Number for flow in pipe is given by;

$$Re = \frac{\mathrm{VD}}{\boldsymbol{\upsilon}}$$ Equation 4

Where,

V= flow velocity in m/s

D= hydraulic diameter of pipe in m

$\boldsymbol{\upsilon}$= Kinematic Viscosity

$$\upsilon = \frac{\mu}{\rho}$$ Equation 5

Where,

$\boldsymbol{\mu}$= dynamic viscosity of the fluid in Kg/(m.s) water at 20°c has dynamic viscosity of 1.0016mPa.s=0.0010016Pa.s=0.0010016Kg/(m.s)

$\boldsymbol{\rho}$= density of the fluid (Kg/m^3) = 1000Kg/m^3 for normal water.

Thus Re can be expressed as, $Re = \frac{\rho \mathrm{VD}}{\mu}$ Equation 6

Re<2000 is laminar flow regime, 2000<Re<4000 is critical flow regime and Re>4000 is turbulent flow regime.

The following formula is used for approximation of friction factor (f) for turbulent flow or Re is 4000 to 10^8.:

$$f = 0.0055[1 + (2\mathrm{x}10^4 \mathrm{x} \frac{\varepsilon}{D} + \frac{10^6}{\mathbf{Re}})^{\frac{1}{3}}]$$ Equation 7

Ɛ is pipe roughness height in mm indicated on moody chart (Figure 7-1) for different pipe material.

D is pipe inner diameter in mm.

Ɛ/D is relative pipe roughness

Re is the computed Reynolds number.

The Colebrook Equation is as given below is also used to compute Darcy friction factor (f) for turbulent flow:

$$\frac{1}{\sqrt{f}} = -2log_{10}(\frac{\varepsilon/D}{3.7} + \frac{2.51}{Re\sqrt{f}})$$ Equation 8

Table 7-1: Dynamic and kinematic viscosity of water

Temperature	Pressure	Dynamic viscosity		Kinematic viscosity
[°C]	[MPa]	[Pa s]	[mPa s]	[m^2/s*10^{-6}]
0.01	0.000612	0.0017914	1.7914	1.7918
10	0.0012	0.001306	1.306	1.3065
20	0.0023	0.0010016	1.0016	1.0035

Temperature	Pressure	Dynamic viscosity		Kinematic viscosity
25	0.0032	0.00089	0.89004	0.8927
30	0.0042	0.0007972	0.79722	0.8007
40	0.0074	0.0006527	0.65272	0.6579
50	0.0124	0.0005465	0.5465	0.5531
60	0.0199	0.000466	0.46602	0.474
70	0.0312	0.0004035	0.40353	0.4127
80	0.0474	0.000354	0.35404	0.3643
90	0.0702	0.0003142	0.31417	0.3255
100	0.101	0.0002816	0.28158	0.2938

The establishment of more accurate friction factors is however still unresolved, and indeed was an issue that needed further work to develop a solution such as that produced by the Colebrook-White formula and the data presented in the Moody chart.

Example: What is the friction factor for steel (Ɛ =0.025mm figure 7-1 Moody Chart) pipe of diameter 300mm conveying water discharge of 100 liter/ second at 20^{o}c? How much is the frictional head loss per 100m length of the pipe according to Darcy Weisbach?

Area of the pipe=A=$\pi D^2/4$=0.07065m^2
Discharge= Q=100l/s or 0.1m^3/s
Thus Flow Velocity by continuity equation= V=Q/A=1.415m/s
Kinematic viscosity of water at 20 ^{o}c is 1.0035x10^{-6} m^2/s from table 7-1.
$Re = \frac{\mathrm{VD}}{\upsilon}$ =(1.415m/s*0.3m)/1.0035x$10^{-6}$$m^2$/s=4.23x$10^5$ turbulent flow.
ε=0.025mm for steel pipe, d=300mm
Thus ε/d or relative pipe roughness=8.33x10^{-5}.
Friction factor using the estimation formula:
$f = 0.0055[1 + (2\mathrm{x}10^4\mathrm{x}\frac{\varepsilon}{D} + \frac{10^6}{\mathbf{Re}})^{\frac{1}{3}}]$ =**0.01425** or using Colebrook equation by trial and error f is found to **be 0.01456**, thus:
Friction head loss in 100m length steel pipe: $\boldsymbol{h_f} = \boldsymbol{f} * \frac{\boldsymbol{L}}{\boldsymbol{D}} * \frac{\boldsymbol{V^2}}{\boldsymbol{2g}}$
h_f=0.01456*(100/0.3)*(1.415^2/19.62)=**0.495m**
Optionally friction factor f can also be read from Moody chart for the calculated Reynolds number **Re** and relative pipe roughness ε/**d**.

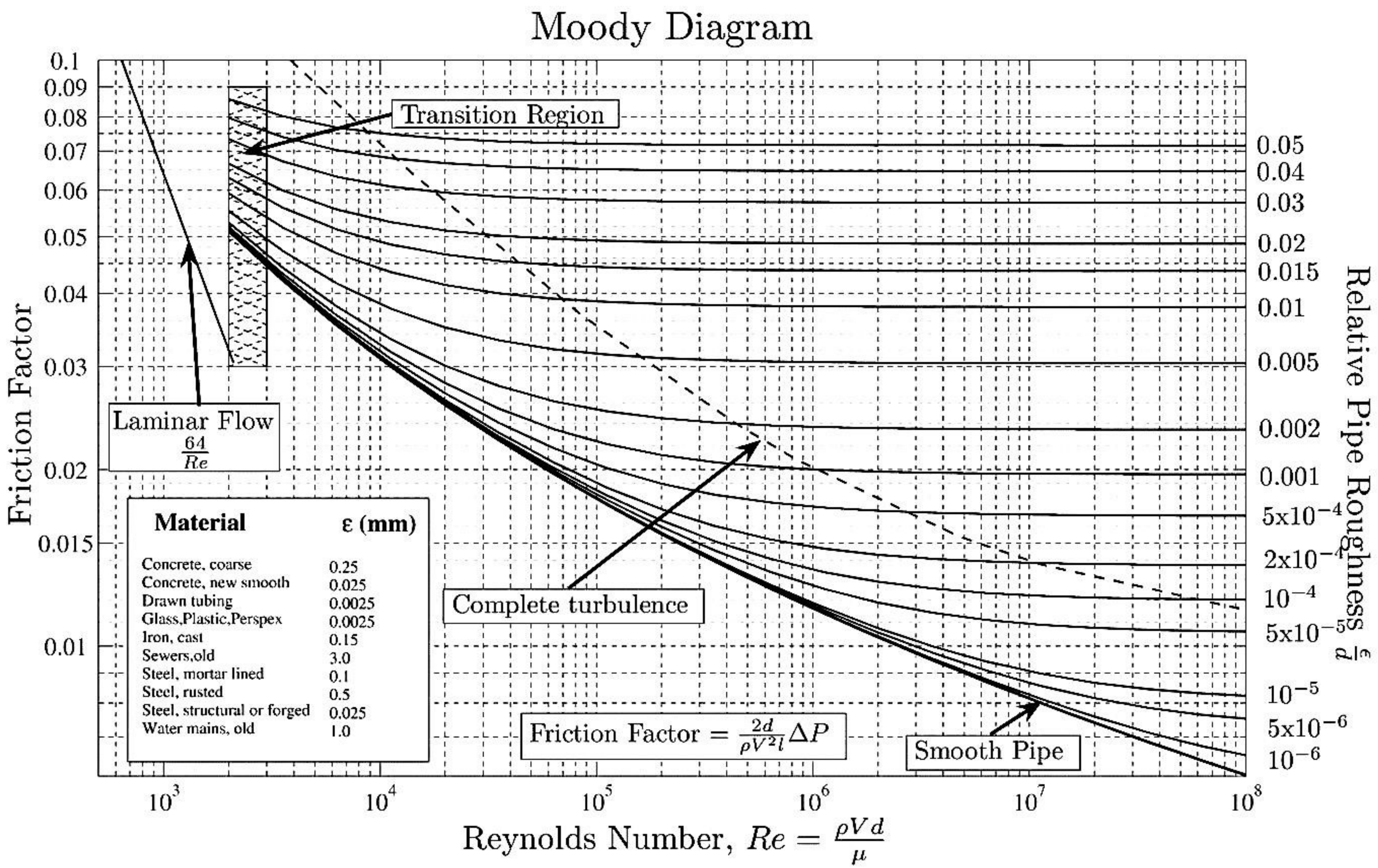


Figure 7-1: Moody diagram

b. Hazen-Williams formula

Before the advent of personal computers the Hazen-Williams formula was extremely popular with piping engineers because of its relatively simple calculation properties.

However the Hazen-Williams results rely upon the value of the friction factor, C, which is used in the formula and the C value, can vary significantly, from around 80 up to 130 and higher, depending on the pipe material, pipe size and the fluid velocity.

Also the Hazen-Williams equation only really gives good results when the fluid is Water and can produce large inaccuracies when this is not the case.

The imperial form of the Hazen-Williams formula is:
Friction loss in pipe is given by Hazen William's formula given as:

$$hf = (L) * [\frac{10.67 * (Q)^{1.852}}{(C)^{1.852}(D)^{4.8704}}]$$ Equation 9

Where,
hf=loss due to friction (m)
L= pipe length (m)
C= pipe friction factor given under table 7-2
Q= flow in pipe (m^3/s)
D= pipe diameter in m
Common Friction Factor Values (C) used for design purposes are given below:

Table 7-2: Common friction factor values

Description	Common Friction Factor	Description	Common Friction Factor
Asbestos Cement	140	Plastic pipe	140
Brass tube	130	PVC pipe	150
Cast-Iron tube	100	General smooth pipes	140
Concrete tube	110	Steel pipe	120
Copper tube	130	Steel riveted pipes	100
Corrugated steel tube	60	Tar coated cast iron tube	100
Galvanized tubing	120	Tin tubing	130
Glass tube	130	Wood Stave	110
Lead piping	130		

These C values provide some allowance for changes to the roughness of internal pipe surface, due to corrosion of the pipe wall during long periods of use and the buildup of other deposits.

Example: Compute friction loss using Hazen Williams formula for the above example.

Friction loss in 100m length of steel pipe, C=120 from table 7-2.
L=100m, D=300mm or 0.30m, Q=0.10 m^3/s from given discharge of 100l/s.

$$hf = (L) * [\frac{10.67 * (Q)^{1.852}}{(C)^{1.852}(D)^{4.8704}}] \text{ hence } hf = (100) * [\frac{10.67 * (0.1)^{1.852}}{(120)^{1.852}(0.3)^{4.8704}}]$$

hf=**0.745m**

Hazen-Williams is simpler to use than Darcy-Weisbach where you are solving for flow rate, pressure drop or velocity dependent on the flow. The Darcy-Weisbach formula is generally considered more accurate and is valid for any liquid or gas. The Hazen-Williams method is very popular, especially among civil engineers, because of its ease of use and its friction coefficient “C” is not a function of velocity or pipe diameter. Its validity depends on the successful selection of the constant friction loss coefficient “C”, which is a function of pipe material, pipe linings and pipe age to indicate the roughness of a pipe interior. The higher the C factor indicate, the smoother the pipe. The Hazen-Williams equation has narrow applicable ranges for Reynolds numbers and pipe sizes. The equation is generally valid for pipe sizes 75 mm and larger, for water flowing at temperatures of between 21° to 24°C and pressures up to 1.2 MPa. The level of error when the Hazen-Williams equation is used outside its data ranges is significant. In this case, the Darcy-Weisbach formula must be used. The Hazen-Williams equation is commonly used for pressure drop calculations in fire sprinkler systems, water distribution systems, and irrigation systems where conditions are mostly constant.

II. Velocity Head

To initiate flow of water to velocity of V, the pump imparts certain amount of energy to the water which is expressed to an equivalent velocity head hv. The water head is given by equation-1.

III. Minor losses

Minor losses caused by various factors are described below:

a. Bend loss

The energy loss when a fluid passes bend section is expressed as bend loss (h_b) and is given by:

$h_b=K_b * hv$ Equation 10

where,

h_v= Velocity head (m)

K_b= the coefficient of head loss due to friction occurring in bend depicted under:

Table 7-3: Loss coefficient due to bend

r/D	1	2	4	6	10	15	20
K_b	0.335	0.19	0.16	0.22	0.32	0.38	0.42

Note:-The value of K_b is the function of the radius r of the curvature and diameter D of the bend. For rough bends, value of K_b may be taken twice that of the above values. Losses in 45 degree bends are usually taken 0.5 times the losses in 180 degree bends.

-Values of K_b for 90 degree smooth bend and for a given pipe type and diameter obtained from laboratory experiments are given in table 7-3.

-The value of the Kb for rough bends is twice the values shown in table 7-3. The values of the K_b for 45 degree bends is half times the values indicated in the table, likewise the head loss values for 180 degree is 1.25 times that of 90 degree bends shown in the same table.

b. Gate Valves loss

The head loss hg occurring during the different apertures openings of the gate valve is given by;

$h_g=K_g*h_v$ Equation 11

Where:

h_v is the velocity head

K_g is the coefficient of head loss due to friction occurring in the gate valve given in table7-4.

Table 7-4 Gate valve losses

Nominal Diameter of valve (mm)	Ratio of height d of valve opening to diameter D of full valve opening (d/D)					
	1/8	**1/4**	**3/8**	**1/2**	**3/4**	**1**
25	310	32	9	4.2	0.90	0.23
50	140	20	6.5	3.0	0.68	0.16
100	91	16	5.6	2.6	0.55	0.14
150	74	14	5.3	2.4	0.49	0.12
200	66	13	5.2	2.3	0.47	0.10
300	56	12	5.1	2.2	0.47	0.09

Source: MoWR Design Manual on Pumping Facilities, 2001

Note:

- The value of K_g much depends on the ratio of the aperture height d and full opening of the gate valve D.
- The value of K_g reduces as the valve opening increases or the value of K_g decreases as the ratio d/D increases.

c. Foot valves and Strainer losses

Head losses due to the foot valve (h_{ft}) and its strainer (h_{st}) are computed as:

$\mathbf{h_{ft}=K_{ft}*h_v}$ and $\mathbf{h_{st}=K_{st}*h_v}$ Equation 12

Where:

h_v is the velocity head; and k_{ft} and k_{st} are the coefficients of head losses due to the foot valve and the strainer respectively. In computation, the values of k_{ft} and K_{st} are usually taken as 0.8 and 0.95 respectively.

d) Head Loss Due to Fittings

Frictional loss due to pipe fittings is expressed in to an equivalent length (EQL) of straight pipeline. Once the equivalent length of straight pipe for a particular fitting is obtained, the head loss due to friction can be further arrived at by using frictional loss standard tables or alternatively using standard Nomo-gram. Frictional loss due to pipe fittings is expressed in to an equivalent length of straight pipeline.

Once the equivalent length of straight pipeline is obtained, the head loss due to friction h_f can be calculated by pipe loss equation-2 (Darcy) or equation-9 (Hazen Williams).

Table 7-5: Equivalent length of straight pipe for different pipe fittings

Type of fitting		Pipe Diameter (mm)						
		25	**50**	**75**	**100**	**125**	**150**	**200**
Regular 90 degree bend	Screwed	1.6	2.60	3.30	4.00	-	-	
	flanged	0.5	0.95	1.30	1.80	2.22	2.71	3.70
Long rad. 90 degree bend	Screwed	0.82	1.10	1.20	1.40	-	-	
	flanged	0.49	0.80	1.03	1.30	1.52	1.74	2.13
Regular 45 degree bend	Screwed	0.40	0.82	1.22	1.67	-	-	
	flanged	0.25	0.52	0.82	1.06	1.37	1.70	2.34
Regular 180 degree bend	Screwed	0.58	2.60	3.35	4.00	-	-	
	flanged	0.49	0.95	1.34	1.80	137.00	1.70	2.34
Straight flow in Tee	Screwed	0.97	2.35	3.66	4.00	-	-	
	flanged	0.30	0.55	0.67	0.85	1.00	1.16	1.43
Branch flow in Tee	Screwed	2.01	3.66	5.20	6.40	-		
	flanged	1.00	2.01	2.86	3.66	4.57	5.49	7.31
Gate valve full open	Screwed	8.84	16.46	24.08	33.53	-	-	

Type of fitting		Pipe Diameter (mm)						
		25	50	75	100	125	150	200
	flanged	13.72	21.34	28.65	38.60	45.72	57.91	79.20
Angle valve	Screwed	5.18	5.49	5.49	5.49			
	flanged	5.18	6.40	8.53	11.58	15.24	19.20	27.40
Check valve	Screwed	3.35	5.80	8.23	11.58	-	-	
	flanged	2.02	5.18	8.23	11.58	15.24	19.20	27.40
Foot valve	Screwed	3.35	5.80	8.23	11.58	15.24	19.20	27.40
	flanged	-	-	-	-	-	-	-

Source: MoWR Design Manual on Pumping Facilities, 2001

e) Head Loss due to Sudden Contraction

When water is made to flow from a larger diameter into smaller pipe, an abrupt increase of velocity or turbulence flow is caused, which is accountable for head loss due to sudden contraction. This is given by:

hc=Kc*hv Equation 13

Were:

• hv is the velocity head and

• Kc is the coefficient of friction of head loss due to sudden contraction.

The values of Kc for few standard combinations of pipe diameters are given in table 7-6.

Table 7-6 Values of Kc for contraction

Velocity in smaller pipe	Ratio of Smaller to Larger Diameter								
	0.1	0.2	0.3	0.4	0.5	0.6	0.7	0.8	0.9
0.6	0.49	0.48	0.45	0.42	0.35	0.28	0.18	0.07	0.03
1.52	0.48	0.47	0.44	0.41	0.37	0.28	0.18	0.09	0.04
3.04	0.46	0.45	0.43	0.40	0.36	0.28	0.18	0.10	0.04
6.08	0.43	0.42	040	0.37	0.33	0.27	0.19	0.11	0.05
12.16	0.36	0.35	0.33	0.31	0.29	0.25	0.20	0.13	0.05

Source: MoWR Design Manual on Pumping Facilities, 2001

f) Head Loss due to Entrance of Water

This is special case of sudden contraction. When water is made to flow from a reservoir into smaller size of a pipe compared to the total water mass, local turbulence occurs that causes a head loss (h_{en}). The magnitude of such loss depends upon the shape of the pipe entrance and projection of the pipe inside the water body.

h_{en}=K_{en}*hv Equation 14

The value of K_{en} is so often taken as 0.5; however, it could be of higher order ranging from 0.5 to 0.9 depending on the length of projection of the pipe into the water body. It is smaller when the pipe is of bell shaped.

g) Head Loss Due to Enlargement

When water is made to flow from a smaller diameter into larger pipe, it causes an abrupt decrease of velocity or turbulence flow that accounts for head loss due to sudden expansion.

This is given by;

he=Ke*hv Equation 15

Where:

• hv is the velocity head; and

• Ke is the coefficient of friction of head loss due to sudden expansion.

The values of Ke for few standard combination of pipe diameters is given in table 7-7

Table 7-7: The values of Ke for sudden enlargement

Velocity in smaller pipe	Ratio of Smaller to Larger Diameter							
	0.1	0.2	0.3	0.4	0.5	0.6	0.8	0.9
0.6	1.00	1.0	0.96	0.74	0.60	0.49	0.15	0.04
1.52	0.96	0.95	0.89	0.69	0.95	0.41	0.14	0.04
3.04	0.91	0.89	0.84	0.65	0.52	0.39	0.13	0.04
6.08	0.86	0.84	0.80	0.62	0.50	0.37	0.12	0.04
12.16	0.81	0.80	0.75	0.58	0.47	0.35	0.11	0.04

h) Head Lose Due to Exit of Water

This is special case of sudden expansion. This occurs when a pipe discharges into a reservoir or a tank full of water tending the velocity energy to reduce instantly to zero thus causing a head loss of energy. The magnitude of such loss is unity since the loss instantly is complete and in practice, this head loss is equal to velocity head. Such loss is given by:

he=Kex*hv Equation 16

Where:

-Kex is the coefficient of friction of head loss due to exit of water which is commonly equal to 1.

-hv is the velocity head.

All head loss discussed above, expressed in to equivalent height of water column, have to be added to the static head to give the total dynamic head under which the pump will have to work

8 DESIGN CONSIDERATIONS IN SUCTION AND DELIVERY LINES

8.1 PIPE SIZING FOR SUCTION AND DELIVERY LINE

Besides material type, pipe over or under sizing may even become a bottleneck for plant operations. A larger size pipeline not only increases the plant cost but also creates operational problems. One should bear in mind that the larger pipeline size than necessary increases plant cost due to pipelines along with the connected valves, fittings, supporting structures, etc. Likewise, the smaller pipe size may consume more energy for fluid movement.

Many factors should be kept in mind before sizing any pipeline. The basic principle of pipeline size is based on the available pressure drop between its two ends. Normally to maintain certain fluid velocity (may be from the available thumb rules), for clear water at pump discharge for the maximum possible fluid flow rate through that pipeline, cross-sectional area (or diameter) of pipeline is calculated. Based on this, the nearest commercially available pipeline size (of inside diameter (DN) closely matching with the calculated value) is selected for application. With these preliminary calculations of pipeline sizing and pipe routing, pressure drop between start and end point, incorporating all fittings, is calculated. Decision of the selection of higher or lower pipeline size is made according to the available pressure drop versus calculated pressure drop.

According to MoWR Design Manual on Pumping Facilities, 2001 velocity of flow is to be about 1.5m/s in suction and 2.0 m/sec in delivery pipe. This velocity limit depends on existing site conditions and scheme type. The velocity both in suction and delivery pipe is usually not less than 1.5m/s and not more than 3.5m/s for the reasons indicated above. The design engineer shall fix the flow velocity based on loss factors, material specification, cost and future operation and maintenance considerations.

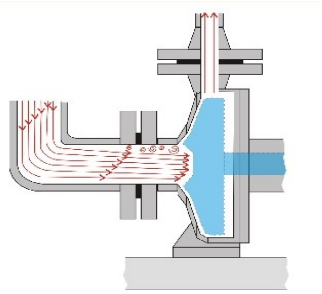
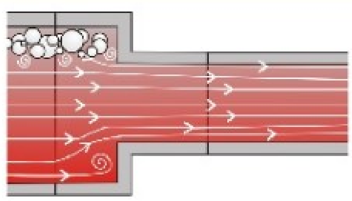
Flow velocity is expressed as:

$$\mathbf{V} = \frac{\mathbf{4Q}}{\boldsymbol{\pi}\mathbf{D^2}} \qquad \text{Equation 17}$$

Where:
V= flow velocity in m/s
Q= Volume of flow in m^3/s
D= Pipe Diameter (m)

8.2 CONDITIONS TO BE ACCOUNTED IN THE SUCTION LINE

Some of the conditions to be avoided in the suction side are cavitation, sudden contractions, and elbow too close to the pump, suspended loads and any air entry / leakage in to the suction pipe. Some of these are illustrated by the following figures. Cavitation is mostly related to the NPSHa which is critical element for smooth operation of the pump.


a. Elbow too close to the pump **b. sudden contractions** **c. Cavitation**

d. Effect of abrasion due to suspended load

Figure 8-1: Conditions to be avoided in suction side

Cavitation can have a serious negative impact on pump operation and lifespan. It can affect many aspects of a pump, but it is often the pump impeller that is most severely impacted. A relatively new impeller that has suffered from cavitation typically looks like it has been in use for many years; the impeller material may be eroded and it can be damaged beyond repair.

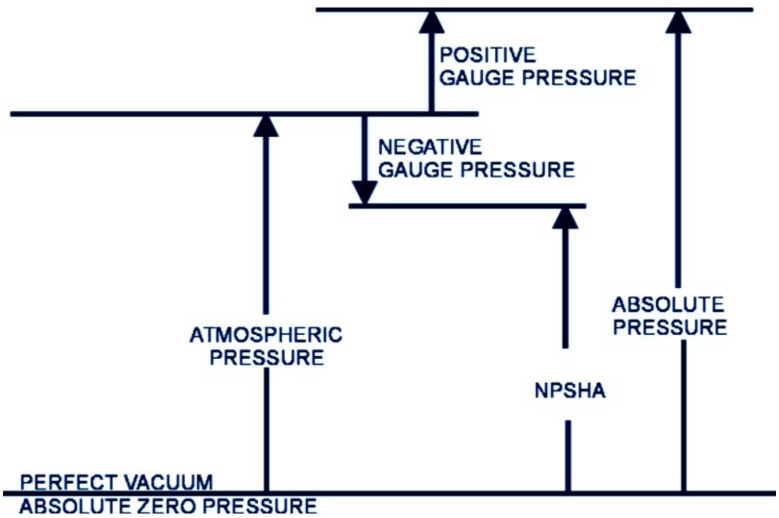
Cavitation occurs when the liquid in a pump turns to a vapor at low pressure. It occurs because there is not enough pressure at the suction end of the pump, or insufficient Net Positive Suction Head available (NPSHa). When cavitation takes place, air bubbles are created at low pressure. As the liquid passes from the suction side of the impeller to the delivery side, the bubbles implode. This creates a shockwave that hits the impeller and creates pump vibration and mechanical damage, possibly leading to complete failure of the pump at some stage.

How to avoid cavitation

Assuming no changes to the suction conditions or liquid properties during operation, cavitation can be avoided most easily during the design stage. The key is to understand Net Positive Suction Head (NPSH) and take it into account throughout the design process.

Net Positive Suction Head Available (NPSHa)

Net Positive Suction Head available (NPSHa) has nothing to do with the pump; it is a system value specific to the system design being considered. NPSHa is the head available at the pump suction flange pipework connection for the pumping system in question, and is completely independent of the pump to be installed there. It is the actual difference between the pressure at the pump inlet flange and the vapor pressure of the liquid for the installation and is determined by the design, configuration and relative levels for the suction side of a particular system.



Source: FAO irrigation Manual Module-5, 2001

Figure 8-2: Schematic presentation of NPSHa

NPSHa = P pump inlet – Vapor pressure (m)

Pressure available at the pump inlet is that which remains after allowances have been made for all the losses as described above.

Properties of water at different temperatures:

By regulating the pressure at which water is subjected its vapor pressure can be changed and it will eventually boil at room temperature. Table 8-1 illustrates a tabulation of temperatures from 5 to 100^{o}C with variations in vapor pressure, density and specific gravity for water.

At 5ºC, the density is 999.9, which an engineer will round to 1000. The specific gravity is 0.9999, or 1. At the bottom of the table one can see that at 100ºC the density has changed to approximately 958, a significant but not a big change. If one looks at the specific gravity, it has changed from 1 to approximately 0.96.

Table 8-1: Properties of water at various temperatures

Temperature ^{o}C	Vapor pressure N/m^2	Density Kg/m^3	Specific gravity
5	871.9	999.9	0.9999
10	1227	999.7	0.9997
15	1704	999	0.9990
20	2337	998.2	0.9982
25	3166	997	0.9970
30	4242	995.6	0.9956
35	5622	994	0.9940
40	7375	992.2	0.9922
45	9582	990.2	0.9902
50	12330	988.1	0.9881
55	15740	985.2	0.9852
60	19920	983.3	0.9833
65	25010	980.4	0.9804
70	31160	977.5	0.9775
75	38550	974.7	0.9747
80	47360	971.8	0.9718
85	57800	969	0.9690
90	70110	965.3	0.9653
95	84530	961.5	0.9615
100	101325	957.9	0.9579

Note: 1 N/m^2= 0.000102 meter of head=1pascal

Vapor pressure (H_{vap}):

At a specific combination of pressure and temperature, which is different for different liquids, the liquid molecules turn to vapor. An everyday example is a pot of water on the kitchen stove. When boiled to 100° Celsius, atmospheric pressure bubbles form on the bottom of the pan and steam rises.

This indicates vapor pressure and temperature have been reached and the water will begin boiling. Vapor pressure is defined as the pressure at which liquid molecules will turn into vapor. It should be noted that the vapor pressure for all liquids varies with temperature. It is also important to understand that vapor pressure and temperature are linked. A half full bottle of water subjected to a partial vacuum will begin to boil without the addition of any heat what so ever.

The influence of atmospheric pressure (H_{atm}):

Atmospheric pressure, sometimes also called barometric pressure, is the pressure within the atmosphere of Earth (or that of another planet). In most circumstances atmospheric pressure is closely approximated by the hydrostatic pressure caused by the weight of air above the measurement point. The pressure exerted by the weight of the atmosphere, which at sea level has a mean value of 101.3 Kilo Pascal or 1.013bar or about **10.13m** of water.

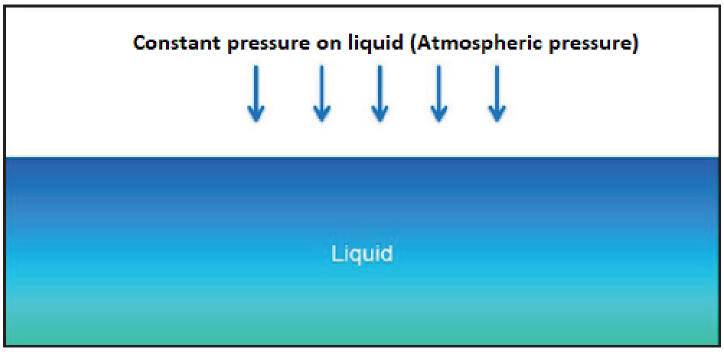


Figure 8-3: Atmospheric pressure on liquid surface

Absolute pressure (H_a):

Absolute pressure is zero-referenced against a perfect vacuum, so it is equal to gauge pressure plus atmospheric pressure. Gauge pressure is zero-referenced against ambient air pressure, so it is equal to absolute pressure minus atmospheric pressure. Negative signs are usually omitted. The altitude in which the pump installed can affect the pump operation. Mainly we use altitude in calculation of NPSH which is related directly to the total head of the pump. Let us see with simple example that the pump is to be installed at an altitude of 1800 masl. Ha is atmospheric pressure at the water surface it is equal to 10.13 meter of water column at 15ºc when measured at the mean sea level. Ha reduces 0.6m per 500m altitude rise.

