

DESIGN AND EXPERIMENTAL STUDY OF VERTICAL TURBINE PUMP

*A thesis submitted towards partial fulfillment of the requirements
for the degree of*

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in Water Resources and Hydraulic Engineering
Course affiliated to Faculty of Engineering & Technology
Jadavpur University

submitted by

DEBASISH SARKAR

Exam Roll No: M6WRP24021

Under the guidance of

PROF. (DR). ASIS MAZUMDAR

Professor & Director

School of Water Resources Engineering, Jadavpur University

DR. GOURAB BANERJEE

Assistant Professor

School of Water Resources Engineering, Jadavpur University

School of Water Resources Engineering

M.E. (Water Resources & Hydraulic Engineering) course affiliated to

Faculty of Engineering & Technology

Jadavpur University

Kolkata 700032, India

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Submitted by

DEBASISH SARKAR

Class Roll No. 202030302030

Registration No. 154693 of 2020-2021

Examination Roll no. M6WRP24021

Under the guidance of

PROF. (DR). ASIS MAZUMDAR,

Professor & Director

School of Water Resources Engineering, Jadavpur University

DR. GOURAB BANERJEE,

Assistant Professor

School of Water Resources Engineering, Jadavpur University

School of Water Resources Engineering

M.E. (Water Resources and Hydraulics Engineering) course affiliated to

Faculty of Engineering and Technology

Jadavpur University,

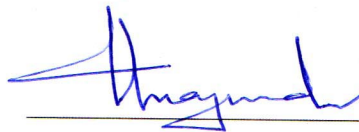
Kolkata 700032, India

(2024)

M.E. (Water Resources & Hydraulic Engineering) course
affiliated to Faculty of Engineering and Technology,
Jadavpur University, Kolkata, India.

CERTIFICATE OF RECOMMENDATION

This is to certify that the thesis entitled “**DESIGN AND EXPERIMENTAL STUDY OF VERTICAL TURBINE PUMP**” is bona-fide work carried out by **DEBASISH SARKAR** under our supervision and guidance for partial fulfilment of the requirement of Master of Engineering (Water Resources & Hydraulic Engineering) in School of Water Resources Engineering, during the academic session 2021-2024.



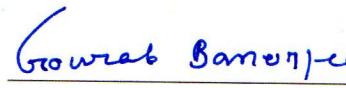
30/05/24

THESIS ADVISOR

Prof (Dr.) Asis Mazumdar

Professor & Director

School of Water Resources Engineering
Jadavpur University, Kolkata-700032



30/05/24

THESIS ADVISOR

Dr. Gourab Banerjee

Assistant Professor

School of Water Resources Engineering
Jadavpur University, Kolkata-700032



30/05/24

Prof (Dr.) Asis Mazumdar

DIRECTOR

School of Water Resources Engineering
Jadavpur University, Kolkata – 700032



30/05/2024

DEAN

Faculty of Interdisciplinary Studies, Law and Management

Jadavpur University

Kolkata – 700032

Dean
Faculty of I.S.L.M., Jadavpur University
Kolkata - 700 032

M.E. (Water Resources & Hydraulic Engineering) course
affiliated to Faculty of Engineering and Technology,
Jadavpur University, Kolkata, India.

CERTIFICATE OF APPROVAL**

This foregoing thesis is hereby approved as a credible study of an engineering subject carried out and presented in a manner satisfactorily to warrant its acceptance as a prerequisite to the degree for which it has been submitted. It is understood that by this approval the undersigned does not endorse or approve any statement made or opinion expressed or conclusion drawn therein but approve the thesis only for the purpose for which it has been submitted.

Committee of the final examination
for the evaluation of the thesis

** Only in case the thesis is approved.

DECLARATION

I hereby declare that this thesis contains a literature survey and original research work by the undersigned candidate, as part of my Master of Engineering in Water Resources & Hydraulic Engineering degree during the academic session 2021-2024. All information in this document has been obtained and presented in accordance with academic rules and ethical conduct. I also declare that, as required by this rules and conduct, I have fully cited and referred all material and results that are not original to this work.

NAME : DEBASISH SARKAR
ROLL NO : 202030302030
EXAMINATION ROLL NO : M6WRP24021
REGISTRATION NO : 154693 of 2020-2021
THESIS TITLE: **DESIGN AND EXPERIMENTAL STUDY OF VERTICAL TURBINE PUMP**

Signature : *Debasish Sarkar*

Date : *30.05.24*

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DATE: 30.05-29
PLACE: JADAVPUR UNIVERSITY

Debasish Sarkar

DEBASISH SARKAR
EXAM ROLL NO: M6WRP24021
REGISTRATION NO: 154693 OF 2020-2021

Abstract

Pump is a device which pressurises the liquid. At its core, a pump adds energy to the liquid, which increases its pressure and/or velocity, allowing it to flow through a system. The main task faced by an engineer while drawing a pump design is with meeting the ends with the economics and the hydraulic operating characteristics. This thesis presents a comprehensive investigation into the design and experimental study of a vertical turbine pump. The study aims to enhance the understanding of vertical turbine pump performance, efficiency, and reliability through theoretical analysis, practical construction, and experimental validation. The research encompasses the development of design methodology, meticulous construction of a standard intake well pump, and rigorous experimental evaluation under varied operating conditions. Key findings and recommendations are provided to advance the field of vertical turbine pump engineering.

Keywords—Vertical Turbine Pump; Performance Testing; Computational Fluid Dynamics (CFD); Efficiency; NPSH; Pump Characteristic curves, Centrifugal Pump.

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LIST OF SYMBOLS

SYMBOLS

b	breadth
d	diameter
g	gravitational acceleration
m	mass
n	rotational speed
r	radius
s	blade thickness, standard deviation, vane thickness
ω	rotational speed
z	number of blades, number of vanes
H	pump head
Q	flow rate
N_s	specific speed
N_r	pump suction specific speed
N_a	system suction specific speed
f	Darcy-Weisbach friction factor
C	Hazen-Williams roughness coefficient
h_f	Head Loss
P	Power
T	torque
V	velocity
α	fluid angle
β	blade angle
Δ	specific diameter
η	efficiency
ρ	density of pumping fluid
τ	torsional stress
π	pi number

CHAPTER 1: INTRODUCTION

1.1 Background of the Study:

Vertical turbine pumps are widely used in various industries for applications such as water supply, irrigation, drainage, and wastewater treatment. Their ability to efficiently transfer fluids over a wide range of flow rates and head conditions makes them indispensable in many engineering systems. However, despite their ubiquity, there is ongoing research aimed at improving the design, construction, and performance of these pumps to meet the evolving needs of modern applications.

The Vertical pump performance can be analysed by testing the physical model in test circuit. But performance testing requires laboratory setup and different types of measuring instruments. The use of CFD enables engineers to obtain solutions for problems with complex geometry and boundary conditions. Investigations of the internal flow structure can be performed with CFD tools in order to correct the design and obtain better performance characteristics from the designed pump. The pump end consists of at least one rotating impeller that is attached to a shaft and directs the well water into a diffuser casing called a bowl. The water enters from the bottom of the pump called suction bell. From suction-bell water moves into the first stage impeller which raises the water's velocity. The diffuser bowl forms the outlet pressure and also it collects the flow from tip of one impeller and directs the fluid into the eye of next impeller located immediately above the bowl, and this process Multistage Vertical Turbine pump continues through all of the stages of the pump. The multistage pump has two or more Impeller and bowl assembly in series for getting high head with improved efficiency. This configuration allows for efficient operation in deep well or submerged conditions, making them suitable for a variety of applications where horizontal pumps may not be practical. Because of the wide variety of applications, pumps have several of shapes and sizes from very large to very small, from handling gas to handling liquid, from high pressure to low pressure, and from high volume to low volume.

1.2 Study Area

Vertical turbine pumps face challenges such as cavitation, mechanical wear, and efficiency losses. The performance of the vertical turbine pump has been studied with the effect of some impeller modification, speed variation and sump water level changes. By combining theoretical knowledge with practical skills in these study areas, the thesis aims to develop a comprehensive understanding of vertical turbine pump design, construction, and performance evaluation. Optimizing the design and construction processes can significantly improve their performance and longevity. The study area can be categorized into the following main sections:

Fluid Dynamics and Hydraulics

- **Flow Characteristics:** Understanding the behaviour of fluid flow through the pump, including velocity profiles, pressure distributions, and turbulence.
- **Hydraulic Performance:** Evaluating the head, flow rate, efficiency, and power consumption of the pump under different operational conditions.
- **Cavitation Phenomena:** Investigating the onset and effects of cavitation on pump performance and longevity.

Mechanical Design and Analysis

- **Impeller Design:** Focusing on the number, shape, and angle of blades to optimize fluid dynamics and minimize losses.
- **Diffuser Design:** Analysing the design of diffuser vanes to enhance pressure recovery and improve efficiency.

Material Science and Manufacturing

- **Material Selection:** Choosing materials for different components based on strength, corrosion resistance, and wear properties.
- **Manufacturing Techniques:** Implementing precision casting, CNC machining, and composite fabrication to achieve high-quality components.

Computational and Experimental Methods

- **Computational Fluid Dynamics (CFD):** Using CFD simulations to model fluid flow and predict the pump's performance, helping to refine the design before physical construction.
- **Theoretical Analysis:** Applying empirical formulas and theoretical models to estimate performance parameters and validate the design.
- **Experimental Testing:** Setting up a test rig to measure actual performance metrics such as flow rate, head, efficiency, and cavitation characteristics under various conditions.

Table 1.1: Pump basic design & Testing data considered

Capacity (m3/hr)	Total Head (m)	Pump Speed (RPM)	Sump depth (m)	Motor Type	Supply voltage	Application	Discharge Head Type	Working Fluid
1200	30	1475	4.2	VSS	415 V, 3 Ph, 50 Hz	Water intake	Surface discharge	Raw Water (S.G 1.0)

1.3 Research Objectives

The primary objective of this thesis is to design, construct, and experimentally study a vertical turbine pump with the aim of advancing the understanding of its hydraulic performance, structural integrity, and operational characteristics. Specifically, the research aims to:

- Develop a comprehensive design methodology for vertical turbine pumps, considering factors such as hydraulic efficiency, structural stability, and material selection.
- Construct a standard VT pump based on the established design criteria, utilizing appropriate manufacturing techniques and quality control measures.
- Testing of VT pump to evaluate the hydraulic performance of the pump under various operating conditions, including flow rate, head, and speed variations.
- Analyze the experimental data to assess the pump's efficiency, reliability, and suitability for different application scenarios.
- Draw conclusions regarding the effectiveness of the design and construction methods employed and identify areas for further improvement and research.

1.4 Proposed Outcome of the Study:

The proposed outcomes of this study on the design, construction, and experimental study of vertical turbine pumps aim to advance the field by providing a thorough, validated approach to pump design. These outcomes will have significant implications for the efficiency, cost-effectiveness, and sustainability of vertical turbine pumps in practical applications.:

- The study is expected to develop an improved design methodology for vertical turbine pumps, incorporating advanced computational tools like to optimize impeller and diffuser geometries.
- This study will also involve in proper guidelines of various regulatory bodies for pump design.
- The results of pump the performance test will be presented
- Through rigorous experimental testing, the study will generate comprehensive performance data for the constructed pump. This data will include measurements of flow rate, head, power consumption, and overall efficiency under various operating conditions.
- The study aims to identify specific design features and construction techniques that contribute to improved pump efficiency. These insights will be valuable for future pump designs, potentially leading to pumps that consume less energy and operate more cost-effectively.

1.5 Outline of the Thesis

This thesis will focus on the design, construction, and experimental study of a single-stage vertical turbine pump to validate the pump performance results. The scope encompasses the entire process, from initial design considerations to final performance evaluation. The thesis is organized into several chapters, each addressing specific aspects of the research:

The thesis is organized as follows: the second chapter is the literature review. This chapter reveals the summery of the most important parameters and phenomena that affect the pump performance. And how the authors study these phenomena either numerically or experimentally. These parameters that affect the pump performance are the pump geometrical parameters, impeller vane angel, impeller diameter, secondary flow, recirculation inside the impeller and hydraulic efficiency.

Chapter three will provide a comprehensive detail of basic informations and working principal of vertical turbine pump.

Chapter four is the pump hydraulic design will detail the process of designing the vertical turbine pump, including selection of design parameters, hydraulic and structural considerations, and material selection.

Chapter five will outline the methodology and experimental setup used to evaluate the pump's hydraulic performance, including instrumentation, measurement techniques, and test conditions. Pump test results are also used to investigate the effect of the impeller modification, speed variation and sump water level changes on pump performance.

Finally, chapter six is the test results and conclusions. This chapter presents the main conclusions of current study and suggest some future works to be continued.

CHAPTER 2: LITERATURE REVIEW

In the literature review, some of the recent published researches dealing with the flow simulation in the Vertical Turbine pump are introduced. In addition, some fluid and geometrical parameters such as fluid viscosity, pump dimensions, friction losses, and secondary flow, and their effect on the pump characteristics have been highlighted.

The pumps are divided into two basic groups, depending on the way in which the liquid is transferred from suction side to the delivery side of the casing as positive displacement pumps and impeller or rotodynamic pumps. The rotodynamic pump and impeller pump terms are firstly introduced by H. Addison. Based on the direction of the flow, the rotodynamic pumps are in the category of cased pumps, the moving element in rotodynamic pumps is the impeller which is the rotor mounted on the rotating shaft and increases the moment of momentum of the flowing liquid in the impeller.

The first known vaned diffuser pump was patented by Osborne Reynolds in 1875 and was called a turbine pump, the pump that is patented by Osborne Reynolds is named as “turbine pump” because of the similarity on the appearance of the pump to the steam turbine, the turbine pumps are first used as lifting water from the small diameter water supplies and irrigation wells. However, they are used in wide range of applications other than lifting water from irrigation wells such as used in circulation systems in the steel industry for cooling, water extraction from boreholes and rivers, sea water services, deep sea mining, extraction water from geothermal wells, city water district systems and etc. Moreover, the main advantage of using the vertical turbine pumps is the ability to assemble the stages in series connection thus increasing the pressure rise across the pump easily.

Jvalant B. Trivedi et. al (2020) studied that the streamline and contours of pressure and velocity in a Vertical Turbine pump (Impeller and Bowl). It is found that the hydraulic performance of the pump is affected by geometrical parameters like Eye Diameter, impeller vane angel, Impeller outlet diameter, impeller outlet width etc.

Malik N. Hawas et. al (2019) studied that pump design parameter that has most effecting in reducing stress can be identified by sensitive analysis. This result evaluates that contact pressure, Mises stresses and the shape parameter are the most affecting parameter. Simulation results shows that there's a positive relationship between specific speed and efficiency. Furthermore, it's found that the blade angle influence on vane performance can be neglected comparing with other design variables. The finding of optimization model shows that, the efficiency of optimized model is higher than the conventional type. Its also concluded that the overall pump performance will enhance by designing optimization process. The minimum pressure coefficient and efficiency can be improved by 0.35 and 0.40 percent respectively.

Sung Kim et. al (2019) found in his study that the two main research objectives namely efficiency and suction performance tend to contradict design optimization to improve the suction performance and efficiency of a mixed-flow pump impeller. He also points out due to the deviation between the pump design condition and its actual operating condition, flow turbulence and recirculation in the pump impellers produced enough vibration excitation forces that caused the mechanical seals to fail prematurely and to leak.

Ravindra Birajdar et. al (2021) studied that the critical hydraulic eccentricity of a Vertical turbine pump impeller by way of numerical simulations and design of experiments to find the vane pitch tolerance using critical hydraulic eccentricity. The effect of impeller vane pitch deviations on hydraulic unbalance is examined for a vertical turbine pump using Design of Experiments. A suitable orthogonal matrix has been selected with vane pitch at different axial locations of an impeller as the control factors. Hydraulic eccentricity, which is the output of the experiments is analysed using S/N ratio, ANOM and regression analysis to find the significant control factor effecting the hydraulic unbalance and hence vibrations. The vane pitch deviation at outlet and inlet of impeller shroud geometry are found to be the most critical factor affecting the pump vibrations.

Sambhrant Srivastava et. al (2014) studied that the design of mixed flow pump impellers of high specific speed is a much tougher than the other types of flow impellers and it becomes more complicated while deciding the blade positioning in the meridional annulus. In this work, natural frequency and deformation of mixed flow pump impeller were evaluated considering two different blade positions in the meridional annulus. ANSYS was used for the investigation of natural frequency and deformation. It was observed that the mixed flow pump impeller with inlet inclined blade position in the meridional annulus was more suitable than the trapezoidal one.

A. Manivannan (2010) studied the potential of enhancing efficiency of mixed flow pump, Computational Fluid Dynamics (CFD) analysis is one of the advanced tools used in the pump industry. From the results of CFD analysis, the velocity and pressure in the outlet of the impeller is predicted. These outlet flow conditions are used to calculate the efficiency of the impeller. From this analysis it is understood that the changes in the inlet vane angle did not change the efficiency of the impeller as much as the changes in outlet angle.

Divya Zindani et. al (2016) found in his study that designing of mixed flow impeller blades is still a cumbersome task, mainly due to the great number of geometric parameters. From the day of its invention various methods have been used for designing the same. Some of the prominent methods are inverse time marching method, the pseudo-stream function method, Fourier expansion singularity method, free vortex method, mean stream line theory method etc. All of these have their own merits and demerits. In this work mean stream line theory has

been used for the designing of blades. It has been observed that this method, although used sparingly, gave results comparable to the other methods.

Grapsas et al. (2008) studied a numerical methodology for hydrodynamic design in vertical turbine pumps considering the possibility to increase the hydraulic efficiency of the pump impeller by improving the blades shape. A parametric study is carried out to examine the influence of some blade design parameters on the performance and the efficiency of the impeller, like the blade length, the inlet height, and the leading-edge inclination. For stability reasons, the algorithm starts from low rotational speed and gradually reaches the nominal speed of 3000 rpm. The optimizer converges after about 100 completed solutions of the flow field to the optimum impeller, which has the following characteristics: blade wrap angle $\theta_w = 74^\circ$, inlet height $b_1 = 1$ & $2 b_2$ and leading-edge inclination angle almost vertical ($\varphi = 2.1$).

Djerroud et al. (2004) investigated numerically the effects of the pertinent design parameters, including the blade height, the blade number, the outlet blade angle, the blade width, and the impeller diameter, have on the steady state liquid flow in a three-dimensional centrifugal pump. Three cases were considered for this study: impeller, combined impeller and volute, and combined impeller and diffuser. The obtained results demonstrate that the pump head and the brake horsepower increase with increasing impeller blade number and impeller blade height, while they decrease with increasing impeller blade width. In addition, the interaction between the impeller and the volute reveals that the decrease of the impeller outer diameter keeping the volute dimensions constant leads to the reduction of the pump head and the brake horsepower. The pump overall efficiency is also influenced by the selected key design parameter. A relatively good agreement was observed comparing the developed numerical approach with the experimental results for the case of the combined impeller and diffuser obtained from a pump manufacturer.

B. Schiavello et. al (2005) investigate that the proper operation of vertical centrifugal pumps is greatly dependent upon the entire pump/piping system, which includes the piping geometry and the system operating conditions. Oftentimes, pumps operate satisfactorily during shop tests but experience problems after they are installed in the field. Here, three large vertical pumps that operated satisfactorily on the test stand experienced excessive vibration after installation. Additionally, pulsation in the system piping was found to be causing unexpected vibration in downstream equipment. It was discovered that the problems were the result of complex interaction between several phenomena. Improper inlet conditions caused suction recirculation that generated broadband turbulence. The turbulent energy excited acoustical resonances of the pump/piping system, resulting in pulsation at several discrete frequencies.

V.G Chhanya (2014) studied and found that the hydraulic performance of the pump was improved by enlarging the section area of volute and reducing the blade inlet angle also the

performance of the pump can be improved by reducing number of blades and increasing radial curvature of the original impeller blade. It is found that optimum design of mixed flow pump is achieved by reducing number of blades from 7 to 5, design of C type curve section reduces the occurrences of vertexes at the head of the blade and increases the blade inlet structure angle properly and reducing the blade outlet structure angle gives optimum performance. It also found that There is no exact pattern for inlet to outlet passage of the impeller. Design of the impeller can be done by assuming lowest possible losses. The inlet geometry is primarily influenced by parameter of suction and outlet geometry is achieved by considering the desired specific energy. The impeller should be designed with respect to the linear distribution of pressure from inlet to outlet.

Miraj S. Desai (2015) studied that the effect of some parameter on the performance of mixed flow pump impeller. These parameters such as inlet blade angle, outlet blade angle. The impeller has been designed and built to study the effect of these parameters with the help of software for analysis. In this paper change the inlet & outlet angle of blade with increase the angle 10%, 20% and decrease the angle 10%, 20% with respect to design angles. From these all simulation was done, we should consider the best inlet & outlet angle to take the best head & efficiency of mixed flow pump.

Chakraborty and Pandey (2011) studied the performance of impellers with the same outlet diameter having different blade numbers for centrifugal pumps is thoroughly evaluated the impeller outlet diameter, the blade angle and the blade numbers are the most critical parameters which affect the performance of centrifugal pumps. The model pump has a design rotation speed of 4000 rpm and an impeller with numbers of blades from 4 to 12 has been considered. The blade number has significant effects of pumps characteristics. With the increase of blade number, the total pressure gradually increases. They concluded that at 7 and 10 numbers of blades the efficiency is more than the other numbers of blades for the pump. The efficiency is dropping at 12 numbers of blades. Therefore, the optimum efficiency for 10-blade number is showing better results compared to other blade numbers.

Anagnostopoulos (2006) investigated the effects of the centrifugal pump impeller geometry on its hydrodynamic performance and hydraulic efficiency. Using Numerical simulation with new meshing technique based on Cartesian Grid. The important advantage from the use of Cartesian grids for complex geometries is the elimination of the computational effort and the automation of the grid generation process. However, the grid lines cannot in general fit in with the flow boundaries, therefore there are cells partly filled by the fluid that must be properly handled. A desirable feature of the method is that the same general form of the linearized equations can be used for all the grid cells, regardless of their blocked portion.

Cheah et al. (2007) investigated the flow field inside the centrifugal pump by using numerical flow simulation and compared the numerical result with experimental data over the wide

flow range. The comparison of the predicted head and efficiency between numerical results and experiment data shows a good agreement. At design point, the internal flow or velocity vector is very smooth along the curvature along the blades. However, at the leading edge due to non-tangential inflow conditions. The single and double vortical flow structures are observed in the volute casing. When operating at off-design load, the flow pattern has changed significantly from the well-behaved flow pattern at design load condition. A strong flow recirculation at the centre of the passage of the impeller can be observed. The stall region developed due to the recirculation is blocking the flow passing through the passage. In this case, the rotational effect is also an important factor to be considered.