$Ha=H_{atm}$-(Altitude*0.60)/500 Equation 18

Thus $Ha = 10.13 - \frac{1800*0.6}{500} = 7.97$ m is the absolute pressure head due to altitude difference.

There are a number of factors which must be considered collectively in order to obtain a complete picture of conditions prevailing at the suction of a pump. Calculation of NPSH involves consideration of fundamental fact that every liquid has a vapor pressure which is a function of the liquid and its temperature. Furthermore, if the pressure acting on a liquid is less than its vapor pressure the liquid will boil.

In any pumping system, there is always an absolute pressure available at the suction source. This pressure is reduced in the suction line due to such factors as static elevation, friction and turbulence losses, and energy expended in accelerating the liquid. Finally, there is a pressure drop within the pump itself caused by an increased velocity at the entrance to the impeller and shock losses in the impeller eye.

In order to prevent the liquid boiling, the suction pressure at the pump suction branch must be at least equal to the vapor pressure of the liquid at pumping temperature plus a margin to overcome losses in the pump. This margin of head available above the vapor pressure of the liquid is the net positive suction head, defined as follows.

Net positive suction head (**NPSH**): total head at the pump suction part over and above the vapor pressure of the liquid being pumped.

NPSH required (**NPSHR**): is a function of the pump design and is the lowest value of NPSH at which the pump can be guaranteed to operate without significant cavitation. There is no absolute criterion for determining what this minimum allowable NPSH should be, but pump manufacturers normally select an arbitrary drop in total dynamic head (differential head) of 3% as the normal value for determining NPSHR.

NPSH available (**NPSHA**): is a function of the system in which the pump operates and is equal to the absolute pressure head on the liquid surface (Ha) plus/ minus the static liquid level (Hs) is below/ above the pump center line (positive if pump axis is below liquid surface or negative if pump axis is above liquid surface) minus the absolute liquid vapor pressure head at pumping temperature (Hvap) minus the suction friction head losses including entrance losses (Hf) minus safety factor (Fs) taken about 0.60m. The **velocity head (Hv)** is usually ignored as it is too small. Thus,

$$\mathbf{NPSHA = Ha \pm Hs - Hvap - Hf - Fs} \qquad \text{Equation 19}$$

For successful operation **NPSHA must be greater than NPSHR**. Ensuring adequate available NPSH Pump suppliers set the NPSH required (NPSHR) for a given pump. The NPSHR takes into account any potential head losses that might occur between the pump's suction nozzle and impeller, thus ensuring that the fluid does not drop below its vapor pressure. Thus NPSHA must exceed the NPSHR set by the supplier.

There are a few options available to increase NPSHA, should it be at or below the NPSHR. Increasing the source pressure or reducing the fluid vapor pressure (by cooling) is rarely feasible. Therefore, there are two process variables remaining that can be adjusted that are the static head and friction losses.

Static head can be adjusted by three methods:

i. Raise the elevation of the source point. This may prove impossible in some cases (e.g., a tank that is set at grade).
ii. Lower the elevation of the pump inlet. This is a less appealing option because pumps are typically located just above ground level, and lowering the inlet may require the suction nozzle to be below grade. This usually results in a much more expensive pump.
iii. Raise the level of fluid in the suction vessel. The acceptability of this approach varies from company to company and should not be used without first consulting company procedures.

Friction losses can be reduced either by increasing the diameter of the pump suction-piping and/or reducing the equivalent length of the suction line. In a grassroots plant, friction losses should already be minimal, so raising static head is more viable. Reducing friction loss is usually more appealing for suction lines in existing plants where throughput has been increased above the original nameplate capacity.

There are also a few options to reduce the NPSHR of the pump, which include using a larger, slower-speed pump, a double-suction impeller, a larger impeller inlet (eye) area, an oversized pump and an inducer, which is a secondary impeller placed ahead of the primary impeller.

Example: Calculate the NPSH available for pump working at 25°C at an altitude of 2000m a.s.l. Pump center line is 2.5m above water surface. The suction pipe is steel pipe of diameter 200mm sucking sediment free water discharging 60l/s to elevated tanker. Suction pipe length is 6m and loss coefficient for entrance, strainer and foot valve is 0.50, 0.95 and 0.80 respectively.

Here NPSHa= Ha - Hs - Hvap – Hf -Fs
Where Ha=absolute pressure head at liquid surface (m)
Ha is dependent on altitude: Ha=10.13- (2000*0.6)/500=7.73m
Hs= the static Liquid level is below pump center line (here it is minus)
Hs= -2.5m
Hvap= absolute vapor pressure at working temperature
Hvap=0.3166m at 25°C (from table 7-3)
Hf= Head loss due to friction including loss at entrance
Hv= velocity head ($v^2/2g$)
Fs= Factor of Safety=0.60m
Friction loss in pipe is given by Hazen William's formula given as:

$$hf = (L) * [\frac{10.67*(Q)^{1.852}}{(C)^{1.852}(D)^{4.8704}}]$$

Where,
hf=loss due to friction (m)
L= pipe length (m)=6m
C= pipe friction factor given under table 7-2=120
Q= flow in pipe (m^3/s)=60l/s=0.06m3/s
D= pipe diameter in m=200mm=0.20m
Thus hf=0.125m but entrance and fitting loss (minor loss) must be included
Minor loss= $K_{loss}*v^2/2g$
V is flow velocity
V=Q/A
Where A= suction flow area =$\pi D^2/4=0.0314m^2$
Thus V=$0.06m^3/s/0.0314m^2$=1.91m/s
Minor loss= (0.50+0.95+0.80)*[1.91^2/(2*9.81)]=0.418m
Hf=0.125m+0.418m=0.543m
Hv=$V^2/2g$=0.186m
Thus NPSHa=7.73m-2.5m-0.3166m-0.543m-0.60m=3.77m
NPSHa computed must be greater than NPSHr provided from manufacturer (supplier) usually provided as pump performance curve.

8.3 THE SUCTION SUMP

The minimum submergence (S) required in suction sump to prevent strong air core vortices is based on a dimensionless flow parameter called the Froude number, defined as:

$$F_D = \frac{V}{(gD)^{0.5}}$$ Equation 20

F_D= Froude number (dimensionless)
V= Velocity at the suction inlet (m/s) = Flow/Area, based on D
D= Diameter of bell or pipe inlet (m)
g= gravitational acceleration (m/s^2)
The minimum submergence in meter unit, S, shall be calculated from (Hecker, G.E., 1987)

$$S = D(1 + 2.3F_D)$$ Equation 21

Example: What must be the minimum submergence in suction sump below pool level for suction pipe of 250mm sucking 90 l/s flow?

Solution:
Pipe diameter D= 0.25m
Flow= 90l/s= 0.09m3/s
Area of flow =0.04906m2, thus V=Flow/Area=1.83m/s
Froude number=1.17
Hence minimum submergence, S=0.92m

Sump sizing and dimensions to be adopted is as indicated on Figure 8-4, however minimum clearance from bottom slab depends on silt conditions, if silt movement under the bed is expected during pump operation minimum clearance from floor must be 0.60m.

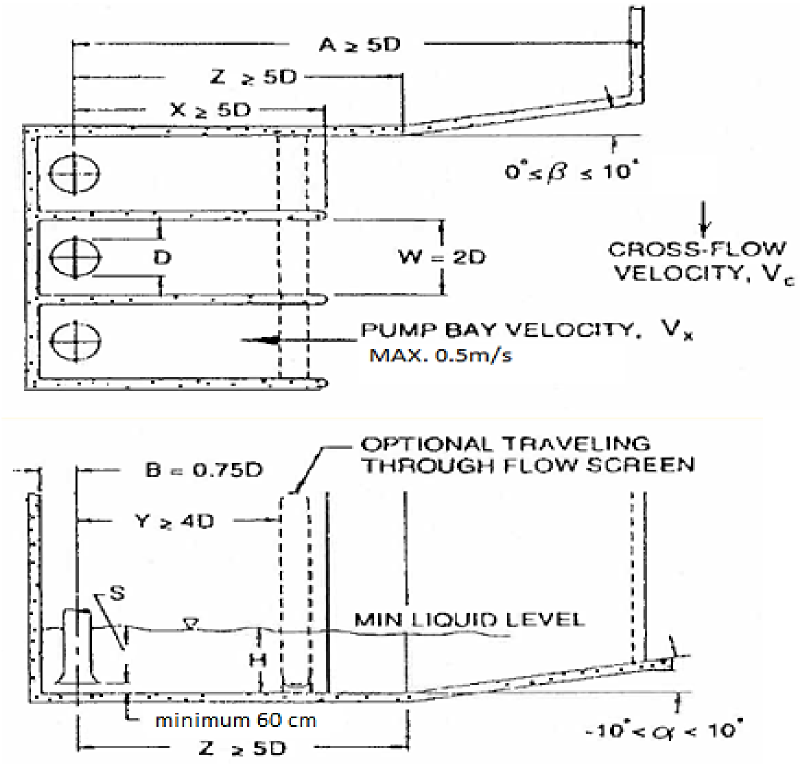


Figure 8-4: Typical suction sump arrangement

Most pump problems are due to suction issues. The instance of a pump problem that was related to the discharge is rare. There are five rules to be followed to avoid pump problems.

Rule-1 Provide sufficient NPSH
Rule-2 Reduce friction losses
Rule-3 Avoid sharp elbows in the suction side
Rule-4 Stop entering air or vapor to suction side
Rule-5 Correct piping alignment

The suction side of the pump is much more important than the piping on the discharge. If any mistakes are made on the discharge side, they can usually be compensated, by increasing the performance capability of the chosen pump. Problems on the suction side, however, can be the source of ongoing and expensive difficulties, which may never be traced back to rules 1 to 5.

8.4 CONDITIONS TO BE ACCOUNTED IN THE DELIVERY PIPE

Too excess loss in the delivery line must be avoided as much as possible. Minimizing losses in the delivery line means minimizing energy. When energy is minimized initial costs consequently reduce and as a result operation and maintenance cost is also reduced. Optimum delivery pipe size has to be systematically decided. Other factors that increase the loss in the delivery pipe are alignments, sharp bends, sharp confluences and the like. In all cases the delivery pipe must withstand the total working pressure as well as the water hammer.

For pumps in parallel the delivery lines (sub-mains lines) may be merged to one delivery line (main line). In any case the angle of confluence of the sub-main lines to main line must not be more than 30^o to minimize confluence losses. In such cases the losses in main and sub-main line shall be calculated individually.

Pipe nominal pressure (PN) must be fixed based on the maximum expected head/pressure in the delivery pipe multiplied by safety factor of 1.3. For instance if total pumping head is 36m or 3.6 bar and if water hammer expected due to sudden shut off is 60m or 6bar then the pipe to be selected is minimum PN7.8 but to nearest standards PN10 is selected. It should be noted that as any pipe gets older nominal specification by manufactured is not expected thus one has to wisely consider the safest value. Anchor blocks are also essential to absorb any shocks particularly for un-buried pipe line. Special consideration/ selection have to be made if the fluid or the soil in contact to the pipe is corrosive. Selected bedding material for delivery pipe line is also essential.

8.5 THE DELIVERY POOL OR BOX

The sizing of the delivery box can be fixed based on the discharge and velocity of flow. For free discharge pipe, velocity of the projectile jet of water at delivery outlet decides the length of the box. If incoming flow and outgoing flow are same delivery box width can be simply half of the length of delivery box obtained. If the delivery lines are more than one the width can be multiplied by the number of delivery pipes. If the discharge velocity is less than 3.5m/s the jet throw length is not significant and thus in that case practical dimensions can be adopted according to the velocity and discharge of specific case. The floor of the delivery box shall be concrete of thickness not less than 20cm to avoid scouring effects and minimum sill of 0.30m shall be provided in the delivery box. The side walls shall be concrete structure of minimum 30cm wall thickness or masonry wall of minimum thickness 40cm well plastered in rich cement mortar to inner face and well pointed to outer face.

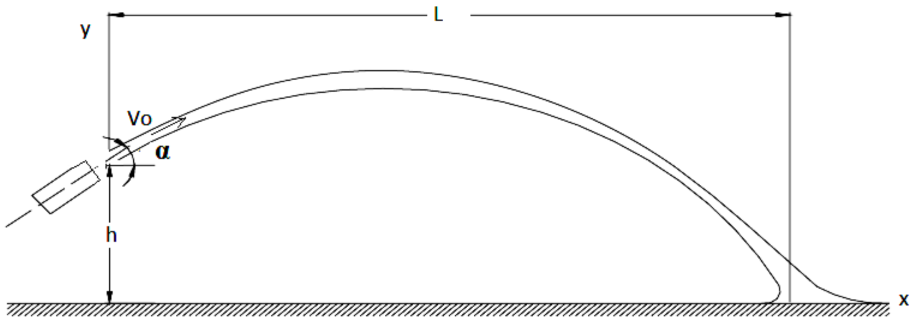


Figure 8-5: Water jet profile

The velocity of the jet (V_o) has two components expressed as:

$(V_o)x=V_o*\text{Cos}\ \alpha$ Equation 22

$(V_o)y=V_o*\text{Sin}\ \alpha$ Equation 23

There for the two coordinate equations for the trajectory are:

$L= (V_o*\text{Cos}\ \alpha)*t$ Equation 24

$h=(V_o*\text{Sin}\ \alpha)*t - ½ *gt^2$ Equation 25

t is time elapsed in the trajectory after liquid leaves the discharge pipe in seconds

By eliminating t and solving for V_o from the two equation we obtain:

$$V_o = \frac{L}{Cos\alpha}\left[\frac{g}{2(L\ tan\alpha - h)}\right]^{\frac{1}{2}}$$ Equation 26

Where,

L = Distance of throw, m

h= height above floor, m

V_o = Initial velocity or velocity at exit, m /s

α = Angle of elevation, degree

g = acceleration due to gravity, m /s^2

Example: What is the throw length (L) for delivery pipe discharging at 5m/s velocity fixed to 30^o and placed 30cm above water surface?

h= height above floor, 0.30m

V_o = Initial velocity or velocity at exit, 5 m /s

α = Angle of elevation, 30 degree

Thus inserting the values in above equation throw length would be:

$$5 = \frac{L}{cos30}\left[\frac{9.81}{2(L\ tan30 - 0.3)}\right]^{\frac{1}{2}}$$

$$5 = \frac{L}{0.866}\left[\frac{9.81}{2(0.577L - 0.3)}\right]^{\frac{1}{2}}$$

And solving for L,

Throw length is 1.37m

Assuming the water jet to fall at center of the box, then box length is 2L=2.74m, and with the consideration that incoming discharge to delivery box and outgoing from the box to be same width of box can be taken 1.37m.

9 SITING AND INSTALLATION OF PUMPS

9.1 SITING OF THE PUMPING STATION

General site selection criteria for pumping facility are indicated under section-4-2. Here specific considerations for siting of pumping station are discussed. Careful selection of a suitable location for a pumping station is very important in irrigation development. Several factors have to be taken into consideration when choosing the site.

Firstly, one has to find out whether the flow is reliable in the case of a river or whether the amount of water stored in the dam is enough to fulfill the annual irrigation requirements for the proposed cropping program. This information is often obtained from the water authority/historical data, physical observations and measurements or from the local farmers' experiences.

Secondly, in the case of river abstraction one has to check the maximum flood level of the river and preferably site the pumping station outside the flood level. With the limitations often imposed by the length of the suction pipe necessary to cater for the net positive suction head, where there are fluctuating flood levels, a portable pumping station is preferable. Such a site, however, should be on stable soil and have enough of water depth for the suction pipe. For permanent pumping stations pumps are installed on concrete plinth or foundation, the size of which varies in relation to the size of the pumping unit. Figure 9-1 shows a typical plinth and its reinforcement for pumps up to 50 kW. The Hydraulic Institute recommends in its Standards that the mass of the concrete foundation should be on the order of five (5) times that of the equipment it is supporting.

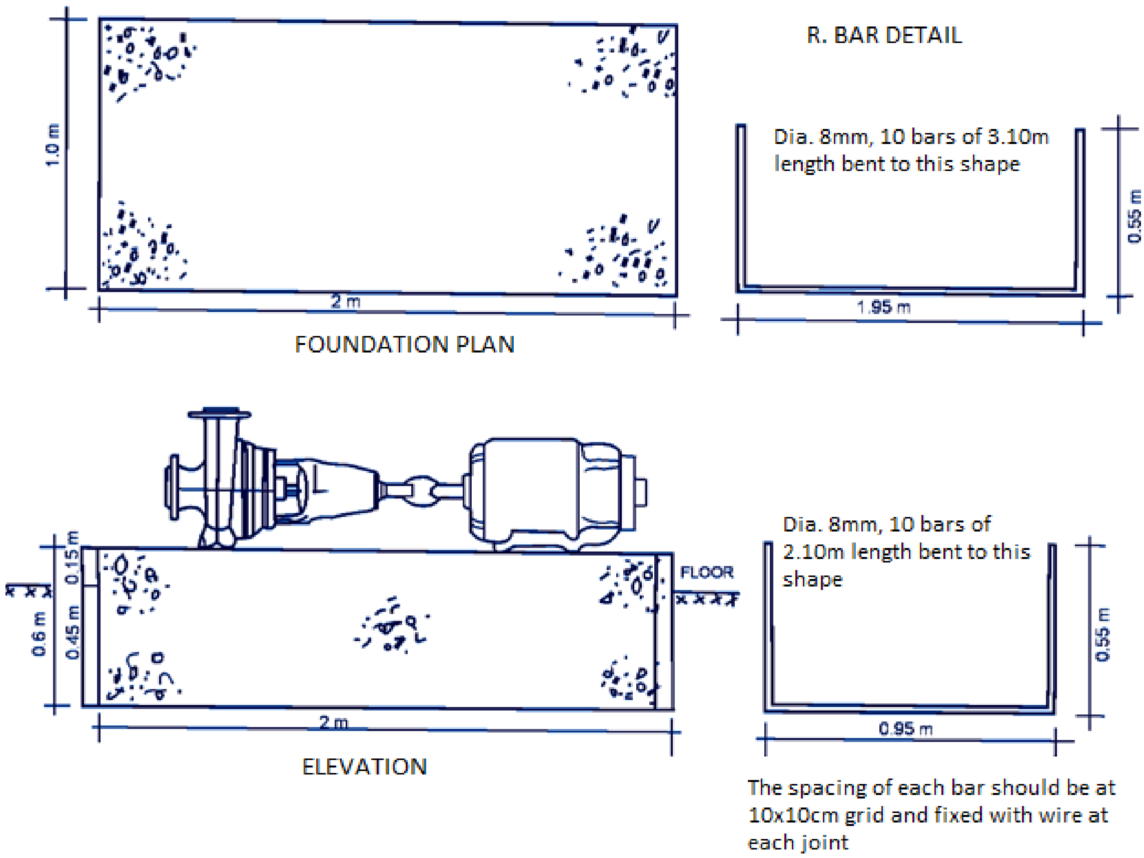


Figure 9-1: Foundation of pumping unit and reinforcement details

Thirdly, the abstraction point should not be sited in a river bend where sand and silt deposition may be predominant. Otherwise, the sand would clog both the suction pipe and pump. Where the river is heavily silted, a sand abstraction system can be developed.

Fourthly, where water is to be pumped from a dam or weir, the site should be outside the full supply level in case of upstream abstraction. In the case of downstream abstraction, the site should neither be too close to nor in line with the spillway.

Finally, as a rule, before a final decision is made on the location of the pumping station, a site visit has to take place to verify the acceptability of the site, taking into consideration the above requirements. It is generally helpful to talk to the local people to get information on the site.
The cost of a pumping station will have to be divided into investment costs, costs of operation and costs of maintenance and repair. These costs will have to be carefully estimated during the various stages of the design process in order to make comparisons for the different options more meaningful.

9.2 INSTALLATION OF PUMP

When the correct type of pump has been selected it must be installed properly to give satisfactory service and be reasonably trouble-free. Pumps are usually installed with the shaft horizontal, occasionally with the shaft vertical (as in wells).

9.3 COUPLING

Pumps are usually shipped already mounted, and it is usually unnecessary to remove either the pump or the driving unit from the base plate. The unit should be placed above the foundation and supported by short strips of steel plate and wedges. A spirit level should be used to ensure a perfect leveling. Leveling is a prerequisite for accurate alignment.

To check the alignment of the pump and drive shafts, place a straightedge across the top and side of the coupling, checking the faces of the coupling halves for parallelism. The clearance between the faces of the couplings should be such that they cannot touch, rub or exert a force on either the pump or the driver.

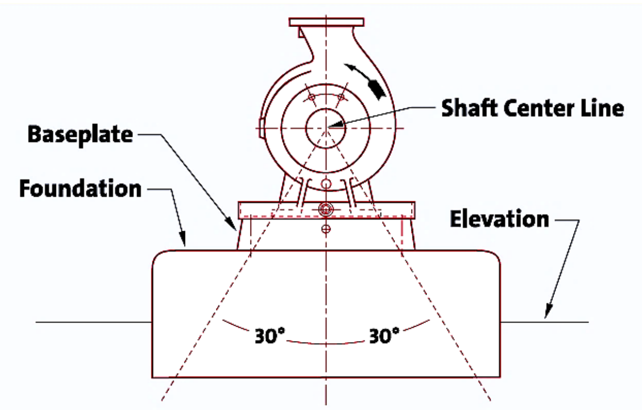


Figure 9-2: Foundation and pump arrangement

9.4 GROUTING

The grouting process involves pouring a mixture of cement, sand and water into the voids of stone, brick, or concrete work, either to provide a solid bearing or to fasten anchor bolts. A wooden form is built around the outside of the bedplate to contain the grout and provide sufficient head for ensuring flow of mixture beneath the only bed plate. The grout should be allowed to set for 48 hours; then the hold-down bolts should be tightened and the coupling halves rechecked.

9.5 PIPE LINES

The suction pipe should be flushed out with clear water before connection, to ensure that it is free of materials that might later clog the pump. The diameter of the suction pipe should not be smaller than the inlet opening of the pump and it should be as short and direct as possible. If a long suction pipe cannot be avoided, then the diameter should be increased. Air pockets and high spots in a suction pipe cause trouble. After installation is completed, the suction pipe should be blanked off and tested hydrostatically for air leaks before the pump is operated. A strainer should be placed at the end of the inlet pipe to prevent clogging. Ideally the strainer should be at least four times as wide as the suction pipe. A foot valve may be installed for convenience in priming. The size of the foot valve should be such that frictional loses are very minimal.

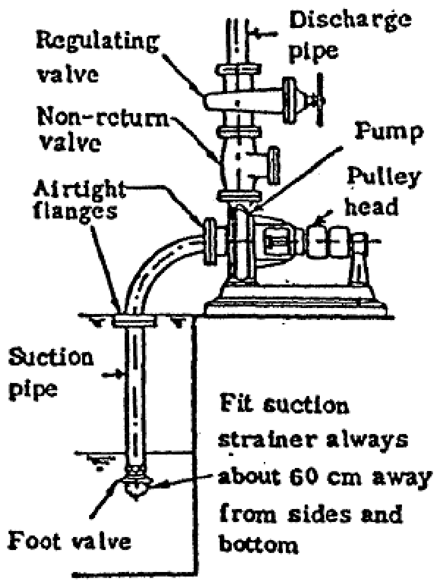


Figure 9-3 Correct arrangement of suction piping and valves on delivery side

Suction pipes must therefore be laid with proper care keeping the following in mind:

- Suction pipes should preferably be as short and as direct as possible.
- When use of long suction line becomes unavoidable it is recommended that the diameter of the pipe be increased by one or two commercial size larger than the diameter of the pump inlet. This will help in containing the frictional loss within the permissible limit.
- When larger diameter of pipe is used, it is essential to install an eccentric reducer between the suction inlet of the pump and the suction pipe. The eccentric side of the reducer is kept facing down thus eliminating any chance of air being trapped in the upper half of the reducer.
- Gate valve should not be placed in the suction line. Should it be placed due to some special reason, the valve handle should point horizontals or downward to prevent

formation of air pocket at the upper half of the valve. Suction pipe should never be throttled when the pump is in operation.
- All joints in the suction line have necessarily to be straight and leak proof and leak proof.
- Foot valve should be of standard design. It should have two times entry area and four times screen area compared to the diameter of the suction pipe and shall be located at the bottom of the suction pipe with a minimum submergence of 60 cm below the pumping water level.
- The end of the suction pipe should be submerged into water minimum by 4 times the pipe diameter.
- Total suction lift without friction should normally be kept within 4.5-6.0 meters.
- The suction pipe should be laid with a uniform rise towards the pump. There should not be any high spot where air pockets are likely to form.
- When use of bends in the suction line become essential, only wide angle (long radius) bend should be used. Should any obstruction need to be crossed; bends in the suction line should cross from under such obstructions so as to prevent formation of air pocket in the line.
- The size of the suction pipe shall be selected in such a way that the velocity of water on the suction side doesn't exceed 3.5 meters per second (as suction above this rate may produce cavitation).
- There shall be ample opening on strainers (below the foot values). The combined area of strainers openings should be about three to four times the area of cross- section of the suction pipe. In order to prevent the strainer from being choked up by mud and other material accumulated at the bottom of the well, it shall be placed at least 0.6m above the bottom of the well.

Like the suction pipe, the discharge pipe should be as short and free of elbows as possible, in order to reduce friction. A gate valve after a check valve should be placed at the pump outlet. The non-return valve prevents backflow from damaging the pump when the pumping action is stopped. The gate valve is used to gradually open the water supply from the pump after starting and to avoid overloading the motor. The same valve is also used to shut off the water supply before switching off the motor.

Delivery pipes must therefore be laid with proper care keeping the following in mind:

- Delivery pipes should be laid as straight as possible.
- A check valve and a gated valve should necessarily be used in the delivery line. The Check valve, which is placed between the pump and gets valve, prevents back flow of water into the pump in case pumping is stopped suddenly. Gate valve, which is placed after the chick valve, is used to control the flow in the delivery line.
- In a long delivery line air release valves are placed at high points. Similarly scour valves are placed at appropriate locations. .Air pipes or Air vents provided in gravity mains.
- A gate valve which should be placed in the lateral pipe, which takes off from the main delivery line. These gate valves control flow into the lateral lines.
- In long and overgrown delivery line where expansion or contraction of pipe may occur due to the variation in ambient temperature, expansion joints, loops, bends flexible steel hose etc. should be used to compensate such expansion or contraction.
- In addition to the pump which carries the normal discharge and use of bends, elbows tees and other fittings is kept to minimum to reduce head losses in the discharge line.
- Necessary supports at suitable intervals should be provided to firmly anchor the pipe in place.
- Provision of flow measuring devices viz. Pressure gauge, manometers, V-notch etc at the delivery end enable estimation of discharge readily. Calibrated pressure gauge may also be fitted in the delivery line to obtain working pressure.

- The use of reflux or non- return valve, fixed on the delivery side close to the pump, is necessary if distant pumping is required to avoid water hammering when pumping is stopped, sluice value can also be fitted immediately after the reflux value to regulate the discharge rate, for the purpose of maintaining controlled working conditions and also for easy disconnections of the pump for inspection and repairs without disturbing the delivery pipe line.

10 PUMP HYDRAULIC CHARACTERISTICS

When a hydraulic pump operates, it performs two functions. First, its mechanical action creates a vacuum at the pump inlet which allows atmospheric pressure to force liquid from the source into the inlet line to the pump. Second, its mechanical action delivers this liquid to the pump outlet and forces it into the hydraulic system.

A pump produces liquid movement or flow: it does not generate pressure. It produces the flow necessary for the development of pressure which is a function of resistance to fluid flow in the system. For example, the pressure of the fluid at the pump outlet is zero for a pump not connected to a system (load). Further, for a pump delivering into a system, the pressure will rise only to the level necessary to overcome the resistance of the load. Here the main focus is on hydraulic characteristics of pumps commonly used for irrigation.

Proper design of an irrigation system requires that the pumping system be precisely matched to the irrigation distribution system. Then the pressure and flow rate required can be efficiently provided by the pumping system.

When an irrigation system is designed or modified to use an existing pumping system, it is necessary to measure the capacity of the existing pump. The irrigation system can be properly designed only if the flow rate and pressure of the pump are accurately measured.

It is not adequate to visually estimate pump capacity or to use the manufacturer's specifications to determine current pump capacity. Visual estimates are normally not accurate, and manufacturer's specifications do not include the effects of site-specific factors such as well characteristics or suction and discharge pipe sizes. Manufacturer's specifications also do not include the effects of age and wear on pumping system performance.

Pump hydraulic characteristics to be considered in design of pumping facility are discussed below. After determination of all the hydraulic characteristics the designer goes for most suitable pump section for the specific irrigation system.

10.1 CAPACITY

Capacity means the flow rate with which liquid is moved or pushed by the pump to the desired point in the process. It is commonly measured in either gallons per minute (gpm) or cubic meters per hour (m3/hr). The capacity usually changes with the changes in operation of the process. The capacity depends on a number of factors like operation conditions and liquid characteristics i.e. density, viscosity, size of the pump and its inlet and outlet sections.

The capacity of a pump has two components, the pump discharge rate and the discharge pressure. The discharge rate is normally measured in gallons per minute (gpm) in English units or liters per second (lps) in metric units. Pressure is normally measured in pounds per square inch (psi) in English units or kilo Pascal (kPa) in metric units. It is necessary to measure both discharge rate and pressure under normal operating conditions in order to determine how the pumping system will operate as a part of an irrigation system.