Jidong Lia, et. al (2006) in his studies of Optimum design on impeller blade of mixed-flow pump based on CFD analysis Under filing of impeller blades at the trailing edge improved the performance of the pump that is designed in this study, as stated in References. Best efficiency points of the pump that is designed in this study moves from 53 l/s to 56 l/s and system efficiency increases 2% for the best efficiency point. The disturbance on the trailing edge of the blade caused by offsetting the designed surface in order to give thickness the blade is reduced, when under filing is performed in the impeller blades, while making the comparison of the CFD results; the convergence of the analyses should be obtained. If the instable analyses are faced by means of convergence, the number of elements should be increased and the analyses should be rerun for the stability. It also important to Select the suitable blade and vane thicknesses, blade swirl and vane swept angles are important from the manufacturing point of view.

Michal Varcholaa et. al (2011) in his studies of Geometry Design of a Mixed Flow Pump Using Experimental Results of on Internal Impeller Flow research of the velocity and pressure flow field in the impeller of the mixed flow pump.

A subject of this paper is a numerical solution of a mixed - flow pump geometry with respect to a distribution of a static reassure in the channel of the pump. The distributions of pressure and velocity fields were obtained through experiments. The blade's design was obtained according to the pressure distribution in the impellers' channel. The hydraulic projection of an impeller is very sensitive in terms of the overall efficiency as well as the position of the best efficiency point. Or said in a different way, the match of optimal flowrate and the specific energy measured and calculated. The pump designed through the described procedure, achieved its peak efficiency at the calculation point, which justifies the validity of the procedure. It can be said that the method used for projection of a blade cut based on the characteristic pressure distribution in the channel of the impeller, seems to be perspective for the prime projection of geometry of the diagonal pump.

Kiran Patel, et. al (2013) in his studies of CFD analysis of mixed flow pump derived that the Head predicted by CFD analysis is higher than the test result at rated point. It also concluded

that Power predicted by CFD analysis is higher at rated point to compare with the test result. Power predicted by CFD analysis is 5 to 10% higher at rated point. To compare with the test result, disc-friction power loss calculated using NEL method the volumetric efficiency is determined. Pump efficiency considering disc friction loss and leakage-loss is predicted and it was found within +5% ranges, at duty point. Efficiency predicted by CFD analysis is higher than the test result. Leakage-loss is predicted using. Efficiency is improved by 1% after matching stator angle and changing hub curve profile. Stator blade loading at hub and shroud has improved.

VASILIOS A et.al (2007) in his studies of CFD analysis of mixed flow pump derived that the mixed flow pumps the numerical results are compared to the measurements, showing good agreement and encouraging the extension of the developed computation methodology for performance prediction and for design optimization of such impeller geometries. A numerical methodology for the calculation of the flow field in centrifugal pump impellers with 2D curvature is developed and validated against corresponding experimental data taken at a Laboratory test rig. The flow is calculated using a two-dimensional approach in order to achieve a fast simulation and the agreement between the numerical results and the measurements is satisfactory. The pressure fluctuation in a vaned diffuser downstream from a centrifugal pump impeller. This is quite encouraging result in order to apply the present numerical model to further flow analysis, as well as, for design optimization purposes in these pump types.

Derakhshan et al. (2013) studied and developed an efficient and original approach and applied this approach to the design of centrifugal pump impellers in order to achieve a higher efficiency. A global optimization method based on the Artificial Neural Networks (ANNs) and Artificial Bee Colony (ABC) algorithm has been Arbitrary design parameters can be used as input data for a pump software in order to generate the impeller geometry. The performance of an impeller depends on various parameters. However, the main parameters are Hub diameter, suction diameter, impeller diameter, impeller width and inlet and outlet blade angles. Numerical results showed efficiency improvement of 3.59% at only 6.89 m increase of total pressure difference for the centrifugal pump. The new impeller geometry presents much more changes in the meridian channel and blade profile. The results indicate reasonable improvement in the optimal design of pump impellers and a higher pump performance when using the ABC algorithms. The optimization by ABC algorithms could be exploited as a beneficial approach in the optimal design of pumps.

The literature survey on the design, construction, and experimental study of vertical turbine pumps covers a wide range of research and advancements in this field. The following is a summary of the key findings and insights from the reviewed literature:

Hydraulic design is critical for achieving high efficiency and performance in VT pump. Research has focused on optimizing impeller design, including blade profile, number of

blades, and blade angle, to enhance flow and pressure characteristics. Computational fluid dynamics simulations are extensively used to refine impeller designs and minimize hydraulic losses. Diffuser design, which converts kinetic energy into pressure energy, also plays a significant role in improving pump efficiency.

Structural integrity is paramount for reliable pump operation. Studies have examined the design of shafts and bearings to withstand operational stresses and reduce wear. The pump housing and column design must endure internal pressures and mechanical loads, making material selection crucial. Materials such as stainless steel and composites are often evaluated for their strength, corrosion resistance, and cost-effectiveness.

Cavitation, caused by vapor bubble formation, can damage pump components and reduce efficiency. Research emphasizes understanding cavitation phenomena and implementing strategies to mitigate it, such as optimizing impeller design and maintaining adequate Net Positive Suction Head (NPSH). Ensuring sufficient NPSH is essential to prevent cavitation and ensure smooth pump operation.

Experimental validation and CFD simulations are vital for pump design and performance optimization. Detailed experimental setups and measurement techniques help in evaluating pump performance under various operating conditions. CFD simulations provide insights into fluid flow patterns, pressure distribution, and help in identifying areas for design improvement.

Ensuring operational flexibility and effective control is crucial for adapting VT pump to varying system demands. The use of Variable Speed Drives and advanced control algorithms helps optimize pump operation, improving energy efficiency and performance.

Despite significant advancements, several research gaps remain. There is a need for integrated design approaches that balance hydraulic performance with structural integrity and material selection. Exploring advanced materials and coatings for enhanced durability and corrosion resistance is also essential. Multi-objective optimization techniques and smart technologies, such as IoT-enabled pumps, present promising areas for future research.

CHAPTER 3: THEORY: BASIC INFORMATIONS AND WORKING PRINCIPAL OF VERTICAL TURBINE PUMP

3.1 Basic definitions of Vertical Turbine Pump

Vertical turbine pump is a vertical axis centrifugal, Francis or mixed-flow type pump comprising stages which accommodate rotating impellers & stationary bowls possessing diffusion vanes. The discharge from the pump unit is co-axial with the shaft and the pumps unit is suspended by the column pipe containing a system of vertical shafting for transmitting power to the impellers, the prime mover being external to the flow stream.

A vertical turbine pump can be a vertical-axis centrifugal type, which can be a multi-stage assembly as well. Also, the pump can be a mixed-flow type vertical pump, or an axial-flow vertical impeller type.

3.2 Elements of a Vertical Turbine Pump

A vertical turbine pump (which can be either self or oil-lubricated) may have impeller that either enclosed or semi-open. Basically, the important elements of the pump are: the main pump element-bowl Assembly, column assembly, and discharge head assembly.

Bowl Assembly: This is either a single stage or multi-stage centrifugal, Francis or mixed-flow type vertical pump with discharge co-axial with the shaft. The bowl assembly consists of rotating impellers, which are housed in stationary bowls having diffusion vanes.

Column Assembly: The column assembly connects the bowl assembly and the head assembly and conveys water from bowl assembly to head assembly. This comprises column pipe, shaft enclosing tube (In case of Oil/Forced water lubricated pump only) and shaft assemblies. The line shaft is supported throughout its length by means of bearings and may be enclosed in the shaft enclosing tube if lubricated with oil/external fresh water or it may be open and lubricated with the water being pumped.

Head Assembly: The head assembly consists of the base from which the column, shaft assembly and the bowl assembly are suspended, the discharge head elbow which directs the water into the delivery piping system, and also support the driver. Surface Discharge In this type of discharge, the discharge elbow or tee is incorporated in the head assembly so that the delivery may be taken at ground level.

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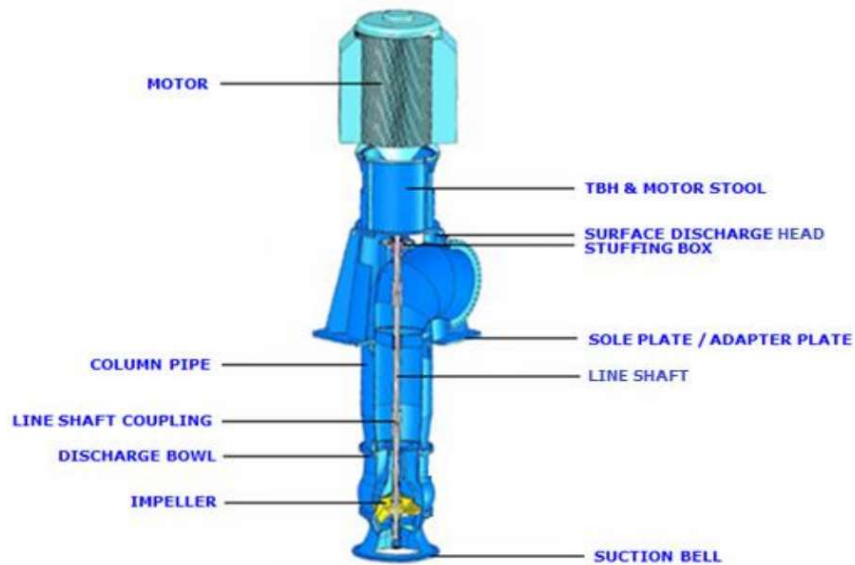


Fig- 3.1: Surface Discharge Vertical Turbine Pump

Under-ground Discharge: In this type of discharge, the discharge elbow or tee is separated from the head assembly and installed in the column pipe at the desired level below the head assembly

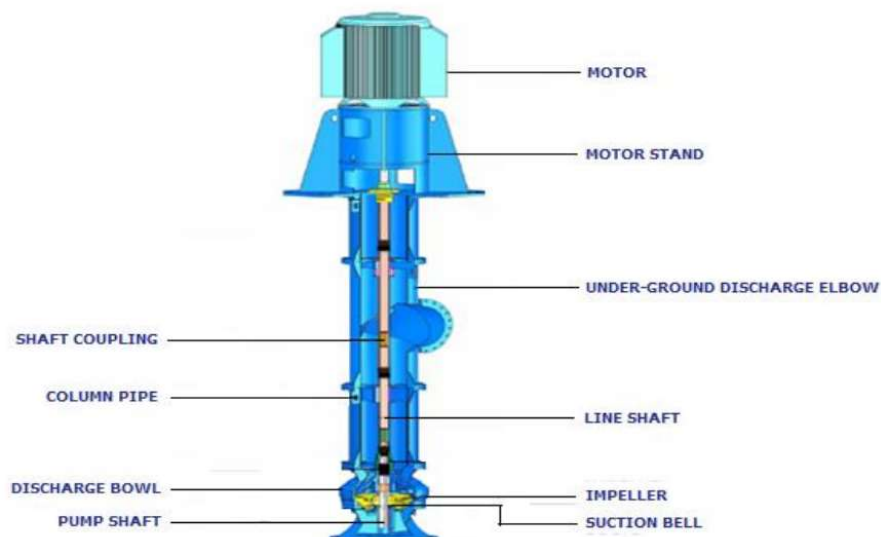


Fig- 3.2: Under-ground Discharge Vertical Turbine Pump

Driver:

The mechanism mounted on the discharge head which transmits power to the impellers through the shaft assemblies. It may or may not be prime mover. The type of drivers are as follows: -

Hollow Shaft Motor: A vertical motor whose shaft is bored hollow for accommodating the head shaft of the pump. The motor contains a bearing or bearings for carrying down-thrust of pumps and also provides for impeller adjustment at its top.

Solid Shaft Motor: The pump head shaft is coupled to Motor Shaft through a flexible coupling which can transmit torque but cannot transmit axial thrust. In this case, a thrust bearing is provided on the pump discharge head to take care of axial thrust generated in the pump.

Belt Driven: A flat or V-belt driven mechanism having a shaft which may be bored hollow to receive the head shaft of the pump. In case of hollow shaft construction, impeller adjustment is made at the top of the pulley head. Belt head is provided with a bearing to carry the total down-thrust as a bearing to carry the belt pull.

Gear Head: A right angle gear-drive arrangement having vertical shaft which, if bored hollow, accommodates the head shaft of the pump. The horizontal shaft of the gear head receives power from the prime mover. Impeller adjustment is possible at the top of the gear head in case of the hollow shaft construction and bearing is provided on the hollow shaft for taking up the down-thrust.

3.3 Vertical Turbine Pump Terminology

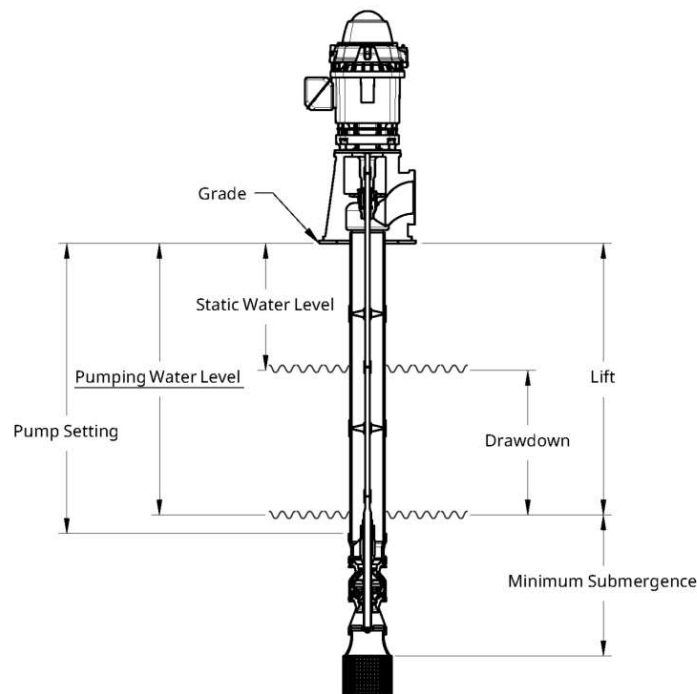


Fig- 3.3: Vertical Turbine Pump Terminology

Grade: The elevation of the surface supporting the pump

Datum: The elevation of the underside of the discharge head.

Setting: Distance in meters from the datum to the column pipe connection at the bowl assembly.

Static Water Level: The level of the free pool when no water is being drawn from it.

Pumping Water Level: The level of the free pool when pump is working.

Draw-Down: The difference between pumping water level and static water level.

Capacity of the Pump (Q): The discharge of the pump at the specified head and speed.

Pump Speed (N): Revolutions per minute of the shaft.

HEADS

Head Below (h_b): The vertical distance in metres from the datum to pumping water level.

- **Head Above (h_a):** The Pressure head above datum plus the velocity head.
- **Velocity Head (h_v):** The kinetic energy per unit weight of water at a given section, and is expressed by, $h_v = \frac{V^2}{2g}$, where 'V' is the average velocity at the cross section of measurement in Mt./s and 'g' is the acceleration due to gravity $g = 9.81 \text{ m/s}^2$.

Total Dynamic Head (h): The head above plus the head below. It is also the bowl assembly head minus the total pump internal frictional head loss. This is the head generally called for in any pump specification. In open-suction installations, it is algebraic sum of the head below the datum and head above datum. In closed-suction installations, it is algebraic sum of the suction head, the distance between suction case flange and the datum, and the head above datum.

Total Bowl Head (H): The head developed by the bowl assembly and is equal to dynamic head plus total pump internal head loss.

Total pump internal head loss is the sum of bell entry head loss (h_{be}), column head loss (h_c) & discharge elbow head loss (h_{de}).

Therefore, $H = h + h_{be} + h_c + h_{de}$ (1.1)

Column Friction Loss - Losses incurred by the flow of water through the pump column assembly.

Shaft Friction Loss - The horsepower required to overcome the friction in the lineshaft bearings.

Total Pump Thrust - The sum of the shaft weight and the hydraulic thrust created by the impellers moving liquid.

3.4 Basic Design Features

The design of vertical turbine pump must consider several elements unique to the method of pump construction that include at a minimum:

- Capacity

- Total Head
- Power, supply voltage
- NPSH (Net Positive suction Head)
- Submergence
- Thrust
- Suction Condition
- Sump details – Well diameter
- Water levels (Static & Pumping)
- Method of lubrication
- Working Fluid condition
- Abrasive (sand) condition
- Driver type and speed

Each of this parameter must at least be considered, although not all are applicable for every condition.

Capacity:

Obviously, as is the case with any pump, the primary consideration is the required capacity and discharge head. Actually, the required flow capacity is usually the primary design criteria, with head coming in second. In a deep well application, the limiting factor of maximum capacity will generally be related to the sump or well diameter. Typically, the maximum efficient capacity of a turbine bowl will be related directly to the outer diameter of the bowl assembly; therefore, the designer must consider the minimum well diameter before the bowl selection can be made.

Well Diameter:

Generally, the well should provide a diameter at least 4 inches larger than the normal small size bowl diameter and for large size it has to be minimum 8 to 10 inches. In other words, if the required flow was 1200 m³/hr, necessitating at least 23-inch bowl diameter, the well diameter should be 28 inch minimum.

This allowance is needed to compensate for any well deviation or misalignment and to permit the pump bowl and shaft to operate straight and plumb and without deflection and binding. In those cases where the well is of ample size regardless of the required flow, it is usually desirable to select a bowl with the largest diameter and the fewest stages that is most efficient for the required flow.

Head Requirements:

Head requirements for vertical turbine pumps are determined in the same way as for any other type of well pump; however, the selection of a specific turbine pump is unique. Curve for VTP are developed based on capacity as the primary element, with head and bowl input

power per stage shown rather than the total head usually found with pre-assembled submersible style pump units.

This difference allows the designer total flexibility during the design of the pump as the bowl can be selected, using the required number of stage and if necessary, impeller trimming for the precise design condition. After calculating the total bowl head, the total head is divided by the head per stage indicated on the curve of the selected bowl assembly to determine the number of stages needed to generate the required head. Trimming of impellers (modifying the outer diameter) can be performed to fine- tune the selection when the total required head falls between the calculated number of stage or standard impeller trim.

Submergence:

With the bowl assembly selected, the next step will be to determine the required pump setting, the size of column and shaft, and type of shaft lubrication method. The pump setting is mainly determined by well factors- chiefly diameter, depth, and pumping levels- as well as the need to maintain sufficient water head over the inlet of the pump (technically refer to as submergence) at all anticipated flow conditions to avoid the introduction of air and possible cavitation during operations.

Net Positive Suction Head (NPSH): NPSH is another pump term while usually not a critical design factor for deep well turbine pump, still requires some limited discussion. Simply stated, as the pump impellers spin, they develop a negative pressure or vacuum (commonly, but incorrectly, referred to as suction) in the impeller eye. There must be adequate pressure, either from atmospheric pressure (head) or generated water pressure to push the water into the impeller eye and relieve this negative pressure.

If insufficient head or pressure exists, the liquid's physical condition will fall below its vapour pressure, resulting in vaporization of the fluid. This is commonly referred to as cavitation. The sound that accompanies the cavitation is actually collapsing these vapour bubbles as they travel through the impeller and reach a point of higher pressure.

Net positive suction head (h_{sv}) is the total suction head (h_{sa}) of liquid in metres absolute, determined at the suction nozzle and corrected to datum, less the vapour pressure of the liquid in metres absolute h_{vps} at the prevailing temperature of liquid.

Therefore, Available NPSH,

$$h_{sv} = h_{sa} - h_{vps} \dots\dots\dots (1.2)$$

$$\text{or, } h_{sv} = h_s - h_{vps} + h_a$$

$$[h_{sa} = h_a \pm h_s, \text{ plus sign for suction head and minus sign for suction lift}]$$

$$h_a = \text{Atmospheric pressure in meters absolute}$$

$$h_s = \text{For vertical pump head between minimum water level in the sump and eye of the impeller}$$

Notes: -

- 1) The available NPSH is a characteristic of the system in which the pump works and the required NPSH is a function of pump design and represents the minimum required margin between the suction head and the vapour pressure at a given capacity.

For satisfactory and smooth run of a pump within a system,

Available NPSH $\geq$ Required NPSH

- 2) NPSH is defined for two conditions: -

- NPSH value when cavitation occurs.
- NPSH value which contains a margin of pressure to ensure operation sufficiently free from cavitation that the efficiency is not prejudiced.

- 3) Altitude/elevation correction if any, needs to be considered, for determination of h_a

Specific Speeds (U.S. Unit):

Pump Specific Speed is a method of characterizing pumping conditions and it may be used to determine the most appropriate pump design for a given application. Often pump characteristics such as NPSH_r, suction lift and pump efficiency will be correlated against the pump specific speed.

Specific speed is the speed (in RPM) a geometrically similar impeller would need to operate in order to deliver 1 L/s of fluid at a head of 1 m (US units: 1 USgal/min at a head of 1 ft).

The common pump specific speed is not dimensionless; however, its dimensions are only important for converting between different unit sets (eg converting and comparing pump specific speeds calculated on a Metric basis and US Customary basis). A truly dimensionless variation on the pump specific speed is available, but is less commonly used

- Pump specific speed, $N_s = \frac{N\sqrt{Q}}{H^{3/4}}$ (1.3)

- Pump suction specific speed, $N_r = \frac{N\sqrt{Q}}{(NPSH_r)^{3/4}}$ (1.4)

- System suction specific speed, $N_a = \frac{N\sqrt{Q}}{(NPSH_a)^{3/4}}$ (1.5)

Here,

Q = Pump discharge in USGPM.

H = Total bowl head per stage in Ft.

N = Pump rotational speed in RPM.

NPSH_r = Net Positive Suction Required in Ft.

$NPSH_a$ = Net Positive Suction Available in Ft.

Notes: -

- 1) For pumps of normal design, value of 'Nr' should not exceed 12000.
- 2) For proper suction conditions, value of 'Na' should be within 8500.

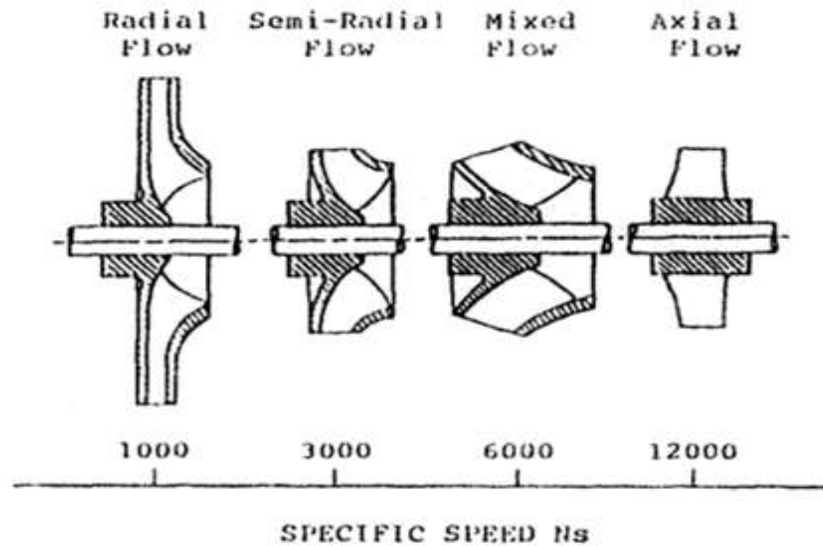


FIG : 3.4 Specific speed and impeller profiles.

POWER:

Bowl Input Power (P_{bi}): The power in kilowatts applied to the impeller shaft. This required power needed by the bowl assembly is also determined by multiplying the number of stages by the power line usually indicated on the bottom-line of the pump curve. Bowl input power can also be calculated by using the following mathematical formula,

$$P_{bi} = \frac{Q \times H \times \text{Sp.Gr.}}{367.2 \times \eta_b} \dots\dots\dots (1.6)$$

Where,

Q = Pump discharge in 'm³/hr.'

H = Total bowl head of pump in 'm'

η_b = Bowl efficiency in '%'

Sp. Gr. = Specific Gravity of the fluid

Pump Input Power (P_{pi}): The power in kilowatts applied to the head shaft and is equal to bowl input power plus total pump internal power loss P_{Loss} .

Mathematically, $P_{pi} = P_{bi} + P_{Loss} \dots\dots\dots (1.7)$

Pump Output Power (P_o): The power in kilowatts delivered by the pump.

$$\text{Mathematically, } P_o = \frac{Q \times h \times \text{Sp.Gr}}{367.2} \dots\dots\dots (1.8)$$

Where,

Q = Pump discharge in 'm³/hr.'

h = Total dynamic head of pump in 'm'

Sp. Gr. = Specific Gravity of the fluid

Total Pump Internal Power Loss (***P_{Loss}***): The power losses in kilowatts within the pump during pump operation.

Internal power losses are of two types: -

- Shaft loss *P_s*
- Thrust bearing loss *P_t*

$$\text{Mathematically, } P_{Loss} = P_s + P_t \dots\dots\dots (1.9)$$

Power Input at Motor Terminal (*P_{di}*): The ratio of pump input to motor efficiency.

$$\text{Mathematically, } P_{di} = \frac{p_{pi}}{\eta_m} \dots\dots\dots (1.10)$$

Note:-

$$\text{Motor loading, } ML\% = \frac{\text{Pump input (} p_{pi} \text{)}}{\text{Motor rating}} \times 100 \dots\dots\dots (1.11)$$

η_m = Motor efficiency at ML% motor loading

Efficiencies

Bowl Efficiency *η_b*: The bowl efficiency in percentage can be expressed as follows:

$$\text{Mathematically, } \eta_b = \frac{Q \times H \times \text{Sp.Gr.}}{367.2 \times P_{bi}} \times 100 \dots\dots\dots (1.12)$$

Where,

Q = Pump discharge in 'm³/hr.'

H = Total bowl head of pump in 'm'

P_{bi} = Bowl input power in 'kw'

Sp. Gr. = Specific Gravity of the fluid

Bowl efficiencies for common flows between manufacturers can also vary widely. As an example, for our earlier flow example 2600 m³/hr, I was able to compare curves for several 30-inch bowl from different manufacturer with efficiencies as low 76 percent upto 88 percent. When designing a vertical turbine pump, a good designer will investigate several selections from various manufacturer and not limit the customer's choice to only those brands he sells regularly.

Pump Efficiency: The ratio of pump output to pump input expressed as percentage.

Mathematically, $\eta_p = \frac{P_o}{P_{pi}} \times 100\%$ (1.13)

Overall Efficiency: The ratio of pump output to drive input expressed as percentage.

Mathematically, $\eta_o = \frac{P_o}{P_{di}} \times 100\%$ (1.14)

3.5 Pump Performance at different Speeds /Affinity Laws

The performance of a pump at any speed can be accurately predicted if the performance at one speed is known. Affinity Laws can be stated as such:

The flow is directly proportional to the speed:

$$\frac{Q_1}{Q_2} = \frac{N_1}{N_2} \text{ (1.15)}$$

The head is proportional to the square of the speed:

$$\frac{H_1}{H_2} = \frac{N_1^2}{N_2^2} \text{ (1.16)}$$

The Power is proportional to the cube of the speed:

$$\frac{P_1}{P_2} = \frac{N_1^3}{N_2^3} \text{ (1.17)}$$

Q_1, H_1, P_1 and N_1 represent the capacity, head, power and speed of a pump at known speed and Q_2, H_2, P_2 and N_2 represent the same items at any other speed for which the performance to be predicted.

Motor Speed

The synchronous speed of A.C. motors are determined by the number of poles and frequency

$$\text{Synchronous Speed} = \frac{120f}{P} \text{ (1.18)}$$

f = Frequency in cycles/Sec

P = Number of poles of the motor

Induction motors will have full load speed of from 2 to 6 percent less than above.

3.6 Working Principle of Vertical Turbine Pump

The vertical turbine type pumps are mainly composed of four subassemblies. These subassemblies are the driver, discharge head, column assembly and the pump assembly. The pump assembly is also composed of several parts which are shown in Figure 3.3.

The power is transmitted from the electric motor or any other type of driver such as diesel engine to the pump. There are different types of electric motors used in different applications. If the pump is driven from the top, the discharge head is used and Vertical Hollow Shaft (VHS) or Vertical Solid Shaft (VSS) electric motors may be applicable for this kind of applications. The length of the column pipe and the intermediate shaft is adjusted for the different installations. If the length of the installation is increased the number of the bearings which are used to hold the intermediate shaft is also increased.

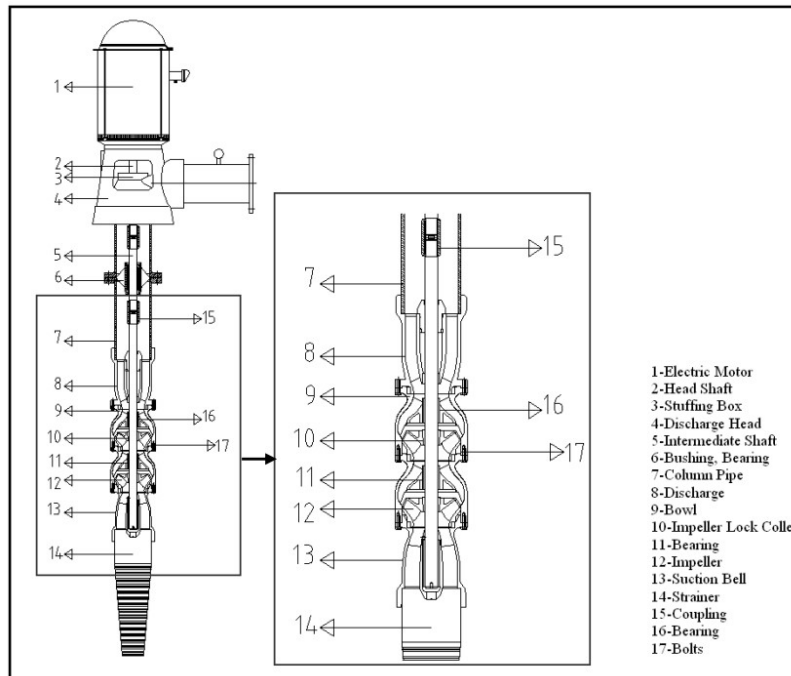


Figure 3.5: Vertical Turbine Pump Parts

In the applications where the driver is located on the top of the discharge head, the power is transmitted to the pump by means of several shaft connections. The reason of using several shafts to transmit the power is due to flexibility of the installation. The head shaft which goes through the discharge head is placed between the intermediate shaft and motor. The stuffing box is placed into the discharge head which is preventing the water coming from the column assembly leak into the motor side. The column assembly is composed of pipes which are connected to each other and the bottom end of the column assembly in the vertical direction is connected to the discharge part of the pump. The intermediate shaft is located in the column assembly and connected to the head shaft and pump shaft by means of coupling connections. However, the intermediate shaft is centred in the column assembly by means of journal bearings. The distance between the bearings are adjusted by concerning the rotational speed of the pump. The pump shaft which is holding the impeller is connected to the intermediate shaft. The impellers are locked on the shafts by means of impeller lock collets or key connections, [3]. The bowls are connected to each other by means of bolt and nut. The impeller is located inside the bowl in vertical turbine pumps and is shown in Figure 5. The bearing inside the bowl is used to align the shafts inside the pump assembly. The manufacturing tolerances are important while producing the bearing inside the bowls due to mechanical efficiency point of view. Nevertheless, the balance occurring due to manufacturing of the impeller is also checked before the impeller is connected to the pump shaft. At the lower end the suction intake is located where the journal bearing inside it, allows the shaft to be aligned from the bottom end of the pump assembly.

3.7 Advantages of Vertical Turbine Pumps

The vertical turbine pump is a trusted and proven workhorse pump found in industrial environments around the world. This pump design is continually improving operations in key global industries such as oil and gas, petrochemical, chemical, power, desalination and mining. Now verticals are frequently selected as a strong solution to common challenges where low flow and high head pumps are required. Despite this, there can be a lack of understanding for design engineers, plant managers and site operators about when a vertical turbine pump is an optimal solution compared to traditional horizontal pumps.

Smaller Footprint

Vertical turbine pumps take up much less floor space and fit into tighter areas while offering a smaller overall footprint. This makes them very popular in situations where space is limited. Naturally, they do take up more vertical space with taller equipment configurations, which is another consideration to keep in mind when considering a vertical pump.

Base Material Upgrades

Base materials in a vertical pump are fairly easy to upgrade. This can reduce wear and tear on the equipment, thus increasing the pump's lifespan and decreasing downtime for your plant.

Scalability

To generate the required positive suction head with a vertical turbine pump, its length can be extended below the ground. Phasing is allowed and adjustments are generally simpler to make, allowing for more system customization and adaptability to different environments and applications.

Efficient Operation

Most vertical turbine pumps are self-priming, which eliminates one common challenge with many horizontal pumps. The operator does not have to fill the pump with liquid, remove air, or create a vacuum. Vertical pumps can operate without priming and the first stage impeller can start pumping with a minimum liquid level.

Motor Control

The motor's location on top of a vertical turbine pump offers convenience in monitoring and controlling. Similarly, motor placement keeps the motor away from flood zones to avoid damage and safety concerns.

Safety

Vertical turbine pumps offer numerous safety advantages, including a submerged pump element, top-mounted drive motor, and steeper head-flow curves. Each feature contributes to a safer operation and reduced cavitation.

Versatility

Vertical turbine pumps utilize hydraulic designs that offer excellent flexibility. They can handle the lowest net positive suction head available (NPSHa), broad specific speed (Ns) range, and pressure by staging. The self-priming capabilities also offer added versatility for a variety of environments and applications.

The vertical turbine pump is easy to install and maintain. It is a popular choice among end-users, contractors, and installers. The technology is easy to use, and it is reliable. The efficiency of the pump is very high, and its low-priced and durable design makes it a good investment. The vertical turbine pump is the best choice for both residential and commercial use. The benefits of the Vertical Turbine Pump are clear.

CHAPTER 4: HYDRAULIC DESIGN AND PRODUCTION OF THE PUMP

4.1 General Information on the Design of the Pump:

The pump that is aimed to be designed in this study is a vertical turbine pump. Hydraulic design of the pump is mainly composed of impeller, bowl and suction bell design. The design procedures in the literature together with the experience and know-how coming from the company are used during the design of these parts. Since there is not enough information on the impeller layout design, a trial and error procedure is applied in order to design the impeller layout. Moreover, the preliminary design and impeller layout design are performed together in order to get the best impeller layout profile for the designed pump. The shaping of the blade is performed by using point by point method described in the related section of the impeller design. The bowl design is performed mainly by considering the absolute velocity distribution inside the vane to vane passage of the bowl. The suction intake is also designed in this chapter.

4.2 Hydraulic Design of Impeller:

In this study, hydraulic design of the impeller consists of three parts. First part is the design of impeller layout profile. Starting with an existing impeller layout profile, new impeller layout is designed in order to meet the hydraulic characteristics of the pump to be designed. The iterations are performed to adjust the best shape of the impeller layout as discussed in design of impeller layout profile section. The preliminary design, which is the second part, is performed coupled with impeller layout profile design for finding out the necessary parameters for the design. The experience and recent design studies are used while eliminating these parameters.

4.2.1 Design of Impeller Layout Profile:

The shape of the impeller layout profile is related to the specific speed of the pump. In order to start the design of the impeller layout profile, non-dimensional specific speed, N_s of the pump to be designed is calculated.

$$\text{Pump specific speed, } N_s = \frac{\omega \sqrt{Q}}{(gH)^{3/4}} \dots\dots\dots (4.1)$$

where, ω is in rad/s, Q is in m^3/s , g is in m/s^2 and H is in m. The non-dimensional specific speed of the pump to be designed is found to be 1.26. Impellers with blades of single curvature are among the simplest. They are used in pumps with low specific speeds ($N < 0.57$) and discharges of up to 140 l/s. Since the specific speed of the pump to be designed is above this limit, impeller with blades of double curvature is decided to be used.

If an impeller already exists, it is always a good idea to analyse the existing design. In order to design the impeller layout, first the impeller layout of the pump which has the closest specific speed at best efficiency point is chosen from the impeller layout library of WPIL LIMITED. The

selected profile from WPIL LIMITED impeller layout library is shown in Figure 4.1 The certain parametric changes in the layout are made in order to attain the hydraulic characteristics of the pump to be designed in this study.

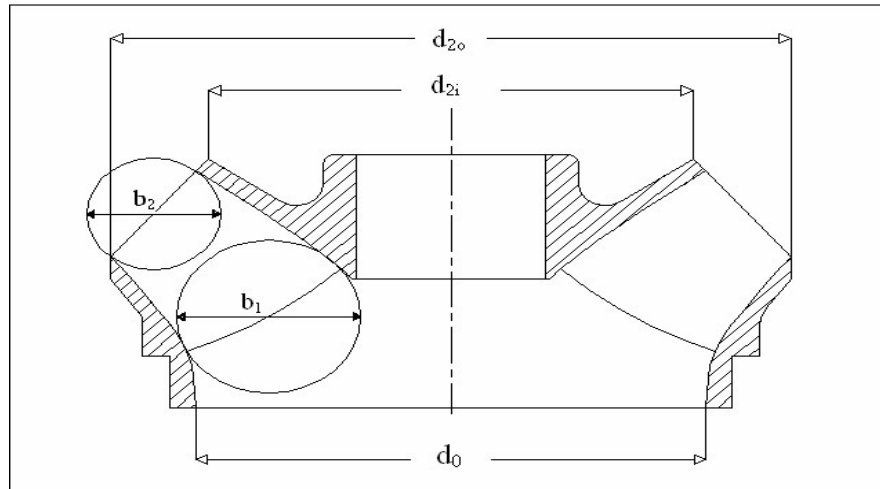


Figure 4.1: Selected impeller layout profile from WPIL LIMITED Library

The design point of the pump is stated again in order to define the specific speed and flow properties in other units. Specific speed in fundamental units is a dimensionless number. However, the more commonly used value is in U.S. units. Specific speed in U.S units is given by:

$$N_s = \frac{n\sqrt{Q}}{H^{3/4}} \quad \text{..... (4.2)}$$

where n is in rpm, Q is in USGPM and H is in feet. The specific speed of the pump in U.S. units is 3432. The non-dimensional specific diameter, Δ , is selected from Figure 4.2- Cordier Diagram.

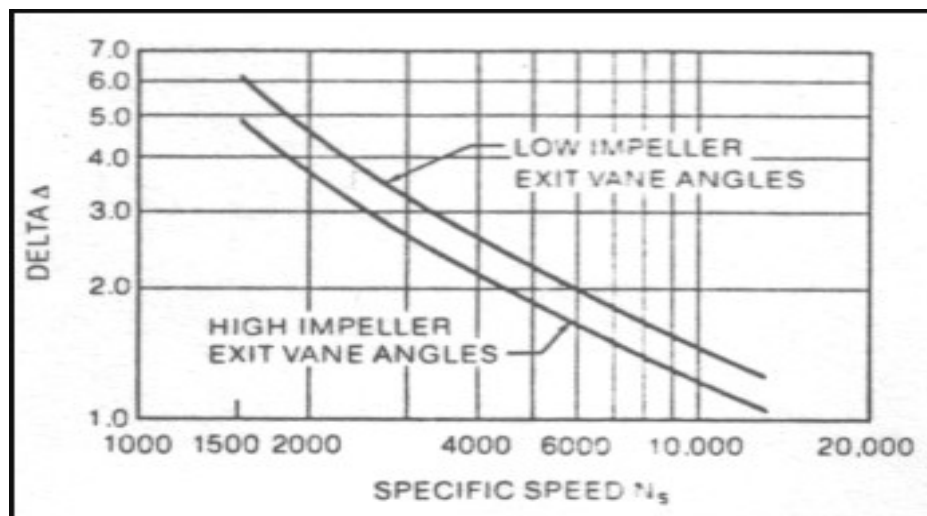


Figure 4.2: Cordier Diagram for selecting Δ value

After determining Δ_{high} and Δ_{low} from Figure 7, the upper and lower values for the median impeller diameter, d_{2m} , are calculated by putting the Δ_{high} and Δ_{low} values to the formula given by,

$$\Delta = \frac{(gH)^{\frac{1}{4}} d_{2m}}{(Q)^{1/2}} \dots\dots\dots (4.3)$$

The dimensional parameters that are evaluated during the design of the impeller layout profile are illustrated in Figure 4.3. Those parameters are the median diameter, d_{2m} , impeller shroud diameter at the trailing edge, d_{2o} , impeller hub diameter at the trailing edge, d_{2i} , impeller inlet breadth, b_1 , impeller outlet breadth, b_2 , impeller hub diameter at the inlet, d_h , and impeller inlet eye diameter d_o .

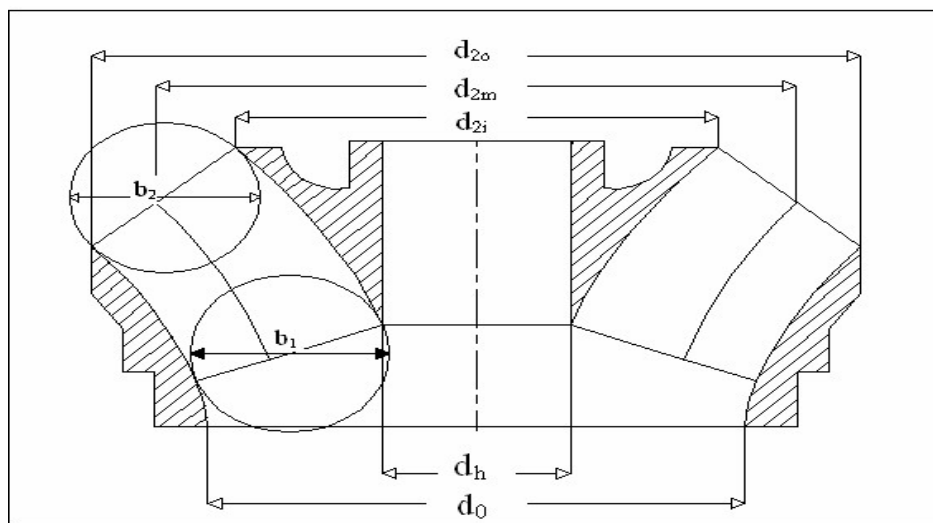


Figure 4.3: Impeller layout profile of pump showing the main geometrical parameters

After determining the, d_{2m} values for high and low impeller exit vane angles the average value of, $d_{2m-high}$ and d_{2m-low} is taken for the first assumption. d_{2o} value is calculated by using the chart given in Figure 4.4:

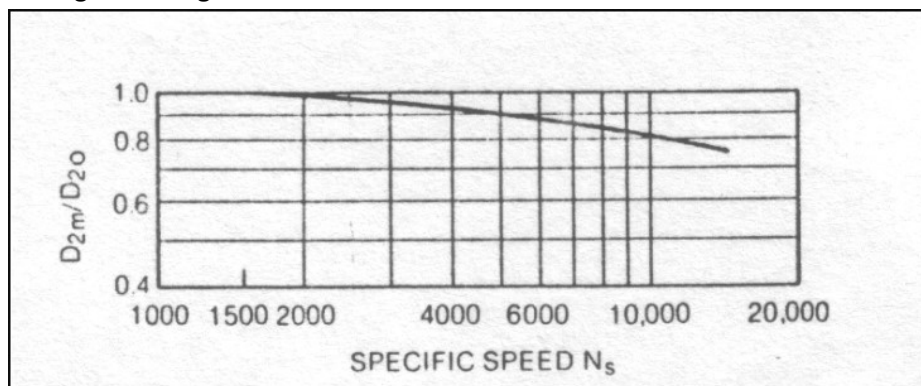


Figure 4.4: Design chart showing d_{2m} / d_{2o} ratio

Found d_{2o} and d_{2m} values are drawn on the impeller layout in order to check the dimensions as shown in Figure 4.5.

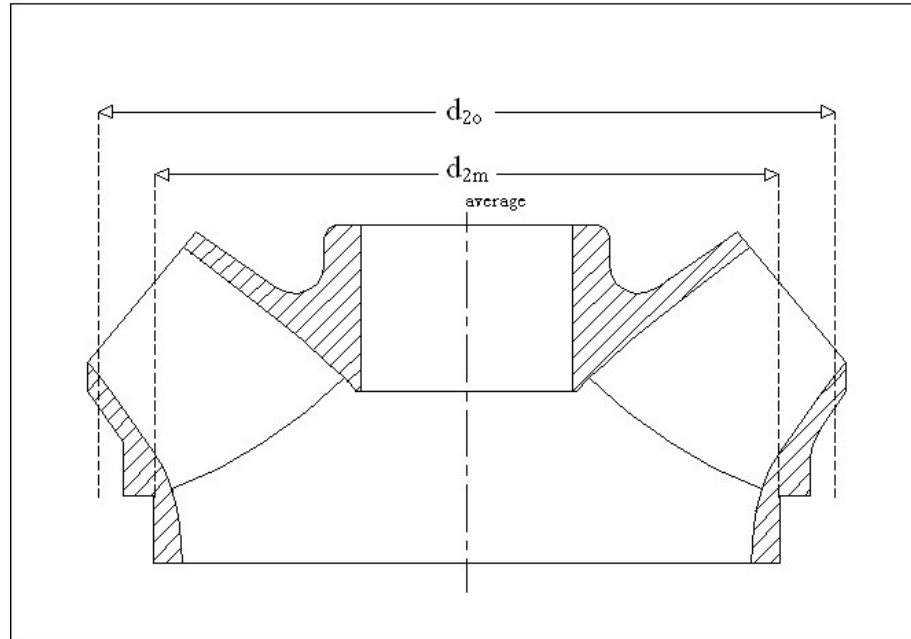


Figure 4.5: –Impeller layout showing found d_{2o} and d_{2m} -average values

d_{2i} value, which is shown in Figure 4.3, is calculated by using the formula:

$$d_{2m} = \sqrt{\left(\frac{(d_{2o})^2 + ((d_{2i})^2)}{2}\right)} \dots\dots\dots (4.4)$$

The outlet breadth, b_2 is determined by using the chart shown in Figure 4.6. The average value of the b_2 is taken for the first guess. However, the value is corrected within the range shown in Figure 4.6 during the iteration procedure as explained in the later steps of the impeller layout design. It is kept in mind that meridional velocity at the exit depends on the value of, b_2 . The relation of b_2 and exit area is given in the impeller exit calculation part of the preliminary design.