10.1.1 Determination of discharge rate in irrigation

One of the basic parameters necessary to be determined in design of pumping facility for irrigation is the discharge rate. The discharge rate for irrigated agriculture means the total demand or supply for irrigation. This design discharge is influenced by the irrigation hours per day or indirectly the pump operation hours per day. The pumping hours per day is usually not more than 8 to 10 hours for engine driven systems. Thus for design purpose it must be referred to the maximum demand of irrigation indicated under the irrigation agronomy report to fix maximum capacity of the pump and stand-by capacity.

10.1.2 Standby capacity

Standby capacity is normally required as an insurance against pump failure as well as an alternative during maintenance operations. This is necessary where the pumping station is the sole source of supply to an area.

There will always be some standby capacity to take care for repairs, breakdowns etc. Generally 100% stand by capacity against average demand, and 33% to 50% standby capacity against the peak demand is considered sufficient and may, therefore, be provided at the pumping stations.

10.2 TOTAL PUMP HEAD

Total Pump Head or Total Dynamic Head (TDH) is the total equivalent height that a fluid is to be pumped, taking into account friction losses in the pipe.

TDH = Static suction lift+ Static delivery or discharge lift + Pipe friction Loss+ minor losses + any extra head required at discharge point.

In irrigation fields, thus the designer has to have topographic data to properly decide the pump location (position or axis), proper pipe line alignments, suction sump (in case of surface pumps) and delivery point. Thus to properly fix pump axis besides the topographic data, geologic formation of the pump site, minimum water level, maximum water level and sediment condition are so important.

Besides the above parameters the pump head is influenced by the sediment condition. If water is silt laden the operating head will definitely increase and this condition affects both the efficiency as well as the life of the pump and the motor.

After computation of total pump head the other most critical factor not to be ignored is the water hammer. The delivery line must withstand the total pump head and the water hammer.

Water Hammer:

Water hammer is the name given to the pressure surges caused by some relatively sudden changes in flow velocity. This can be caused by valve opening or closing, pump starting or stopping, cavitation or the collapse of air pockets in pipelines, filling empty pipelines or, of most concern to irrigation applications, a power outage, which suddenly shuts down all the electric pumps on the pipeline. When the velocity in the pipeline is suddenly reduced, the kinetic energy (velocity head) of the moving column of water is converted into potential energy (pressure head), Compressing the water and stretching the pipe walls. These disturbances then travel up and down the pipeline as water hammer waves. The reader has probably experienced the banging and rattling of household pipework resulting from opening and closing a tap too rapidly – this is water hammer.

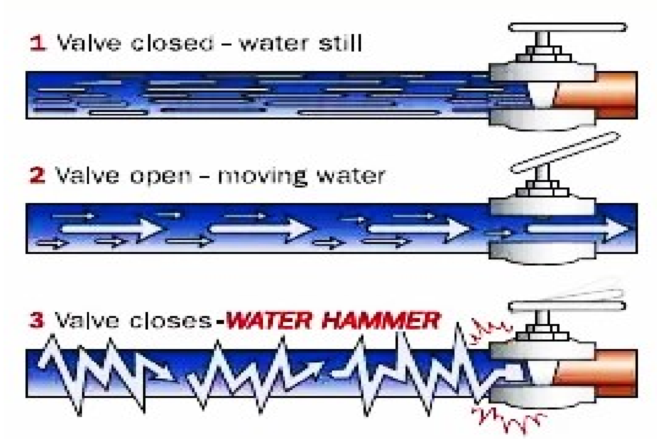


Figure 10-1: Water hammer due to sudden valve closure

The pressure surges may either be positive or negative, i.e. the pressure may either rise above or fall below the operating pressure (static pressure,) by an amount equal to the maximum surge pressure, or surge head. Events that cause water hammer include:

- Valve closure or opening (in full or in part),
- Pump speed change (trip or start up)
- Relief valve cracking open
- Rapid tank pressurization
- Periodic pressure or flow conditions

When the velocity of flow in a pipe is suddenly reduced, kinetic energy is converted into potential energy as the pressure increases, the liquid is compressed and the pipe wall is stretched. The disturbances so produced travel up and down the pipeline as water hammer waves. The changes in velocity can be caused by a wide range of disturbances such as valve operation, pump operation, and so on. The common circumstances which give rise to water hammer of engineering significance are those associated within pumps; normal starting and stopping of pumps and stoppages caused by power failure. In the design of pipe systems, it is necessary to take into account the magnitudes of pressure surges associated with water hammer phenomena and, consequently, it is important that these water hammer effects be calculated with the appropriate accuracy.

The pressure change Δp_{Jou} in a fluid caused by an instantaneous change in flow velocity Δv is calculated by:

$$\Delta p_{Jou} = \rho . a . \Delta v \qquad \text{Equation 27}$$

Where:
Δv: Flow velocity change in m/s
ρ: Density of the fluid in kg/m^3
a: Wave propagation velocity through the fluid in the pipeline in m/s
Δp_{Jou}: pressure change in N/m^2

The Δp_{Jou} formula is referred to as the Joukowsky equation. The relationship only applies to the period of time in which the velocity change Δv is taking place. If Δv runs in opposite direction to the flow, the pressure will rise, otherwise it will fall. If the liquid pumped is water, i.e. $\boldsymbol{\rho}$ = 1000 kg/m^3, equation 27 will look like this:

$$\Delta hJou = \frac{a}{g} . \Delta V \approx 100. \Delta V$$ Equation 28

Where,

g: Acceleration due to gravity 9.81 m/s^2
Δh_{Jou}: Pressure head change in m.

The wave propagation velocity (a) is one of the elements of the Joukowsky equation and, therefore, a vital parameter for defining the intensity of a surge. It is calculated by solving equation 29.

$$a = \sqrt{\frac{1}{\frac{\rho}{Ef} + \frac{\rho . D\,(1-\mu^2)}{Ep.t}}}$$ Equation 29

Where,

$\boldsymbol{\rho}$: Density of the fluid in kg/m^3
Ef: Modulus of elasticity of the fluid in N/m^2
Ep: Modulus of elasticity of the pipe wall in N/m^2
D: Inside pipe diameter in mm
t: Pipe wall thickness in mm
μ: Transverse contraction number or poison's ratio=0.40 for PVC, 0.30 for steel pipe

Equation 29 produces a range of values from approximately 1400 m/s for steel pipes to around 300 m/s for ductile plastic pipes. Characteristics of the pipe, such as temperature, pipe material and the ratio of the diameter of the pipe to its wall thickness, affect the elastic properties of the pipe and will ultimately have an impact on the speed at which the shock waves travel up and down the pipe.

Table 10-1: Typical values of modulus of elasticity (Larock et al., 2000)

Material	Modulus of Elasticity, E in pa or N/m^2
steel	2.077×10^{11}
copper	1.10×10^{11}
bronze	1.00×10^{11}
Asbestos cement	2.30×10^{10}
Fiber glass reinforced	9.00×10^{9}
PVC	2.8×10^{9}
Polyethylene	8.00×10^{8}
Water (bulk)	2.2×10^{9}

It may seem inconsistent that Δp_{Jou} in the Joukowsky equation 28 seems to have nothing to do with the mass of the flow inside the pipeline. For example, if the water hammer had been based on a pipe diameter twice that of the diameter used, $A = \pi D^2/4$ would have caused the fluid mass and its kinetic energy to turn out four times as large. What seems to be a paradox is instantly resolved if one considers the force exerted on the shut-off valve, i.e. force $F = \Delta p \cdot A$, the defining parameter for the surge load. Because of A, it is now in actual fact four times as large as before. This shows that one must also consider the fluid mass to judge the risk of water hammer, although that does not seem necessary after a superficial glance at Joukowsky's equation.

At the same time, this explains why the pressure surges occurring in domestic piping systems with their small diameters and lengths are usually negligible. In these systems, the kinetic energy levels and fluid masses are very small. In addition, it is practically impossible to close a valve within the very short reflection time of a domestic water system.

Return time of wave

The time taken by the wave to go to peak and come back is given by the following equation:

$$T_r = \frac{2 \,.\, L}{a}$$ Equation 30

Where:

T_r	:	Return time of the wave (seconds)
L	:	length of the pipe (m)
a	:	speed of pressure wave (m/s)

Example: what is the time of return of the wave for water under pumping at rate of 100l/s in 800m long steel pipe of inner diameter 300mm and wall thickness 6mm if pump is suddenly stopped? Calculate time of return for case of PVC pipe having 16mm wall thickness under same operation conditions.

Solution:

L=800m

ρ: Density of the fluid in kg/m^3=1000kg/m^3

Ef: Modulus of elasticity of the fluid in N/m^2=2.2x10^9 N/m^2 (table-----)

Ep: Modulus of elasticity of the pipe wall in N/m^2=2.077x10^{11}N/m^2

D: Inside pipe diameter in mm=300mm

t: Pipe wall thickness in mm=6mm

µ: Transverse contraction number or poison's ratio=0.30 for steel pipe

Inserting the values in above formula: a=1219m/s

Thus Tr in steel pipe= (2*800)/1219=1.31 seconds

If the pipe material is PVC then µ=0.40, Ep=2.8x10^9, t=16mm thus a=406m/s.

Tr in PVC pipe is equal to 3.94seconds.

Effect of the closure time:

The closure time (T_c) is the time during which the flow varies. For example, in the case of a power cut, the pump will not stop instantaneously but will gradually reduce its flow depending on its inertia and water pressure. Similarly, valves need some time to be closed. The way the flow is reduced is usually not linear, for instance, gate valves have their flow reduced mainly during the last 20% of the closure run.

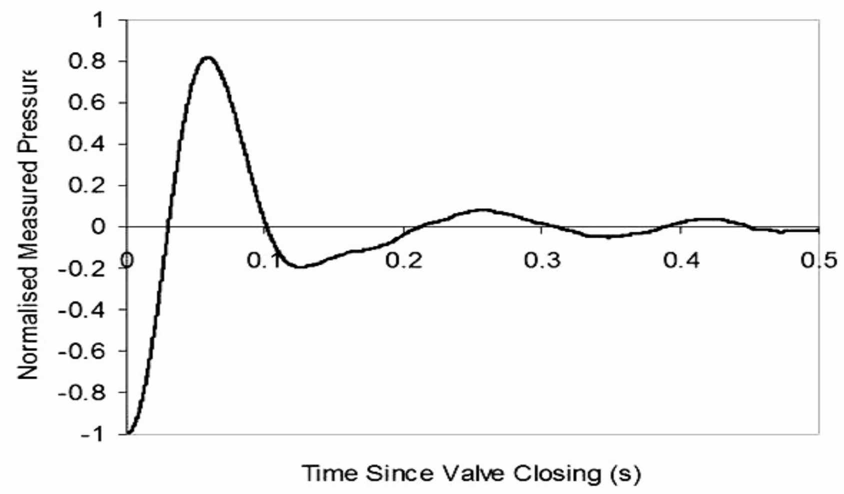


Figure 10-2: Typical pressure wave caused by sudden valve closure

The return time is an important parameter of water system, allowing setting closure time for valve or shutdown time for pump soft starters avoiding water hammer.

Water hammer must be considered during the design stage. But if it happens due to missed design considerations the solutions to avoid water hammer depend on the circumstances of each situation. They can include the following:

- Remove the cause of the hammer.

Some causes can be resolved by arranging for the elimination or control of the problem item. Apart from the items previously discussed, this might include vibrating pressure relief valves, fast emergency shutdown valve closures, and some manual valve closures eg butterfly valves.

- Reduce the pumping velocity.

This can be done using a larger pipe diameter or lower flow rate.

- Make the pipe stronger.

This can be expensive but might be a solution if the pipe specification is only slightly exceeded.

- Slow down valves, or use ones with better discharge characteristics in the pipe system.
- Use surge tanks

These allow liquid to leave or enter the pipe when water hammer occurs, and are normally only seen on water systems.

- Use pressure relief valves

These are not suitable with toxic materials unless a catch system is provided.

- Use air inlet valves

These are not suitable if ingress of air or other possible external materials is not permissible.

The Joukowsky equation can be used to calculate simple estimates of water hammer. Let's consider these examples.

Example-1:
In a DN 500 pipeline, L = 8000 m, a = 1000 m/s and v = 2 m/s, a gate valve is closed in 5 seconds. Calculate the pressure surge. Calculate the force exerted on the gate.

Solution:
5 second < Tr =(2xL/a) or 16 second, i.e. Joukowsky's equation may be applied. If the flow velocity is reduced from 2 m/s to zero as the valve is closed, Δv = 2 m/s. This gives us a pressure increase $\Delta h = 100 \cdot 2 = 200$ m or approximately $\Delta p = 20 \cdot 10^5$ N/m^2, which is 20 bar.
The valve cross-section measures $A = D^2 \cdot 0.25 \cdot \pi \approx 0.2$ m^2. The force acting on the gate is p·A = $0.2 \cdot 20 \cdot 10^5 = 4 \cdot 10^5$ N= 400 kN.

Example-2:
A pump delivers water at Q = 300 l/s and a head Δh = 40 m through a DN 400 discharge pipe measuring L = 5000 m into an overhead tank; a = 1000 m/s. The inertia moments of pump and motor are negligible. Is there a risk of liquid column separation, i.e. macro-cavitation, following pump trip? If so, what is the anticipated pressure increase?

Solution:
Q = 300 l/s in a DN 400 pipeline roughly corresponds to a flow velocity v = 2.4 m/s. As a result of pump trip and the loss of mass inertia moment, the pump comes to a sudden standstill, i.e. Δv = 2.4 m/s. According to the Joukowsky equation, this causes a head drop of $\Delta h = -100 \cdot 2.4$ m = -240 m. Since the steady-state head is just 40 m, vacuum is reached, the liquid column collapses and macro-cavitation sets in. Following the liquid column separation near the pump outlet, the two liquid columns will recombine with great impact after some time.

For reasons of energy conservation, the highest velocity of the backward flow cannot exceed the original velocity of the steady-state flow of 2.4 m/s.
Under the most unfavorable conditions, the cavitation-induced pressure rise will, therefore, be $\Delta h = 100 \cdot 2.4 = 240$ m, which is the equivalent to 24 bar.

10.3 POWER AND EFFICIENCY

Power in pumping system is the total energy needed to run the pump/s under given capacity and total dynamic head. Horsepower is the unit of power to define hydraulic or water horsepower. In International Systems (SI) it is expressed in kilowatts (KW). 1 horsepower = 746 watts or 0.746KW. The input power delivered by the motor to the pump is called brake horsepower (bHp). The difference between the brake horsepower and hydraulic power is the pump efficiency.

Water Horse Power (Theoretical hp):

$$P_w = \gamma Q H \qquad \text{Equation 31}$$

Where,
P_w= power in Kw
H is the total lift head in m,
Q is the pump discharge in m^3/s.
γ=unit weight of water which is approximately 10KN/m^3.

Shaft horse power, for given pump efficiency of ηp:

$P_s = P_w / \eta_p$ Equation 32

Brake horse power for given motor efficiency ηm.

$P_b = P_s / \eta_m$ Equation 33

The efficiency of electric motor, which in most cases ranges between 70-95%, is generally higher than that of diesel engine 65-80%. Therefore, an installation whether the prime mover is electric motor or diesel engine, the efficiency of the system selected should be as high as possible so that cost incurred is minimized. For design purpose efficiency of electric mover (η_m) can be taken 85% where as that of diesel engine 70% and the efficiency of the coupled centrifugal pump (η_p) 70% can be considered. Thus overall pump efficiency (η_o) of 60% for electrical pump and 49% for diesel pump can be adopted for design purpose. In all cases the performance curve of the manufacturer is to be referred to select most efficient pump for the specific site.

Table 10-2: Typical electric motor and pumping plant efficiencies by motor size

Electric motor (Kw)	Full load motor efficiency (η_m) in %	Matched pump efficiency (η_p) in %	Overall pump efficiency (η_o) in %
2-4	80 – 86	55 – 65	44 – 56
5-7.5	85 – 89	60 – 70	51 – 62
10-22	86 – 90	65 – 75	56 – 68
30-45	88 – 92	70 – 80	62 – 74
>55	90 – 93	75 – 85	68 – 79

Source: North Carolina Cooperative Extension Service, Publication Number: AG 452-6

Example: Calculate the brake horse power of centrifugal pump installed to discharge 80l/s silt free water to 30m total lift head. Consider if pump coupled to Diesel engine and electric motor.
Solution:
Unit weight of water=10KN/m^3
Discharge Q=80l/s=0.08m^3/s
Head=30m
Water Horse Power [in KW]= $P_w = \gamma QH$
=10*0.08*30=24KW
Shaft horse power, Consider pump efficiency η_p =70%
$P_s = P_w / \eta_p$ =24/0.7=34.3KW
Brake horse power,
For diesel engine η_m=70%
For electric motor η_m=85%
Thus brake horse power (KW)=34.3/0.7=49KW or **66hp** for pump coupled to diesel engine
brake horse power (KW)=34.3/0.85=40KW or **54hp** for pump coupled to electric motor.

Motor or engine de-rating

Motors or engines are designed fabricated for specific ambient temperature and altitude above sea level. For instance a motor can be designed and manufactured for 40°C ambient temperature and altitude 1000m a.s.l then manufacturer may give the following de-rating factor for other temperature and altitude indicated in Table 10-3. Similarly for diesel or petrol engines de-rating is given for temperature and humidity differences. In above example if the electric motor is working at 2000m a.s.l and temperature 45°C then de-rating factor by this manufacturer for power requirement is 0.9. Thus power required is 54hp/0.90=60hp.

Table 10-3 Sample ambient temperature and altitude reduction factor for power requirement

Altitude above sea level (m)	Ambient temperature (°C)						
	30°C	35°C	40°C	45°C	50°C	55°C	60°C
1000	1.06	1.03	1.00	0.96	0.92	0.87	0.82
1500	1.03	1.00	0.97	0.93	0.89	0.84	0.80
2000	1.00	0.97	0.94	0.90	0.86	0.82	0.77
2500	0.95	0.93	0.90	0.86	0.83	0.78	0.74
3000	0.91	0.89	0.86	0.83	0.79	0.75	0.71
3500	0.87	0.84	0.82	0.79	0.75	0.71	0.67
4000	0.82	0.79	0.77	0.74	0.71	0.67	0.63

In diesel engines according to Pair et al. (1983), de-rating is approximately 1% per 100 m increase in altitude and 1% per 5.6°C increase in air temperature from the published maximum output horsepower curve. On the top of that, an additional 5-10% for reserve should be deducted. If the continuous output curves are used, only the 5-10% deduction is applied.

Most engineers multiply the power requirement by a factor of 1.2 and use the engine continuous output rating curve. In other words, they de-rate an engine by 20%. The total de-rating on the continuous output curve is 10-15% for V-belt or gear (5-10% de-rating for continuous output and 5% for the transmission losses).

10.4 SPECIFIC SPEED

Specific Speed is a number characterizing the type of impeller in pump in a unique and coherent manner. Specific speed is determined independent of the pump size and can be useful when comparing different pump designs.

Low-specific speed radial flow impellers develop hydraulic head principally through centrifugal force. Pumps of higher specific speeds develop head partly by centrifugal force and partly by axial force. An axial flow or propeller pump with a specific speed of 10,000 or greater generates its head exclusively through axial forces. Radial impellers are generally low flow/high head designs whereas axial flow impellers are high flow/low head designs. In theory, the discharge of a "purely" centrifugal machine is tangential to the rotation of the impeller whereas a "purely" axial-flow machine's discharge will be parallel to the axis of rotation. There are also machines that exhibit a combination of both properties and are specifically referred to as "mixed-flow" machines.

Centrifugal pump impellers have specific speed values ranging from 500 to 10,000 (English units), with radial flow pumps at 500-4000, mixed flow at 2000-8000 and axial flow pumps at 7000-20,000. Values of specific speed less than 500 are associated with positive displacement pumps.

As the specific speed increases, the ratio of the impeller outlet diameter to the inlet or eye diameter decreases. This ratio becomes 1.0 for a true axial flow impeller.

$$\mathbf{N_s} = \frac{\mathbf{n}\sqrt{\mathbf{Q}}}{\mathbf{H}^{3/4}} \qquad \text{Equation 34}$$

Where:

Ns : is specific speed (dimensionless)
n : is pump rotational speed (rpm)
Q : is flow rate (l/s) at the point of best efficiency
H : is total head (m) per stage at the point of best efficiency

Note that the units used affect the specific speed value in the above equation and consistent units should be used for comparisons. Pump specific speed can be calculated using British gallons or using metric units (m^3/s or L/s and meters head).

Speed variation

In discussing pump characteristic curves, no mention of speed was made. A typical manufacturer's characteristic curve provides several TDH-Q, EFF-Q and BP-Q curves. This is because the same pump can operate at different speeds. A change in the impeller speed causes a shift of the Q-H characteristics in the diagram. It is a shift upwards and to the right with increasing speed and downwards and to the left when the speed is decreased. The BP required power also changes.

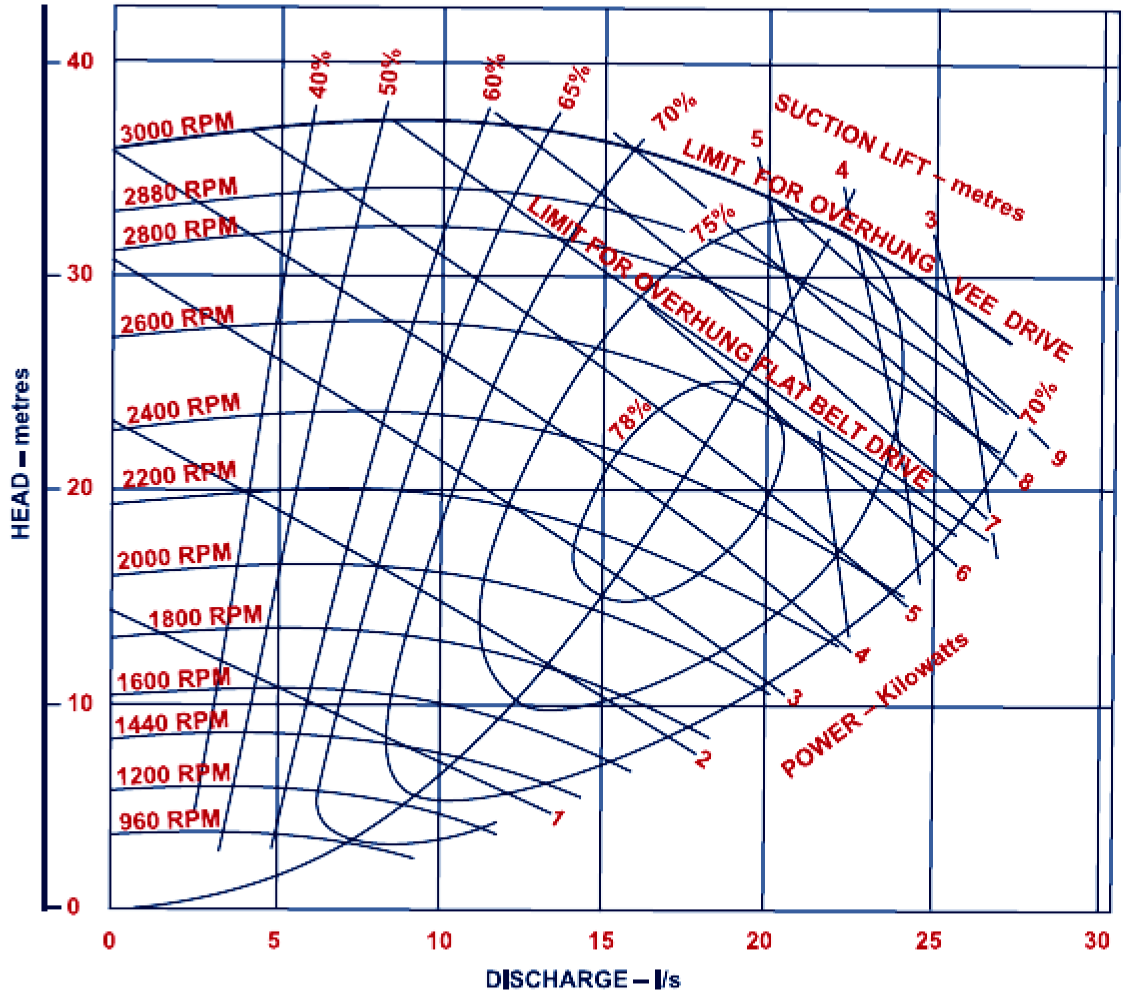


Figure 10-3: Pump characteristic curves (variable speed)

The relationship between speed, on the one hand, and discharge, head and power on the other is described by Euler's affinity laws in the Hydraulics Handbook of Colt Industries (1975) as follows:

The discharge Q varies in direct proportion to the speed:

$$\frac{Q1}{Q2} = \frac{N1}{N2} \qquad \text{Equation 35}$$

The head H varies directly with the square of the speed:

$$\frac{H1}{H2} = \frac{N1^2}{N2^2}$$ Equation 36

The break power BP varies approximately with the cube of the speed:

$$\frac{BP1}{BP2} = \frac{N1^3}{N2^3}$$ Equation 37

Where:
Q1 = discharge and H1 = head and BP1 = brake power at N1 speed in revolutions per minute (rpm)
Q2 = discharge and H2 = head and BP2 = brake power at N2 speed in revolutions per minute (rpm)

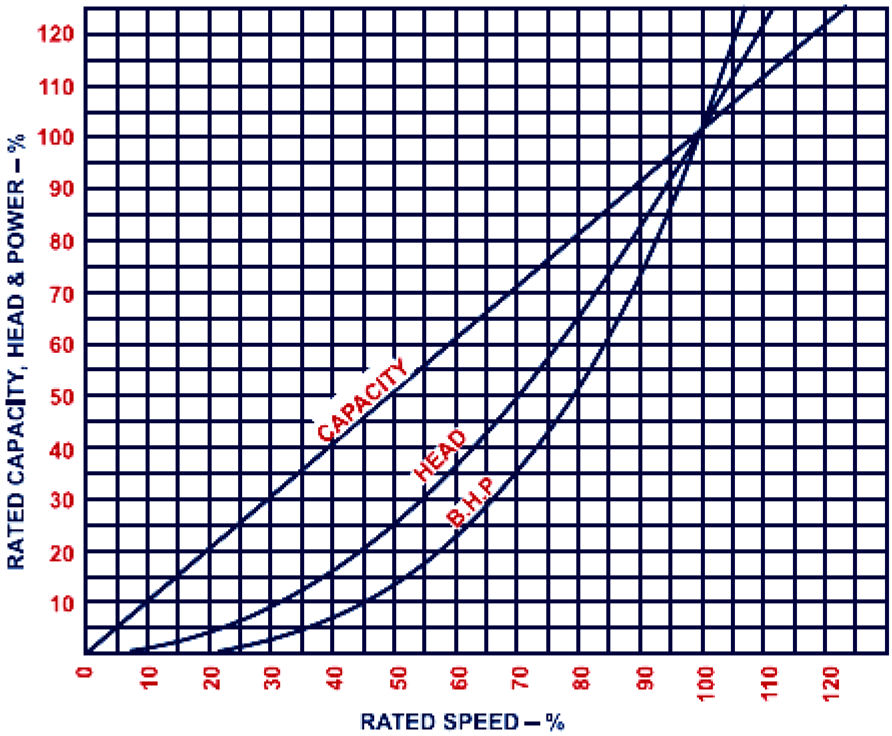


Figure 10-4: Effect of speed change on centrifugal pump performance

As a rule, most pump characteristic curves are presented with one speed only. Hence the need to use Euler's affinity laws in deriving performance at different speeds. Example below clarifies the process.

If the speed of the pump is changed from 1200 rpm to 2000 rpm, the discharge, head and brake power will change from 40 l/s to 66.7 l/s, 32 m to 88.9 m, and 16.8 to 77.7 kW respectively. However, the affinity laws make no reference as to how the pump efficiency is affected by speed changes. As a rule, pumps that are efficient at one speed would be efficient at other speeds.

Example:
If a pump delivers 40 l/s at a head of 32 m and runs at a speed of 1200 rpm, what would be the discharge and head at 2000 rpm? What would the brake power of the pump be if it were 16.78 kW at 1200 rpm?

$$\frac{Q1}{Q2} = \frac{N1}{N2} \rightarrow \frac{40}{Q2} = \frac{1200}{2000} \rightarrow Q = 66.7l/s$$

Using Equation 36 the new head would be:

$$\frac{H1}{H2} = \frac{N1^2}{N2^2} \rightarrow \frac{32}{H2} = \frac{1200^2}{2000^2} \rightarrow H2 = 88.90m$$

BP2 is calculated using Equation 37 as follows:

$$\frac{BP1}{BP2} = \frac{N1^3}{N2^3} \rightarrow \frac{16.78}{BP2} = \frac{1200^3}{2000^3} \rightarrow BP2 = 77.70KW$$

10.4.1 Affinity law

Pump Affinity Laws

Turbo machines affinity laws can be used to calculate volume capacity, head or power consumption in centrifugal pumps with changing speed or wheel diameters.

The Affinity Laws of centrifugal pumps or fans indicates the influence on volume capacity, head (pressure) and/or power consumption of a pump or fan due to change in speed of wheel - revolutions per minute (rpm).

Note that there are two sets of affinity laws:

- Affinity laws for a specific centrifugal pump - to approximate head, capacity and power curves for different motor speeds and /or different diameter of impellers.
- Affinity laws for a family of geometrically similar centrifugal pumps - to approximate head, capacity and power curves for different motor speeds and /or different diameter of impellers.