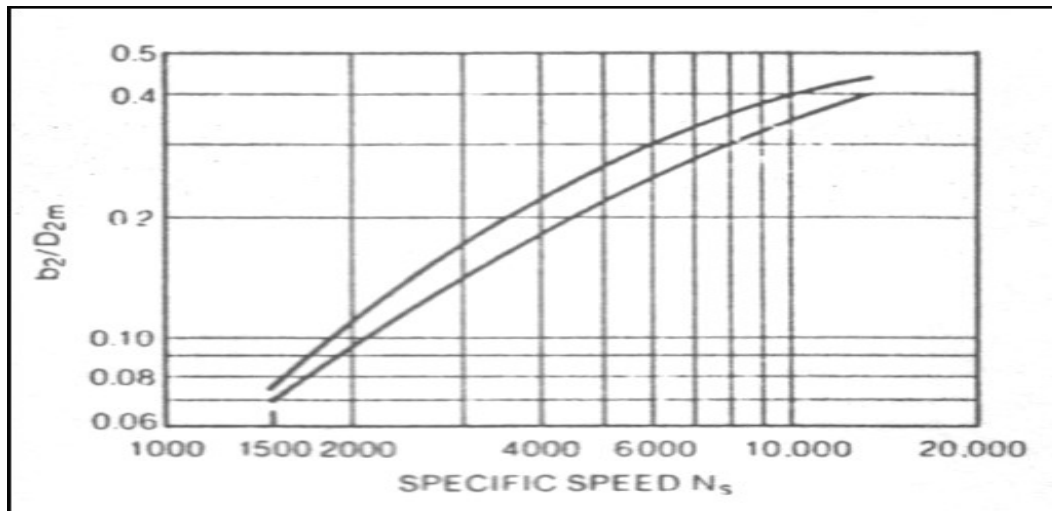


Figure 4.6 – Design chart showing b_2/d_{2m} ratio

After determining the b_2 value, the circle which has the radius of b_2 is drawn. The center of this circle lies at the intersection points of the d_{2o} and existing outlet breadth of selected profile from the library. The circle which is drawn in order to construct the outlet breadth of the pump to be designed is shown in Figure 4.7.

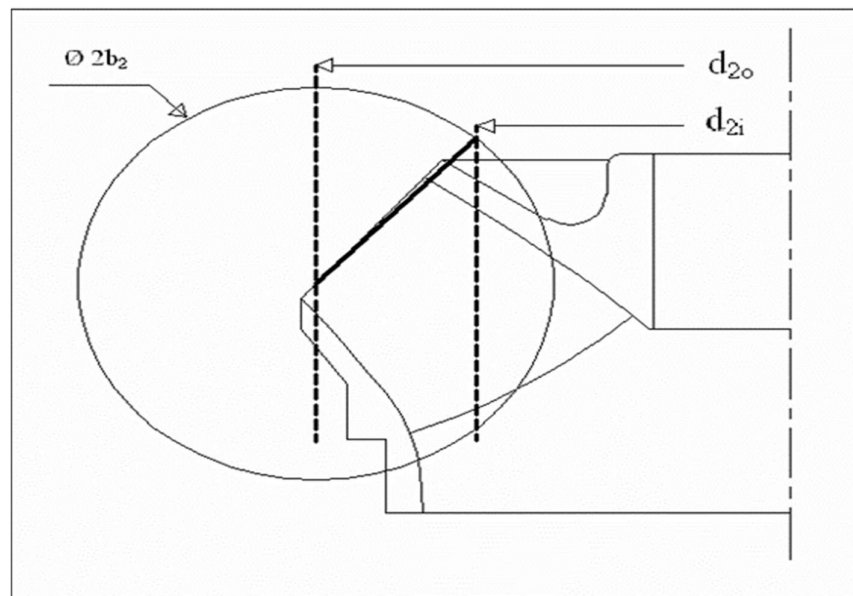


Figure 4.7 – Impeller layout showing found d_{2o} and d_{2i} values

However in order to construct the starting point of the arc for the hub side; the shaft size and the hub diameter should be determined. The shaft size of the impeller is calculated for multiple stages. The selection of the stage number is made by considering the demand for the pump to be designed from the previous sales statistics and adding several stages to this value as safety factor. Indeed, the shaft material is also important as seen in Equation (v). Available

shaft materials in the company and future demands are analyzed and suitable material is selected. The selected material is ASTM A276 SS410. The diameter of the shaft is calculated,

$$d_s = \sqrt[3]{\left(\frac{36000 \cdot P_m}{\tau_{\text{torsion}} \cdot n}\right)} \dots\dots\dots (4.5)$$

where d_s is the shaft diameter in cm, τ_s is the permissible torsion stress in kPa/cm^2 , P_m is the transmitted power in Metric H.P and n is speed in rpm,

The transmitted power for multiple stages, P_m , is defined as the multistage pump power which is calculated as:

$$P_m = \frac{\rho \cdot Q \cdot g \cdot H}{\eta} \cdot m \dots\dots\dots (4.6)$$

where, m is the number of stages.

The inlet diameter is calculated from the empirical formula [1].

$$d_o = K_0 \cdot \left(\frac{Q}{n}\right)^{1/3} \dots\dots\dots (4.7)$$

K_0 term in Equation (vii) varies between 4 and 4.5. “ K_0 is selected close to the 4 where better pump efficiency is desired and K_0 is selected close to the 4.5 where better cavitation performance is desired during the design”, [20]. The K_0 term is selected close to 4 for the pump to be designed in terms of concerning better efficiency in the design. Impeller inlet eye diameter is also shown in Figure 4.3.

The hub and shroud profiles are constructed by fitting the smooth arcs between the impeller inlet breadth, b_1 , and impeller exit breadth, b_2 . The blade meridional profile is extended through the impeller inlet eye in order to lower the blade number of the impeller. The blade number of the impeller is selected to be 6 from the recent design experience, know how coming from the company and the literature [1] and [21]. If the blade number is higher than these values, it is hard to produce and assemble the cores in order to obtain the impeller. The design of impeller layout then completed with a trial and error procedure which is shown as a flow chart in table 4.1. The main aim of this process is to construct the best impeller layout in order to satisfy the flow characteristics of the pump to be designed. The unknown parameters are eliminated by the iterative procedure coupled with the preliminary design which is explained in the following sections. In deed the impeller layout design is highly dependent on the previous studies and experience. Heuristic and intuitive approaches become mainly the most important bases of the impeller layout design. However these approaches have to be supported by theoretical approaches in fluid dynamics, numerical experimentation and experimental results. Once the desired pump characteristics are achieved by numerical experimentation and validated by the experimental results, the

designed impeller layout is added to the impeller layout library. The aim of creating the library is shortening the design periods of the pumps by means of impeller layout profile design. The designer may design the impeller layout which lies between close specific speed layouts easily if a library exists. Nevertheless the blade passage of the impeller layout may be enlarged and used if higher flow rates are expected from an existing pump layout. This process is named as simply building up a pump family which uses the same bowl for different impellers with different flow rates or heads. The extension procedure is mainly done by adjusting the exit breadth and inlet breadth of the impeller layout.

4.2.2 Types of Blades:

Backward-curved blades, $\beta_2 > 90^\circ$

When $\beta_2 > 90^\circ$, $\cot\beta_2$ is negative and hence with increase in mass flow rate, the head decreases. The head capacity characteristic has a negative slope.

Radial Blades $\beta_2 = 90^\circ$

For radial vanes $\beta_2 = 90^\circ$ and $\cot\beta_2 = 0$, so head does not vary with flow rate.

Forward-Curved Blades, $\beta_2 < 90^\circ$

For forward curved blades $\beta_2 < 90^\circ$ and $\cot\beta_2$ is positive. Hence, with increase in mass flow rate head also increases, the head capacity characteristic has a positive slope.

As shown from Figure 4.8 for forward blades impeller, the fluid leaves the impeller with relatively high speed which means that the major part of the energy gained is kinetic energy, this type of impeller requires a very good diffuser to convert this kinetic energy to pressure energy. In practice, it is difficult to construct this kind of diffuser; also, it is usually more efficient to convert pressure energy to kinetic energy rather than converting kinetic energy to pressure energy.

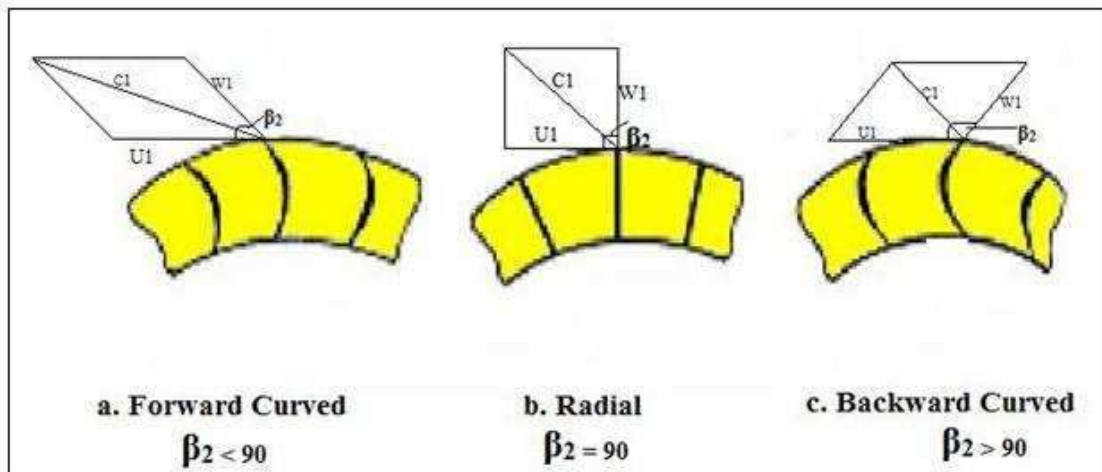


Fig.4.8 Types of Impeller blade base on curvature of blade

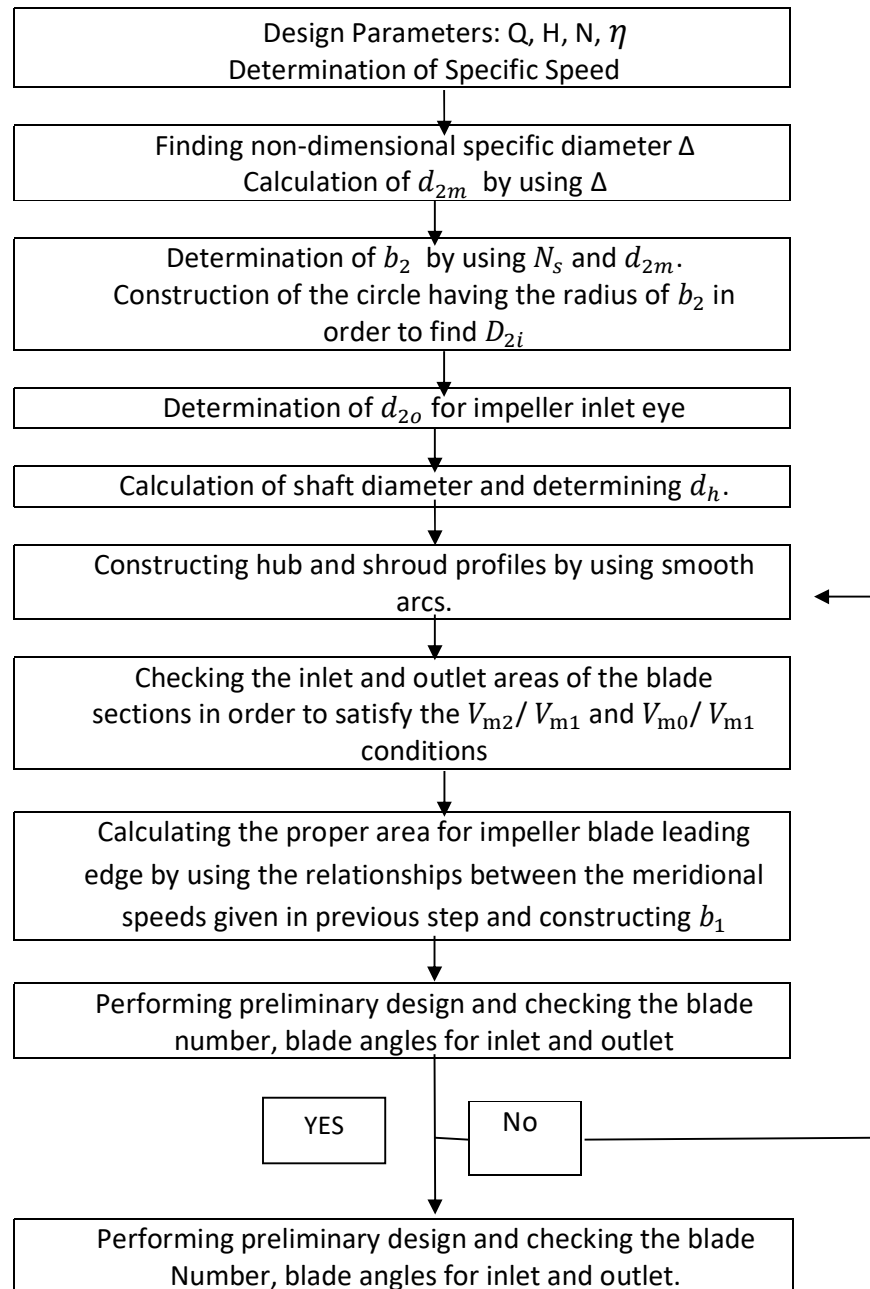


Figure 4.9 – Flow Chart showing impeller layout design

The designed impeller layout which is used to design the impeller is shown in Figure 4.10. The schematic representation of designed and selected impeller layout profile is also shown in Figure 4.11

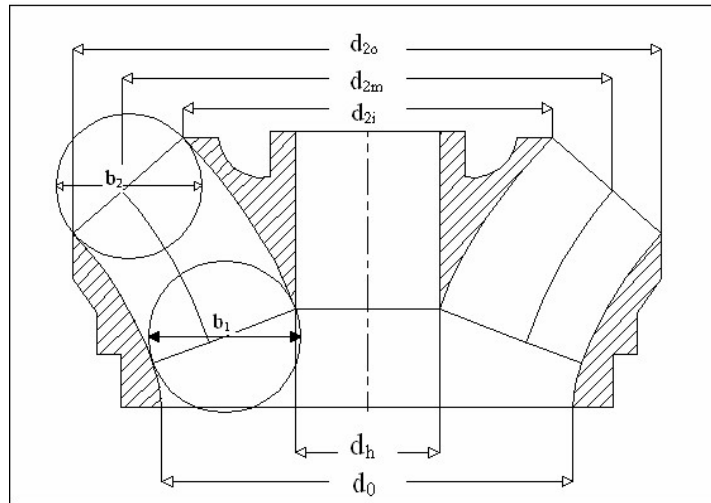


Figure 4.10: Designed impeller layout profile

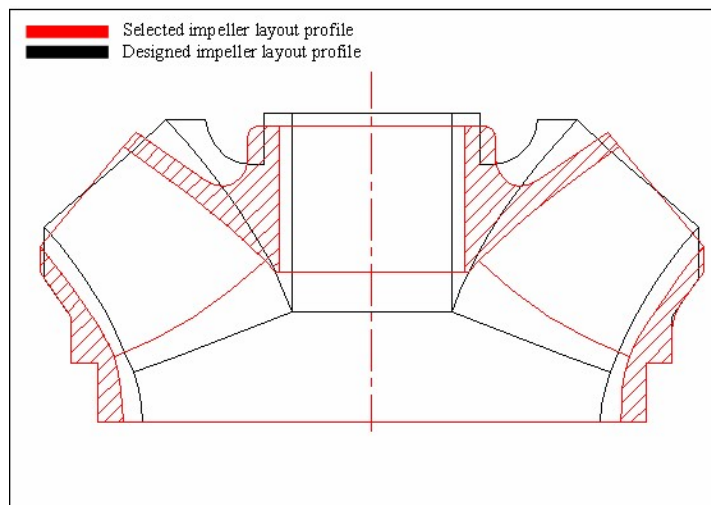


Figure 4.11: Schematic representation of designed and WPIL LIMITED selected impeller layout profiles

4.2.3 Impeller Inlet & Outlet Velocity Diagram:

The flow rate, Q_i which is mainly defined as the volumetric flow rate passing through the impeller blade passage is calculated using the volumetric efficiency, η_v and the design point flow rate Q . The formula is given as:

$$Q_i = \frac{Q}{\eta_v} \dots\dots\dots (4.8)$$

The net area in the impeller inlet eye, A_0 , is calculated with considering the shaft in the impeller inlet eye as:

$$A_0 = \frac{\pi \cdot (d_0^2 - d_s^2)}{4} \dots\dots\dots (4.9)$$

The net impeller inlet eye diameter, d_0 , is taken from the designed impeller layout. The axial velocity in the impeller inlet eye which is also named as meridional velocity is given by:

$$V_m = \frac{Q_i}{A_0} \dots\dots\dots (4.10)$$

Assumptions for Velocity Diagram of Pump Impeller:

- Liquid enters the impeller eye in radial direction.
- No Energy losses in impeller due to friction & eddy formation.
- Liquid enters without shock.
- Uniform velocity distribution in the passage between two adjacent vanes.

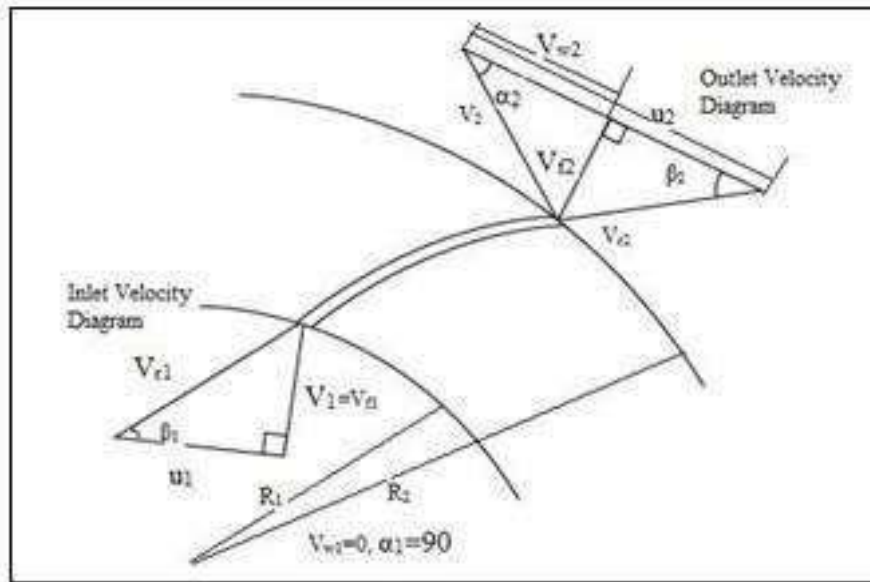


Figure: - 4.12 Impeller Velocity Diagram

Now,
$$U_1 = \frac{\pi D_{2i} N}{60} \dots\dots\dots (4.11)$$

$$U_2 = \frac{\pi D_{2o} N}{60} \dots\dots\dots (4.12)$$

Mass Flow Rate,

$$Q = \frac{\pi D_{2i}^2 V_1}{4} \dots\dots\dots (4.13)$$

Hence,
$$V_1 = \frac{4Q}{\pi D_{2i}^2}$$

From Inlet Velocity Triangle, Inlet Blade Angle,

$$\tan \beta_1 = \frac{U_1}{V_1}$$

and
$$V_{r1} = \sqrt{(V_1^2 + U_1^2)} = V_{r2} \dots\dots\dots (4.14)$$

From Outlet Velocity Triangle,

$$Q = \pi D_{2o} B_2 V_{f2}$$

$$V_{f2} = \frac{Q}{\pi D_{2o} B_2}$$

$$V_{r2}^2 = V_{f2}^2 + (U_2 - V_{w2})^2 \dots\dots\dots (4.15)$$

And
$$V_2 = \sqrt{(V_{w2}^2 + V_{f2}^2)} \dots\dots\dots (4.16)$$

Outlet Blade Angle,
$$\tan \beta_2 = \frac{V_{f2}}{(U_2 - V_{w2})}$$

Head Generated by Impeller,
$$H = \frac{V_{w2} U_2}{g} - \frac{V_2^2}{2g} \dots\dots\dots (4.17)$$

Overall Efficiency of Impeller,
$$\eta = \frac{\rho \cdot Q \cdot g \cdot H}{power} \dots\dots\dots (4.18)$$

The hub diameter is taken from the impeller layout and the blade inlet area without blade thicknesses, A_1 is calculated by revolving the blade inlet breadth, b_1 , around the rotating axis. The blade thickness is the important parameter in the design. The thickness of the blade is highly dependent on the production techniques used in the company. If the thickness is taken too small and not adjusted with the other wall thickness in the impeller, casting of the impeller may not be achieved in the desired manner

Table 4.1: GEOMETRIC PARAMETER OF DESIGN IMPELLER

Parameter	Size
Impeller inlet diameter (di)	266.7mm
Impeller outlet diameter (do)	384 mm
Blades number	7
Inlet angle (β_1)	22.7°
Outlet angle (β_2)	35.5°
Blade thickness (t)	7.9 mm
hub / Shaft diameter	63 mm
thickness leading edge	5.76 mm
thickness trailing edge	5.76 mm
Blade inlet height (Li)	78.3

4.3 Design of the Bowl:

The main function of the bowl used in vertical turbine pump that is designed in this study is, changing the direction of flow of the liquid leaving the impeller and directing the flow along the axis of rotation. However, while changing the direction of flow; the design should minimize the losses inside the bowl. In order to satisfy this need, the bowl should be designed not too long that may increase the friction losses and not too short that may cause the flow not to be directed properly to the second stage.