Pump affinity laws for a specific centrifugal pump:

Capacity

The capacity (discharge) of a centrifugal pump can be expressed like

$$\frac{q1}{q2} = \left(\frac{n1}{n2}\right)\left(\frac{d1}{d2}\right) \qquad \text{Equation 38}$$

Where:

q = volume flow capacity (m^3/s, gpm, cfm, ..)

n = wheel velocity - revolution per minute - (rpm)

d = wheel diameter (m, ft)

Head or Pressure

The head or pressure of a centrifugal pump can be expressed like

$$\frac{dp1}{dp2} = \left(\frac{n1}{n2}\right)^2\left(\frac{d1}{d2}\right)^2 \qquad \text{Equation 39}$$

Where,

dp = head or pressure (m, ft, Pa, psi, ..)

Power

The power consumption of a centrifugal pump can be expressed as

$$\frac{p1}{p2} = \left(\frac{n1}{n2}\right)^3 \left(\frac{d1}{d2}\right)^3$$ Equation 40

Where,

P = power (W, bhp,)

Changing Wheel Velocity

If the wheel diameter is constant - change in pump wheel velocity can simplify the affinity laws to Equations 38, 39 and 40 indicated above:

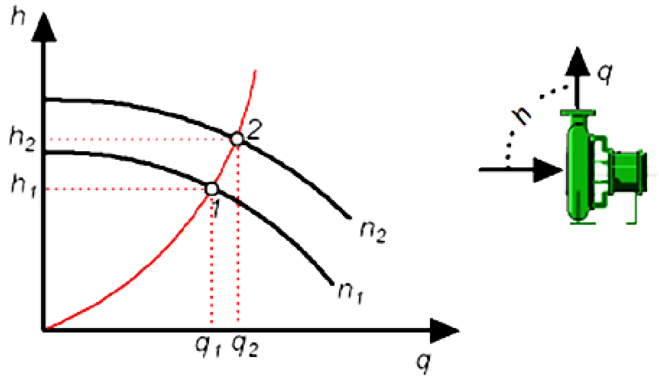


Figure 10-5: Changing wheel velocity

Note: If the speed of a pump is increased with 10%, the volume flow increases with 10%, the head increases with 21% and the power increases with 33 %.

If we want to increase the volume flow capacity of an existing system with 10% we have to increase the power supply with 33%.

Changing the Impeller Diameter

If wheel velocity is constant a change in impeller diameter simplifies the affinity laws to:

Capacity

$$\frac{q1}{q2} = \left(\frac{d1}{d2}\right)$$ Equation 41

Head or Pressure

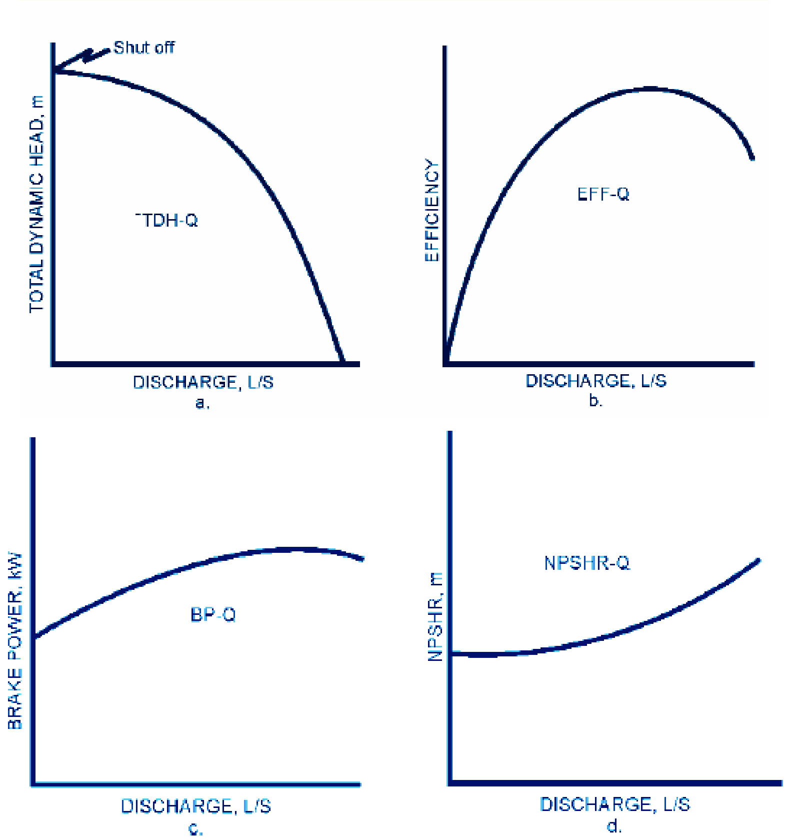
$$\frac{dp1}{dp2} = \left(\frac{d1}{d2}\right)^2$$ Equation 42

Power

$$\frac{p1}{p2} = \left(\frac{d1}{d2}\right)^3$$ Equation 43

10.4.2 Pump characteristic curve

Most manufacturers provide four different characteristic curves for every pump: the Total Dynamic Head versus Discharge or TDH-Q curve, the Efficiency versus Discharge or EFF-Q curve, the Brake Power versus Discharge or BPQ curve and Net Positive Suction Head Required versus Discharge or NPSHR-Q curve. All four curves are discharge related. Figure 10-6 presents the four typical characteristic curves for a pump, with one stage or impeller.



Source: FAO irrigation Manual Module-5, 2001

Figure 10-6" Pump characteristic curves

Total dynamic head versus discharge (TDH-Q)
This is a curve that relates the head to the discharge of the pump. It shows that the same pump can provide different combinations of discharge and head. It is also noticeable that as the head increases the discharge decreases and vice versa.

The point at which the discharge is zero and the head at maximum is called shut off head. This happens when a pump is operating with a closed valve outlet. As this may happen in the practice, knowledge of the **shut off head** (or pressure) of a particular pump would allow the engineer to provide for a pipe that can sustain the pressure at shut off point if necessary.

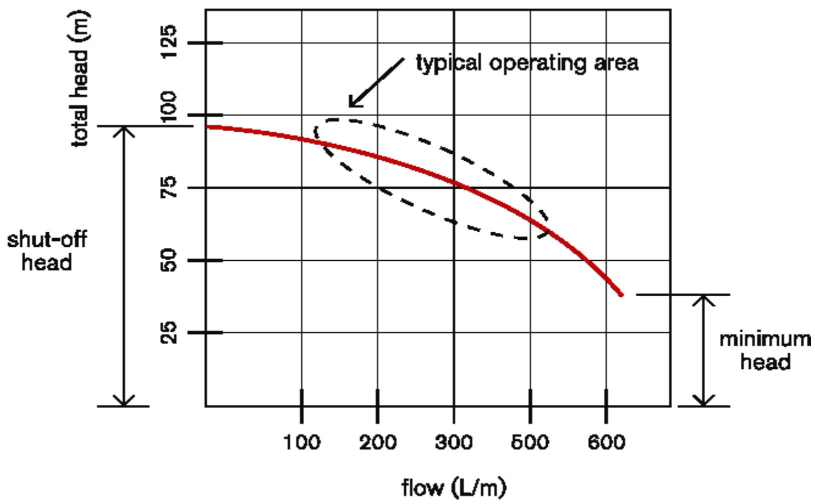


Figure 10-7: Pump operating area

Efficiency versus discharge (EFF-Q)

This curve relates the pump efficiency to the discharge. The materials used for the construction and the finish of the impellers, the finish of the casting and the number and the type of bearings used affect the efficiency. As a rule larger pumps have higher efficiencies. Efficiency is defined as the output work over the input work.

E_{pump} = Output work/ Input work = WP/ BP = (Q x TDH)/ (C x BP) Equation 44

Where:

E_{pump} = Pump efficiency

BP = Brake power (kW or HP = 1.34 x kW): energy imparted by the prime mover to the pump

WP = Water power (kW): energy imparted by the pump to the water

Q = Discharge (l/s or m^3/hr)

TDH = Total Dynamic Head (m)

C = Coefficient to convert work to energy units – equals 102 if Q is measured in l/s and 360 if Q is measured in m^3/hr

Brake or input power versus discharge (BP-Q)

This curve relates the input power required to drive the pump to the discharge. It is interesting to note that even at zero flow an input of energy is still required by the pump to operate against the shut-off head. The vertical scale of this curve is usually small and difficult to read accurately. Therefore, it is necessary that BP is calculated using Equation 45, which can be found by rearranging Equation 44:

$$BP = \frac{Q \times TDH}{C \times E_{pump}}$$ Equation 45

Net positive suction head required versus discharge (NPSHR-Q)

At sea level, atmospheric pressure is 100 kPa or 10.13 m of water. This means that if a pipe was to be installed vertically in a water source at sea level and a perfect vacuum created, the water would rise vertically in the pipe to a distance of 10.13 m. Since atmospheric pressure decreases with elevation, water would rise less than 10.13 m at higher altitudes.

A suction pipe acts in the manner of the pipe mentioned above and the pump creates the vacuum that causes water to rise in the suction pipe. Of the atmospheric pressure at water level, some is lost in the vertical distance to the eye of the impeller, some to frictional losses in the suction pipe and some to the velocity head. The total energy that is left at the eye of the impeller is termed the Net Positive Suction Head.

The amount of pressure (absolute) or energy required to move the water into the eye of the impeller is called the Net Positive Suction Head Requirement (NPSHR). It is a pump characteristic and a function of the pump speed, the shape of the impeller and the discharge. Manufacturers establish the NPSHR-Q curves for the different models after testing.

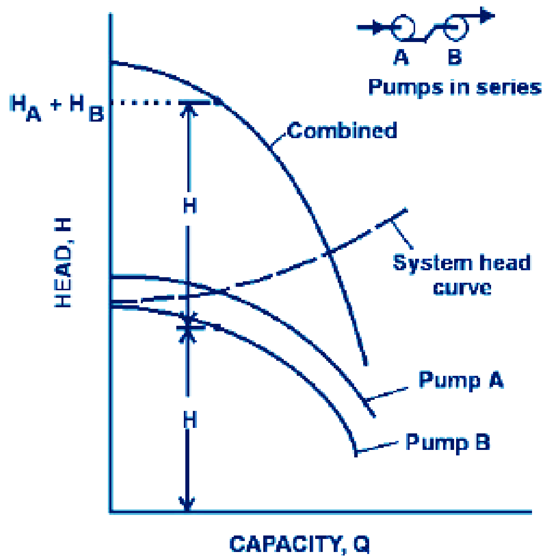
If the energy available at the intake side is not sufficient to move the water to the eye of the impeller, the water will vaporize and the pump will cavitate. In order to avoid cavitation the NPSHA should be higher than the NPSHR required by the pump under consideration.

10.4.3 Pump arrangements

Pumps in series

A good example of connecting pumps in series is where a centrifugal pump takes water from a dam and pumps it to another pump, which in turn boosts the pressure to the required level. Another example is the multistage turbine pump. In fact, each stage impeller represents a pump. In general, connecting pumps in series applies to the cases where the same discharge is required but more head is needed than that which one pump can produce.

For two pumps operating in series, the combined head equals the sum of the individual heads at a certain discharge. Figure 10-8 shows how the combined TDH-Q curve can be derived. If pumps placed in series are to operate well, the discharge of these pumps must be the same.



Source: FAO irrigation Manual Module-5, 2001

Figure 10-8: TDH-Q curve for pumps operating in series

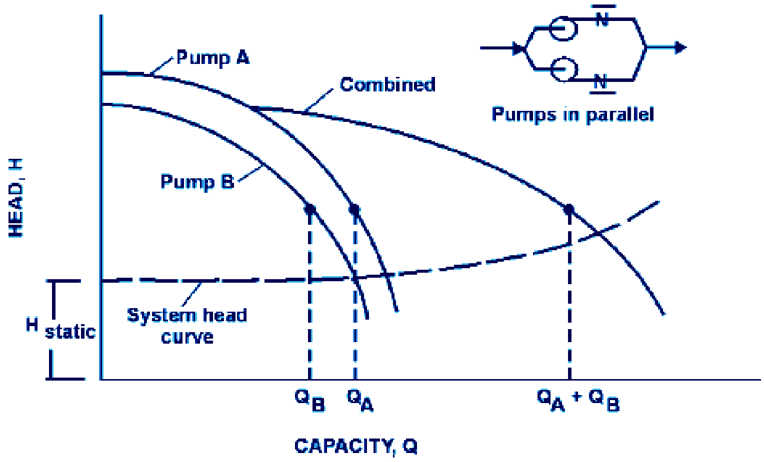
The following equation from Longenbaugh and Duke (1980) allows the calculation of the combined efficiency at a particular discharge.

$$\mathbf{Eseries} = \frac{\mathbf{Qx(TDHa+TDHb)}}{\mathbf{Cx(BPa+BPb)}}$$ Equation 46

Where:
E = Efficiency
Q = Discharge (l/s)
TDH = Total Dynamic Head (m)
C = 102 (coefficient to convert work to energy units)
BP = Brake power (kW)

Pumps in parallel

Pumps are operated in parallel when, for roughly the same head, variation in discharge is required. A typical example would be a smallholder pressurized irrigation system with many users. In order to provide a certain degree of flexibility when a number of farmers cannot be present, due to other unforeseen obligations (for example funerals), several smaller pumps are used instead of one or two larger pumps. This has been practiced in a number of irrigation schemes. Figure 10-9 shows the TDH-Q combined curve, for two pumps in parallel.



Source: FAO irrigation Manual Module-5, 2001

Figure 10-9: TDH-Q curve for two pumps operating in parallel

The equation for the calculation of the combined efficiency of pumps in parallel is as follows:

$$\mathbf{Eparallel} = \frac{\mathbf{(Qa+Qb)x(TDH)}}{\mathbf{Cx(BPa+BPb)}}$$ Equation 47

Where:
E = Efficiency
Q = Discharge (l/s)
TDH = Total Dynamic Head (m)
C = 102 (coefficient to convert work to energy units)
BP = Brake power (kW)

It should be noted from this equation that each of the pumps used in parallel should deliver the same head and this has to be a criterion when selecting the pumps. At times, engineers are confronted with a situation where pumping is required from a number of different sources at different elevations. In this case each pump should deliver its water to a common reservoir and not a common pipe in order to avoid the flow of water from one pump to another.

Operators should be careful when operating pumps in parallel to ensure that the minimum flow requirement is met for each pump. Using several pumps in parallel broadens the range of flow that can be delivered to the system. It should also be noted that pumps used at the same time should have identical performance characteristics. Pumps with differing performance curves operating in parallel cannot both operate at their BEP and the likelihood of system failure is greatly increased.

10.4.4 Pump combinations

Using a number of smaller pumps rather than one larger unit may sometimes be an economical alternative. This will allow a smaller pump operating at high efficiency to be used during low demand periods which resultants saving in running costs. The change from single pumps to multiple pump operation may affect individual pump efficiencies and maintenance costs. This aspect must be satisfactorily resolved to ensure a multiple pump installation is feasible.

A single pump is unable to consistently operate close to its BEP with a wide variation in system requirements. Multiple pumps consisting of several smaller pumps in combination can be used to serve the pumping requirements of a system, particularly those with large differences between the flow rate required during normal system operation and that required during maximum system flow conditions.

The advantages in using combinations of smaller pumps rather than a single large one are:

- Operation flexibility
- Redundancy in case of a pump failure
- Lower maintenance requirements due to pumps operating near their BEP
- Higher efficiency

Generally the number of pumps to be installed for the specific purpose is decided based on the capacity of pump/s and anticipated demands, the efficiency and cost consideration during initial installation, operation and maintenance for the specific number of pumps considered. This needs good engineering judgment to fix proper number of pumps to satisfy the same demand.

10.4.5 Pump curve and system curve

Pump curve

Performance curve or characteristic curve is discussed under section 10.4.2 above. The parameters to be indicated in pump curve are also discussed under same section. A combined typical pump curve is indicated under Figure 10-10.

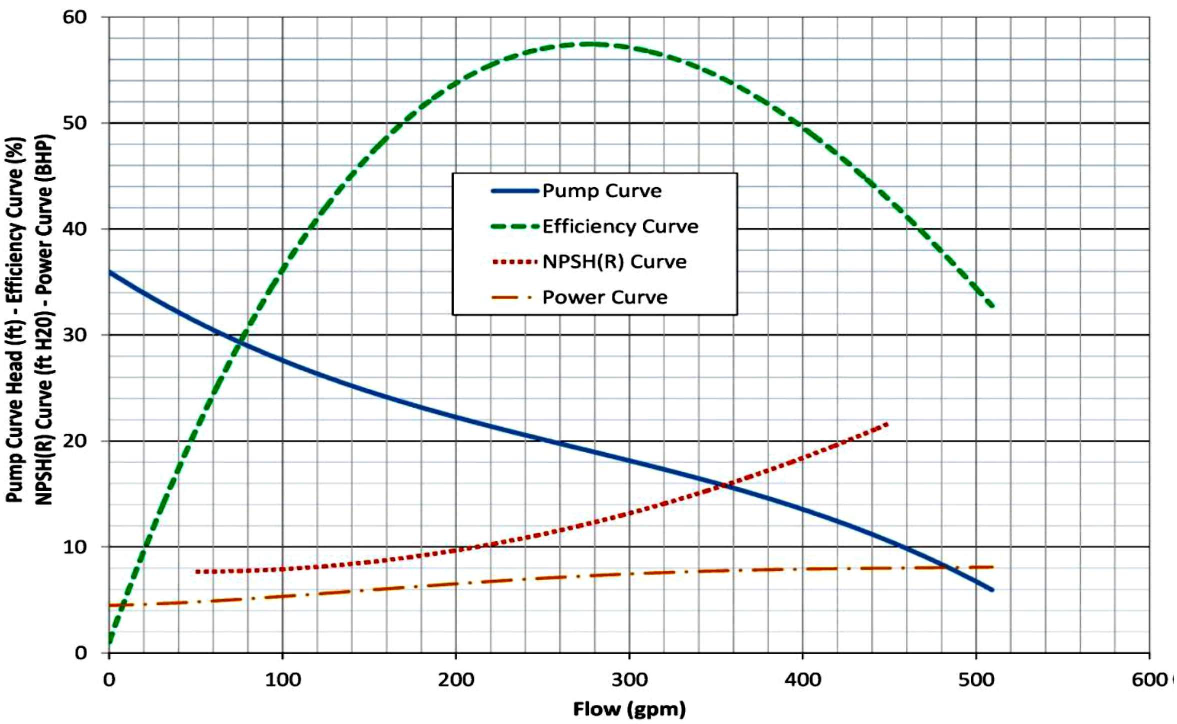


Figure 10-10: Typical pump curve

System curve:

- The system curve is a graphical representation of the relationship between discharge and head loss in a system of pipe.
- The system curve is completely independent of the pump characteristics.
- The basic shape of the system curve is parabolic because the exponent on the head loss equation is 2.0 or nearly 2.0
- The system curve start at zero flow and zero head if there is no static lift. Otherwise the curve will be vertically offset from the zero head value.
- Most sprinkler irrigation system have more than one system curve because either the sprinklers moves between sets (periodic move system) move continuously or stations (blocks) of laterals are cycled on and off.
- The intersection between the system and pump characteristic curves is the operating point (duty point)

Interaction of the system curve with the pump curve

Pumps operate where the pump curve meets the system curve. Ideally, pumps should be sized to run as closely as possible to its best efficiency flow rate. This not only makes the pump more efficient, but also improves its reliability. Correct sizing requires that both pump curves be fairly accurate. Minor variances of the manufacturer's tolerances may affect the pumps performance, but all curves have a tolerance of approximately ±3%. System curves have a much wider range of inaccuracy due to variations in pipe and fitting friction losses between various manufacturers. The total head at zero flow is the maximum head also called the **shut-off head**, the total head decreases as the flow increases.

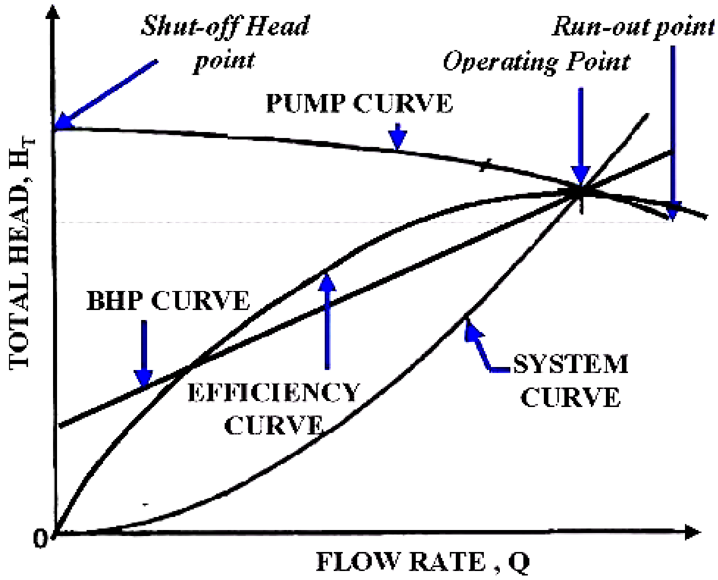


Figure 10-11: Interaction of pump curve and system curve

Pump performance is known after test. The pump test can either be done at the factory or in the field with the appropriate equipment. Head, capacity, current, power input and voltage are all items that are to read by the test equipment. Power factor, motor efficiency, motor input horsepower (EHp), brake horsepower (BHp), hydraulic horsepower (WHp), total efficiency, and pump efficiency must be given in catalogue of the manufacturer.

10.4.6 Best Efficiency Point (BEP)

The Best Efficiency Point is defined as the flow at which the pump operates at the highest or optimum efficiency for a given impeller diameter. When we operate a pump at flows greater than or less than the flow designated by the BEP, we call this “operating pumps away from the Best Efficiency Point”

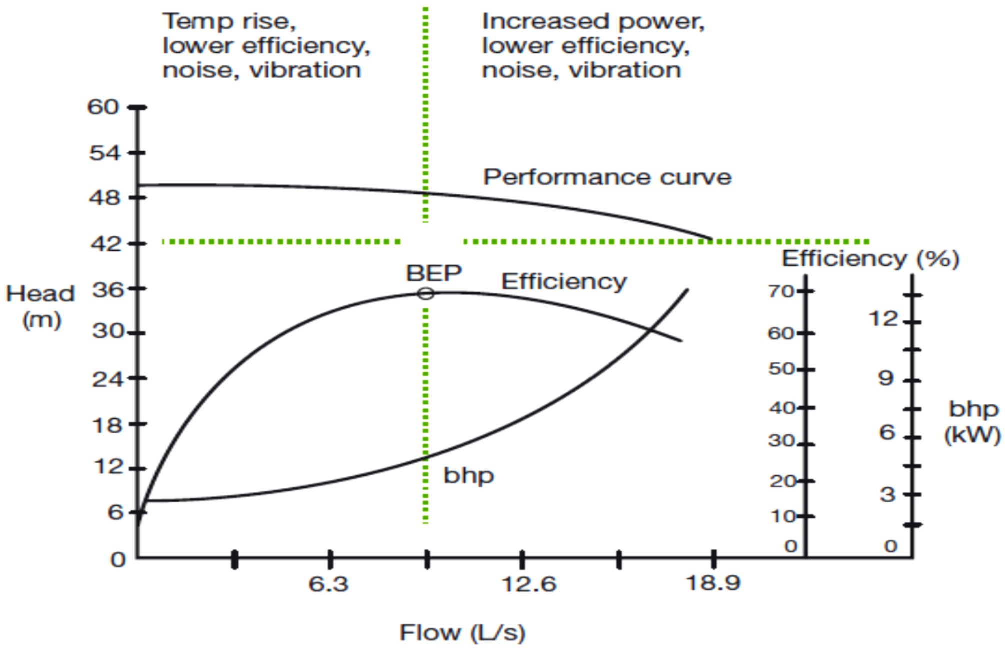


Figure 10-12: Best efficient points

10.5 PUMP SELECTION CRITERIA

The selection of pumps requires the use of manufacturers' pump curves. As a first step, by looking at the various pump curves we can identify a pump that can provide the discharge and head required at the highest possible efficiency. Following the identification of the pump, the NPSHR-Q curve is checked and evaluations are made to ensure that its NPSHA is higher than the NPSHR.

Example:
Let us assume that a designed sprinkler system would require a Q = 40 m^3/hr at an H = 60 m. What would be the best pump to select?

Looking at various performance curves provided by manufacturers (Figures 10-13 and 10-14) the curve of Figure 10-14 was selected, as it appears to provide the highest efficiency (65%) for the required discharge and head requirements, compared to an efficiency of 45% given by curves of Figure 10-13. Ideally we would have preferred a pump where the required head and flow combination falls on the right-hand side of the efficiency curve. With age, the operating point will move to the left, then we would be able to operate with higher efficiency. This pump should be equipped with the 209 mm impeller, as shown in the curve.
Looking at the NPSH-Q curve in Figure 10-14, the NPSHR of this pump is 1.2 m.
Assuming the following data for the site:

- Elevation: 2000 m
- Static suction: 2 m
- Suction pipe friction losses: 0.5 m
- Maximum temperature: 35°C

NPSHA = (10.33 - 0.00108 x 2000) - 2.0 - 0.5 - 0.58 = 5.09 m
Since NPSHA (5.09 m) is higher than the NPSHR (1.2 m) of the selected pump no cavitation should be expected.

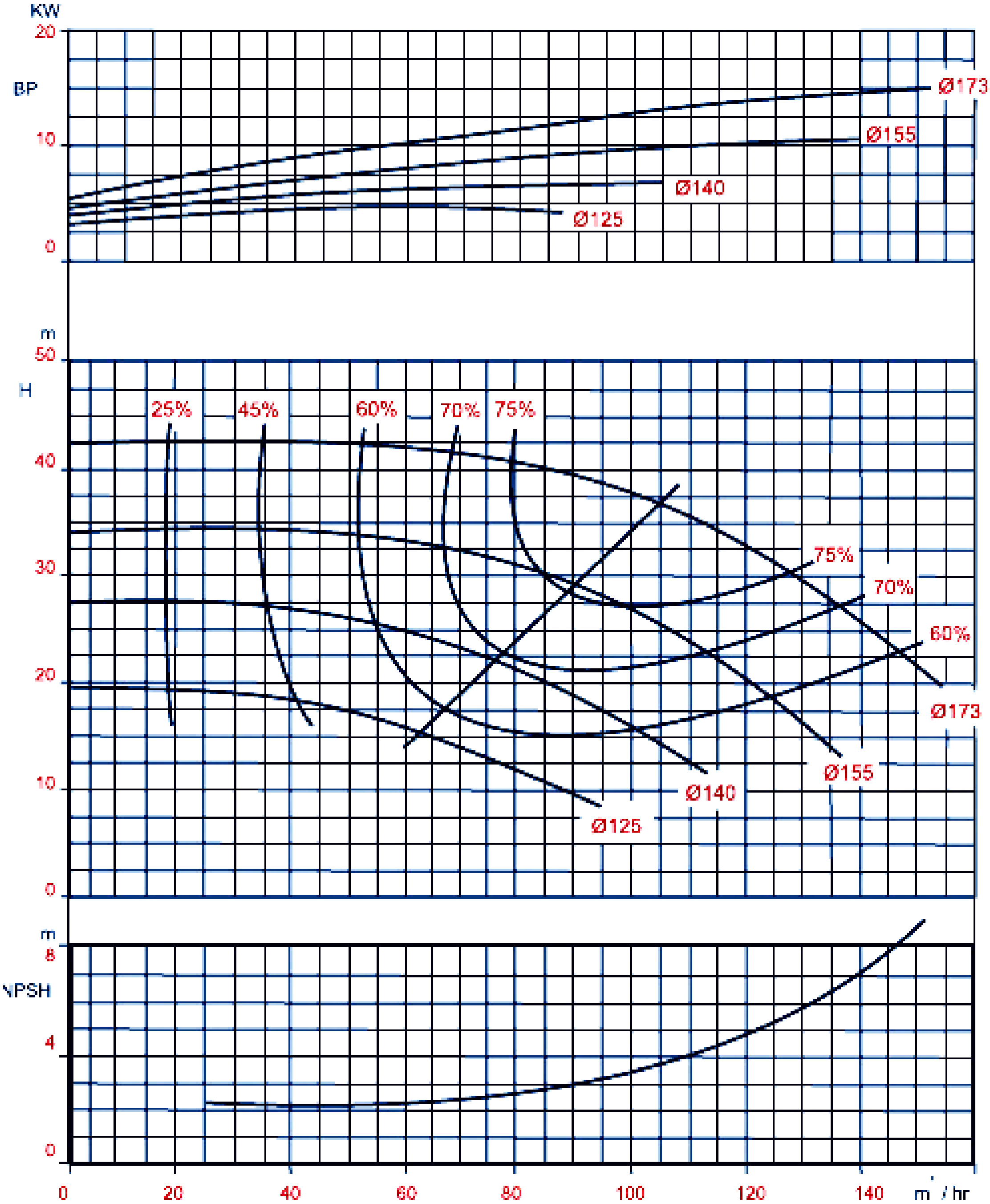


Figure 10-13: Performance curve of pump (a)

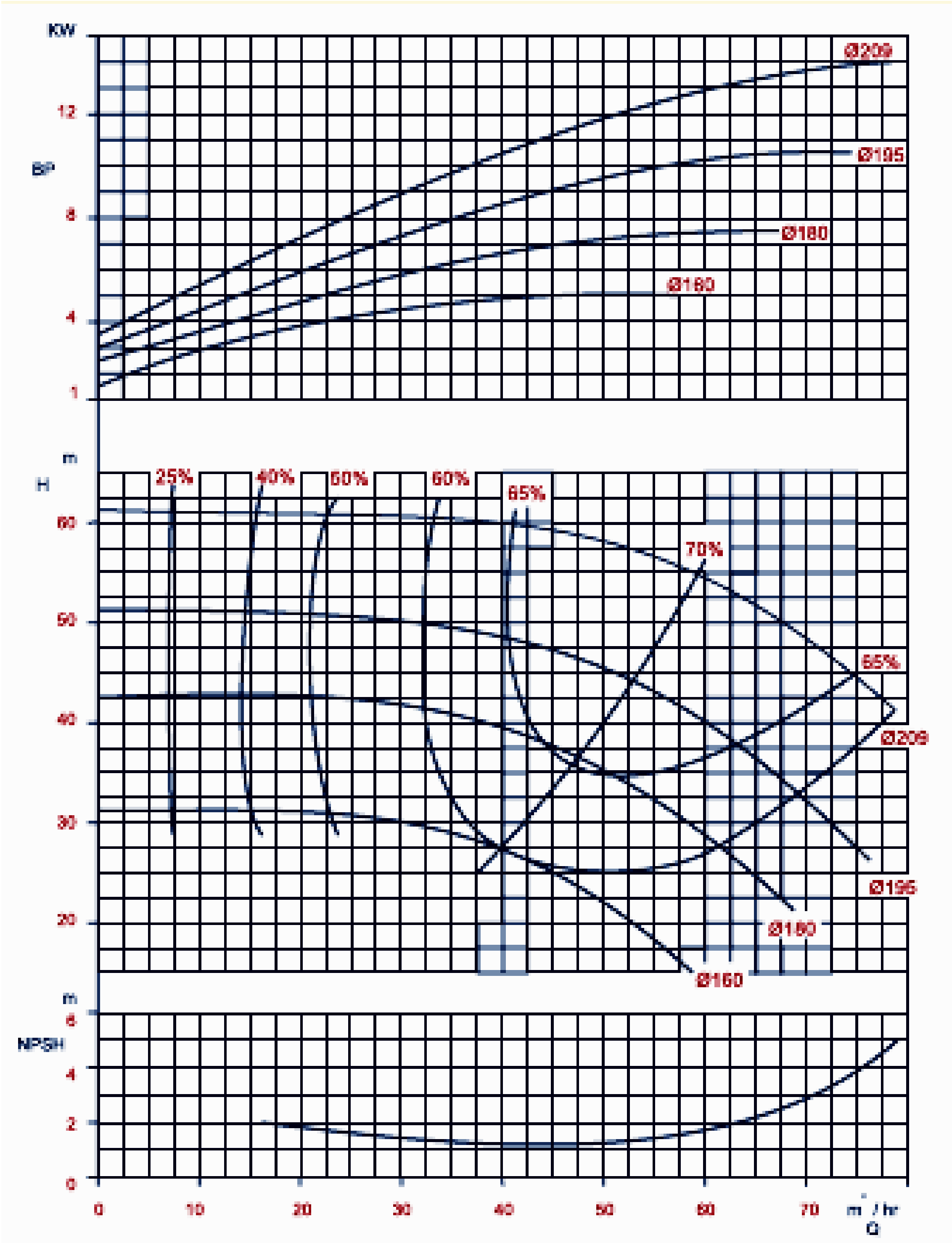


Figure 10-14: Performance curve of pump (b)

The general guidelines for proper pump selection are:

- Select the pump based on rated conditions.
- The BEP should be between the rated point and the normal operating point.
- The head/capacity characteristic-curve should continuously rise as flow is reduced to shutoff (or zero flow).
- The pump should be capable of a head increase at rated conditions by installing a larger impeller.