The length of the bowl is also important from the manufacturing point of view because of the raw material used in the casting. The bowl layout used in the design is shown in Figure 4.13

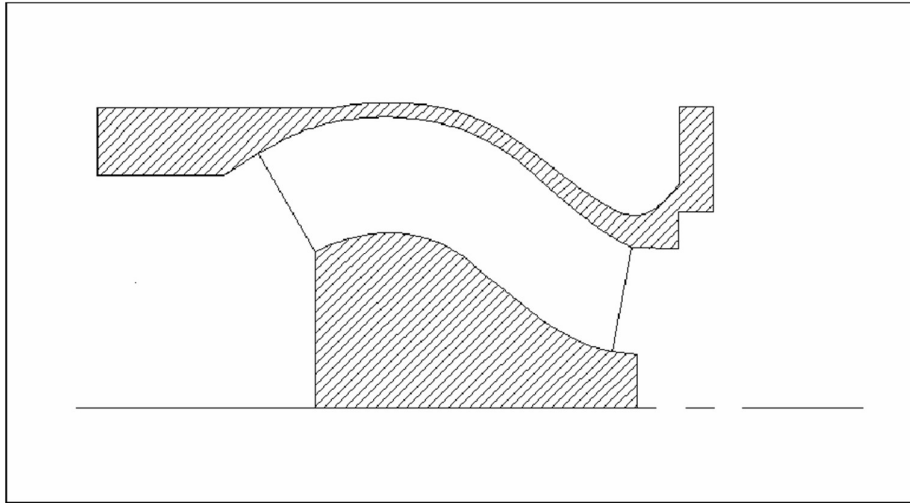


Figure 4.13 – The bowl layout used in the bowl design

Firstly, the bowl inlet and outlet breadth dimensions are checked. “The breadth of the diffuser at the inlet is about 10 to 12 % larger than the breadth of the impeller.”, [1]. The purpose of this adjustment is placing different impellers of a pump family into the same bowl. However, in this study the length of the inlet breadth of the bowl vane, b_3 , is selected close to the exit breadth, b_2 of the impeller because of not concerning a pump family. The exit breadth of the bowl vane, b_4 is adjusted to the inlet breadth of the impeller, b_1 .

While designing the outlet part of the bowl especially the exit flange of the bowl, the clearance is also adjusted. The clearance is important from the assembling point of view of the pump. However, the gap between the exit of the bowl vane and leading edge of the impeller blade should be taken as small as possible in order to get close to no inlet whirl assumption in the impeller blade design.

The mid-streamline is drawn for bowl design, A_3A_4 is shown in Figure 4.14.

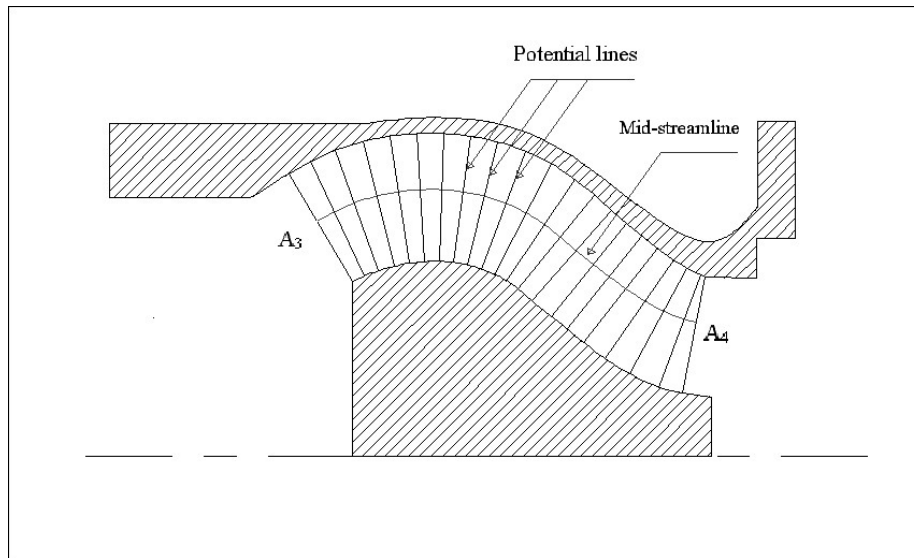


Figure 4.14 – Mid-streamline A_3A_4 and potential lines for the design of the bowl vane

The mid-streamline is divided into segments of equal length ΔS and potential lines are drawn. The potential lines are the arc segments where the axial component of the absolute velocity, V_y , is along the normal of the surfaces obtained by revolving these potential lines around the axis of the bowl. The potential lines are also shown in Figure 4.14.

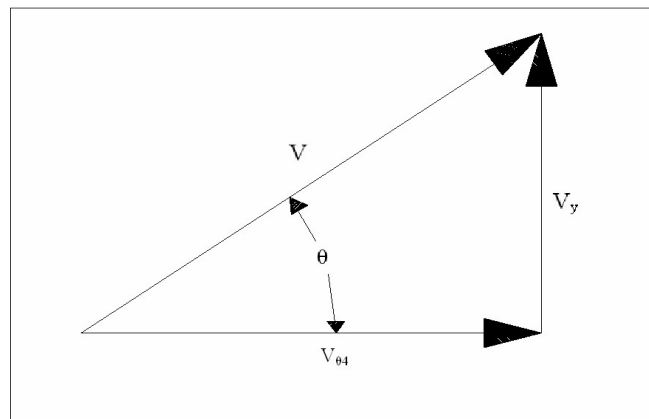


Figure 4.15 – Velocity vectors for the vane inlet of the bowl

Point by point method needs to follow for shaping the vane of the bowl. Since the flowing liquid enters the impeller in the second stage or discharge case with no inlet whirl condition, the flow is in the direction of axis of rotation which means the tangential component of the absolute velocity, should be zero at the exit of the bowl vane. The vane of the bowl is shaped by considering the distribution of absolute velocity throughout the equal distanced segments from inlet to outlet. On the other hand, the distribution of absolute velocity between the inlet and exit of the bowl vane is adjusted by considering the swept angle. Swept angle is the angle between the inlet and outlet of each streamline between the inlet and outlet from the top view. The main reason of considering the swept angle is due to manufacturing.

4.4 Design of the Suction Bell:

The suction bell for a multi stage pump has two main functions, [3]. The first one is the mechanical purpose which is supporting the bearing housing for the shaft at the lower end of the pump. The strainer or suction pipe is attached to the suction bell in different applications. In this study, the standard strainer is attached to the suction intake in order to prevent big size particles to get into the pump. The second function of the suction intake is building up a hydraulic passage before the impeller inlet eye in order to increase the fluid velocity gradually with the proper area change. In this study, the suction bell is designed for the intake element of the pump. “A conservatively designed high specific speed pump has a suction bell diameter large enough so that the inlet velocity is from 1.4 to 1.5 m/s”, [3]. The velocity at the inlet of the suction bell is taken as 1.4 m/s and the area distribution is adjusted from suction bell to the impeller inlet eye. The designed suction bell is shown below in Figure 4.16. In order to visualize the area distribution, the height of the suction bell from inlet to the outlet is divided into 10 equal segments and area distribution is shown in Figure 4.17.

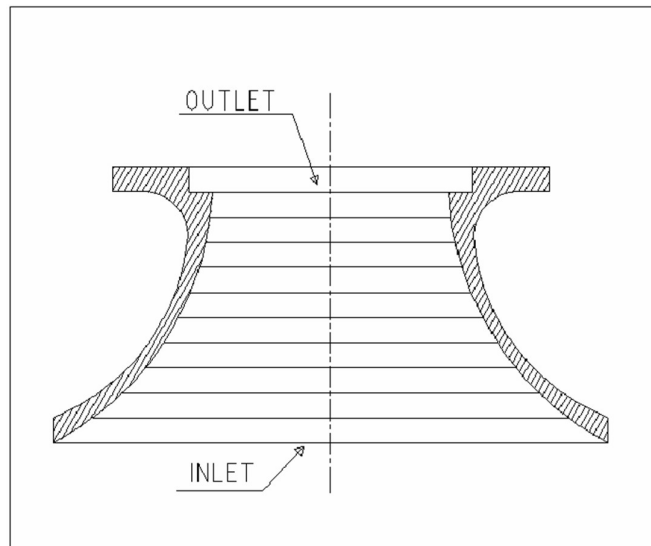


Figure 4.16 – Suction bell

The area distribution of suction bell is adjusted in order not to disturb the flow from inlet to the exit of suction bell as seen in Figure 4.16

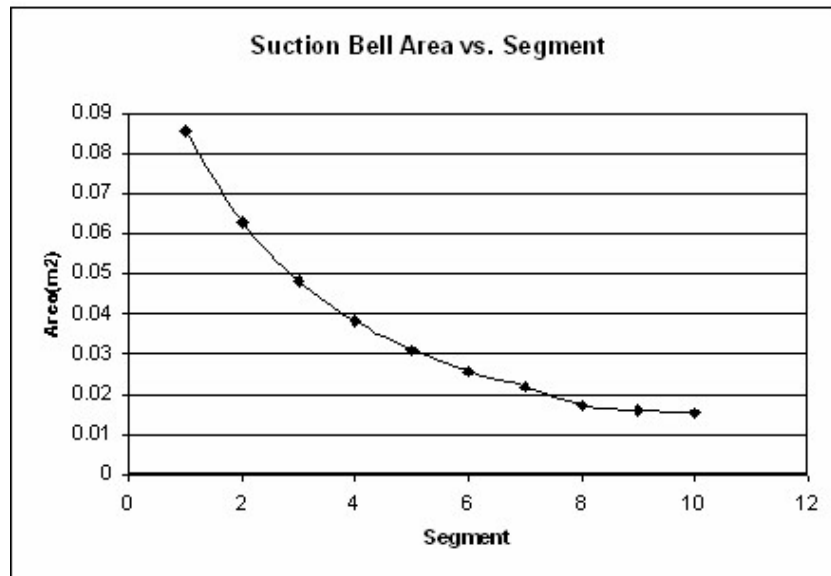


Figure 4.17 – Area distribution of the suction bell [Ref: WPIL LIMITED]

4.5 Minimum Submergence Required:

Minimum submergence is crucial in vertical turbine pumps to prevent cavitation and ensure efficient operation. Minimum submergence refers to the minimum depth of liquid required above the pump's suction inlet to prevent cavitation. Proper submergence ensures the pump operates efficiently and avoids issues such as air entrainment and vortex formation, which can lead to cavitation and damage to the pump.

Factors Influencing Minimum Submergence:

- **NPSH Required (NPSH_r):** The minimum pressure required at the pump's suction to avoid cavitation.
- **NPSH Available (NPSH_a):** The actual pressure available at the pump's suction, which must be higher than NPSH_r.
- **Water Depth:** The depth of the water source relative to the pump's suction inlet.
- **Flow Velocity:** Higher velocities can reduce pressure, increasing the risk of cavitation.
- **Impeller Design:** Impellers designed for higher flow rates or heads may require greater submergence.
- **Pump Speed:** Higher rotational speeds increase the risk of cavitation, necessitating greater submergence.

Calculation of Minimum Submergence as per HIS guidelines (Table 4.2)

a) Minimum submergence calculation at rated flow

Rated Flow, Q	1200	M ³ /Hr
Suction Bell diameter, D	584.2	Mm
Velocity at bell mouth, V	1.243057	m/sec
Froude no., $Fr = \frac{V}{(gD)^{0.5}}$(4.19)	0.519249	
Minimum submergence required to prevent vortex, $H = D * (1+2.3 Fr)$ (4.20)		
Minimum Submergence required, H	1282	Mm

b) Minimum submergence calculation at maximum flow

Run Out Flow, Q (125% of rated Flow)	1500.00	M ³ /Hr
Suction Bell diameter, D	584.2	Mm
Velocity at bell mouth, V	1.553821	M/sec
Froude no., $Fr = \frac{V}{(gD)^{0.5}}$	0.649061	
Minimum submergence required to prevent vortex, $H = D * (1+2.3 Fr)$		
Minimum Submergence required, H	1456	mm

4.6 Calculation of NPSH Available (NPSHA):

Net Positive Suction Head (NPSH) is a critical parameter in pump systems, including vertical turbine pumps. It ensures that the pump operates without cavitation, which can damage the pump and reduce its efficiency. There are two types of NPSH: NPSH Available (NPSHA) and NPSH Required (NPSHR). NPSHA is the actual pressure head available at the pump suction, while NPSHR is the minimum pressure head required by the pump to avoid cavitation.

The formula to calculate NPSHA is: (Ref.: Hydraulic Institute Standards for Vertical Pumps)

$$NPSHA = H_s + H_a - H_v - H_f \quad \text{..... (4.21)}$$

Where,

H_s = Static suction head (the vertical distance between the surface of the liquid source and the pump impeller centreline).

H_a = Atmospheric pressure head (at the liquid surface). This is usually given in meter of liquid

H_v = Vapor pressure head of the liquid being pumped. This is the pressure at which the liquid will start to vaporize at the pumping temperature, usually given in meters of liquid.

H_f = Friction losses in the suction line, usually given in meters of liquid.

Table No. 4.3: Calculation of NPSHA

Data:		
H_v vapour pressure of water at 40°C =	0.553	MWC
H_a Atmospheric pressure head (at sea level)	10.330	MWC
H_f Friction losses in the suction line	0.024	MWC
ze, distance from bell tip to impeller eye =	0.308	M
G, distance from pump house floor to bell tip =	0.675	M
Bottom of pump house Level =	-4.200	M
Minimum water level =	-2.240	M
Maximum water level =	-0.500	M
Normal water level =	~	M
h_l height of water column above pump house floor	1.960	M
$H_s = h_l - G - ze =$	0.977	M
Difference between Min. W.L. & Max. W.L.	1.740	M
Difference between Min. W.L. & Nor. W.L.	~	M
NPSHA (w.r.t Min WL)=	10.730	M
NPSHA (w.r.t Max. WL)=	12.470	M
As per performance curve,		
NPSHR AT DUTY POINT (50 Hz)	7.30	M
NPSHR AT MAX FLOW (50 Hz)	11.00	M
So, the NPSHA for the vertical turbine pump at duty point condition is 10.73 meters.		
Hence, NPSHA at Min. Water Level > NPSHR at Duty Point		

4.7 Column Diameter Selection and Column Loss Calculation:

Selecting the appropriate column diameter for a vertical turbine pump is crucial for optimizing pump performance, minimizing friction losses, and ensuring efficient operation. Column loss in vertical turbine pumps refers to the head loss due to friction and turbulence as water flows through the column pipe. Accurate calculation of column loss is essential for determining the total bowl head and ensuring efficient pump operation. For column pipe diameter selection working fluid velocity in column also needs to consider. Basically, fluid velocity tries to keep below 3.5 m/sec to mitigate pressure and flow turbulence issues.

Factors Affecting Column Loss:

- Pipe Diameter and Length: Smaller diameters result in higher velocities and greater friction losses. Longer columns increase the frictional resistance and thus the head loss.
- Higher flow rates increase the velocity of the fluid, leading to increased friction losses.
- Pipe Material and Roughness: Different materials have different roughness coefficients, affecting friction losses. Increased roughness leads to higher friction losses.
- Fluid Properties: Higher viscosity fluids experience greater friction losses. Density affects the Reynolds number, influencing the flow regime (laminar or turbulent).

Calculation Methods:

Darcy-Weisbach Equation:

The Darcy-Weisbach equation is commonly used for calculating head loss due to friction in pipes:

$$h_f = f \cdot \left(\frac{L}{D}\right) \cdot \frac{V^2}{2g} \dots\dots\dots (4.22)$$

h_f = Head loss (m)

f = Darcy-Weisbach friction factor (dimensionless)

L = Length of the pipe (m)

D = Diameter of the pipe (m)

V = Flow velocity (m/s)

g = Acceleration due to gravity (9.81 m/s²)

Hazen-Williams Equation:

The Hazen-Williams equation is another method used, particularly for water:

$$h_f = 10.67 L \cdot \left(\frac{Q}{C}\right)^{1.852} \cdot D^{-4.87} \dots\dots\dots (4.23)$$

h_f = Head loss (m)

L = Length of the pipe (m)

C = Hazen-Williams roughness coefficient (dimensionless)

D = Diameter of the pipe (m)

Q = Flow rate (m^3/s)

Table 4.4: Column Diameter Selection and Column Loss Calculation sheet

INPUT DATA :		
Col. Pipe dia(D_1) =	400	MM
Col Pipe length (PL) =	2.520	Mtr
Flow (Q) =	1200	M^3/HR
Shaft Dia (D_2)	43	mm
Suction Bell Dia.	584	mm
Suction bell Velocity	1.25	m/sec.
Column loss as per Hazen-Williams Equation:		
$h_f = V^{1.852} \times L / (0.74 \times C^{1.852} \times r^{1.167})$		
$r = ((D_1 - D_2) / (1000 \times 4))$	0.089284375	Mtr.
$A = 3.14 / 4 \times (D_1^2 - D_2^2)$	0.124157803	Sq. Mt.
$V = (Q / 3600) / A$	2.684755415	Mt./Sec.
Column length, $L = PL$	2.52	Mtr.
C (friction co-eff)	110	
h_f col. pipe =	0.059	
Add Margin	20%	
Total Column Loss	0.070707813	
A Column Loss Considered	0.071	Mtr.
Total Column Loss/mtr	0.028174603	
Column Loss Per 100 Mtr	2.817460317	
B. Discharge Head Loss ($0.3V^2/2g$)	0.108	Mtr
C. Suction bell entry loss ($.05V^2/2g$)	0.004	Mtr.
D. Total Dynamic Head	30.00	Mtr.
Total Bowl Head (A+B+C+D)	30.18	Mtr.

4.8 Production of the Pump: Construction Techniques

Production of pump provides an in-depth exploration of the construction techniques used in the development of vertical turbine pumps. Understanding these techniques is crucial for ensuring the structural integrity, performance, and longevity of the pump. The topic covers material selection, fabrication processes, assembly methods, and quality control measures.

4.8.1 Material Selection:

Material selection is vital for the durability and efficiency of vertical turbine pumps. Key criteria include:

- **Mechanical Strength:** Materials must withstand high pressures and mechanical stresses.
- **Corrosion Resistance:** Essential for pumps handling various fluids, especially in harsh environments.
- **Wear Resistance:** Important for components subject to friction and wear, such as impellers and bearings.
- **Cost-effectiveness:** Balancing performance with economic feasibility.

Common Materials Used:

- **Cast Iron:** Commonly used for pump casings due to its good mechanical properties and cost-effectiveness.
- **Stainless Steel:** Preferred for impellers and shafts due to its excellent corrosion resistance and strength.
- **Bronze and Brass:** Often used for wear rings and bearings for their good wear resistance and low friction properties.
- **Composite Materials:** Increasingly used for certain components to reduce weight and improve corrosion resistance.

4.8.2 Fabrication Processes

Casting:

Casting is a widely used process for creating pump casings and large components.

Sand Casting: Commonly used for its flexibility in producing complex shapes and its cost-effectiveness for large parts.

Investment Casting: Used for smaller, precision components where higher accuracy and finer details are required.

Machining:

Machining is essential for achieving the precise dimensions and surface finishes required for pump components.

Turning and Milling: Used for shaping shafts, impellers, and diffusers.

Grinding and Polishing: Employed to achieve the smooth surface finishes necessary for reducing friction and wear.

Welding and Brazing:

These processes are used to join different components and ensure leak-proof construction.

Welding: Typically used for assembling the pump casing and other large components.

Brazing: Often used for joining smaller parts and for repair work.

4.8.3 Assembly Methods

Pre-assembly Preparation

Before assembly, all components are cleaned and inspected to ensure they meet design specifications. This step includes:

- **Dimensional Inspection:** Verifying that all parts conform to the specified dimensions.
- **Surface Finish Inspection:** Ensuring that all surfaces meet the required finish standards.

Assembly Sequence

The assembly process involves a precise sequence to ensure proper alignment and functioning:

Shaft Assembly: The shaft is assembled first, ensuring it is straight and free from defects.

Impeller Installation: Impellers are mounted on the shaft, typically starting from the bottom stage upwards.

Casing Assembly: The assembled shaft and impellers are then placed inside the pump casing, and the casing is sealed.

Tolerances and Alignments

Maintaining tight tolerances and proper alignment is crucial for pump performance:

- **Radial and Axial Alignment:** Ensuring the shaft is correctly aligned to prevent vibrations and premature wear.
- **Clearances:** Maintaining proper clearances between moving parts to ensure smooth operation and prevent contact.

4.8.4 Quality Control Measures

Inspection Techniques

Various inspection techniques are employed to ensure the quality of the constructed pump:

- **Non-Destructive Testing (NDT):** Methods such as ultrasonic testing, radiography, and dye penetrant inspection are used to detect internal and surface defects.
- **Hydrostatic Testing:** The assembled pump is subjected to high-pressure water tests to check for leaks and verify structural integrity.

4.8.5 Challenges and Solutions in Construction

Common Challenges

- **Material Defects:** Issues such as casting defects and material inconsistencies can impact pump performance.
- **Alignment Issues:** Misalignment of components can lead to increased wear and reduced efficiency.
- **Leakage:** Ensuring leak-proof construction, especially at joints and seals, can be challenging.

Mitigation Strategies

- **Enhanced Quality Control:** Implementing stringent quality control measures at every stage of construction.
- **Precision Machining:** Using advanced machining techniques to achieve high precision and accuracy.
- **Improved Sealing Techniques:** Employing better sealing materials and methods to prevent leakage.
- **Exploring 3D printing** for creating complex components with high precision and reduced material waste.
- **Incorporating automation and robotics** in the assembly process to enhance precision and reduce human error.

CHAPTER 5: METHODOLOGY: TEST SETUP AND PROCEDURE

This chapter outlines the methodology for designing, constructing, and experimentally studying the vertical turbine pump (VTP). It details the test setup and procedures used to evaluate the pump's performance, including the design specifications, construction steps, and experimental protocols.

5.1 Summary:

Summary of methodology of this thesis work is describe below in the form of flow chart:

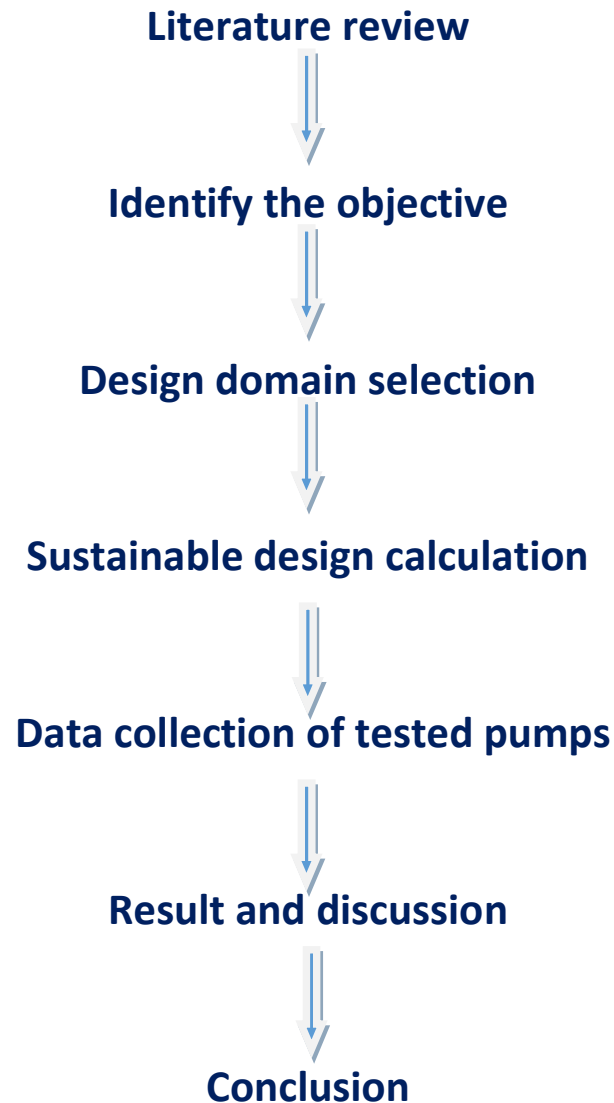


Figure 5.1: Flowchart of methodology

5.2 Literature Review

The literature review methodology involves systematically collecting, analyzing, and synthesizing existing research relevant to the design, construction, and experimental study of vertical turbine pumps (VTPs). This process provides a comprehensive understanding of the current state of knowledge, identifies gaps, and establishes a foundation for the research objectives. The following sections outline the key steps in the literature review methodology.

Design Principles: Research emphasizes the importance of optimizing impeller geometry to enhance efficiency and reduce hydraulic losses. CFD simulations are widely used to model fluid flow and predict performance. Studies have shown that blade number, blade angle, and diameter significantly impact pump performance.

Construction Techniques: Common materials for VTP components include stainless steel and composites for their durability and corrosion resistance. Advances in manufacturing techniques, such as precision casting and CNC machining, have improved component quality and performance.

Experimental Testing: Performance testing typically involves measuring head, flow rate, efficiency, and cavitation characteristics. Experimental setups often include calibrated flow meters, pressure sensors, and data acquisition systems. NPSH testing is crucial to understand cavitation limits and ensure reliable operation.

The literature review methodology provides a structured approach to understanding the current knowledge landscape on vertical turbine pumps. By systematically searching, analyzing, and synthesizing existing research, this methodology identifies critical insights and research gaps, guiding the design, construction, and experimental evaluation phases of the study.

5.3 Define Research Questions and Objectives

The primary objective of this thesis is to design, construct, and experimentally study a vertical turbine pump with the aim of advancing the understanding of its hydraulic performance, structural integrity, and operational characteristics. Specifically, the research aims to:

- What are the critical design parameters and considerations for VTPs?
- What construction materials and methods are commonly used for VTPs?
- How are VTPs experimentally tested and evaluated for performance?
- How to Construct a standard VT pump based on the established design criteria, utilizing appropriate manufacturing techniques and quality control measures.
- Testing of VT pump to evaluate the hydraulic performance of the pump under various operating conditions, including flow rate, head, and speed variations.

5.4 Pump Technical Data Sheet & Performance Curves

Table No. 5.1 : Pump Technical Data Sheet			
Sl.No.	Item	Unit	Sl. No 1
1	Pump performance		
1.1	Rated capacity	m ³ / hr	1200
1.2	Total Bowl head	mtr	30.18
1.3	Total Dynamic Head	mtr	30.00
1.4	Speed	RPM	1475
1.5	Bowl efficiency	%	86.00
1.6	Bowl input at duty point	KW	114.69
1.7	Pump input at duty point	KW	115.26
1.8	NPSH required at duty point	mtr	7.3
1.9	Liquid Handled		Water
1.1	Sp.gravity (Assumed)		1.00
1.11	Specific Speed :	US Unit	3416.3
1.12	Suction Specific Speed :	US Unit	9905.1
2	PUMP SPECIFICATION		
2.1	Discharge outlet size	mm	400
2.2	Column size X Shaft Size :	mm X mm	400 X 43
2.3	TYPE OF PUMP		Vertical Turbine Pump
2.4	Bowl assembly model		D20TC (WVT-32-20)
2.5	No. of stages		1
2.6	Maximum bowl diameter	mm	584
2.7	Pump suspension length	mtr	4.00
2.8	Type of lubrication		Self Water
2.9	Impeller Dia (Max/Rated/Min) :	mm	400 / 384 / 375
2.10	Doward Axial Thrust at Duty Point :	Kg	2115.9
2.11	Min. submergence required from bottom of bell to prevent vortex	mtr	1.282
2.12	Bottom clearance	mm	200
3	PUMP MATERIAL OF CONSTRUCTION :		
3.1	Bowl casing :		CAST IRON (ASTM A48) CLASS 30
3.2	Impeller :		SS410 (ASTM A743) CA15
3.3	Shaft, Sleeve :		ASTM A276 SS410
3.4	Pump Column -Flanged :		MILD STEEL
3.5	Bolts & Nut :		SS304
3.6	Discharge Head :		MILD STEEL
3.7	Thrust bearing :		Antifriction Type
3.8	Coupling Type :		Flexible Type.
3.9	Basket Strainer :		MILD STEEL
3.10	Flexible Coupling :		Cast IRON
3.11	Motor Stand :		MILD STEEL
3.12	Bowl & Line Shaft Bearings :		Bronze / Rubber with Bronze Shell
3.12	Gland Packing :		Impregnated TEFLON
3.13	Foundation Bolts With Nut :		Carbon Steel
3.14	Sole Plate :		MILD STEEL
4	ELECTRIC MOTOR (IE3)		
4.01	Rating	Kw	132
4.02	Synchronous Speed	rpm	1500
4.03	Rated Supply		415 Volt , 3 Ph , 50 Hz , AC
4.04	Make		ABB / WEG / Marathon or Equivalent
4.05	Type :		VSS
4.06	Enclosure		TEFC
4.07	Area Classification :		Safe
4.08	Degree of Protection		IP55
4.09	Class of Insulation		F
4.10	Design Ambient Temperature :	°C	50
5	Scope of supply :		
	Pump with Motor stool, Thrust bearing assembly , Discharge Head, Column Assembly, Bowl Assembly (with Basket Strainer) and Drive Electric Motor.		
	Note :		
	1) Pump Performance Tolerance as per ISO 9906 Gr 2B.		
	2) Electrical Motor will be VFD compatible.		
	3) DISCH. FLNG. DRILLING STD. : ANSI B16.5 CL 150 FF		
	4) Pump Painting will be Synthetic Enamel Color.		

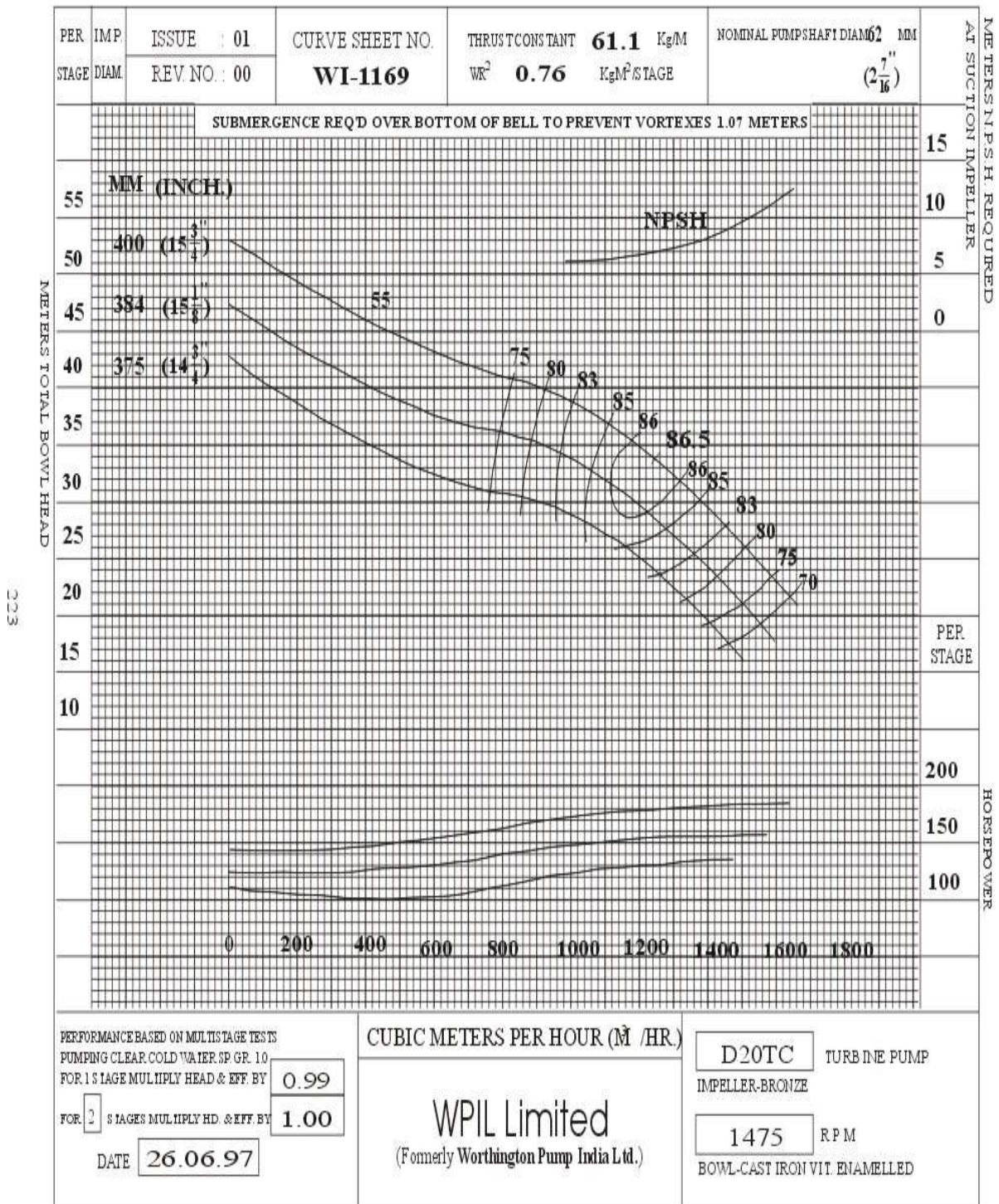


Figure 5.2 WPIL Selected Model Performance Curve

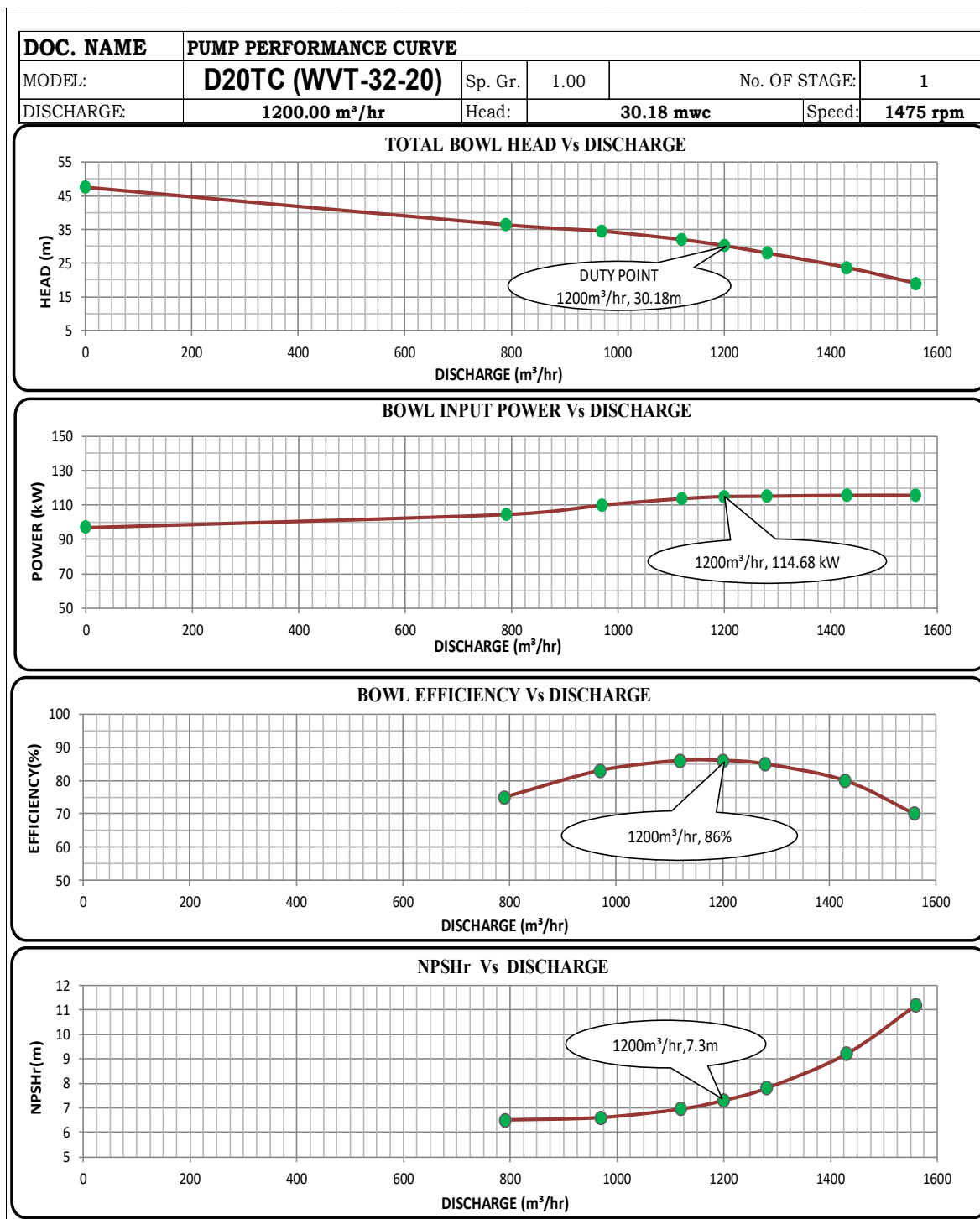


Figure 5.3 Pump Performance Curve at rated duty point

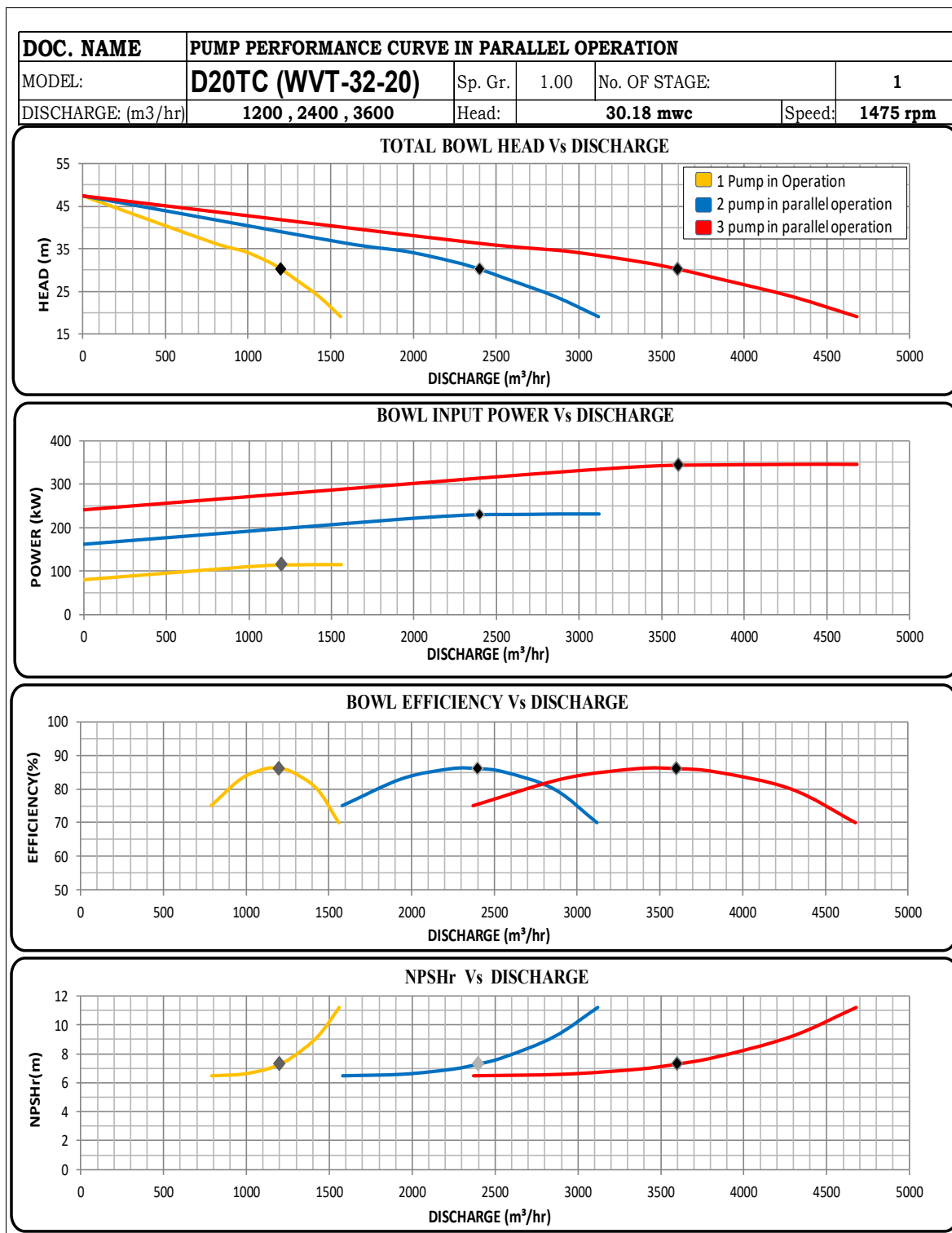


Figure 5.4 Pump Performance Curve in Parallel Operations

PUMP MODEL- 1-Stg, D20TC (Variable Speed Curve at 1475 R.P.M, 1560 R.P.M, 1420 R.P.M & 1250 R.P.M)