- The pump should not be operated below the manufacturer's minimum continuous flow rate.

Critical Points for Selection of Pumps:

- Fluid characteristics: A detailed study of the fluid characteristics is usually the most important factor for the proper selection of pumps. Right type of pump can only be selected based on the experience and knowledge of the selecting engineer and the environment.
- The suction head and the available Net Positive Suction Head (NPSH) both decreases with an increase in liquid viscosity for the same pumping rate. At the same time, discharge and total head both increases with an increase in liquid viscosity for the same pumping rate. In other words, the power requirement also increases with liquid viscosity.
- Specific Gravity: It affects the pump life along with performance of the pump.
- Temperature: The operating temperature at the pump is an important factor affecting overall performance of the pump. While considering temperature, the combined ambient and liquid temperature along with the temperature rise due to evaluation of heart from the resistance in the system shall be taken into consideration. As per general experience, pumps can perform efficiently with trouble-free operation over an approximate temperature of up to 80°C.
- Lubricating quality of liquid handled.
- Space available for pump: It helps in selecting the pump, ie, horizontal or vertical. It also influences the model and size of the pump.
- Ease in operation and maintenance
- Lubricating system
- Availability of spare parts
- Cost factor
- Reliability
- Environmental-friendly
- Efficiency of pump at duty point
- Arrangement for minimising gland leakage, etc
- Material of construction, etc
- Drive system and reliability.
- Selection of Pumps Based on Practical Experience.

11 DESIGN OF PUMPING FACILITY

During preliminary design, the designer will confirm the suitability of the site for pump station construction, evaluate facility alternatives, and confirm station features as a function of capacity and risk. These activities will be accomplished through a geotechnical investigation, site survey, environmental investigation, and preliminary design report. Each is described in other guidelines for Topographic data, Hydro metrological data, Geological data and Agronomic data.

11.1 DATA REQUIREMENT

Type of data requirements shall be identified first to be used in the process of design activity in order to have practical design report. Basic data required to the study and design of the proposed scheme has to be properly collected, analyzed and simulated using proper and practicable methodologies. The following major activities, but not limited are supposed to be considered to meet the study and design needs of the project.

The complete information about the project site like location of source, farm area, roots of pipe system etc. are needed. Detail survey works both at the head work site and irrigable area shall be conducted and a topographic map of the scheme area has to be prepared at a specified scale.

Topographic map of the head work at a scale not more than 1:500 and at a contour interval of not more than 0.5, showing all the features upstream and downstream, right and left of the proposed site including observation pits shall be prepared. Cross-sections at a scale of both vertical and horizontal shall be 1:100 indicating the pertinent features to head work shall be prepared. Topographic map of the irrigable area at a scale of not more than 1:1000, at a contour interval of not more than 1m for steeply area (25%) and not more than 0.5m for planning and undulating areas shall be established. The topographic map should also show every major feature in the irrigable area such as traditional canals, watering points/springs, settlements, foot path, cattle crossings, gullies, trees, benchmarks, hills, graves, etc.

All the required benchmarks and stations shall be established using stable features and shall be properly connected with the national grid stations and benchmarks of the proposed pump site.

The hydrological analysis is an essential engineering component that has to be investigated thoroughly meeting the two basic elements safety and economy of the hydraulic structures. Thus, proper estimation of extreme flows (high and low flows) is vital for the design of hydraulic structures and planning and management of irrigation water. The two parameters crucial for design purpose are the **lean flow and the peak flood**. These levels must be clearly indicated or marked (as meter elevation a.s.l.) on site and on topographic map and sections.

The estimation is crucial for better planning and developing the given irrigable area and well-being of the ecosystem in the downstream portion of the river reach. For the purpose of irrigation usually 80% flow exceedance should be adequately meet the crop water needed. The estimation of this flow exceedance could only possible when direct measurements of flow are available in the case of gauged river or it can be estimated using a regionalization approach or different techniques. There are different methods or techniques available for the estimation of the lean flow in the river. The automated base flow separation technique, graphical method, tracer methods using isotopes, and rainfall runoff models are some of the methods applicable if the river is gauged.

Variety of crops, crop plan, crop demand, type and characteristic of soil are some of the necessary data. The main concern in this guideline is to obtain necessary data for pump design. Basically the monthly demands after water balance consideration and the **maximum duty of crops** is mandatory for design purpose. For details please refer Guideline for Irrigation Agronomy.

Desirable quantity and quality of geological and geotechnical investigations are required for safe and economical design of the pumping facility. The detail about how geological date is to be collected and used is given in Guideline for Geology and Engineering Geological Study.

11.2 DESIGN PROCEDURE

- The design process starts with determination of discharge required for irrigation.
- The next step in design is determination of the head or elevation difference and the lengths of suction and delivery lines.
- Next select type of pipe materials and size the components.
- Once the flow rate and pipe sizes are known, the friction head loss can be determined.
- Additional energy losses are encountered as the fluid passes through bends, constrictions, valves, etc. These are referred to as minor losses.
- Calculation of the system losses at several different flow rates will yield a system curve. System curves represent a loss of energy in systems with a variation in the flow rate. Or, stated differently, the amount of energy the pump must generate to operate at a given flow rates. Consider de-rating and other factors.
- The total dynamic head (TDH) of a pumping system is calculated TDH is the head the pump must overcome to move the fluid to its destination. At the same time determine the NPSH and the water hammer expected.
- By calculating multiple TDH values based on incrementing flow values, and plotting these on a graph of TDH vs flow rate, the system curve is determined.

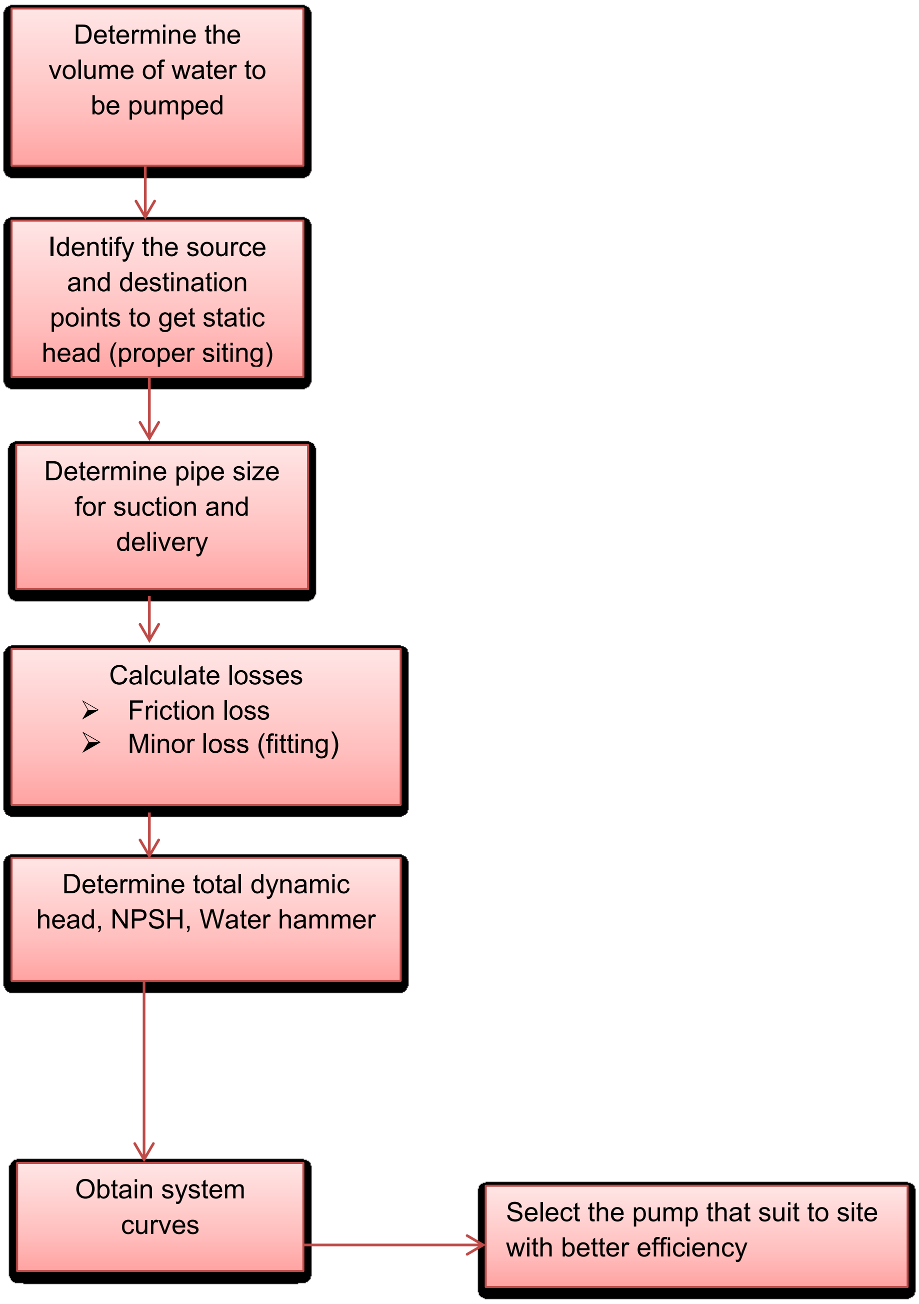


Figure 11-1: Pump design procedure

11.3 DESIGN EXAMPLE ON SURFACE IRRIGATION SYSTEM

Consider the peak irrigation demand of a project is found to be 42 l/s for 24 hours of irrigation application according to agronomy reports. The level difference of the center of pump and the minimum water level was obtained from the developed stage discharge curve and the source (river) cross-sectional data. The minimum water level as extracted from the stage discharge curve is 2352.30 m a.s.l. The pump site is situated above the design flood level and the center of the pump is 2355.40 m a.s.l. The delivery elevation is 2375.00 m a.s.l. Total length of delivery line is 300m. Length of suction pipe is 6m. Thus design for proper pipe sizing and pumping unit considering electric power supply at site.

Solution:

Note that pumping hours per day determines the number as well as the size of the pumping unit. If the pump working hours per day is fixed to 8 hours then the discharge to satisfy the above demand for 24 hours indicated as 42l/s must be satisfied.

Option-1 Two pumps working for 8 hours are considered then the discharge of each pump will be 31.5l/s. This is found from [(24/16)*42]/2.

Option-2 One pump working for 8 hours per day discharging 63l/s.

Design option-1:

Number of pumps =2

Discharge each pump=31.5l/s= 0.0315m^3/s

Level difference in suction side=pump center-minimum water level=2355.40-2352.30=3.10m

Level difference in deliver side=delivery level-pump center=2375.00-2355.40=19.60m

In sizing the suction pipe the two main elements coming to mind are the suction velocity limit and the NPSH. The possible problems in suction side pipework often have damaging consequences for the pump and can be avoided by following the following guidelines:

- Ensure that conditions do not favor cavitation
- Position the feed pipe to minimize entrainment of air/vapour and solids.
- Minimize friction and turbulence by choosing appropriate pipes and components:
- Use pipes with a diameter twice that of the pump's suction side flange.
- Use an eccentric reducer orientated to eliminate air pockets.
- Keep the pipe suction velocity below 2m/s.

Also in sizing delivery pipe conditions previously discussed must be accounted. Based on the recommendations cited above, the pipe size has been fixed keeping the flow velocity in suction pipe as 1.7m/s and delivery pipe velocity of about 2m/s is considered for this design. Hence, applying continuity equation:

$$\mathbf{Q = A * V} \qquad \text{Equation 48}$$

Where:

Q = Pipe capacity (m^3/s),

A = Pipe cross-sectional area(m^2)

V = Allowable flow velocity in the suction pipe (m/s).

Suction pipe size is thus:

$$\boldsymbol{Q = \frac{\pi d^2}{4} * V \quad thus\ d^2 = \frac{4*Q}{\pi*V}} \qquad \text{Equation 49}$$

$$d = \sqrt{\frac{4*Q}{\pi*V}} \quad d = \sqrt{\frac{4*Q}{\pi*V}} = d = \sqrt{\frac{4*0.0315}{3.14*1.7}} = 0.154\text{m}$$

Hence, the nearest diameter size for suction pipe available in the market which is 150mm has been adopted. Accordingly suction velocity is computed to be 1.78m/s and suction velocity head=v^2/2g =0.16m.

Delivery pipe size for 1.8m/s is thus:

$$d = \sqrt{\frac{4 * 0.0315}{3.14 * 1.8}}$$

D=0.149m thus practical diameter to be adopted for delivery pipe is 150mm. Thus delivery velocity is 1.78m/s with delivery velocity head 0.16m.

Consider steel pipe for delivery and reinforced rubber pipe for suction.

Now let's compute the total suction lift of option-1:
Length=6m
Static suction lift or head =3.10m as indicated above
Friction loss given by Hazen Williams formula: C= 130 for reinforced rubber pipe considered for suction.

$$hf = (L) * [\frac{10.67 * (Q)^{1.852}}{(C)^{1.852}(D)^{4.8704}}]$$

Inserting all values to above equation suction side friction loss hf=0.13m
Minor losses:
hm= K.hv
Where K= coefficient of loss and hv is velocity head =0.16m.
Suction side (6m long):

- Loss in foot valve and strainer=K=1.80
- Loss in large smooth 90o bend=k=0.34
- Entry loss for bell mouth entry=K=0.45

Total suction side minor loss coefficient=2.59 and thus minor loss in suction pipe=Khs=0.41m
Total suction lift head =Static lift head + friction loss +minor loss +velocity head
=3.1+0.13+0.41+0.16=**3.80m**

Net Positive suction Head (NPSH):
NPSHa = Ha ± Hs - Hvap - Hf -Fs

Ha = $10.13 - \frac{2355 * 0.6}{500}$ = 7.30m is the absolute pressure head due to altitude difference.

- Ha=absolute pressure at 2355m a.s.l (pump centre line)=7.30m
- Hs=static suction lift=3.10m
- Hf=friction loss including fitting loss=0.54m
- Hvap=Vapour pressure at 25oC average temperature=0.32m (from table-8-1)
- Fs= safety factor=0.60m
- NPSHa= Ha - Hs - Hvap - Hf -Fs
- NPSHa=7.30-3.10-0.32-0.54-0.60=2.74m

Available **NPSH=2.74m** Thus Required NPSH from the supplier shall be less than the Available NPSH.

Let's compute for delivery side (300m long):
Here also two options for pipe line can be considered:

a. **Independent pipe line 150mm diameter for both pumps**
b. **Two sub-main lines merging to one main line**

If pipe lines merge to one main line then size of main line shall be fixed: This is similar as that of the sub-main line with flow velocity of 1.78m/s.

$$\mathrm{d} = \sqrt{\frac{4 * 0.063}{3.14 * 1.78}}$$

D=0.212m

Thus nearest nominal size=200mm. Accordingly velocity in 200mm main pipe will be 2m/s. and velocity head will be 0.20m in main pipe line.

a. Independent pipe line for both pumps

Friction loss in 300m long pipe: Consider steel pipe for delivery, C=120

$$hf = (L) * [\frac{10.67 * (Q)^{1.852}}{(C)^{1.852}(D)^{4.8704}}]$$

hf=7.70m

Minor losses in delivery line:

- Check valve K=1 in No=1.00
- Gate valve (fully open) K=0.50
- 90o elbow, 2 in No=0.90x2=1.80
- Sudden enlargement dnXDn, K=0.26
- Coupling losses K=0.02*n where n=50 for 6m standard pipe length K=50*0.02=1.00
- Exit K=1.00

Total delivery line minor loss coefficient=5.56 and hence minor loss in delivery pipe=0.89m

Total delivery lift head =Static delivery head + friction loss +minor loss +velocity head

=19.60m + 7.70m + 0.89m + 0.16m

=28.35m

Total Lift head=H= Total suction lift head + Total delivery lift head

H=3.80m + 28.35m= 32.15 say **33m**

Water hammer:

Water hammer is given by:

$$\Delta \mathrm{hJou} = \frac{\mathrm{a}}{\mathrm{g}} \,.\Delta \mathrm{V}$$

Thus flow velocity in delivery line =2m/s under sudden shut off ∆V=1.78m/s. a is wave propagation velocity given by:

$$\boldsymbol{a} = \sqrt{\frac{\mathbf{1}}{\frac{\boldsymbol{\rho}}{\boldsymbol{Ef}} + \frac{\boldsymbol{\rho} . \boldsymbol{D}\,(\mathbf{1} - \boldsymbol{\mu}^{\mathbf{2}})}{\boldsymbol{Ep.t}}}}$$

Where,

a= wave propagation velocity (m/s)

L= length of delivery pipe= 300m

ρ: Density of the fluid in kg/m^3=$1000kg/m^3$

Ef: Modulus of elasticity of the fluid in N/m^2=$2.2x10^9$ N/m^2

Ep: Modulus of elasticity of the pipe wall in N/m^2=$2.077x10^{11}N/m^2$

D: Inside pipe diameter in mm=150mm

t: Pipe wall thickness in mm=6mm assumed

μ: Transverse contraction number or poison's ratio=0.30 for steel pipe

Inserting the values in above formula: a=1332m/s

Thus water hammer=(1332/9.81)*1.78=242m or 24bar. Thus the pipe must withstand this head besides the working head.

Power Requirement: (For option-1 two pumps with independent delivery line)
Water Horse Power (Theoretical hp)=WHP=P[w] =γQH where H is the total lift head=33m, Q is unit pump discharge=0.0315m3/s and γ=unit weight of water which is approximately 10KN/m3.
Thus P[Kw]=10.40KW
Shaft horse power, assuming pump efficiency of 70% will be 14.86KW.
Brake horse power for motor efficiency of 85% will be **17.50KW**.

b. Two sub-main lines merging to one main line

Out of 300m delivery line, let 18m is sub-main line (150mm) and the rest 282m is main line (200mm Diameter).Thus using Hazen Williams formula friction los in sub-main line=0.46m and in main line=6.43m thus total friction loss in delivery line=6.89m.

Minor loss in delivery (consider previously calculated loss in delivery + confluence loss) =0.89m + $0.95*2^2/19.62$= 1.08m. Here confluence loss co-efficient is taken 0.95 for confluence angle not more than 30^o.
Velocity head in delivery=0.20m
Total delivery lift head =Static delivery head + friction loss +minor loss +velocity head
=19.60m + 6.89m + 1.08m + 0.20m
=27.77m
Total Lift head=H= Total suction lift head + Total delivery lift head
H=3.80m + 27.77m= 31.57 say **32m**

Power Requirement: (For option-1 two pumps with merged delivery line)
Water Horse Power (Theoretical hp)=WHP=P[w] =γQH where H is the total lift head=32m, Q is unit pump discharge=0.0315m3/s and γ=unit weight of water which is approximately 10KN/m3.
Thus P[Kw]=10.08KW

Shaft horse power, assuming pump efficiency of 70% will be 14.40KW.
Brake horse power for motor efficiency of 85% will be 16.94KW.
Here pump power requirement is nearly the same for independent as well as merged delivery line thus **two in number 18KW** pumps can be considered under option-1.

Design option-2:
Number of pump =1
Discharge each pump=63 l/s= $0.063m^3/s$
Level difference in suction side=pump center-minimum water level=2355.40-2352.30=3.10m
Level difference in deliver side=delivery level-pump center=2375.00-2355.40=19.60m

Suction and Delivery pipe size for 1.8m/s is:

$$d = \sqrt{\frac{4 * 0.063}{3.14 * 1.8}}$$

d=0.211m thus practical diameter to be adopted for delivery pipe is **200mm**. Thus delivery velocity is 2.00m/s with delivery velocity head 0.20m.

Consider steel pipe for delivery and reinforced rubber pipe for suction.

Now let's compute the total suction lift of option-2:
Length=6m
Static suction lift or head =3.10m as indicated above
Friction loss given by Hazen Williams formula: C= 130 for reinforced rubber pipe considered for suction.

$$hf = (L) * [\frac{10.67 * (Q)^{1.852}}{(C)^{1.852}(D)^{4.8704}}]$$

Inserting all values to above equation suction side friction loss hf=0.14m
Minor losses:
hm= K.hv
Where K= coefficient of loss and hv is velocity head =0.20m.
Suction side (6m long):

- Loss in foot valve and strainer=K=1.80
- Loss in large smooth 90o bend=k=0.34
- Entry loss for bell mouth entry=K=0.45

Total suction side minor loss coefficient=2.59 and thus minor loss in suction pipe=Khs=0.52m
Total suction lift head =Static lift head + friction loss +minor loss +velocity head
=3.1+0.14+0.52+0.20=**3.96m**

Net Positive suction Head (NPSH):
NPSHa = Ha $\underline{+}$ Hs - Hvap - Hf -Fs
Ha = 10.13 – $\frac{2355*0.6}{500}$ = 7.30m is the absolute pressure head due to altitude difference.

- Ha=absolute pressure at 2355m a.s.l (pump centre line)=7.30m
- Hs=static suction lift=3.10m
- Hf=friction loss including fitting loss=0.66m
- Hvap=Vapour pressure at 25oC average temperature=0.32m (from table-8-1)
- Fs= safety factor=0.60m
- NPSHa= Ha - Hs - Hvap - Hf -Fs
- NPSHa=7.30-3.10-0.32-0.66-0.60=2.62m

Available **NPSH=2.62m** Thus Required NPSH from the supplier shall be less than the Available NPSH.
Friction loss in 300m long pipe: Consider steel pipe for delivery, C=120

$$\mathrm{hf} = (\mathrm{L}) * [\frac{10.67 * (\mathrm{Q})^{1.852}}{(\mathrm{C})^{1.852}(\mathrm{D})^{4.8704}}]$$

hf=6.84m
Minor losses in delivery line:

- Check valve K=1 in No=1.00
- Gate valve (fully open) K=0.50
- 90o elbow, 2 in No=0.90x2=1.80
- Sudden enlargement dnXDn, K=0.26
- Coupling losses K=0.02*n where n= number of couplings =50 for 6m standard pipe length K=50*0.02=1.00
- Exit K=1.00

Total delivery line minor loss coefficient=5.56 and hence minor loss in delivery pipe=1.11m

Total delivery lift head =Static delivery head + friction loss +minor loss +velocity head
=19.60m + 6.84m + 1.11m + 0.20m
=27.75m
Total Lift head=H= Total suction lift head + Total delivery lift head
H=3.96m + 27.75m= 31.71 say **32m**

Power Requirement: (For option-2 one pump only)

Water Horse Power (Theoretical hp)=WHP=P[w] =γQH where H is the total lift head=32m, Q is unit pump discharge=0.063m3/s and γ=unit weight of water which is approximately 10KN/m3.
Thus P[Kw]=20.16KW
Shaft horse power, assuming pump efficiency of 70% will be 28.80KW.
Brake horse power for motor efficiency of 85% will be **34KW.**

By assuming 20% for electric motor overload, the pump motor will be
Pm = Ph *(1+0.2), KW
Pm = 18* 1.2 = 21.6 KW (two pump case)
Pm = 34* 1.2 = 40.80 KW (one pump case)
Hence, the motors shall be selected from **motor rating standards**. Say 22KW for two pump case and 41 KW for one pump case.
Additional power requirement for:
Illumination and others…………………………..5KW,
Allowance……………………………………….. 5KW,

Thus actual total power requirement (KW) at site is 54KW for two pump case and 51KW for one pump case.

The apparent power to be supplied (KVA) is computed by dividing the real power to power factor (say 0.80).

Note:

- The pump must be equipped with priming facility.
- The designer shall fix number of pumps based on considerations cited in the guideline.
- Stand-by capacity shall be considered by the designer as per the guideline.
- Transformer size is fixed considering the power required at start of the largest motor.

11.4 DESIGN EXAMPLE ON GROUND WATER IRRIGATION SYSTEM

The ground water is extracted from two boreholes around the irrigable area. The two submersible pumps will discharge to the collection chamber. Finally booster surface pumps are utilized to lift water to canal system. Details are given below.