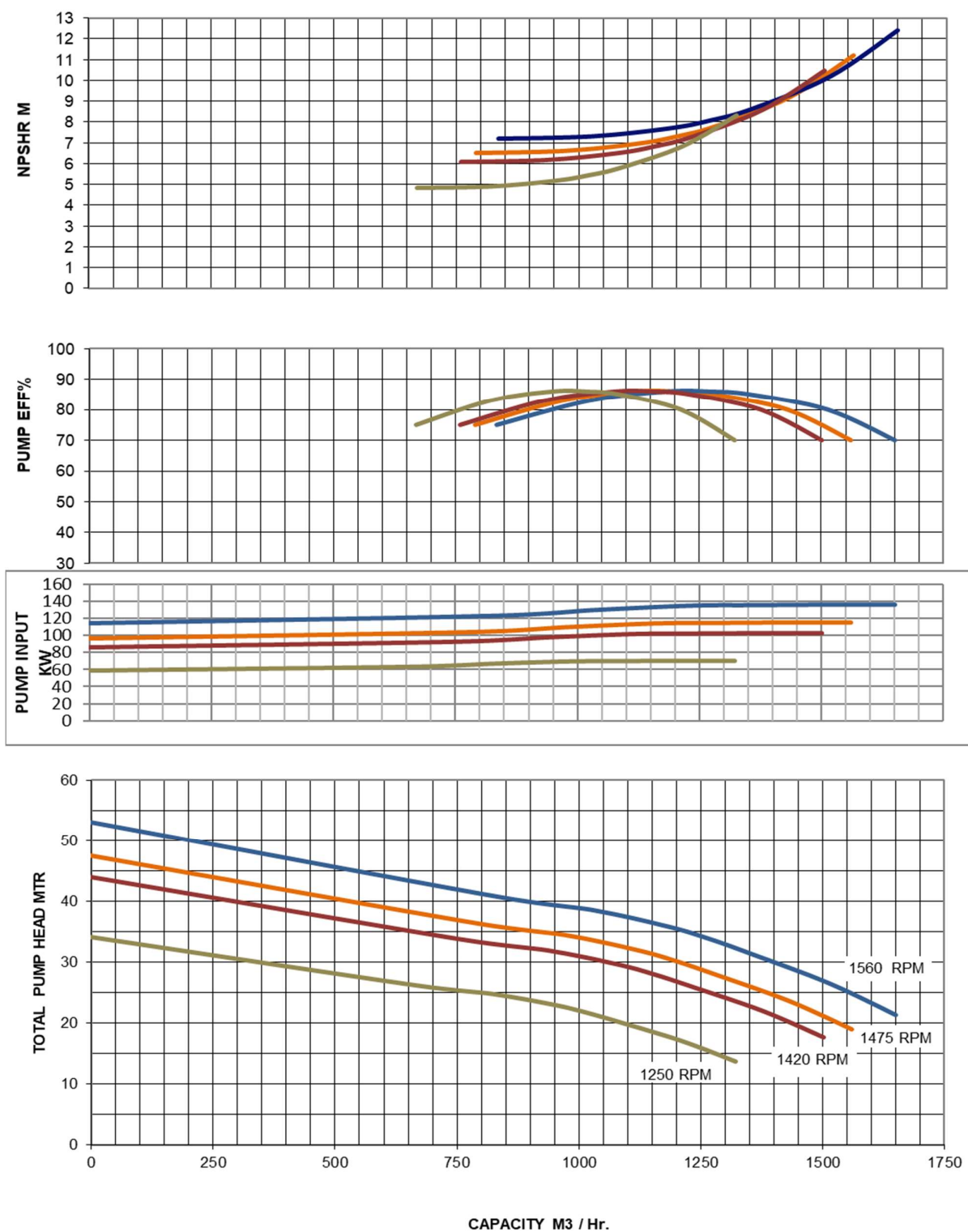


Figure 5.5 Pump Performance Curve in Variable Speed Operations

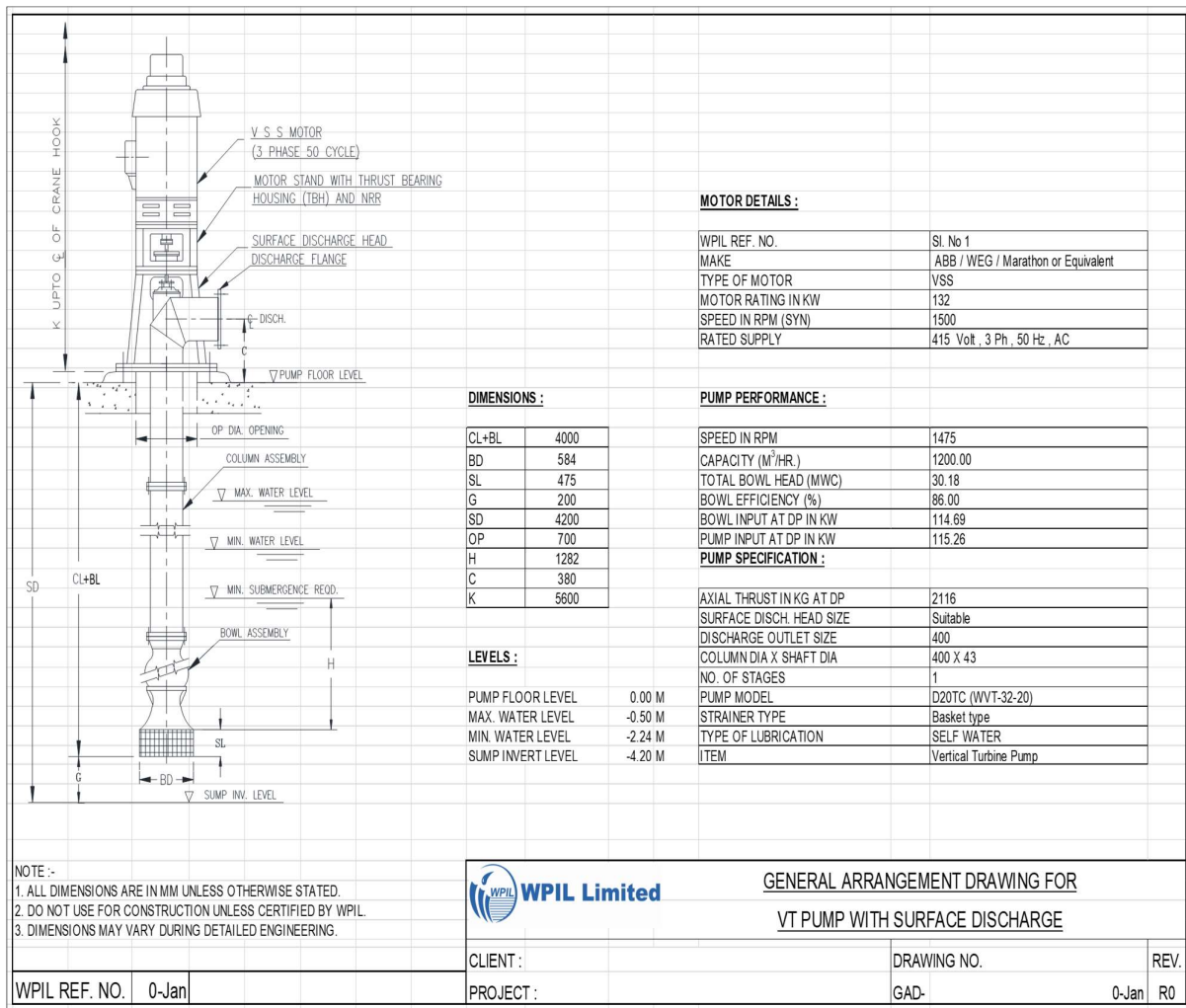


Figure 5.6: PUMP GENERAL ARRANGEMENT DRAWING

5.5 Pump Test Set Up and Procedure

5.5.1 Test Stand:

The pump that is designed in this study is tested in the test stand of WPIL Limited- Kolkata factory. The test stand is accredited by Indian Standards Institute. The national standards are applicable in the test stand and it has measurement traceability.

There are six wells in the test stand. Those wells are opened to a pool which has a depth of 6 m and maximum diameter of 3 m. The butterfly types of valves are used to discharge and control the flow for pumps to be tested in each well. The flexibility of pump testing by means of discharge flange connection to the butterfly valves is performed by short intermediate pipes between discharge flange and the butterfly valve with different reduction ratios. The

reduction in the intermediate pipes is adjusted by taking the flow rate limitations for each pipe into consideration. The crane is used to carry the pumps inside the test stand and position them to the suitable well. The capacity of the crane is 25 tons.

The butterfly valves are connected to the control panel which is located in the control room of the test stand. Outlet pressure of the pumps is adjusted by these valves by the help of joysticks installed on the control panel. The control panel is also connected to the main electric panel in order to start and stop the electric motor that drives the pump. The technical person is responsible from the whole testing procedure and uses the control panel to test the pumps. The test stand has a maximum capacity of 4000 kW as electrical input power to the motor. There is also a frequency control drive in the test stand.

The pumps up to 160 kW of motor input power is tested at any alternative current frequency lower than 50 Hz. However, in cases where the limitation on the torque required to run the pump is exceeded cannot be tested by using frequency control drive. This problem may occur when high capacity pumps with large impellers are tried to be tested in the test stand.

There are 11 manometers and 2 pressure transducers which are capable of measuring pressures up to 50 bars. The flow rate of the pump to be tested is measured by electromagnetic flow meters that are installed on different discharge pipes. The diameters of the discharge pipes are DN100, DN200, DN300, DN400, DN600, DN800, DN1200. This system is capable of measuring flow rates upto 38000 m³/hr. The periodic calibration of the devices used in the test stand for tests are calibrated by the certified calibration agencies.

The pumps are tested by using “cold clean water” as working fluid. The cold clean water is described in standard as water having a maximum temperature of 45°C, 1.75×10^{-6} m²/s in kinematic viscosity, 1000 kg/m³ in density, 2.5 kg/m³ in non-absorbing solid particles. The limitations of the described clean cold water are satisfied by the test stand working fluid.

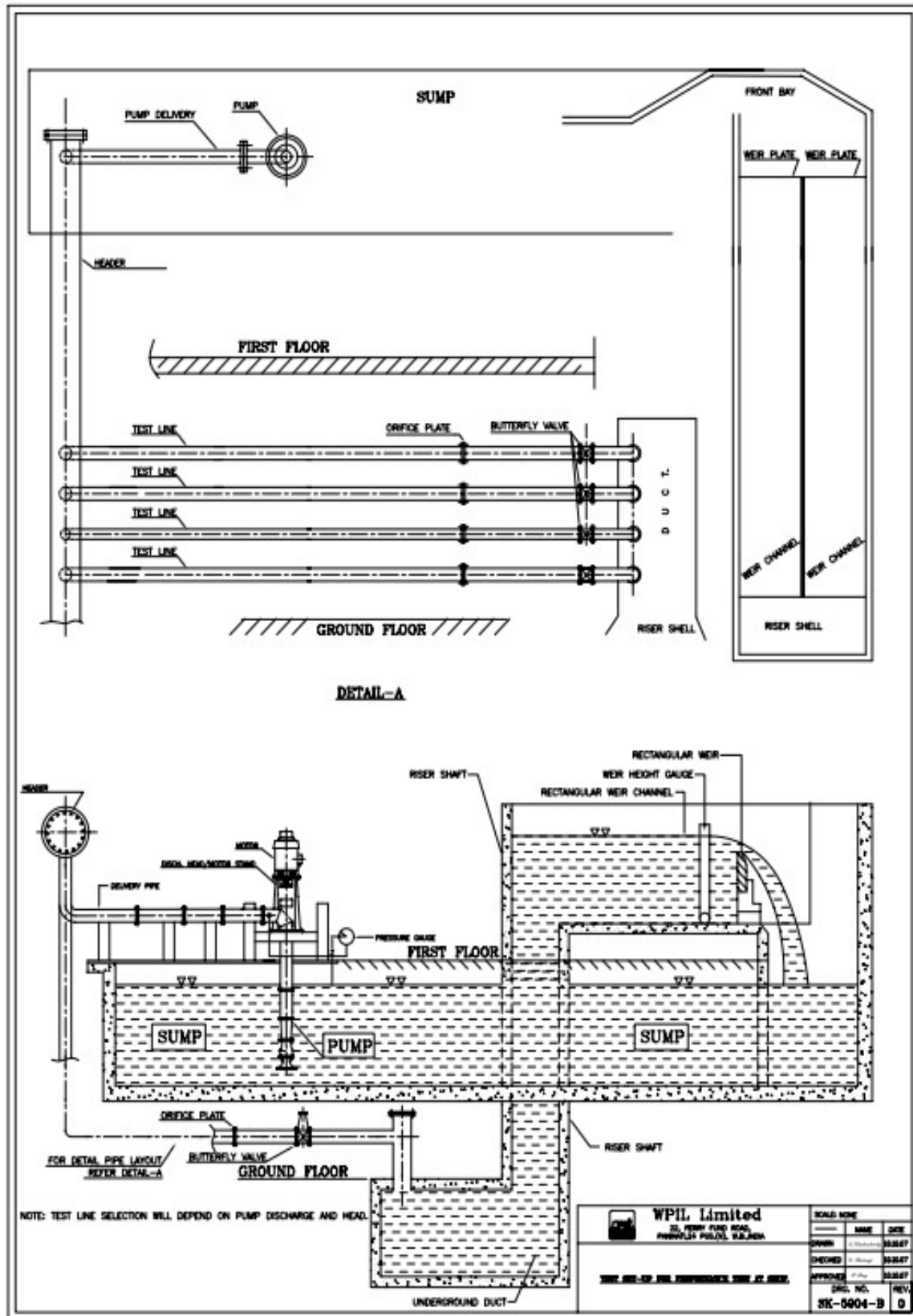


Figure 5.7 Test Set Up for performance Test at Shop

5.5.2 Test Setup & Performance Data Collection:

The pump that is designed in this study is tested as line shaft pump. In this test setup which is shown in Figure 5.7, Vertical Solid Shaft (VSS) motor is used to drive the pump. Before the pump is carried into the test stand, the parts are assembled in the assembly line of the company. The clearances are controlled in order to check whether the pump is assembled correctly or not. When the pump is carried into the test stand, first of all the strainer is mounted to the bowl assembly which consists of impeller, bowl, suction bell and discharge case. The standard discharge case which is suitable for the upper flange of the bowl is used during tests of the pump. The power is transmitted to the pump from the motor by the shaft connections which go through the water lubricated column assembly. The shaft connection consists of several separate shafts in order to make the setup procedure flexible. The intermediate shaft is placed between the pump shaft and head shaft which is connected to the pump shaft by a coupling on one end and to the head shaft on the other end. The head shaft passes through the stuffing box which is placed in the discharge head and goes into the VSS motor. The stuffing box is used to prevent water to pass through to the motor side by the rotation of the shaft. Moreover, the discharge head is connected to the other end of the column assembly. All connections are performed with flange connections which are fastened with bolts and nuts to the each other. This arrangement which consists of strainer, pump, column assembly and discharge head is placed to the suitable testing line and well position. Positioning the impeller and bowl is performed by fastening the nut which is placed on the upper part of the motor. When the pump assembly is placed in the pump setup, the impeller is in the lower position. The movement of the shaft on the upper direction designates the clearance between the impeller and bowl. However, before the clearance is adjusted the impeller which is connected to the pump shaft is lifted all the way up by the help of the nut in order to control the total clearance between the impeller and bowl.

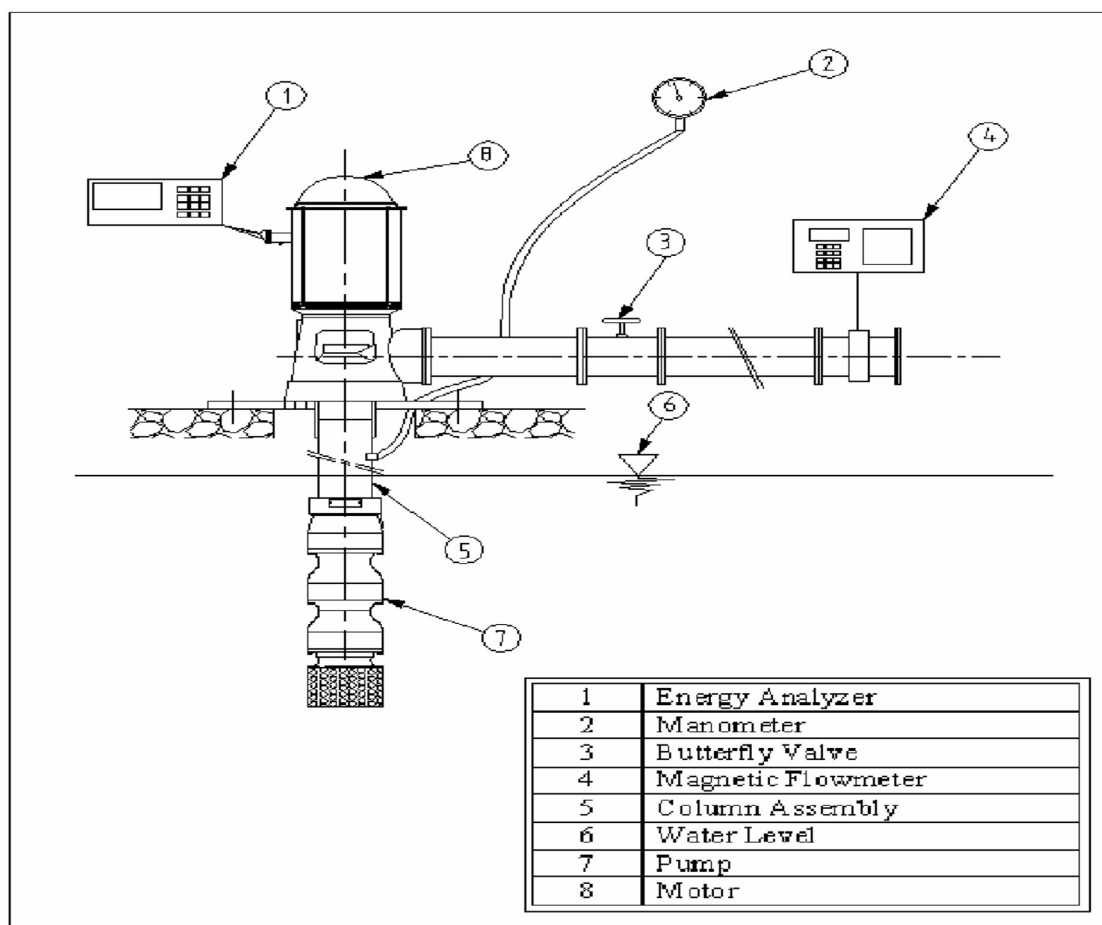


Figure 5.8 – In-line shaft Test Setup

5.5.3 Test Procedure:

The pump will be tested as per IS 1710 / IS 5120 in the Test Laboratory of M/s WPIL Ltd., Panihati, West Bengal works. The pump will be driven by one of the job motors or suitable shop motor.

Total five readings shall be taken during the test including shut - off point. The five points are

- 1) duty point of guaranteed values i.e. 100%,
- 2) 110% of rated discharge,
- 3) maximum flow available at test lab,
- 4) 80% of rated discharge and
- 5) shut off head

The procedure which is given below is followed during the pump tests performed in this study.

1. The pump is assembled and prepared for the test.
2. Test setup is prepared and column assembly is filled with water.

3. The motor is connected to the suitable electrical panel by looking at the motor electrical properties and the protection relay in the panel is set to the required value by technician.
4. The protection relay is adjusted to the required value.
5. The rotation direction of the pump is checked and corrected by simply starting the pump for a short period of time.
6. The connections to the butterfly valve from the intermediate pipe is controlled and fastened in order to prevent the water leakage from the connections.
7. The manometer and flowmeter are selected by looking at the maximum expected head and flowrate of the pump to be tested respectively. The chosen Manometer is connected to the discharge pipe in the column assembly. Required valve configuration is set to direct the flow from collector pipe to the chosen flowmeter.
8. Pump is started. If the frequency converter is used in order to operate the pump at different speeds than the motor speed, the frequency is adjusted to the desired working speed.
9. After the pump is started the exit pressure of the pump is adjusted by using butterfly valve. When the pressure is in the desired level, no readings are taken for a while before the pump reaches steady state operation at this point. The decision of reaching the equilibrium is made by looking at the oscillations on both flowmeter and manometer. When the oscillations damp out, the recording is performed by means of flowrate and pressure readings from the displays of flowmeter and manometer, respectively. The procedure described above is repeated for a number of data points that are sufficient to inspect the hydraulic and power consumption characteristics of the pump.
10. The readings of Head, Power and Discharge shall be taken at rated flow and four more points as near as possible at shut off (for closed impellers only), 80%, 110% and 130% of rated flow. The time interval between two consecutive readings shall be 5 minutes minimum.
11. The pump performance will be evaluated by drawing following characteristics at rated speed
 - a) Capacity vs. Pump head
 - b) Capacity vs. Pump efficiency
 - c) Capacity vs. Pump input power

12. Vibration and noise level readings shall be taken for record purpose only. However, noise and vibration shall be guaranteed at site only.

13. The pump is then stopped. The electrical connections are unplugged and the safety regulations are followed to reassemble the pump from the test stand.

5.5.4 Processing the Test Data:

The procedure and instrumentation of measurement of different parameters during the testing are detailed hereafter: Reading of head, power and discharge will be taken at different pump discharge conditions on both sides of the rated discharge value of the pump

Capacity measurement:

Flow in test line will be measured by magnetic type flow meter / orificemeter installed in the delivery pipe line. For flow meter, direct flow reading shall be displayed through PLC connected with the system. For flow measurement from orificemeter, differential head across the orifice plates will be sensed by the Differential Mercury Manometer. The flow rate through all the delivery line(s) will be controlled by throttling Butterfly valve(s) installed downstream of the orificemeters. The orificemeters conform to the requirements of BS EN ISO 5167- 2:2003 or JIS B 8302: 2002. In case of orifice, flow = $K \sqrt{L_1 + L_2}$ where K = Orifice constant, L_1 & L_2 are height of mercury column measured from manometer.

Total Pump Head measurement:

Total pump head will be calculated as follows:

$$\text{Total Pump Head } H = H_p + H_b + H_v$$

Where,

H_p = Pressure gauge/transmitter reading from the column/discharge pipe tapping,

H_b = Level difference between the pressure gauge/transmitter centre line and water level in the sump,

H_v = Velocity head at the pressure gauge/transmitter tapping point in column pipe, $= V^2 / (2g)$

V = Velocity of water at the section where the pressure gauge/transmitter tapping is made,

g = Acceleration due to gravity.

$$\text{Total Bowl Head } H_{\text{bowl}} = \text{Total Pump Head } H + H_{\text{loss}}$$

Where,

H_{loss} = Head loss will be calculated by using the Hazen Williams Formula

Speed Measurement:

Rotational speed of the pump will be measured by using an Optical Digital Tachometer.

Pump Power Input Measurement

To arrive at the pump input power, motor input power is measured by Two - Wattmeter method and then pump input power is calculated after multiplying it by motor efficiency.

Pump Input Power = Motor Input Power x Motor Efficiency

If W_1 and W_2 are the readings of the two Wattmeters, the power input to the motor will be given by:

Motor Input Power = $(W_1 X_1 + W_2 X_2)$ C.T. Factor x P.T. Factor / 1000

P.T. Factor = current transformer's input/output current ratio, P.T. Factor = power transformer's input/output voltage ratio

Where, X_1 and X_2 are individual Wattmeter factors

Alternatively, motor input power shall be directly taken from PLC panel

Motor efficiency will be obtained from the in-house Motor calibration record (for lab motor) or motor type test certificate in case of job motor.

Motor output power = motor input power x motor efficiency = pump input power.