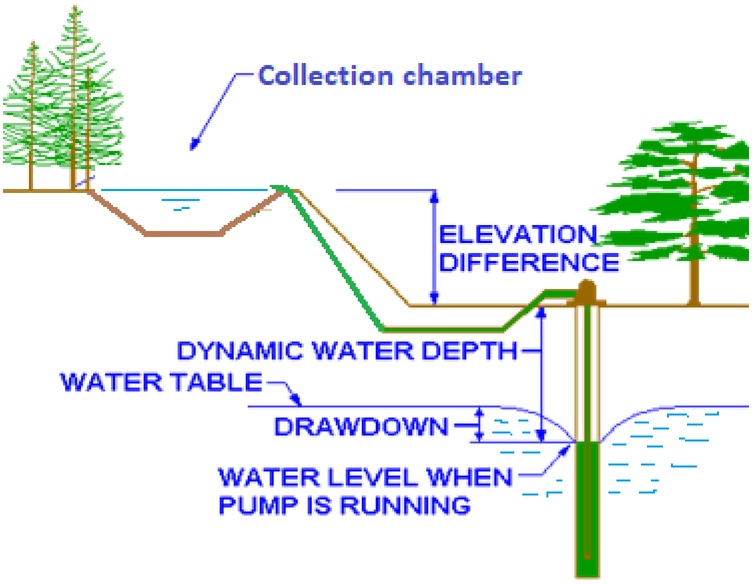


Figure 11-2: Typical well pumping system

Borehole-1 Hydraulic Design:

Given Data for Bore hole-1 (BH1)

- Delivery tank inlet level = 2300masl
- Elevation of borehole = 2216.65masl
- Dynamic Water level (drawdown) = 58.2m
- Collector Pipe Length DN250 = 1762m
- Transmission Pipe Length DN400 = 2452m
- Discharge (safe yield of well) = 30l/s
- Pump position = 144m

Calculated Values from the Give field Data

- Collector Pipe diameter = DN250
- Static head ((2300-2216.65)+58.2) = 141.55m

Determination of Riser Pipe Diameter

Assuming the flow velocity inside the riser pipe to be limited to 2.5m/s and applying the equation:

Q = V * A

Where Q : flow rate [m^3/s] = 30/1000=0.03

V : flow velocity [m/s] =2.5

A : cross-section area [m^2] = $\frac{\pi D^2}{4}$

$$D = \sqrt{\frac{Q*4}{V*3.14}}$$

$$D = \sqrt{\frac{0.03*4}{2.5*3.14}} \quad =0.12364m$$

Based on the above calculation the appropriate riser pipe diameter is selected to be **DN125mm.**

Dynamic head Loss analysis for transmission pipe line, riser pipes & fittings

Using Hazen Williams formula:

$h_f = 10.7 (Q/C)^{1.85} \times L / (D)^{4.87}$ for pipes and
$h_{ff} = k*v^2/2*g$ for valves and fittings
Where,

C - coefficient of Roughness for 120 for GS, 130 for DCI and 150 for PVC pipe
D - Pipe diameter (m)
Q - Flow Rate (m^3/s)
h_f - pipe friction loss (m)
L - pipe length (m)
h_{ff} - fitting loss

All the fittings in pipe line are listed below in the table to calculate the friction head loss or dynamic loss H_f using the above formula. Let us calculate dynamic loss for the 3 elbows and check valve

90° bends

$H_f = k*v^2/2*g = 1.5*(0.95493047)^2/(2*9.81) = 0.068409 *3 = 0.205227$

Check valve

$H_f = k*v^2/2*g = 1.9*(0.95493047)^2/(2*9.81) = 0.086510$

Likewise, substituting the known values in the above equation, the total calculated pipes and fittings frictional loss is equal to 14.67m. See Table 11-1 below for the analysis:

Table 11-1: Dynamic loss for BH1

Description	Unit	Qty	Type	Discharge [M^3/S]	Diameter [M]	Velocity [M/S]	Roughness	Friction Coefficient	Formula	Dynamic Loss [M]
90^0 bends	PCs	3	GS	0.03	0.2	0.95493047		1.5	$K * v^2 / 2g$	0.205226
Check valve	PCs	1	GS	0.03	0.2	0.95493047		1.9	$K * v^2 / 2g$	0.086510
Flow meter	PCs	1	GS	0.03	0.2	0.95493047		0.024 bar		0.24
Gate valve	PCs	1	GS	0.03	0.2	0.95493047		0.4	$K * v^2 / 2g$	0.018242
Tee	PCs	1	GS	0.03	0.2	0.95493047		0.37	$K * v^2 / 2g$	0.016871
Riser pipe	mts	144	GS	0.03	0.125	2.44462199	120	0.05802579	K * L	8.355714
Collector pipe	mts	2452	DCI	0.12	0.4	0.95493047	130	0.00225446	K * L	5.527936
90^0 bends	PCs	2	GS	0.15	0.4	1.19366308		1.5	$K * v^2 / 2g$	0.217864
Total Dynamic Loss										**14.67**

A) Duty Point Calculation for Submersible Pump1

The total pumping head is calculated from:

$$H_T = H_S + H_D \qquad \text{Equation 50}$$

Where H_T : total head
H_S : static head
H_D : dynamic head loss

Substituting the known values in the above equation:
The total pump head, H_T = 141.55+14.67= 156.25m
Therefore, the pump duty point is selected to be 160m considering allowance of 3m for losses not considered (Transmission main fittings loss). **Duty Point: 108m^3/h at 160m**

B) Pumping Unit Summary

- No of Pump : 1 Submersible Pump
- Capacity : 108m^3/h which is equal to 30l/s
- Head : 160m
- Medium to be pumped : Ground water;
- Pump speed : 2900 rpm

<u>BH2 Hydraulic design</u>:
Given Data

- Tank/collection chamber inlet level = 2300masl
- Elevation of borehole = 2242masl
- Dynamic Water level = 63.4m
- Collector Pipe Length DN250 = 1930m
- Transmission Pipe Length DN400 = 2400m
- Discharge = 40l/s
- Pump position = 132m

Calculated Values from the Give field Data

- Collector Pipe diameter = DN250
- Static head((2300-2242)+63.4) = 121.4m

C) Determination of Riser Pipe Diameter

Assuming the flow velocity inside the riser pipe to be limited to 2.5m/s and applying the equation:

Q = V * A

Where Q : flow rate [m^3/s] => 40/1000=0.04
V : flow velocity [m/s] =>2.5
A : cross-section area [m^2] = $\frac{\pi D^2}{4}$

$$D = \sqrt{\frac{Q*4}{V*3.14}}$$

$$D = \sqrt{\frac{0.04*4}{2.5*3.14}} = 0.142766\text{m} = 142.77\text{mm}$$

Based on the above calculation the appropriate riser pipe diameter is selected to be **DN150**mm.

D) Dynamic head Loss analysis for transmission pipe line, riser pipes & fittings

Using Hazen Williams's formula:

$h_f = 10.7\ (Q/C)^{1.85} \times L / (D)^{4.87}$ for pipes and

$h_{ff} = k*v^2/2*g$ for valves and fittings

All the fittings in pipe line are listed below in table 11-2 to calculate the friction head loss or dynamic loss H_f using the above formula. Let us calculate dynamic loss for the riser pipe and Tee.

Rise pipe:

$H_{fr} = k*L = 0.05802579*132 = 7.6594$

Tee:

$H_{ft} = k*v^2/2*g = 0.37*(0.95493047)^2/(2*9.81) = 0.017196744$

Similarly, substituting the known values in the above equation, the total calculated pipes and fittings frictional loss is equal to 13.76. See Table 11-2 below for the analysis:

Table 11-2: Dynamic loss for BH2

Description	Unit	Qty	Type	Discharge [m^3/s]	Diameter [m]	Velocity [m/s]	Roughness	Friction Coefficient	Formula	Dynamic Loss [m]
Check valve	PCs	1	GS	0.03	0.2	0.95493047		1.9	$K * v^2 / 2g$	0.086510
Flow meter	PCs	1	GS	0.03	0.2	0.95493047		0.024 bar		0.24
Gate valve	PCs	1	GS	0.03	0.2	0.95493047		0.4	$K * v^2 / 2g$	0.018242
Tee	PCs	1	GS	0.03	0.2	0.95493047		0.37	$K * v^2 / 2g$	0.017196
Riser pipe	mts	132	GS	0.03	0.150	2.44462199	120	0.05802579	K * L	7.659400
Collector pipe	mts	1930	DCI	0.12	0.4	0.95493047	130	0.00225446	K * L	5.527936
90^0 bends	PCs	2	GS	0.15	0.4	1.19366308		1.5	$K * v^2 / 2g$	0.217864
Total Dynamic Loss										**13.76**

Duty Point Calculation for Submersible Pump2

The total pumping head is calculated from:

$$H_T = H_S + H_D$$

Where H_T : total head
H_S : static head
H_D : dynamic head loss

Substituting the known values in the above equation:

The total pump head, H_T = 121.4+13.76= 135.16m

Therefore, the pump duty point is selected to be 138m considering allowance of about 3m for losses not considered (Transmission main fittings loss)

Duty Point: 144m^3/h at 138m

Pumping Unit Summary

- No of Pump : 1 Submersible Pump
- Capacity : 144m^3/h or 40l/s
- Head : 138m
- Medium to be pumped : Ground water;
- Pump speed : 2900 rpm

Booster Pumping Station Hydraulic Design:

General

Water extracted from the two boreholes (BH1 and BH2) is pumped to the irrigation where reservoir is proposed to feed the canals by gravity. For this purpose two identical surface pumps with single pump capacity of 70l/s are selected on one duty and one standby arrangement to lift the required water. The 1 + 1 pump arrangement is selected from technical and economical point of view.

Design Data

- Reservoir inlet level = 2322masl
- Minimum Water level = 2219.86masl
- Rising Pipe diameter = DN200 DCI
- Rising Pipe Length DN200 = 620m
- Discharge = 70l/s
- Static head = 102.14m (2322-2219.86)

Determination of Suction Pipe Diameter

Assuming the flow velocity inside the suction pipe to be limited to 1m/s and applying the equation:

$$Q = V * A$$

Where Q : flow rate [m^3/s]
V : flow velocity [m/s]
A : cross-section area [m^2] = $\frac{\pi D^2}{4}$

$$D = \sqrt{\frac{Q*4}{V*3.14}}$$

$$D = \sqrt{\frac{0.07*4}{1*3.14}} = 0.29862\text{m} = 0.29862\text{m} * 1000 = 298.6\text{mm}$$

Take the next higher standard pipe size as suction pipe **DN350.**

Determination of Discharge Pipe Diameter

Assuming the flow velocity inside the discharge pipe to be limited to 2.5m/s and applying the equation:

Q = V * A

Where Q : flow rate [m^3/s]

V : flow velocity [m/s]

A : cross-section area [m^2] = $\frac{\pi D^2}{4}$

$$D = \sqrt{\frac{Q*4}{V*3.14}}$$

Substituting the known values in the above equation

$$D = \sqrt{\frac{0.07*4}{2.5*3.14}} = 0.188862m = 0.188862m * 1000 = 188.86mm$$

the pump discharge pipe diameter is selected to be **DN200.**

Determination of actual velocity for suction and discharge pipes V_s and V_d respectively.

Applying the equation:

Q = V * A

Where Q : flow rate [m^3/s]

V : flow velocity [m/s]

A : cross-section area [m^2] = $\pi D^2/4$

V_s = Q / A

= (70/1000*4) /(3.14*0.35*0.35)

= 0.28 /0.38465

= 0.73 m/s is suction velocity

V_d = Q / A

= (70/1000*4) /(3.14*0.2*0.2)

= 0.28 /0.1256

= 2.23 m/s is the discharge velocity

Dynamic head Loss analysis for transmission pipe, suction and discharge pipes & fittings

Using Hazen Williams's formula

$h_f = 10.7 (Q/C)^{1.85} \times L / (D)^{4.87}$ for pipes and

$h_{ff} = k*v^2/2*g$ for valves and fittings

Where C- coefficient of Roughness for GS pipe = 120 & for DCI pipe = 130

D - Pipe diameter (m)

Q - Flow Rate (m^3/s)

h_f - pipe friction loss (m)

L - pipe length (m)

h_{ff} - fitting loss

Substituting the given values in the above equation as shown in the Table 11-3 below, the total calculated pipes and fittings frictional loss is equal to 3.37m.

Table 11-3: Fitting losses

Description	Unit	Qty	Type	Discharge [m^3/s]	Diameter [m]	Velocity [m/s]	Roughness	Friction Coefficient	Formula	Dynamic Loss [m]
Suction										
Strainer	PCs	1	GS	0.07	0.35	0.727934		0.15	$K * v^2 / 2g$	0.004051
Suction pipe	m	30	GS	0.07	0.35	0.727934	120	0.00153538	K * L	0.046061
90^0 bend	PCs	4	GS	0.07	0.35	0.727934		1.5	$K * v^2 / 2g$	0.040511
Gate valve	PCs	1	GS	0.07	0.35	0.727934		0.4	$K * v^2 / 2g$	0.010803
Gate valve	PCs	1	GS	0.07	0.35	0.727934		0.4	$K * v^2 / 2g$	0.010803
Reducer	PCs	1	GS	0.07	0.20	0.727934		0.1	$K * v^2 / 2g$	0.002701
Discharge										
Check valve	PCs	1	GS	0.07	0.20	2.229299		1.9	$K * v^2 / 2g$	0.481273
Gate valve	PCs	1	GS	0.07	0.20	2.229299		0.4	$K * v^2 / 2g$	0.101321
Flow meter	PCs	1	GS	0.07	0.20	2.229299		0.024 bar		0.24
Gate valve	PCs	1	GS	0.07	0.20	2.229299		0.4	$K * v^2 / 2g$	0.101321
90^0 bend	PCs	2	GS	0.07	0.20	2.229299		1.5	$K * v^2 / 2g$	0.379952
Tee	PCs	1	GS	0.07	0.20	2.229299		0.37	$K * v^2 / 2g$	0.093722
Trans. pipe	m	620	DCI	0.07	0.20	2.229299	120	0.00294195	K * L	1.824009
Total Dynamic Loss										**3.37**

Note:- The above fitting and pipe quantities are taken from the layout and mechanical drawings.

Duty Point Calculation

The total pumping head is calculated from:

$$H_T = H_S + H_D$$

Where H_T : Total head

H_S : Static head

H_D : Dynamic Head Loss

Substituting the known values in the above equation: The total pump head,
HT = 102.14+3.365m = 105.505m

Therefore, the pump duty point is selected to be 108m considering allowance of 3m for losses not considered (Transmission main fittings loss).

Duty Point: 252m^3/h at 108m

Pumping Unit Summary

- No of Pumps : 2 Surface Pumps
- Capacity : 252m^3/h each
- Head : 108m
- Medium to be pumped : Ground water
- Pump speed : 1500 rpm

Net Positive Suction Head

The value of NPSHav mainly depends on the site condition, position of pump, suction pipe and suction sump arrangement, and is given by:

$NPSH_{av} = H_{abs} \pm H_s - h_L - H_{vap} - s.f$

Data at booster station:

Atmospheric pressure $P_{atm.}$ = 7.781m at elevation of 2220

Vapour pressure $P_{vap.}$ = 0.576 at a temperature of 35^{0}C

Suction head Hs= 2m above water surface
Head loss, h_l = 0.5m

Safety factor S.f. = 0.6m

Substituting the above values, the available $NPSH_{av}$.= 4.105M

The required NPSH must be less than 4.105m.

11.5 PUMP MOTOR POWER, STARTING METHODS AND CABLE SIZING

The duty pints of the pumps are given in Table 11-4 below:

Table 11-4: Pumps duty points

Item No	Description	Discharge (Q l/s)	Head (Hm)
1	Borehole 1	30	160
2	Borehole 2	40	138
3	Surface pump	70	108

a) Motors Power Determination

Taking the calculated duty point of pumps:-

Pump discharge (Q) and total head (m) & assumed value of overall pump and motor efficiency η, the required pump and motor power in KW can be calculated using the formula:-

$$P_P\ [KW] = \frac{Q*Ht}{102*\eta}$$

Where Q : discharge [l/s]

Ht : total head [m]

η : overall efficiency (assumed as 65%)

Substituting Q and H data in the above formula, the absorbed pump power and the selected motor power are shown in the following Table 11-5.

Table 11-5: Selected motor power

Description	Discharge [l/s]	Head [m]	Pump power [kw]	Selected motor power [kw]
Borehole 1	30	160	72.40	85
Borehole 2	40	138	83.26	98
Surface pump	70	108	114.03	134

The motor power is determined assuming 15-20% safety factor over pump power.

b) Proposed Motor Starting Method

For motors, with a power of more than 7HP or 5.2kw star-delta starting method is proposed. This starting method is widely used to start low and medium power motors and it is easy to manage.

c) Cables Sizing and Selection

Power and control cables are sized and selected based on their current carrying capacity, site conditions and method of installation.

Accordingly, rated motor currents are calculated using the formula shown below and proper size of motor cables are selected from standard cable catalogue with consideration of necessary factors.

$$\textbf{Current (I)} = \frac{\textbf{Power x 1000}}{\sqrt{3}\textbf{xV x Cos}\Phi} \qquad \text{Equation 51}$$

Where:

P = Power

V = Voltage=380V for 3 phase motors

CosΦ = Power factor =0.8

Calculate the current (I) for **borehole 1** pump with the motor power of 85KW as it is shown in the above table

$$\text{Current (I)} = \frac{\text{Power x 1000}}{\sqrt{3}\text{xV x Cos}\phi}$$

$$= \frac{85 \text{ x } 1000}{1.73 \text{ x } 380 \text{ x } 0.8} \quad = 161\text{A}$$

From the cable manufacturer catalogue shown below (Table 11-6) sixth column we get 174A and on the same raw under the title “nominal cross-sectional area of conductor” we get $50mm^2$ and select the corresponding cable size of $3x50mm^2 + 25mm^2$.

Table 11-6: Standard cable sizes

Table A.52-10 (52-C9) – **Current-carrying capacities in amperes for installation methods E, F and G of table A.52-1** (52-B1) – **PVC insulation/Copper conductors**
Conductor temperature: 70 °C/Reference ambient temperature: 30 °C

Nominal cross-sectional area of conductor mm²	Installation methods of table A.52-1						
	Multi-core cables		Single-core cables				
	Two loaded conductors	Three loaded conductors	Two loaded conductors touching	Three loaded conductors trefoil	Three loaded conductors, flat		
					Touching	Spaced	
						Horizontal	Vertical
	Method E	Method E	Method F	Method F	Method F	Method G	Method G
1	2	3	4	5	6	7	8
1,5	22	18,5	–	–	–	–	–
2,5	30	25	–	–	–	–	–
4	40	34	–	–	–	–	–
6	51	43	–	–	–	–	–
10	70	60	–	–	–	–	–
16	94	80	–	–	–	–	–
25	119	101	131	110	114	146	130
35	148	126	162	137	143	181	162
50	180	153	196	167	174	219	197
70	232	196	251	216	225	281	254
95	282	238	304	264	275	341	311
120	328	276	352	308	321	396	362
150	379	319	406	356	372	456	419
185	434	364	463	409	427	521	480
240	514	430	546	485	507	615	569
300	593	497	629	561	587	709	659
400	–	–	754	656	689	852	795
500	–	–	868	749	789	982	920
630	–	–	1 005	855	905	1 138	1 070

NOTE Circular conductors are assumed for sizes up to and including 16 mm². Values for larger sizes relate to shaped conductors and may safely be applied to circular conductors.

Calculate the current (I) for borehole 2 pump with the motor power of 98KW as it is shown in the above table

$$\text{Current (I)} = \frac{\text{Power x 1000}}{\sqrt{3}\text{xV x Cos}\emptyset}$$

$$= \frac{95 \text{ x } 1000}{1.73 \text{ x } 380 \text{ x } 0.8} = 186\text{A}$$

From the Table 11-6 sixth column we get 174A and on the same raw under the title “nominal cross-sectional area of conductor” we get 50mm² and select the corresponding cable size of 3x50mm2+ 25mm2

Calculate the current (I) of the Surface pump with the motor power of 134KW as it is shown in the above table

$$\text{Current (I)} = \frac{\text{Power x 1000}}{\sqrt{3}\text{xV x Cos}\emptyset}$$

$$= \frac{134 \text{ x } 1000}{1.73 \text{ x } 380 \text{ x } 0.8} = 255\text{A}$$

From Table 11-6; sixth column we get 275A and on the same raw under the title “nominal cross-sectional area of conductor” we get 95mm² and select the corresponding cable size of 3x95mm²+ 50mm².

12 POWER UNITS

Most irrigation pumps are powered either with electric motors or diesel engines. In some countries, natural gas, propane, butane and gasoline engines are also used to drive pumps. Wind and solar driven pumps are also used for pumping water, but mostly for human and animal purposes. Manual pumps like treadle pump, rope and washer pump, hand pump etc. are also used as lifting mechanism for small discharge rates and low heads.

In earlier parts of the guide line it is described how to compute the size of the power unit. For centrifugal pumps and turbine pumps up to 20 m deep it is not necessary to compute the energy required to overcome bearing losses in the pump. For turbine pumps that are more than 20 m deep, the manufacturer's literature should be consulted on line shaft bearing losses.

12.1 ELECTRIC MOTORS

For most centrifugal pumps the motors are directly coupled to the pump. This results in the elimination of belt drives and energy loss due to belt slippage, and safety hazards.
Most centrifugal pumps used in Eastern and Southern Africa are coupled to the motor shaft through a flexible coupling.

In the past it was common practice to overload motors by 10-15% above the rated output without encountering problems. However, because of the materials currently used, motors can no longer stand this overloading. Therefore, they should be sized to the needed and projected future output. For sustained use of a motor at more than 1100 m altitude or at temperatures above 37°C de-rating may be necessary. Manufacturer's literature should be consulted for the necessary de-rating. An example of the de-rating of diesel engines is shown in the following section.

Electric motor
If the brake horsepower of a centrifugal pump exceeds the safe operating load of the motor, the motor may be damaged or burned out. The shape of the power characteristics curve and the system head characteristics will determine if pump operation may exceed the safe loading of its electric motor. Attention must also be paid to the shape of the speed -torque curve of the motor and the voltage supply of the power system. Any supply voltage lower than the rated voltage would be unable to take the load. The motor should be able to take the loads over the entire range of operating conditions.

Power factor
In AC circuits, the power factor is the ratio of the real power that is used to do work and the apparent power that is supplied to the circuit. The power factor can get values in the range from 0 to 1. When all the power is reactive power with no real power (usually inductive load), the power factor is 0. When all the power is real power with no reactive power (resistive load), the power factor is 1.

a) Direct current motor
Direct current (dc) motors are available in three types viz. shunt sound, series wound and compound wound. But dc motors are not very suitable for centrifugal pumps as the full load speed of DC, motors may vary by 5 to 7.5% from the rated speed. A dc motor to be used for a centrifugal pump has therefore to be specially ordered to match the pump speed. Such use is possible only

for special requirements. A DC motor is useful in situation where duty of the pump changes periodically.

b) Alternating current motor

Single-phase motors are normally used as directly coupled with small centrifugal pumps. Uses of single phased motors are restricted up to 7.5 HP applications. Motors above 7.5Hp are 380 volts, 3 phase 50 cycles/sec. 2) Three Phase Motors Squirrel cage motors are simplest type of poly-phase motors and are most widely used for irrigation pumps. These motors have a primary winding (stator) and a secondary squirrel cage winding (rotor) which takes power from the primary winding by transformer action. Centrifugal pumps do not require motors with high starting torque.

Hence normal torque, normal starting current (National Electrical Manufacturers Association (NEMA) class A) or normal torque low-starting current (NEMA class B) motors are adequate. Ac motors run at a constant speed. The Standard rpm ranges from 1450 to 1500. Other motors are wound rotor motors and synchronous motors.

Motors can also classified into horizontal or vertical depending upon their housing design, slow speed or high speed, constant speed or Multi speed motors depending upon their speed characteristics. Motors can be run in either direction by simply changing any two leads of a 3 phase motor.

C) Feature of electric motors

- Long life
- Low operating and maintenance cost
- Dependable service
- Easy operation
- Practically automatic
- Ineffective when power supply is not dependable
- Extension of power lines and installation of transformers etc for a new scheme cost extra and cause inordinate delay
- Standard motor sizes are 3, 5, 7.5, 10, 15,20,25,30,40,50,60KW
- Closed motors have advantage over open motors in the sense that they are protected from dust, drip and rodents.
- Motors have a built in service factor of 10 to 15% at air temperature 40 degree centigrade or less. Present bay motors can withstand up to 70 degree centigrade without much damage.
- Present day motors are smaller in size than earlier models due to the use of high quality insulation.
- Sometimes voltages in all the three phase may not be equal. A 3.5 % imbalance may cause about 25% increase in temperature.
- Similarly a reduction of 10% voltage in each line will cause temperature increase of about 16%.
- Motors are air cooled by the surrounding air and are designed to operate safe up to an altitude of 1000 m above sea level, for altitudes above 1000 m de-rating curves are used. It should be observed that, for higher altitude operations, special motor is therefore, necessary.
- Standard motors are designed to operate satisfactorily at 5% reduction of rated frequency.
- The normal efficiency of an electric motor ranges from 70% to 95 %

d) Starting methods of Motors

Motors require **two to three times more current** to start than when it is running of full load.
Motor starters are classified in various ways depending on their method of:

- Control (automatic or non- automatic);
- Operation (manual, electromagnetic, motor operated, pneumatic, and electro pneumatic;
- Interruption Mode (air-break, oil-break);
- Degree of protection provided by the enclosure), etc (see all parts of IS 8544 or equivalent standards for details)

The most usual way and simplest method of starting a squirrel cage motor is means of direct on line starting. A push button starters or stroke switches are normally applied for direct on line starting. If the conditions are such that the starting current must be limited, the star-delta starting method should be employed. The starting current, and also the starting torque, are then limited to about one third of the values obtained with direct on line starting. This lower starting torque is quite adequate in many cases, since the driven machine during the start is often unloaded or running under a relatively low load. The use of an auto-transformer starter has the advantage that the starting torque can be more closely matched with the actual requirements through a suitable choice of the turn ration.

Starter of electric motors also contain built in circuit breaker as a protective device. Good quality starter therefore ensures motor safety. Motor may also get damaged if there is no current in one of the three phases due to faulty connection. To overcome this single phasing, the control of motors is used to prevent the motor from over loading and power fluctuation. Controls like time delay fuses are available to start pump automatically after power supply is resumed after a power failure. Suitable alarms, light indicators etc. are available to indicate whether a motor is running.

NOTE:

The choice of starling method depends on the actual rating of the motor, the specific work requirements or application or the duty of the motor, etc. In some countries, for instance in India the following requirement is enforced:

Motors up to 7.5 HP are normally started by Direct on Line (DOL) starter. Motors ranging from **7.5-50 HP are started on "Star-Delta" starter**. As the motor picks up speed the star connection is changed over to delta shaped circuit either automatically or manually. Star and delta means the shape in which the leads gets arranged within the starter by different shifting contacts and the circuit reassemble either a triangular star or a delta. Motors of 50 HP and above are started by using an auto transformer in line. Therefore, users are advised to observe the electrical wiring regulations issued by pertinent national organizations.

e) Installation

In situation where it is difficult to calculate the exact mead correctly and in order to avoid continuous overloading of electric motor, a safely margin over the calculated HP is recommended at the following rate:

- Add 50% for pumps requiring up to 2 HP
- Add 30% for pumps requiring 2-5 HP
- Add 20% for pumps requiring 5-10 HP
- Add 15% for pumps requiring 10- 20 HP
- Add 10% for pumps requiring over 20 HP

In large schemes, instead of having a single motor or having provision for standby motors, parallels operation is more advantageous. The recommended break lip of motors is as below.

- Up to 200 HP: 2 numbers
- 200 -300 HP: 3 numbers
- Above 300 HP: 4 numbers

Decision on the number of motors, however, is guided more by the required capacity of each pump, the capacity of power supply and availability of motors.

f) Electric connection

Load up to 100 kW can be handled by Low Tension (LT) lines of 380 V. For higher load, High-Tension (HT) line is required.

- Selection of electric motor to the rated power of pumps, proper installations and use of suitable controls for the operation and protection of the motor are the core factors that could affect proper installations.
- Wiring of motors shall follow the technical regulations issued by the
- Ethiopian electric agency, Ethiopian Electric Power Corporation, or Ministry of Works & Urban Development; or the Ethiopian Standards issued by the Quality and Standards Authority of Ethiopia. Except the standards, the wiring regulations issued by the Ethiopian Electric Agency shall have precedence and prevalence over those of the remaining regulations.
- The wires to be used shall have sufficient current carrying capacity selected on the maximum allowable voltage drop set ill the wiring regulation.

Alternating current supply of 380 V 50 Hz four wire, three phase system or 220 V 50 Hz single-phase system may be used for lighting of pump station All three phase loads need to be equally divided between all three phases system. The service entrance of the installation shall be grounded. Alternating current supply of 380 V 50 Hz four wire, three phase system or 220 V 50 Hz single-phase system may be used for lighting of pump station All three phase loads need to be equally divided between all three phases system. The service entrance of the installation shall be grounded.

g) Accessories

The following accessories are used in electrical installation

- Energy meters,
- Volt meters and ammeters,
- Indicator lamps,
- Main switch fuses,
- Starters,
- For single-phase motor a hard operated switch is used for controlling the starting of the motor, and
- Starters are usually designed and provided with overload (magnetic and thermal) adjustable or fixed releases.

12.2 DIESEL ENGINES

As a rule, petrol engines drive very small pumps. For most irrigation conditions, the diesel engine has gained popularity. It is more robust, requires less maintenance and has lower overall operation and maintenance costs.

The common types of diesel engines are:

- Horizontal engines: These are less in use in the present days.
- Lister type: These are slow speed vertical engines.
- High-speed vertical engine: These are most commonly used these days.

Diesel engines for agriculture purpose are available, for instance, in 5, 6, 7, 8, 9, 10, 12, 14 HP range. Diesel engines may be either air-cooled or water-cooled. Air-cooled engines are available normally up to 25 HP. They are improved over water-cooled engines. They are less messy than the water-cooled engines where the coolant (water) is to be properly disposed off to prevent the site turning muddy. Water-cooled engines however are made in all sizes. The user or purchaser shall specify the standard reference operating or testing conditions for internal combustion engines indicated below. For such proposes one can refer to ISO or equivalent standards.

- Mean barometric pressure
- Atmospheric temperature (air temperature)
- Relative humidity
- Charge air coolant temperature
- Intake air depression and exhaust back pressure
- Auxiliaries
- Fuel
- Air inlet temperature
- Fuel supply system
- Fuel inspection equipment
- Governor
- Lubrication system

Most literature on engines uses English units of measurement. To convert kilowatts to horsepower a conversion factor of 1.34 can be applied. Horsepower versus speed curves (Figure 12-1) illustrate how output power increases with engine speed. However, there is a particular speed at which the engine efficiency is highest.

This is the point at which the selected engine should operate. The continuous rated curve indicates the safest continuous duty at which the engine can be operated. Care should be taken to use the continuous rated output curve and not the intermittent output curve.

Manufacturer's curves are calculated for operating conditions at sea level and below 30°C. It is therefore necessary to de-rate the engines for different altitudes and temperatures where the operating conditions are different. According to Pair et al. (1983), de-rating is approximately 1% per 100 m increase in altitude and 1% per 5.6°C increase in air temperature from the published maximum output horsepower curve. On the top of that, an additional 5-10% for reserve should be deducted. If the continuous output curves are used, only the 5-10% deduction is applied.

Example: What will be the output of a diesel engine with a speed of 2600 rpm at 2000 m altitude and a temperature of 35°C?
Referring to Figure 12-1, the maximum output at 2600 rpm, by interpolation, would be around 114 hp, which falls outside the limits of this curve. By applying the above rule for 2000 m altitude and 35°C, a deduction of 20% should be applied for elevation and 1% for temperature. An additional 10% should be applied for reserve. Therefore, the total deduction should be 114 x 0.31 = 35.3 HP, resulting in an output of 78.7 HP (= 114 - 35.3).
If we apply the 10% deduction on the continuous rating curve then the output will be 80 - 8 = 72 HP. This is a more conservative approach.

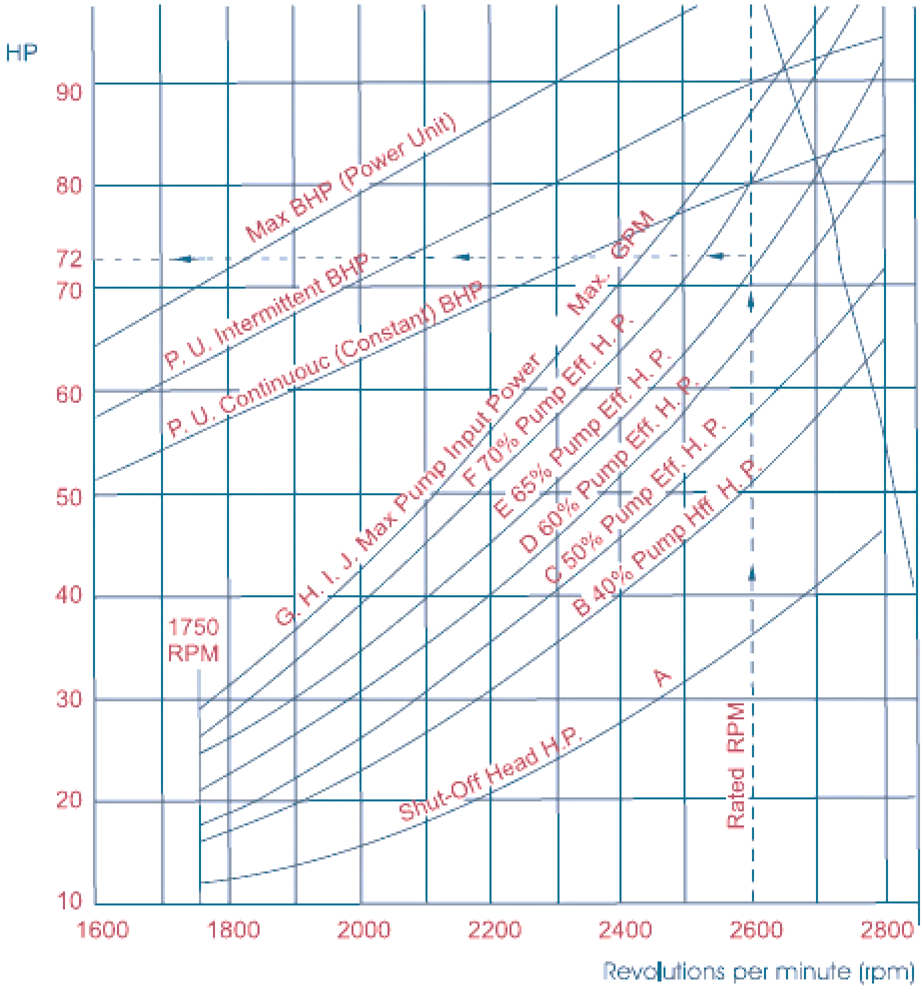


Figure 12-1: Rating curves for engine (Source: Irrigation Association, 1983)

12.3 POWER TRANSMISSION

There are four types of transmission usually applied to irrigation pumps: direct coupling, flat belt, V-belt and gear. Direct coupling generally implies negligible or no loss of power. The loss of power through flat belt varies from 3-20%. Transmission losses for V-belt and gear drive, as a rule, do not exceed 5%.

Referring to the example, if we use direct coupling of engine to pump, the HP would remain 72 HP. If we use gear or V-belt drive then the power available to the pump would be 68.4 HP (0.95 x 72). This should satisfy the input power requirements of the pump as calculated using Equation 3 and multiplying the result by 1.34 to convert to horse power gear or V-belt drive then the power available to the pump would be 68.4 HP (0.95 x 72). This should satisfy the input power requirements.

12.4 SOLAR POWER

A solar powered pump works like any other available and commonly used water pumps.
The main difference is that solar powered pumps run on solar energy and do not require any fuel (diesel, kerosene, gas, etc.) or external source of electricity (from an electricity company) to deliver water.
A solar powered pumping system has consists the set consists of the following parts:

Solar array: The solar array is a set of solar modules which are to be connected in series and possibly strings of modules connected in parallel to get the required power to operate the pump.

Controller: The controller is an electronic device that matches the power output from the solar array to the pump motor and regulates the operation of the pump according to the input from the solar array.

Pump: The pump comprises of the motor which drives the movement (prime mover) and the pump impeller which moves the water under pressure. Additionally, the solar powered pump set might include accessories like cabling and fittings, a battery, and an inverter.

Solar pumps are used for both surface pumps and submersible pumps. The most important criteria for choosing between a surface and a submersible pump are the groundwater level and the type of water source. If the well is a bore well and total head (suction + delivery) is greater than 10-15 meters, a submersible pump should be used. If the water source is an open well, pond, canal, etc. then a surface pump is more feasible. As a thumb rule, a surface pump can be used if the water level is less than 10 meters; for levels more than 10 meters, the installation of a submersible pump is required. Currently the solar pumps are widely applied to areas that require small energy to run the system.

Typical Technology options and accessories: Thanks to technology now a days invented smart solution-PSk2 Smart Solution introduces hybrid power solutions to the PSk2 system. Using a combination of the SmartPSUk2 and Smart Start the PSk2 system can be powered by solar, by AC power from a generator or grid supply or with a mix of solar and AC power. Thus;

I. The PSk2 controls the Smart Solution components to deliver water depending on power available, time or specific water requirements.
II. To connect PSk2 to an AC 3-phase grid power supply then a SmartPSUk2 is used

Solar-diesel system sizing considerations: The SmartPSUk2 is able to switch between solar and diesel as well as blend both power sources, e.g. to obtain a desired flow rate or pressure irrespective of sunshine. It can operate on diesel generators with a constant power rating higher, equal or smaller than the power rating of the pump motor. The constant power rating of the diesel generator should be selected considering the desired flow or pressure.

SmartSolution supports the following diesel generator specifications:

380/ 400/ 415 V
50/ 60 Hz
Remote start function (if use of SmartStart is desired)

To connect PSk2 to an AC 3-phase generator then a SmartPSUk2 and SmartStart is used.

Sample calculation for solar energy

1. GF 270

The GF is a polycrystalline solar module. The module is equipped with MC plugs for easy connection and comes as pieces per pallet without individual packing. It must be mounted on a support structure tilted at an angle ensuring optimum utilization of the solar energy.it can be installed at Range of ambient temperature 233-358K which is suitable for our country

Electrical data module
Maximum power point voltage = 31.6V
Open circuit voltage = 38.4V
Max power point current = 8.76 A
Module shortcut current = 9.11 A
Maximum power output 270W

To get the required power for a single pump we need 148 solar modules adopt 150 which is installed as Number of solar modules in series: 19, in parallel: 10 this will result in

Solar array rated power: 51.3 kW
Solar array rated volts: 600.4 V

2. RSI 3x380V IP66 37 kW 72A

Renewable Solar Inverter RSI is a sourced off-grid solar inverter to enable us to expand the family of solar submersible pumping systems. RSI is configured specifically to be compatible with 3 phase 400V SP submersible pumps. With the built-in MPPT and various protection software, it delivers an efficient and reliable renewable system solution as shown in table below.

Table 12-1 Electrical data RSI 3x380V IP66 37 kW 72A

Electrical data	Qty	Qty
Rated power-P2		KW
Phase:		
Enclosure class (IEC 34-5)		
Rated voltage output AC		V
Voltage input DC:		V
Rated current output AC		A
Udc		V

Summary of products

Table 12-2 Summary of main products for solar power supply

No	Product	Unit	Qty
1	GF 270 solar modules	pcs	
2	RSI 3x380V IP66 37 kW 72A	pcs	
3	surge protection, DC	pcs	
4	circuit breaker, DC	pcs	
5	Panel Support steel Structure	ls	
6	Cables and accessory	ls	
7	Transport service and installation	ls	
	Total		

Note: the installation of the panel requires large area than other power system

13 CONTROL PANELS AND OTHER UNITS

13.1 CONTROL PANELS

Indoor or outdoor control panels will have provision for energy meter, main switch, starter, single phase preventer, capacitor, etc. some of the components embedded in a control panel are as follows:-

a) Switches: A main switch of adequate capacity to disconnect power supply shall be provided after the meter. This will enable to disconnect the service immediately in case of any emergency or for maintenance purpose.
b) Starter: Starter with over load relay is provided to start and stop the motor and to protect it against any over load. Over load may be either electrical or mechanical.
c) Capacitor:- Installation of capacitor of suitable rating in the motor circuit will improve the power factor and reduce energy consumption. The running cost of the motor will also be reduced. The recommended capacitor ratings can be given by control panel manufacturer.
d) Single phase preventer: - In three phase circuit, three fuses are provided (1 for each phase). If in any one phase fuses were to blow or any one phase is disconnected form service during running of the motor, the motor keeps running drawing excess current from the two lines and hence causing damage to the motor. If a single phase preventer is provided in the circuit, it will sense the operating coil and trips the starter and protects the motor from burning.
e) Voltmeter and three phase ammeters: - These meters will indicate whether system voltage is within permissible range for the motor or to know whether motor is drawing current equally on all three phases. The functioning of Voltmeter is very important, voltage being low in villages damaging the motors.

13.2 TRANSFORMER AND STANDBY GENERATOR SIZING

13.2.1 Transformer sizing

Transformers for the station are sized taking in to consideration motor starting currents and other station loads. Generally transformers have high impact withstanding capability against induction load starting/ motor loads.

To determine transformer capacity:-

- Add up all site electrical loads
- Multiply the total power by 1.25 (for big plant)
- Consider starting of the biggest motor (if motors are two or less)

Example: In a borehole site there is a power demand of 85KW for motor; 3KW lighting purpose and other loads about 1.5KW determine the site Transformer capacity?

Table 13-1: Transformer size

Name	Motor capacity[kw]	Motor capacity at start[kw]	Other station loads[kw]	Total station loads[kw]	Selected transformer capacity[KVA]
Pump station	85	147	3+1.5	151	190

Example: Electrical surface pumps are installed to pump in different farm lands. 3 pumps 20KW each; 4 pumps 35KW each and 2 pumps 50KW each and 1 dewatering pump of 10KW. Calculate the transformer capacity assuming 4KW of site utility loads?

Table 13-2: Total power demand

Name	Motor capacity[kw]	Remark]
Type 1	3*20 = 60	
Type 2	4*35 = 140	
Type 3	2*50 = 100	
Dewatering pump	1*10 = 10	
Other	4	
Total	314	
Total of 1.25%	392.5	25% for reserve
Transformer	490KVA	KVA=KW/0.8

13.2.2 Standby generator sizing

In designing of the diesel generator the following assumptions are considered:

1. The proposed motor starting method is star-delta starting system.
2. Typical starting power for star-delta starting system is taken as 2.5 x rated full load
3. Considering sequential starting of motors, and by starting the biggest motor first and other small loads later, the stand by generator is sized using the following empirical formula:-

$$\mathbf{S = (\sum P + 2.5(P_m))} \quad \text{Equation 52}$$

Where,

S= Stand-by generator size in KVA.

$\sum P$ - is the sum of other loads,

P_m – is the biggest motor load

14 ECONOMICS OF PUMP IRRIGATION SYSTEM

14.1 GENERAL

Basic laws of economics are as much applicable to pump irrigation system as in any other activities. Production of a particular crop depends upon a number of factors, the most important of which are:

- Soil condition;
- Meteorological condition; and
- Application of inputs in which water comprises single most important input for crop production.

Economics of Pump irrigation system therefore centers on studying the crop yield in relation to the application of water at a fairly reasonable cost that should bring a direct benefit in the form of revenue. Cost is that expenses incurred directly and indirectly in applying water to the crop, which is of two types - one is the initial or capital cost and the other is operating cost that comprises of both the fixed and variable cost.

14.1.1 Planning for pump irrigation system

Planning for feasibility:
Pump irrigation system planning involves socio economic consideration as to the feasible course of investment and other aspects of social developments.
When planning factors that deserve consideration are discussed below:-

a) Dependability
Dependability of sources and Source types, period of availability (with a minimum quantity of supply)
Quality of water (whether these is an adverse effect on crops, salinity in upstream course, whether these is an adverse effect if there are industrial effluents, etc)

b) Location of Command Area
Location of command area and distance of delivery point from the source decides the cost of the pump irrigation system. Initial capital costs with too high lifting cost and long pipelines may be considered' acceptable but the high operating cost due to the use of higher horsepower may completely upset the economic viability of LIS scheme.

c) Viability of pump irrigation system
Viability of pump irrigation system shall be decided either on the basis of unit cost or socio-economic consideration (as applicable, to that specific scheme.)

d) Shape of command area
Emphasis should be given to the design and distribution layout (whether gravity flow or other type) depends on the topography of the irrigable area.

e) Social conditions
Factors such as: soil thickness, texture and water holding capacity fertility, shall be supported by chemical analysis.

f) Agro- climatologically condition
Data on rainfall reliability, extent of precipitation, runoff, etc is of utmost importance. Further analysis to determine: effective rainfall and potential evapotranspiration are required.

g) Copping pattern
Study of existing and proposed cropping pattern needs to be considered with respect to the cost benefit analysis (Agronomy GL)

h) Source Sharing
Number and type of schemes operating in the up and down stream sides of the sources (river, canal, stream, etc) shall be considered so as not to disturb the existing water sharing practice.

i) Source of energy
The choice of prime movers and the source of energy for the prime movers (diesel and electric maintenance practice) ought to be studied for economic viability.

j) Land use
The ownership of the land and use pattern to be affected (grazing area, etc) should be Studied.

k) Availability of infrastructure or material
Availability of standard construction material pumping machineries, maintenance facilities, source of energy, agricultural input, etc must be looked into.

l) Social demands
Pump irrigation system that does not meet technical or economic viability maybe acceptable from the point of view of social desirability, in such cases, subsidy under appreciable policy or financing scheme ought to be assessed.

14.1.2 Alternatives in designing pump irrigation systems

Designing pump irrigation system involves selection of possible alternatives and preparation of specific details concerning the details of; Development of water source; Assessment on pumping equipment; Decision on pumping equipment and distribution layout; and Preparation of cost estimates.

What is sought from a good designed lift irrigation system is not only least unit cost of construction but also it must ensure smooth and trouble free operation over the designed project life. Thus, the design engineer be adequately familiar with the aspects of; Development of water sources; Water requirement of crops Selection of pumps and pipes Installation, operation and maintenance, and Cost and benefit analysis of techniques of such schemes; Intake structures; Pump house; etc.

14.2 COST CONSIDERATIONS

14.2.1 Capital cost

All costs involved during the implementation of the scheme. Capital cost is ascertained easily after the scheme is implemented. Once the technical design of the scheme is finalized, a rough estimate of the capital cost could be made by following standard methods of cost estimates. Cost in a pump irrigation system is generally incurred in the following items:

- Water intake structure (tube well, open well, jack well, intake well etc)
- Pump and prime mover with suction and delivery pipes
- Pump house
- Field level distribution system
- Electric installations

Notes:
1. Realistic costs to the different components are assigned based basically on actual market situation.
2. Experienced person shall do standard methods of cost estimation.

14.2.2 Economics of cost of pumping

Estimates of cost of pumping are necessary to compare the relative costs of different types of pumping installations and evaluate the economics of irrigation. Cost of pumping includes fixed costs and operating or variable costs.

a) Fixed Cost
Operating cost has both the fixed and variable cost components. Whether the scheme is operational or not a scheme would entail certain fixed costs.
The annual fixed costs are:

Interest;
Interest is calculated on the average value of the installation at the prevailing interest rate. Annual interest cost = (Value of installation minus salvage value) multiplied by interest rate and divided by a factor of two = (Capital or investment cost – salvage value) x interest rate)/ 2

- • Taxes and insurances;
- • Fixed payment for electric installation;
- • Depreciation of equipment or plant is the loss in the value of the pumping plant due to operation or age.

Table 14-1 Useful life of pumping equipment

Item	Useful life in years or hours
Centrifugal pump	16 years or 32000 hours
Turbine pump bowl assembly	8 years or 16000 hours
Turbine pump column assembly	16 years or 32000 hours
Diesel engine	14 years or 28000 hours
Electric motor	25 years or 50000 hours
Mild steel pipe	25 years
Tube well casing	20 years
Power transmission:	
Gear head	15 years or 30000 hours
V-belt	3 years or 6000 hours
Flat belt, rubber and fabric	5 years or 10000 hours
Flat belt, leather	10 years 20000 hours

b) Variable or Operating Cost
The direct cost incurred in pumping is a direct function of available water and the type of prime mover used e.g. whether an electric motor or a diesel engine. Higher cost implies that higher quantity of pumped water or variable cost. The cost of pumping could be obtained either = (Original cost-salvage value)/Useful life in years In other words depreciation of equipment is calculated on the basis of the effective life span of the equipment/ plant. Annual depreciation in the

simplest method (straight line depreciation) is obtained by dividing the cost of the equipment with its estimated life span. The useful lives of different items used in pump irrigation system are different. Depreciation of each item is therefore obtained separately as presented in Table 14-1.

Energy cost of diesel engine is two to three times the energy cost of electrical pumps. The energy cost per KW-hour can be taken 0.80-1.00ETbirr under current price. The cost of diesel can be estimated based on the following fuel consumption table by multiplying by current price of gasoline. HP generated by a diesel engine increases proportionally with its speed (rpm). Engines consuming more than 200 gram/BHP-hr should normally be avoided. To convert gram to liter, specific gravity of gasoline can be taken 0.835. Consumption of lubricating oil should normally be 1 % of diesel consumption but may however be accepted as much as 3-4% depending upon the type of engines. The recommended specific fuel consumption (SFC) for standard diesel engine at different speed is presented in table 14-2.

Table 14-2 Recommended specific fuel consumption of diesel engine

Rated engine speed (RPM)	Maximum gram/BHP-hr	SFC gram/KW-hr
Up to 500	245	332.50
500-1000	203	275.50
1000-2000	185	251.75
above 2000	227	308.75

Example: Calculate the brake horse power of centrifugal pump installed to discharge 80l/s silt free water to 30m total lift head. Consider if pump coupled to Diesel engine and electric motor. Calculate the seasonal energy cost for both cases if seasonal pumping hours are 2000 and RPM is 1800 consider specific fuel consumption 250gram/KW-hr.

Solution: The brake horse power is computed and found to be =49KW or **66hp** for pump coupled to diesel engine and brake horse power (KW=40KW or **54hp** for pump coupled to electric motor.

Electric motor:
Seasonal Cost of energy=40KW *2000hr*0.90ETB/KW-hr =72,000 ETB
Diesel Engine:
Seasonal Cost of energy = KW x SFC x hours/ Sp. gravity x 1000 x cost per liter of fuel
Seasonal Cost of energy =[49*250*2000/(0.835*1000)]*20ETB=586,826ETB.

c) Discharge Capacity

The pump discharge should meet the peak demand of water for the selected cropping pattern requirement (which differs as the area and crop types, rotation period (intervals between successive irrigation of a crop) and duration of pumping. The discharge capacity of the pump under consideration would have direct effect on cost of pumping.

d) Power Requirements and Efficiency

To determine the horse power rating of the prime mover, it is necessary to know the efficiency of the pump, the type of drive, the type of power unit, the head under which the pump will operate and losses (in the piping system, pump, etc). Pump power and efficiency are other significant components influencing the cost of pumping.

d) Profitability

Profitability study should be undertaken both at the time of planning the project with estimated cost and anticipated benefit and also verify the same after the actual production is started as certain amount of capital is invested in a pump irrigation system as a means of production. If the rate of return received from a Pump irrigation system is found to be less than the existing interest rate then naturally a thorough economic review of the entire activity would be

15 TECHNICAL SPECIFICATION FOR PUMPING FACILITIES

The technical specification shall be descriptive and having full detail of the equipment that we desire to buy. Let us see below typical sample technical specification.

15.1 MECHANICAL

The following details relating to the Mechanical Plant are to be provided.

- General arrangement drawings showing equipment positions and pipe work routings
- Equipment, valve and pipe work schedules. These should include the overall size and weight of individual items as broken down for delivery and installation
- Schedule of painting and protection systems for all plant
- Process and Instrumentation diagrams, detailing in full pipeline diameters, instrument functions, I/O requirements, control panel interfaces etc.
- General arrangements of individual items of plant. These drawings must be suitably detailed to enable the Engineer to satisfy himself that the Plant is well designed, complies with the Specification and is suitable for the intended purpose.
- Sectional drawings showing component parts, associated materials and sufficient details to determine that each item of plant can be operated, maintained, dismantled, assembled, and adjusted in a safe and efficient manner
- Detailed manufacturing drawings (workshop drawings) for each item of plant
- Hydraulic calculations
- Plant sizing calculations
- Pump and motor performance curves for main pump sets
- A list of the special tools that are to be provided for the operation of the Permanent Works

15.2 ELECTRICAL POWER TECHNICAL SPECIFICATION

The following details relating to the Power Supply are to be provided

- Equipment location plan;
- Single line diagrams for the site electrical distribution;
- A load schedule calculating the maximum demand for each transformer and bus bar section.
- Drive schedule;
- Switchgear/MCC/control panel general arrangement drawings (internal and external), including fixing and mounting details in plan and section;
- Cable routing diagrams;
- Circuit schematics providing full details of the proposed individual power and control circuits. Full rating details of all fuses, overloads, CTs, VTs, contactors, relays etc. to be shown. All relay contacts used shall be fully cross referenced within each schematic using an approved grid and table reference system;
- Cable schedules;

Power system study comprising:

- Cable calculations for power cables (cable calculations generated by proprietary computer software must be fully substantiated manually i.e a sample of the manual calculations shall also be submitted).
- Full system calculations detailing worst case voltage drops at motor terminals and transformer secondary terminals when operating on the mains supply and the standby generator supply.

- Co-ordinated protection study, incorporating full details of all fault levels, graded protection curves and calculations.
- Protection relay setting schedule
- Emergency shutdown schedule, detailing location of all emergency stop pushbuttons and their interfaces with proprietary shutdown relays and associated plant.

15.3 IRRIGATION PUMP SETS

Pumps shall be multistage centrifugal ring section type and shall be long coupled with its electric motor and mounted horizontally on a fabricated steel base plate.

The pump shall have both suction and delivery radial nozzles with rolling element bearings on both the drive and suction side. The orientation of both the suction and delivery nozzles shall suit the layout as shown on the Tender Drawings.

The first stage shall have a special suction impeller designed to minimise the required NPSH.
Shaft seals shall be provided with un-cooled gland packing or mechanical shaft seal and shaft protection sleeves made of stainless steel.

Casing wear rings shall be provided that is easily replaceable. Pumps shall be fitted with pressure gauges, complete with valves, pipe and fittings, on both discharge and suction flanges.

15.3.1 Irrigation pump motor sizing and de-rating

The motor sizing shall take account of maximum possible run out operation, de-rating due to both the high ambient temperatures in the pumping station and the high altitude of the pumping stations.

The maximum ‘specified run out’ absorbed power shall be the absorbed power when only one pump is running under the system curve condition of minimum static head.

The sizing shall be calculated as follows:

Maximum ‘specified run out’ absorbed power	P kW
Add a 10% margin	(P+10%) kW
Factor up by 12% for altitude de-rating	(P+10%) x 1.12 kW
Factor up by 4% for temperature de-rating	(P+10%) x 1.04 kW

Hence the selected motor size must equal or be greater than 1.26P kW

15.3.2 Irrigation pump materials

Materials of construction shall be selected with due regard to the water being pumped and the risk of corrosion, cavitation, and metal-to-metal galling that can occur within a pump in all its modes of operation. The following table outlines the general material construction of the pump.

Table 15-1: Material of construction for pump parts

Multistage Centrifugal Ring Section Type	
Pump Parts	**Material**
Suction Casing	Cast iron
Inter-stage casing	Cast iron
Discharge casing	Cast iron
Diffuser	Cast iron
Bearing Housing	Cast iron
Stuffing box Housing	Cast iron
Shaft	Stainless steel
Impeller	Cast bronze or zinc free bronze
Casing and diffuser wear rings	Bronze or stainless steel
Shaft sleeve	Chrome nickel steel
Shaft protecting sleeve	Chrome nickel steel
Gland packing	Teflon + graphite
Mechanical shaft seal	Standard material

15.3.3 Supply of irrigation pump set technical information

The following technical Information, in the general specification is required.

(i) Motor de-rating Calculations

(ii) A set of curves showing the system curves and all the duty pump H/Q curves operating in parallel.

(iii) A set of combined pump set performance curves which shall include:

- The standard head versus flow over the complete working range for the duty impeller diameter and both the maximum and minimum impeller diameters. (Impeller diameters shall be specified)
- The pump efficiency
- The NPSH required
- The pump absorbed power
- The motor efficiency
- The motor input power

The guaranteed duty point and all other specified duty points shall be shown on these curves.

15.3.4 Factory performance tests of irrigation pump sets

All pumps shall be tested individually in accordance with standard spec BS EN ISO 9906 Grade 1 using clean water.

The pump performance guarantee duty point shall relate to the flow rate, the total head and the efficiency of the pump.

The tests shall be conducted as follows:

One pump set of each pump type shall be subject to a full test as follows:

- Pump sets shall be tested at their guaranteed duty point, closed valve, the specified run out point, pump maximum run out point, maximum permissible head and five other points on the performance curve to be determined by the Engineer.

- Temperature sensors shall be connected to all bearings and motor windings and the temperatures recorded throughout the test
- Vibration of the pump sets shall be tested in accordance with the American Hydraulic Institute Standards (in range of 10 – 1000 Hz).
- The pump sets shall be subjected to a noise test and the maximum 'A' noise level shall be not more than 90 dB at 1 metre from the surface of the pump set.
- All remaining pump sets shall be subject to an abbreviated test as follows:
- Pump sets shall be tested at their duty point, closed valve, specified run out point, maximum permissible head and five other points on the performance curve to be determined by the Engineer

A full NPSH test shall be conducted on one pump set of each type at each pumping station.
All the above pump tests shall be carried out with their electric motors and assembled on their base plate.

Tests shall provide information for performance curves to be drawn for: head/quantity, efficiency/quantity, power absorbed/quantity and net positive suction head/quantity. All the specified duty points in the Technical Schedules shall be shown on the performance curves.

All pump sets shall operate at their guaranteed design points within the 'acceptance' tolerances for flow rate and total head laid down in BS EN ISO 9906. The pump sets will not be accepted if they are outside the accepted tolerances.

Vibration of the pump sets or any point inside or outside the bearings shall be no more than 0.71 mm/s and the vibration amplitude (crest value) shall be no more than 0.05 mm when the pump sets run normally.

Pump casings shall be subject to a pressure test at 1.5 times the pressure obtained with the delivery valve closed. The positive suction head shall be taken into account in determining this pressure.

Pump tests in sub paragraphs (a) and (c) shall be witnessed by the Engineer or his nominated inspection agent.

15.4 IRRIGATION PUMPING STATION STANDBY GENERATING PLANT

15.4.1 General

The supply and installation of the standby power diesel generating plant, must be included together with all other services, switchgear, cabinets, cables etc., as specified below. The requirements shall include for the manufacture, supply and complete installation of all materials, equipment, plant and components necessary for the provision of fully operational and functioning standby power diesel plant, associated switchgear and services, as specified. The diesel plant and equipment shall include:

- Diesel generating plant
- Fuel tanks and pipe work
- Switchgear
- Cables
- Earthing
- All accessories and other necessary items

The compound requires plant and associated equipment to be to the highest possible standard and requiring the minimum of maintenance. The Contractor shall therefore ensure that the equipment offered meets all relevant British or to alternative widely accepted Standards and is of a type and manufacture such that spares are readily available for immediate and long term use.

The diesel plant, while providing power source, shall be continuously rated and of a type suitable for continuous operation: The sets shall be rated for the pump and auxiliary duties specified together with all starting currents, surges and the like, due allowance being made for the altitude and climate conditions specified.

It is intended that the plant is sized/ rated to cater for the total electrical load at compound and sites specified. The details of the load capabilities of each set shall be clearly identified on the set and as part of the operating manual supplied.

15.4.2 Starting surges and sequence starting

Starting surges are particularly important as certain pump units are large and form the basis of the main power demand. The Contractor shall size/rate the diesel plant to suit the finally agreed starting method for the pumping sets. The effects of light load running must be taken into account, the Contractor including for the provision of any dummy loads necessary. Full details of how the sets have been rated shall be provided.

The start-up of pumping plant is also to be considered by the Contractor. The restarting of the pumping plant must be carried out in a controlled sequence so that the diesel plant can be started against a load within its capability.

Load/supply alarm indicator shall be provided to advise operators that the mains network is not sufficient to meet the compound/site loads. Details shall be agreed.

15.4.3 General description

The diesel engine generators shall be as set out below:

Diesel engine:	
Number required	1 for Booster station,
Minimum power at site	to suit the alternator
Ignition	Compression
Starting medium	Electric 24 volts heavy duty
Starting method	to suit the individual pumps
	Manual push button
Cycle	4 strokes
Cooling	Water cooled with radiator
Lubrication	pressure
Shut down	Manual push button and automatic
	Shutdown safety features
Engine speed	1500 revolution per minute

Alternator:	
Number required	one for one engine
Coupling	Direct
Generating voltage	400 ACV

Phase	Three (3), 4-wire system
Connection	Star with neutral earth point
Pole No.	4
Frequency	50 Hz
Rating Power	250 KVA
Duty	Continuous
Overload	10% for 1 hour in any 12hr period
Excitation	Self

The plant shall be designed and/or selected to operate under the specified conditions prevailing at the site. In this respect it has to be considered for operation in an unheated, minimum ventilated building and dust laden atmosphere. The sizing of the diesel plant has to suit the rating of the alternator which in turn shall be sized and rated to suit the pumping plant, starting surges, equipment loads and auxiliaries at each of the compounds. Full details are to be submitted with the Tender.