Bowl Input Power

Bowl Input Power = Pump Input Power - (Line Shaft Loss + Thrust Bearing Loss)

Line shaft loss + thrust bearing loss is calculated from Pump input power - bowl input power from data sheet

Bowl Efficiency

Bowl efficiency will be derived from the following relationship:

$$\text{Bowl Efficiency} = \frac{\text{Bowl Output}}{\text{Bowl Input}}$$

Where, Bowl Output Power in KW = $Q (M^3 / Hr) \times H (MWC) / 367.2$

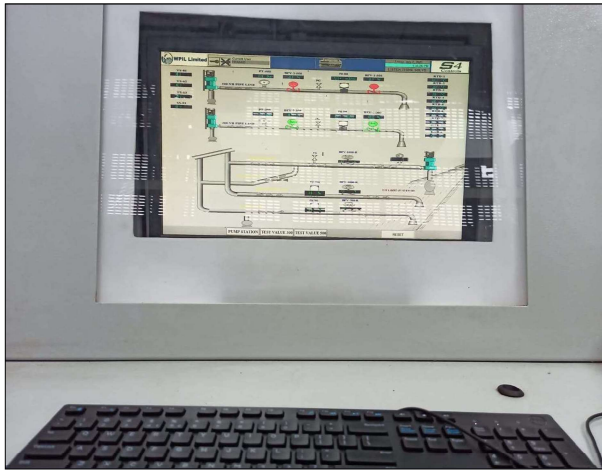
Table 5.2: Accuracy of measuring instruments:

Sl. No	Parameters	Instrument	Class of Accuracy
1	Capacity	Orificemeter/Flowmeter	+/- 1.0 %
2	Pressure	Pressure gauge/Transmitter	+/- 0.5 %
3	Power	Wattmeters/Power meter	+/- 0.75 %
4	Speed	Optical Tachometer	+/- 1 rpm

5.9 TEST SET UP AND INSTRUMENTATION PHOTOS:



TEST PIPE LINE



PLC BASED DATA ACQUISITION SYSTEM



FLOWMETER



REGULATION VALVE ACTUATOR



Digital sound level meter



Digital Tachometer



5.5.5 Performance Test Report:

PERFORMANCE TEST REPORT

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OBSERVATIONS

S. No	Gauge to Water Level height (m)	Del. Gauge Reading (kg/cm2)	Discharge Q (m3/hr)	Current			Watt. Mtr. Reading		SPEED (RPM)	Voltage (volts)
				A1 (Amps)	A2 (Amps)	A3 (Amps)	W1 (Watts)	W2 (Watts)		
	A	B	C	D	E	F	G	H	I	J
1	1.500	1.85	1316.0				95	60	1349	396
2	1.500	2.10	1206.0				93	61	1349	392
3	1.500	2.35	1097.0				90	60	1351	392
4	1.500	2.55	990.0				88	58	1353	396
5	1.500	2.90	763.0				78	58	1352	396
6	1.500	4.00	0.0				70	45	1356	396

CALCULATIONS

CT Factor= 80

Watt Meter factor= 8

S. No	Dis. Velocity (m/sec)	Vel. Head (m)	Total Head (m)	Discharge (m3/hr)	Bowl Out-put (KW)	Current (amps)	Motor Input (KW)	Bowl Input (KW)	Bowl Efficiency (%)	SPEED (RPM)
	$C/3600 / (\pi r d^2/4)$	$(Vd^2)/2g$	$(A+10XB+K)$	DIRECT READING	$[L \times C] / 367.2$	$[D+E+F]/3 \times CTF$	$[G+H] \times CTF \times WMF/1000$	$P \times \eta_m$	$N/Q \times 100$	Direct Reading
	Vd	K	L	C	N	O	P	Q	R	I
1	2.91	0.43	20.43	1316.0	73.22		99.2	89.21	82.08	1349
2	2.67	0.36	22.86	1206.0	75.09		98.6	88.64	84.72	1349
3	2.43	0.30	25.30	1097.0	75.58		96.0	86.33	87.55	1351
4	2.19	0.24	27.24	990.0	73.45		93.4	84.03	87.41	1353
5	1.69	0.15	30.65	763.0	63.68		87.0	78.28	81.35	1352
6	0.00	0.00	41.50	0.0	0.00		73.6	66.19	0.00	1356

RESULTS AT RATED RPM 1484

S. No	Total Head (m)	Discharge (m3/hr)	Bowl Out-put (KW)	Motor Input (KW)	Bowl Input (KW)	Bowl Efficiency (%)	Guaranteed Duty Point	
	$(\text{Rated Speed} / I) \times L$	$(\text{Rated Speed} / I) \times C$	$(\text{Rated Speed} / I) \times N$	$(\text{Rated Speed} / I) \times P$	$V \times \eta_m$	$U/W \times 100$		
	S	T	U	V	W	X	Duty Point	
	m	m3/hr	KW	KW	KW	%		
1	24.73	1447.7	97.48	132.06	118.76	82.08	Duty Point	Head (mtr)
2	27.67	1326.7	99.96	131.21	118.00	84.72		Capacity (mtr ³ /hr.)
3	30.53	1205.0	100.18	127.24	114.42	87.55		Bowl Efficiency (%)
4	32.78	1085.9	96.92	123.29	110.88	87.41		Speed (rpm)
5	36.92	837.5	84.21	115.10	103.51	81.35		Bowl Input (kw)
6	49.70	0.0	0.00	96.47	86.76	0.00		115.30

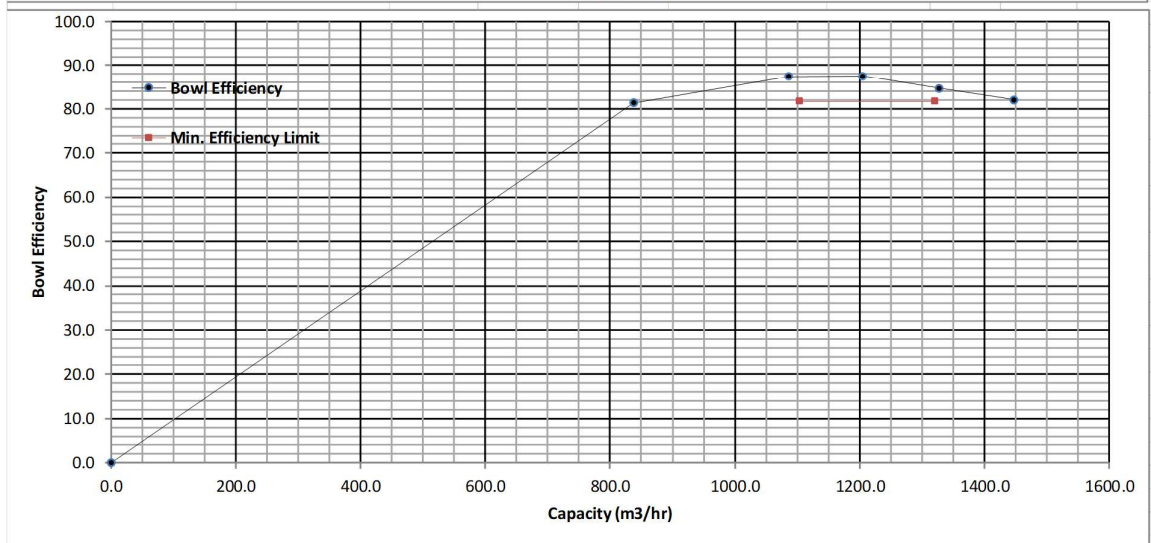
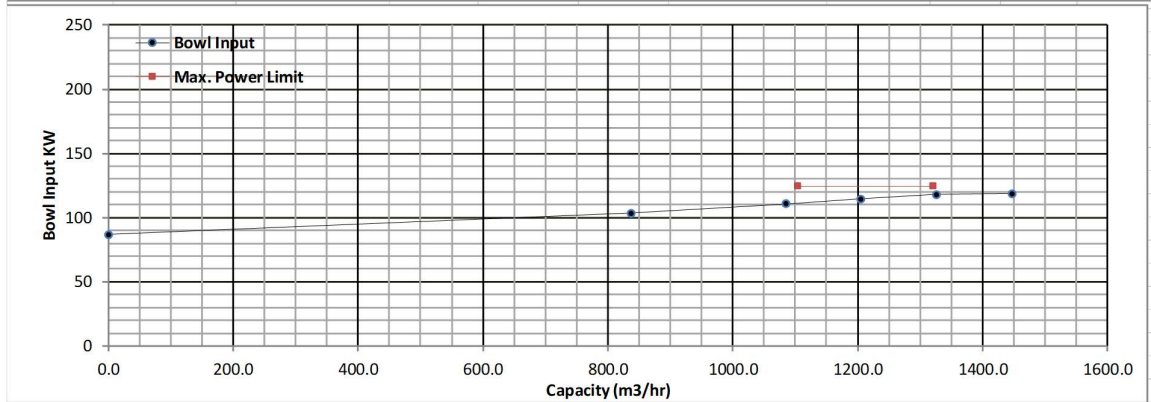
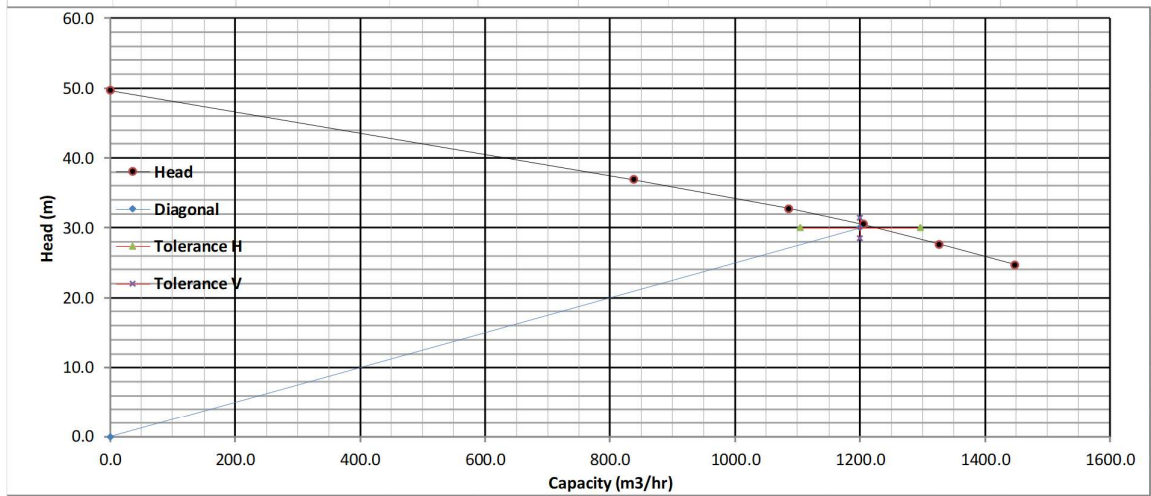
CONCLUSION Performance OK as per HI 14.6 Gr 2B

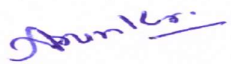
Noise (db)	Vibration (mm/sec)		
	X	Y	A
80.0	4.7	2.4	4.8

7-Feb-24	<i>Amir K.</i>									
Date		Tested By				Witnessed by				

 WPIL Limited	Kolkata, India	PUMP TEST CURVES--MODEL - WVT 32-20 (D20TC)	STAGE- 1	VBS- 50091
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Head (mtr)	Capacity (mtr ³ /hr.)	Bowl Efficiency (%)	RPM	Bowl Input	Test class	HI 14.6 Gr 2B
30	1200	86	1484	115.3		



7-Feb-24						
Date	Tested By		Witnessed by			

5.5.6 Sample Test Result Calculation:

Table 5.4: Performance Test sample Calculation

Observations at duty point

Gauge to Water Level height (m)	Del. Gauge reading (kg/cm ²)	Discharge Q (m ³ /hr)	W1 (Watts)	W2 (Watts)	Speed (RPM)	CT Factor	Watt meter factor	Delivery size (m)
1.500	2.35	1097.0	90	60	1351	80	8	0.400

Total Head Calculations

Total Head = Discharge Gauge reading+Gauge height above water level+velocity Head

A	At Duty point Discharge gauge reading is	2.35 Kg/cm ²	
B	Gauge height above water level is	1.500 m	
C	Velocity in discharge line = Discharge/Area of delivery pipe		
D	At Duty point Discharge reading is		
E	Area of delivery line = $\pi(Dd)^2/4 =$	$\pi/4 \times 0.16$	0.125714286 m ²
F	Hence velocity =	(Discharge/area)/3600, as vel. is in m ³ /hr =	2.42 m/sec
G	Velocity Head =	(Velocity) ² /2g =	0.30 m
H	Hence Total Bowl Head =	(Ax10)+ B + G, as gauge reading is in Kg/cm ²	25.30 m @test speed

Calculation of Head at Rated speed

	Rated speed =	1484	
	Test speed =	1351	
I	Ratio of Rated speed toTest speed =	1.09845	
J	Total Bowl Head at rated speed = (Ratio of Rated speed toTest speed) ² X Head at test speed		30.5 m

Capacity calculation

	It is direct reading of flow meter =	1097.0 m ³ /hr @test speed	
	Capacity at rated speed = (Ratio of Rated speed toTest speed) XCapacity at test speed		1205.0 m ³ /hr

Power calculation

	Power input to Bowl =	Power input to motor x Motor efficiency	
	Power input to Bowl =	(Sum of both wattmeter reading) x Watt meter factor x CT Factor	
K	At Duty point first wattmeter reading is	90 watts	
L	At Duty point second wattmeter reading is	60 watts	
M	Watt meter factor is	8	
N	CT Factor is	80	
O	Motor Efficiency is	89.93%	
	Hence power input to Bowl is	(K+L)X M X N X O/1000 KW, as power is in watts =	86.33 KW @test speed
P	Bowl input at rated speed =	(Ratio of Rated speed toTest speed) ³ XBowl input at test speed	114.42 KW

Bowl Efficiency calculation

Q	Bowl assembly output =	Total Bowl Head X Discharge/367.2 KW =	100.17 KW
	Bowl Efficiency in percent =	Bowl output/Bowl input = (Q/P) X 100 =	87.55 %

CHAPTER 6: RESULTS AND CONCLUSION

6.1 Test Results and Discussion:

This chapter presents the experimental findings from the performance testing of the vertical turbine pump (VTP) and discusses their implications in the context of the design objectives and literature. The first part in this chapter is the discussion of the effects of the design parameters on the hydraulic performance of the pump. Then the results are analysed in terms of performance metrics such as flow rate, head, efficiency, and cavitation characteristics, and are compared with theoretical predictions.

In this thesis, a vertical turbine type pump is designed, manufactured and tested in WPIL LIMITED Pumps Company in Kolkata. The tests of the pump are performed with an electric motor having the nominal speed of 1500 rpm. Since the design of the impeller is started with existing impeller layout, necessary changes are made on this profile in order to meet the design point of the pump. The methodology that is given in Chapter 4 regarding to the design of the impeller layout is a trial and error procedure. In deed once the profile is constructed, the preliminary design is performed and several points are checked in order to continue with shaping the blades of the impeller. First check point is the blade number of the impeller. The number of the blades with respect to the specific speed and the experience coming from the company is followed while controlling the blade number. It is found that the hydraulic efficiency is highly affected by finding out the correct value of the blade number for the desired characteristics of the pump.

The blade thickness of the impeller is directly affecting the hydraulic performance of the pump. However, from the production point of view the blade thickness is selected according to the casting capabilities. Although the meat thicknesses of the impeller should be selected close to the thickness of the blade in order to cast the impeller properly.

Performance Curves Head vs. Flow Rate

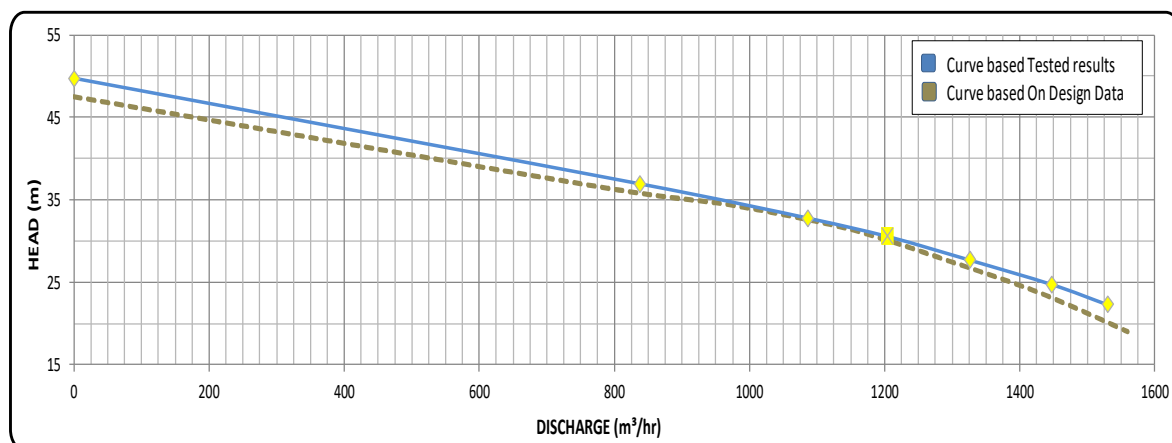


Fig: 6.1 TOTAL BOWL HEAD Vs DISCHARGE CURVE

Results:

- The head developed by the pump was measured at various flow rates. The data was used to plot the head-flow characteristic curve (Figure 6.1).
- At lower flow rates, the pump generated a higher head, which decreased as the flow rate increased.
- The head-flow curve indicates the pump's ability to convert mechanical energy into hydraulic energy. The curve's shape matches the expected trend for vertical turbine pumps, demonstrating higher heads at lower flow rates due to lower hydraulic losses.
- Deviations from the predicted curve at higher flow rates can be attributed to increased hydraulic losses and potential cavitation effects. The results highlight the importance of optimizing the impeller and diffuser design to minimize these losses.

Efficiency vs. Flow Rate

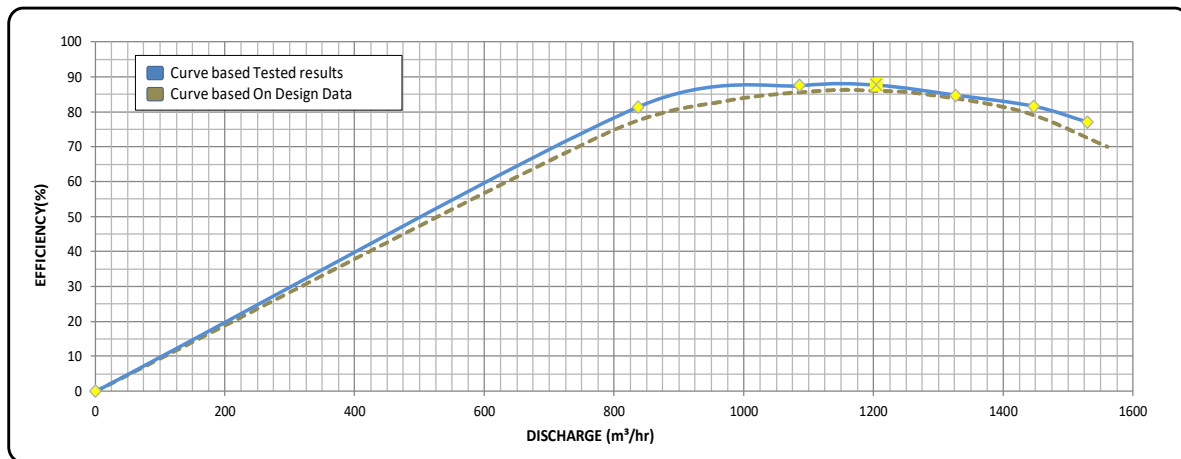


Figure 6.2: BOWL EFFICIENCY Vs DISCHARGE

Results:

- Efficiency measurements were taken at various flow rates, and the efficiency curve was plotted (Figure 6.2).
- The pump's efficiency peaked at a certain flow rate, known as the best efficiency point (BEP), and decreased at flow rates higher and lower than the BEP.
- The efficiency curve illustrates the pump's bowl efficiency, highlighting the BEP where the pump performs most effectively.
- Efficiency decreases at lower and higher flow rates can be linked to increased hydraulic losses and off-design operation. Best efficiency points of the pump that is designed in this study moves from 1170 m³/hr to 1210 m³/hr and system efficiency increases 1.55% for the best efficiency point. Optimization of the impeller blade shape and angle reduce hydraulic losses and improve fluid flow.
- The experimental efficiency values are compared with theoretical predictions.

Bowl Input Power vs. Flow Rate

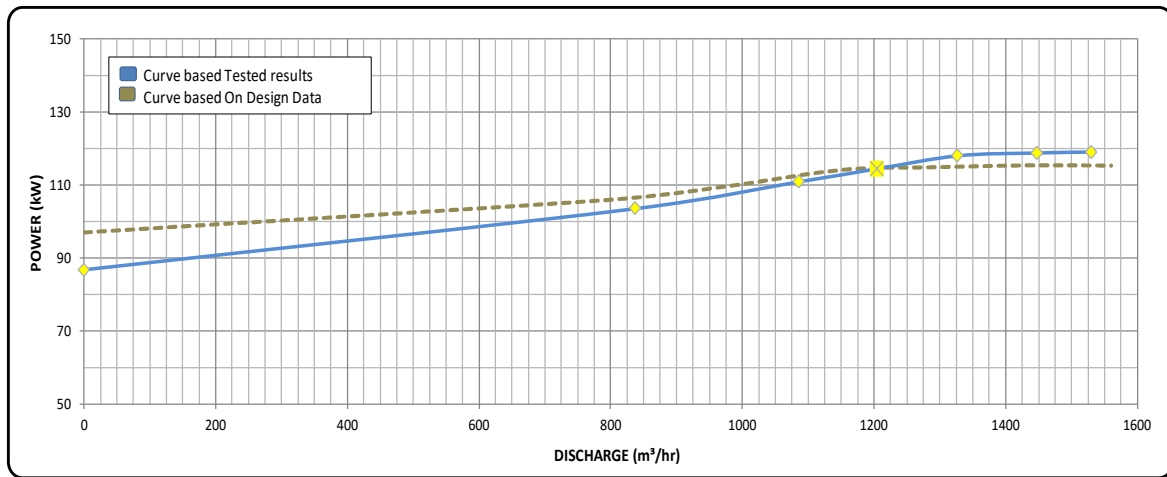


Figure 6.3: BOWL INPUT POWER Vs DISCHARGE CURVE

Results:

- The input power required by the pump was recorded across different flow rates and used to plot the power curve (Figure 6.3).
- The power consumption increased with flow rate, demonstrating the relationship between flow rate and the energy required to maintain it.
- The power curve provides insights into the pump's energy consumption characteristics, crucial for understanding operational costs.
- The increase in power consumption with flow rate is consistent with the pump's design, as more energy is needed to overcome higher hydraulic loads.
- Comparing these results with theoretical ensures that the power requirements align with design expectations.

Cavitation Characteristics:

Cavitation tests were conducted by reducing the submergence level until cavitation was observed. The Net Positive Suction Head required (NPSHr) was measured. The critical submergence level was determined and compared with the theoretical NPSH available (NPSHa).

- The cavitation tests show the pump's susceptibility to cavitation under different operational conditions. NPSHr values provide critical information for safe pump operation.
- Ensuring NPSHa exceeds NPSHr is essential to prevent cavitation, which can cause damage and reduce efficiency.
- Comparing the experimental NPSHr with theoretical values helps in validating the pump design and ensuring it meets operational safety margins.

6.2 Conclusion:

The experimental study of the vertical turbine pump demonstrates the effectiveness of the design and construction methodologies in achieving desired performance metrics. The results validate the theoretical calculation, confirming the pump's efficiency and operational reliability. This research contributes valuable insights into VTP design, offering a foundation for future innovations and improvements in pump technology.

Underfilling of impeller blades at the trailing edge improved the performance of the pump that is designed in this study, as stated in References [19]. Best efficiency points of the pump that is designed in this study moves from 1170 m³/hr to 1210 m³/hr and system efficiency increases 1.55% for the best efficiency point. The best efficiency point of the pump moves towards to the design point when the blades are underfilled. The disturbance on the trailing edge of the blade caused by offsetting the designed surface in order to give thickness the blade is reduced, when underfilling is performed in the impeller blades.

On the hydraulic design of the impeller, the impeller layout is designed starting with an existing layout. Building an impeller layout library and using this library for the further designs is an important tool from the design point of view. However, if the designed impeller layout for a specific design does not satisfy the design parameters, the reverse process may be followed.

The manufacturing of the pump is important as the hydraulic design studies and numerical experimentations. While producing the pump, the cores and core boxes should be manufactured by following the original design. However, the designs should be robust in order to manufacture the pumps easily and getting the closest pump characteristics when the test results are compared with the CFD results and hydraulic design of the pump. Selecting the suitable blade and vane thicknesses, blade swirl and vane swept angles are important from the manufacturing point of view.

By adopting a comprehensive approach that combines design improvements, material advancements, optimal operational practices, and proactive maintenance, the efficiency of vertical turbine pumps can be significantly enhanced. This not only reduces operational costs but also contributes to sustainability by lowering energy consumption and extending the lifespan of the equipment.

6.3 Future Work:

In order to improve the design procedure that is followed in this thesis, there are several points may be checked. The pressure distribution may be determined on the designed streamlines by using CFD. The plots of the pressure distribution versus meridional distance on the pressure and suction sides of the impeller blade give an idea about the theoretical head of the pump that is aimed to be designed. Moreover, the distribution of the velocity components on the designed streamlines may be drawn by the help of CFD and these distributions can be compared with the velocity distributions on the shaping of the blades part of the design.

From the production point of view, the casting technique that is used to cast the products in this study is simple gravity die casting. In this method, the thicknesses that could be cast are limited. However, using another casting technique such as investments casting may allow the designs to be cast with smaller thicknesses. Decreasing the thickness of the blade increases the pump performance. However, decreasing the thicknesses of the blades allows the impellers face with wear when they are operating in environments where the impurities exist in the working fluid such as sand. The thicknesses should be adjusted by considering the overall operating life of the pumps.

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