The plant and equipment shall be suitable for continuous operation and shall be rated to Suit the prime power requirements of the complete installation including starting currents, Surges and the like.

The ratings specified for the diesel plant are approximate only. It has to be determined precisely what loadings should be used to size the diesel plant, all details being agreed before the commencement of manufacture to avoid difficulties in respect of light loads. Also consider the load situation over a full 24 hour cycle when sizing and considering light loads. All components shall be fully tropicalized and protected against mould growth.

15.4.4 Alternator power output

The alternator power output of the diesel set shall be of sufficient rating to provide power for the items of plant finally identified, particularly the pumping plant. The final rating of the diesel plant shall therefore be determined to suit the pumping plant and auxiliaries, together with de-rating factors for temperature and altitude and an allowance for starting surges and transient voltage dip associated with the starting of the large pump sets. Detailed load analysis calculations shall be done, the voltage dip being to a maximum of 15% on application of the impact load when the largest pump set is started.

It shall be ensured that the plant offered and supplied is properly and adequately ventilated and suitable for the site operating conditions. In addition, the plant shall be sized and rated to suit the load demand, altitude and temperature, the necessary de-rating factors being taken into account to suit the plant characteristics.

The final loads associated with the pumping plant must be determined considering starting characteristics, plant load requirements and auxiliaries so that the final size and rating of the diesel plant meets the load requirements associated with the compound.

The diesel control panel shall be determined considering, the maximum load capability of the plant such that operators can add loads to the set and be fully aware of the set loadings. It is necessary to clearly identify the minimum load at which the set can operate for prolonged periods without causing damage to the set through "light load running". In addition to being indicated on the control panel, these limits shall also be clearly described within the operation/maintenance manual.

15.4.5 Control panel (generator)

A control cubicle shall be provided for each alternator and shall accommodate the following equipment, in addition to all other items necessary to the installation and operation of the plant. The control panels shall comprise totally enclosed steel cubicles, the numbers and layout of compartments being dictated by the additional switchgear requirements. A hinged door(s) on the panel shall be provided for access to all components. The panel shall be free standing on a 100 mm raised base. The final external colour of the panel shall be subject to an agreement. The inside shall be white and all instruments black on a white background.

The control panel shall include:

- a manually/automatically operated air or molded case circuit breaker with overload and under voltage protection to control the generator load
- Tripped indication
- Engine off, start, stop controls
- Audible fault/stop alarm with mute and reset provision
- Earth bar and neutral link
- Mains and control cable terminal boxes for incoming and outgoing supplies
- a single phase switch fuse which shall be wired to a pair of incoming terminations and associated distribution fuses for supplying any required mains operated equipment on the set (i.e. heaters, battery charger and the like)
- An automatic voltage regulator of the type which will maintain its adjustment for long periods without attention.
- Where required, a hand field regulator complete with a " Hand/Auto" switch. The hand field regulator shall give stable control of the voltage under the specified operating conditions. If the hand field regulator is to be left in a precise position when the set is under the control of the automatic voltage regulator, this position shall be clearly marked.
- A hand-auto-test switch shall be provided to facilitate operation and maintenance checks. The Contractor shall recommend the frequency with which the sets shall be test operated on load/no load.
- control equipment as necessary to fulfil the functional requirements associated with the engine start, stop, governor control and the like as necessary for each station
- Voltmeter
- Voltmeter selector switch
- three ammeters, reading current in each leg
- polyphase maximum demand indicator meter
- hours run meter
- frequency meter of the vibrating reed type
- power factor meter
- kilowatt hour meter with maximum demand pointer
- battery trickle charge ammeter
- low oil pressure indicator
- high oil temperature indicator
- High water temperature indicator
- over speed indicator
- high vibration indicator
- overload indicator
- starter failed indicator

- set failed indicator

The panel shall also include other lamps as required for the correct operation of the generating set together with alarms and switches. Automatic shutdown shall occur with the operation of any of the first five alarm, protection and indicating circuits together with overload after reaching an agreed predetermined load value, and the operation of the thermistor protection relay.

15.4.6 Enclosures for electrical and control equipment

Enclosures for electrical and control equipment shall be drip proof and dust protecting with adequate front access as necessary for maintenance and repair and to IP51 classification. Special attention shall be given to the method of construction and to the mounting of the components to minimize the effect of vibration. Diagrams of connections in durable form shall be mounted inside the enclosures.

Full consideration shall be given to the fixing/supports and the running/connection of all cables. Full details shall be submitted for approval, the Contractor taking into consideration all details including cable terminations, access to these terminations, cable bending radii etc.

15.4.7 Finishes

All ferrous metalwork shall be either painted or processed to give a rustproof coating. Ferrous metalwork to be painted shall first be either shot blasted or thoroughly wire brushed to remove all scale and oxide and immediately given one brushed coat or two sprayed coats of primer. After not less than four hours, one brushed or two sprayed undercoats followed by one brushed or two sprayed finishing coats of heat and oil resisting quality paint shall be applied.

Successive coats of paint shall be slightly differing shades. The interior service of electrical equipment enclosures shall be finished white and all external surfaces shall be finished to the BS 4800 to suit the manufacturer's standard colour. The engine crank case shall not be painted internally unless the plant is resistant to the lubricating oil.

15.4.8 Drawings of pumping facilities

The following drawings are essential:

- Building drawings showing details of cable entries, pipe entries and ducts required.
- General arrangement of the diesel engines.
- Details, supports and general requirements associated with the exhaust system.
- General arrangement of the alternator and exciter showing terminal markings, polarity and phase rotation.
- General arrangement of the electrical control panel
- Schematic and wiring diagram of the electrical control panel

15.5 PUMP MOTOR CONTROL PANEL SPECIFICATION

15.5.1 General

A. The motor control panel shall be assembled and tested by a controls system manufacturer meeting the Standards of UL 508A for industrial controls and be UL labeled and serialized accordingly. The motor control panel shall be assembled and tested by the manufacturer so as to insure suitability in matching controls to motors and to insure single source responsibility for the equipment.

B. The panel shall contain all components required by the pump manufacturer for starting and protecting the motor as well as features required by the pump manufacturer for warranty of the pumps. Items such as thermal overload detection or seal failure detection shall be included when required.

C. Incoming pump power shall be single/three -phase, 50 Hz, 230/380 volts AC.
D. Incoming control/alarm power shall be single-phase50 Hz, 230 volts AC.

E. The control panel shall incorporate three (3) normally open; mercury or mechanically-activated control switches with pipe clamps. Floats shall be labeled in the panel as stop, start, and alarm. Floats shall be SJE-Rhombus control switches or approved equal.

15.5.2 Construction

A. The controls for the pump shall be housed in an engineered thermoplastic enclosure meeting NEMA 4X requirements with a hinged door and neoprene gasket. The enclosure shall have provisions for a padlock.

B. A nameplate shall be permanently affixed to the panel. A ratings label shall include the model number, voltage, phase, frequency, ampere rating and horsepower rating and shall be affixed to the inside of the enclosure. A warning label against electric shock shall be permanently affixed to the outer door. The interior of the enclosure shall have a clear envelope with "as built" schematics located within.

C. A removable aluminum back plate shall be provided for mounting all circuit breakers, motor starters, etc. All components mounted to the back plate shall be secured by type 25, self-tapping screws in extruded holes. Rivets shall not be acceptable for securing any component to the back plate.

D. A simplex pump controller shall be provided for control logic. The controller shall utilize a printed circuit board to avoid conventional wiring. The printed circuit board of the pump controller shall be manufactured using UL listed materials. There shall be separately fused control and alarm circuit protection. A run light and hand-off-auto switch shall be provided for the pump circuit. The run light and hand-off-auto switch shall be mounted on the printed circuit board. The run light shall be green.

E. A circuit breaker shall be used as branch circuit protection for the pump. The circuit breaker shall be thermal magnetic and sized to meet NEC requirements for interrupt capacity and amp rating.

F. The magnetic motor starter shall be general purpose type rated for the pump horsepower and include a contactor with a minimum mechanical life of 500,000 operations and a minimum contact life of 100,000 operations. Pump overloads, if not included in the pump, shall provide overload protection for the pump circuit and shall be sized to meet NEC requirements for the pump full load ampere rating specified.

G. A high-level alarm condition shall activate the main alarm light (red, mounted on the top of the panel) and alarm horn. The alarm light shall remain illuminated until the problem is corrected. The alarm horn shall be rated 83-85 dB minimum. A Test-Normal-Silence toggle switch labeled and placed adjacent to the horn, shall be included.

H. H. Wire ties shall be used to maintain panel wiring in neat bundles for maintenance and to prevent interference with operating devices. All grounding conductors shall be securely connected to assure a proper ground.

15.5.3 Irrigation pump motor starter

Starter sections shall be provided for the irrigation pumping station. Each starter shall be complete with the following features:

- Incoming door interlocked fuse-switch disconnect
- Control circuit bypass “test” switch
- Set of three contactors for star/delta starting
- Star/delta changeover timer relay
- Motor protection relay incorporating instantaneous overcurrent, earth-fault, thermal overload, delayed start, current unbalance, loss of phase and under voltage
- Integral 380/110 V transformer for AC control supply
- Power factor correction to 0.96 when the load is at rated value
- 110 V AC operated “external trip” relay (de-energise to trip/energise to permit running)
- 110 V AC operated “start/stop” command relay (energise to start)
- Thermistor relay
- Front of panel mounted start, stop and reset pushbuttons
- Front of panel mounted hand/off/auto selector switch (auto selects control from the C&I section)
- Front of panel mounted indicator lamps for running, stopped, motor protection relay fault, motor winding over temperature alarm, motor winding over temperature trip, and “external trip operated”
- “Power-up” auto-reset timer relay
- Anti-condensation heater complete with fuse protection and thermostat
- Cable termination features for multiple outgoing multicore power cables
- Volt-free changeover contacts for remote indication of running status, internal trip conditions, external trip relay operated, control selection and “available”
- Auxiliary terminals and contacts as necessary

The external trip relay shall be powered from circuitry in the Control & Instrumentation section and will be energised when external hydraulic conditions permit operation of the pump set. If the external trip relay is de-energised it shall latch out and power to the pump shall immediately be cut-off. The pump shall remain unavailable for service until the trip condition has been cancelled and the relay has been manually reset, or reset by a power-up auto reset function following power failure.

The “available” signal shall be generated when no internally detected trip conditions exist and the “external trip” relay is energised.

16 INSTALLATION, MAINTENANCE AND OPERATION OF E&M EQUIPMENT

Generally, Electro-mechanical equipment experiences the most serious problems with operation and maintenance and lags in cost recovery aspects. Hundreds of project built infrastructure deteriorates after the project's termination. Therefore, it is imperative to plan for operation and maintenance, with a planned withdrawal of external support as local ownership builds. This document is intended for managers and planners who are concerned with the challenging problem of how to implement effective operation and maintenance plan in irrigation projects.

16.1 SAFETY INSTRUCTIONS WHILE MAINTENANCE & SERVICING A PUMP

Before attempting any maintenance on a pump particularly if it has been handling any form of liquid, it should be ensured that the unit is safe to work on. The pump must be flushed thoroughly with suitable cleaner to purge away any of the product left in the pump components. This should be carried out by the plant operator and assurance of cleanliness obtained before starting work. To avoid any risk to health it is also advisable to wear protective clothing as recommended by the site safety officer especially when removing old packing which may be contaminated.

Check and ensure that the pump operates at below the maximum working pressure specified in the manual or on the pump nameplate and before maintenance, ensure that the pump is drained down. Wear a suitable mask or respirator when working with packing and gasket components which contain fibrous material, as these can be hazardous when the fibrous dust is inhaled. Be cautious, if other supplier's components have been substituted for genuine parts, these may then contain hazardous materials.

16.2 INSTALLATION, MAINTENANCE AND OPERATION OF PUMP SET AND POWER SUPPLY

16.2.1 Submersible pump installation

- Before installation, must check whether the cable is damaged, scratched, broken, etc. If they are faulty, must consult the supplier.
- The pumps are normally supplied with motor and pump disconnected; connect the coupling and pump-motor suction lantern.
- Clean the surface to be coupled.
- Put the suction lantern of the pump in correspondence of the motor studs.
- Couple the grooved joint of the pump to the motor shaft.
- Screw in the nuts to the suction lantern, and then fix the crosswise starting from the one opposed to the cable.
- The torque recommended is 10 Nm for 4' motor.
- Connect the cable to the pump with the cable guard and place the filter on the suction lantern.
- When threaded connections are used, delivery pipes must be tightened to avoid any risks of the pump falling into the well.
- A safety rope or chain on non-perishable material should always be used to secure the pump.
- Attach the power cables to the delivery pipe with cable clamps placed at intervals of approximate 3 meter.
- Lower the pump into the well, making sure the feed cables are not damaged in any way during the operation.

- Install the pump with its axis placed at least 0.5 meter above the bottom of the well.

16.2.2 Power supply installation

From safety point of view all work of an electrical system must be carefully carried out by skilled person only. Thus note that:

- Make sure the frequency and mains voltage correspond with the name plate data.
- Check whether the power supply is conformed to the stipulation of name plate before installation.
- Power supply connected with the pump must be assembled with electric cable, and the voltage fluctuation must be controlled within ± 10% of rated.
- For connection of cables in the well, use thermo-shrinking insulation sheathes or other system used for submerged cables.
- The operating temperature limiting the drop in voltage to at most 3%.
- The live wire should made dead before making the connection. The connection between the pump and the other part must be made using suitable material for submersed cables and in compliance with the current regulation from local electricity authority.
- All checking of connection should be checked by appropriate instruments.
- Important While installing the motor pump, avoid impact, rubbing or anything else that can damage the power cable or puncture the cable leading to short-circuit.

16.2.3 Operation and maintenance of pumps

Daily checks

- Pressure gauge reading.
- Voltage and current.
- Pump and motor bearing temperature.
- Vibration and noise.

Periodical checks

- Replenishing of the grease to the pump antifriction bearing and motor bearing after proper intervals.
- Check the vibrations.
- Check the liquid level controller for its functioning.
- Calibration of measuring instruments.
- Check the level of motor stool as described in assembly procedure after taking out motor at regular intervals of 6 months.
- Clean the tank, if there are chances of deposition of the contents of the liquid handled.

Overhauling

- With continuous daily operations spell, the pump will be due for overhaul after 10000 working hours. This work should be carried out by specialised and experienced fitters.
- While ordering spare parts, the details of the nameplate must be quoted in full. Particularly the name of the pump, order number, name of the part and quantity required.
- Keep the sufficient stock of spare parts in order to meet the emergency requirement.

Follow the following procedure while dismantling the pump.

- Disconnect the delivery pipe connections above support plate. Unscrew the fasteners holding support plate on the flange of tank.
- Disconnect the motor power connections. Unscrew the nuts of motor stool and take out motor stool along with motor.

- Remove coupling star and take out pump half coupling after loosening the set screw. Use suitable puller. Remove coupling key.
- Unscrew the two bearing nuts
- Remove the bearing cover along with oil seal Use release bolts if necessary.
- Remove spacer for thrust bearing adapter and split key for unit pumps).
- Unscrew nuts holding bearing holder on stuffing box housing and remove bearing holder.
- Lift entire pump unit vertically upwards with the help of crane having slings
- Supported on eye bolts screwed on support plate.
- Fit clamp to the top column pipe and keep two girders or strong wooden logs across the opening of the tank.
- Rest the clamps on these two girders of wooden log.
- Unscrew flange and pipe nut.
- Unscrew bolts holding bearing carrier support plate and top column pipe together.
- Remove key for thrust bearing carrier Take out bearing holder along with oil seal, angular contact ball bearing and thrust bearing carrier Use release bolts.
- Push thrust bearing carrier so that it will come out of bearing holder along with angular contact ball bearing. Remove angular contact ball bearing from thrust bearing carrier only if found to be damaged.
- Lift the unit vertically up after engaging the shelling on the arms of the pump.
- Fit another pair of clamps on column pipe next below the top column pipe.
- Allow the unit to lower down so as to rest the clamps of the girders or wooden logs.
- Unscrew the bolts holding top column pipe, bearing spider and column pipe together.
- Take out top column pipe connect the head shaft using suitable spanner.
- Disconnect the lubricating pipe for bearing spider by unscrewing nut.
- Take out the bearing spider.
- Unscrew grub screw at the junction of intermediate bearing bush and bearing spider. Remove the bearing bush out of bearing spider only if it is found damaged.
- Unscrew the screwed coupling from intermediate shaft or impeller shaft
- Follow the procedure given till you remove last intermediate shaft last bearing spider and last screwed coupling.
- Disconnect the rising pipe at support plate while dismantling the column

Preventive maintenance usually consists in checking pump functionality and cleaning the pump and site daily, greasing weekly, checking all parts of the pump stand monthly, and taking the whole pump apart for a check, cleaning the parts with clean water and painting the pump stand annually. Pump rods that show bad corrosion must be replaced. Under normal conditions, a galvanized steel pump rod needs replacement every five to six years. Rising mains consisting of galvanized iron have to be removed and checked and pipes with badly corroded threads must be replaced. Small repairs are the replacement of bearings, cup seals and washers, straightening bent pumping rods, etc. Major repairs may involve the replacement of the plunger, foot valve, and cylinder, pump rods, rising main; pump handle, fulcrum, etc. With open-top cylinder pumps, all preventive maintenance activities can normally be executed by a village pump caretaker. For major repairs and problems, external support may be needed. Closed-top cylinder pumps often need special lifting equipment to pull up the rising main and cylinder for maintenance of parts down in the hole.

In addition to the above the following checks, on-site modifications, re-arrangements and preventive maintenance are necessary:

- Check and repair any leakage in piping or through valves.
- Flush for time of 2-3min for each line to prevent sedimentation on the inner pipes walls.
- Clean the canals of the system thoroughly before every irrigation.
- Check the air and check valves periodically for proper functioning.

- Inspect plastic equipment, valves and devices for cracks and other physical damage.
- Conduct systematic checks to spot malfunctioning equipment affected by physical deterioration and other possible damage by machinery, animals, etc.
- Make frequent visual checks of the system to ensure that it is in good condition and operating efficiently.

The following checks and inspections are recommended for most engine or electric motor driven pumps:

- Noise;
- Vibration;
- Leakage;
- Temperatures of bearings and windings;
- Fuel/power consumption;
- Capacity and output (water discharge and dynamic head);
- Ventilation screens, clean where necessary;
- Oil pressure;
- Oil, lubrication, change where necessary

16.2.4 Maintenance of electric motor

- Clean all debris accumulated during the storage period.
- Change motor bearing oil with special type of lubricant, do not overfill, use grease gun to lubricate bearings.
- Change oil in reduced voltage starters.
- Check that motor ventilation vents are open; clean dust and dirt from all moving parts of motor and panel.
- Check and tighten all electrical connections, replace overheated connections with new material; test all coils and heaters for continuity and shorts; clean all magnet surfaces; check for spare fuses of proper size; ensure all conduits or shielded cables are in good condition; check that all conduct points are corrosion free.
- Ensure service cabinet interior is moisture free.
- Operate all moving parts by hand before applying power.

Table 16-1: Fault and solutions for electric driven pumps

FAULT	POSSIBLE CAUSES	REMEDIES
Pump does not start	• Too low voltage; • Impeller blocked; • Stator winding burn up; • Capacitor damaged; in case of single phase motor. • Absent phase; in 3Ø • Too large resistance of cable	• Adjust voltage to ±10% of the rated; release it by inserting screw driver into the shaft at the fan side to remove the sundries; • Repair; • Replace capacitor; • Check switch and cable connection etc; • Check 3Ø wires. • Use the proper cable;
Pump delivers insufficient water or no water	• Highly delivery head; • The water level is low. • Air locked; • Pump worn off; • Wrong rotation.	• Lower the head; • Throttle pump output or reset pump at lower depth if possible. • Normal delivery may resume if pump is started and stopped at one minute interval; • Replace pump or repair; • Inverse two phase.

16.3 INSTALLATION, MAINTENANCE AND OPERATION OF DIESEL ENGINE

Installation

Diesel engines are to be installed by qualified technician following the manufacturer procedure.

Operation

The engine must be operated by a trained caretaker. Every engine has its own typical operating instructions. Before starting it, the levels of fuel, oil and cooling water (if not air-cooled) are checked. If these levels are low, extra fuel, oil or water has to be added. During operation, the fuel level, oil pressure, and engine speed are checked and also the functioning of the pump or generator. Some moving parts may need manual lubrication. When the engine is operated at very low speeds, its efficiency is low and carbon builds up rapidly in the engine, increasing the need for servicing. All data on liquid levels and running hours are written down in a log book.

Maintenance

Every day the outside of the engine must be cleaned, and in dusty conditions the air filter must be checked and cleaned. Some parts may need manual lubrication. In moderately dusty conditions, oil-bath air filters are cleaned once a week, dry-paper air filters a little less frequently. The engine is serviced for preventive maintenance according to the number of hours it has run. Every 50 hours, the clutch (if present) must be greased. Every 250 hours, clean all filters (replace if necessary), change oil, check nuts and bolts and exhaust pipe. Every 1500 hours, major service overhaul with decarbonizing, adjusting valve clearance, etc. If the engine is connected to a pump or generator with a V-belt, this will regularly need replacement. Once a year the engine house must be painted and occasionally repaired. If a generator is present it will have its own maintenance needs. The Table below shows only the most important O&M activities.

16.4 INSTALLATION, MAINTENANCE AND OPERATION OF PIPES

The maintenance of distribution pipes consists in checking for leaks, corrosion, and scale. Each time a line is opened, the interior of the piping can be observed. The life of more than one system has been greatly extended by workmen who have reported pipe damage when making a house connation.

Due to differential thermal expansion rates between metal and plastic, transmittal of pipe vibration, and pipe loading forces DIRECT INSTALLATION OF METAL PIPE INTO PLASTIC CONNECTIONS IS NOT RECOMMENDED. Wherever installation of a plastic duplex strainer into a metal piping systems is necessary, it is recommended that at least 10 pipe diameter in length of plastic pipe be installed upstream and downstream of the plastic duplex strainer to compensate for the factors mentioned above

16.5 MAINTENANCE OF IRRIGATION NETWORK

The procedure for the irrigation network checkup is as follows:

- Inspect for possible damage to the network and repair it.
- Open fully and drain completely all valves.
- Remove dirt, corrosion and other foreign material from the component parts.
- Check E&M equipment for possible, damage, wear and signs of deterioration, and replace where necessary.

- Store all spare parts and unused parts in a dry clean place on shelves away from fertilizers, chemicals, oil, grease and lubricants.
- Examine the condition of air and check valves.
- Clean all equipment in the pumping point.
- Check condition of gaskets and seals; remove, clean and store in a dry place.
- Inspect all portable metal pipes for any kind of damage and consult suppliers for repair;
- store properly away from power lines and wiring.
- Drain completely all pipes left in the open
- These troubleshooting and corrective action guidelines are provided as a sample listing of the problems that may be encountered. The system has been installed and a checkout performed that showed the system to be operating correctly. Tailor the guidelines, through deletions and additions, to each specific site. List the probable causes for each problem in the order of priority in which they should be evaluated to minimize expensive or unnecessary repair work.

Table 16-2: Diagnostic and trouble shooting of pumping system

Item No	Problem	Cause	Corrective Action
1	Your pump delivers little or no water	A. Water level in a low production well drops too low while pump is operating, causing it to air lock. (Resulting in loss of prime and possibly serious damage to the pump) B. Intake screen is partially plugged. C. Check valve(s) may be stuck. D. Voltage is too low; the motor runs slowly, causing low discharge pressure (head) and high operating current draw.	A. Lower the pump further into the well, but make sure it is at least five feet from the bottom of the well. Install a control valve in the discharge pipe between the pump and pressure tank. Use the control valve to restrict the flow until the discharge rate does not exceed well recovery rate. B. Lime or other matter in the water may build up on screen. Pull pump and clean screen. C. Make sure that the built-in check valve in the pump and any check valves in the discharge line are free to open properly. D. Have a certified electrician verify voltage at the electrical disconnect box (2 wire) or control center (3 wire) while the pump is operating. If the voltage is low, the power company may need to raise it or installation may require larger wire. Discuss this with the power company or a licensed electrician.
2	Air or milky water discharges from your faucets	Well water may be gaseous	If your well is naturally gaseous and your system has a standard tank, remove the bleeder orifices and plug the tees. If the condition is serious, check with certified well professionals
3	Pump starts too frequently	A. Leak in the pressure tank or plumbing. B. Pressure switch is defective or out of adjustment. C. Check valve is leaking. D. Tank is waterlogged.	A. Check all connections with soapsuds for air leaks. Fix any leaks you find. Check the plumbing for water leaks. Fix any leaks you find. B. If necessary, replace switch. C. Inspect valves and replace if necessary. D. Captive Air® Tanks: Check the tank for leaks; correct if possible. Precharge tanks to 18 PSI with a 20-40 PSI switch, 28 PSI for a 30-50 PSI switch, 38 PSI for a 40-60 PSI switch, etc. Standard tanks: Check the tank for leaks;

Item No	Problem	Cause	Corrective Action
		E. Drop pipe leaking. F. Pressure switch is too far from the tank.	correct if possible. Check bleeder orifices and clean bleeders; replace if necessary. E. Raise one length of pipe at a time until the leak is found. When water stands in the pipe there is no leak below this point. F. Move the pressure switch within one foot of the tank.
4	Fuses blow or overload protector trips when the motor starts	A. Fuses or wires are too small. B. Low or high voltage. C. Cable splices or motor windings grounded, shortened, or open. D. 3-wire only; Cable leads may be improperly connected in pump control box, pressure switch or fused disconnect switch. E. 3-wire only; There may be a broken wire in the pump control box. F. 3-wire only; Starting or running capacitor in control box may be defective or vented (blown out).	A. Replace with correct wire sizes. B. While motor is running, voltage should not exceed plus 5% or minus 5% or rated voltage shown on motor nameplate. Call the electric power company to adjust line voltage if not within these limits. C. Consult certified electrician or service technician. D. Check wiring diagram on pump control box and color coding of drop cable. E. Employ certified electrician examine all connections and wiring in control panel. If necessary, repair them. F. Inspect capacitors. Employ a certified electrician to check capacitors and replace them if necessary. WARNING! Hazardous voltage, can shock, burn or cause death. Capacitors may still carry voltage charges even after being disconnected from wiring. Have them checked by a certified electrician.
5	Motor will not start but does not blow fuses. WARNING! Hazardous voltage. Can shock, burn or cause death. Employ a qualified electricians should work on electrical service.	A. No voltage to motor. B. Cable splices or motor windings may be grounded, shorted or open-circuited. C. Open circuit in pump control box (3-wire only); faulty connections; faulty wires. D. Faulty pressure switch. E. 3-wire only; Cable leads improperly connected in the control center.	A. With a voltmeter check; 1) fuse box to make sure full voltage is available; 2) pressure switch terminals, to make pressure switch is passing voltage correctly; and 3) terminal strips in pump control box or disconnect switch box to make sure voltage is available there. On 1-1/2 through 3 HP: Push red overload reset button(s) on the bottom of control center. B. Consult certified electrician or service electrician. Do not attempt to disassemble pump or motor. C. Examine all connections and wires; examine terminal strips in the control center (3-wire only); repair if necessary. D. Check pressure switch; replace if necessary. E. Check wiring diagram on control center panel and color coding of drop cable.
6	Pressure switch fails to shut off pump	A. Voltage is too low; motor will run slowly, causing low discharge pressure (head) and high operating current	A. Have a certified electrician verify voltage at the electrical disconnect box (2-wire) or the pump control box (3-wire) while the pump is operating. If the voltage is low, your power

Item No	Problem	Cause	Corrective Action
		draw. B. Faulty pressure switch. C. Drop pipe is leaking. D. Water level in the well may become too low when pump is running	company may require larger wire. Discuss with the power company or a certified electrician. Check voltage with a recording meter if trouble recurs. B. Replace switch. C. Raise one length at a time until the leak is found. When water stands in the pipe, there is no leak below this point. D. Lower pump further into well, make sure it is between five and ten feet from the bottom of the well. Install a valve into the discharge pipe between the pump and the pressure tank. Use the valve to restrict flow until discharge rate does not exceed the well recovery rate. WARNING! To prevent the possibility of dangerous high pressure, install a relief valve in the discharge pipe between the pump and flow restriction valve. The relief valve must be capable of passing full pump flow at 75 PSI
7	Fuses blow or overload protector trips when motor is running	A. Low or high voltage. B. 3-Wire only: High ambient (atmospheric) temperature. C. 3-Wire only: Pump control box is wrong horsepower or voltage for installation. D. Wire size is too small. Improperly connected in the pump control box. E. Cable splices or motor windings may be grounded, shorted or open-circuited	A. While the motor is running, voltage should not exceed plus 5% or minus 5% of rated voltage shown on motor nameplate. Call your power company to adjust line voltage if it is not within these limits. B. Make sure the pump control box is installed out of direct sunlight. C. Compare horsepower and voltage rating of motor (from motor nameplate) with those of the pump control box (from pump control box nameplate). These numbers must match. Make sure the wire sizes match specifications in the Table. E. Consult certified electrician or a service technician to determine if this is the cause of the problem or not. Do not attempt to disassemble pump or motor.

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