PERFORMANCE EVALUATION OF AXIAL FLOW PUMP AS TURBINE

A DISSERTATION

Submitted in partial fulfillment of the requirements for the award of the degree
of
MASTER OF TECHNOLOGY
in
ALTERNATE HYDRO ENERGY SYSTEMS

By
DEEPAK S.

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JUNE, 2010

CANDIDATE'S DECLARATION

I hereby declare that the report which is being presented in this dissertation work entitled "**PERFORMANCE EVALUATION OF AXIAL FLOW PUMP AS TURBINE**" in partial fulfillment of the requirements for the award of the degree of **Master of Technology** in "**Alternate Hydro Energy Systems**", submitted in **Alternate Hydro Energy Centre**, Indian Institute of Technology, Roorkee is an authentic record of my own work carried out during a period from July 2009 to June 2010 under the supervision of **Dr. R.P. Saini**, Associate Professor. I have not submitted the matter embodied in this report for the award of any other degree or diploma.

Date: JUNE 30, 2010

(DEEPAK S)

Place:Roorkee

This is to certify that the above statement made by the candidate is correct to the best of my knowledge.

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Dated: June , 2010 **(DEEPAK S)**

Place: Roorkee

ABSTRACT

Energy is the basic requirement for economic development. Every sector of a country's economy needs input of energy. As the nonrenewable fossil energy sources continues to deplete and realizing the summits held at Brazil , Kyoto, Copenhagen to reduce the greenhouse gas emissions, hydropower has moved towards the top power development option to meet the increasing energy demand.

Pump as Turbine has been effectively used in micro hydro power sector for a quite a long time. The use of Pump as Turbine in lieu of conventional custom made turbine has brought down the electro mechanical cost, which contributes a major share in the installation cost of micro hydro power plants and thereby making micro hydro power plants more economically feasible. A lot of work has been carried out in the area of Pump as Turbine especially in analyzing the performance of pumps in turbine mode. A few works has been directed towards suggesting methods to improve the efficiency of pump in the turbine mode of operation. It was however found out that very little work is available in the literature on the use of axial flow pump in turbine mode. These pumps are used for low head and high discharge.

In the Himalayan region a number of water mill sites exist where the head range is of the order of 3-5 m with a capacity of up to 5 kW. The potential of these sites can be harnessed effectively by the use of efficient equipment. Use of axial pumps as turbines can be a good option for such sites. Keeping this in view the present study has been carried out.

Under the present investigation an axial flow pump which can be used in the micro hydro range is dimensioned and fabricated .Computational Fluid Dynamic analysis using the commercially available FLUENT package was carried out to simulate the flow in the turbine mode operation of the pump. The pump was tested in the turbine mode to determine its operating characteristics and the results obtained from the experimentation was compared with simulation results which are found to be in good agreement.

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CHAPTER 1: INTRODUCTION AND LITERATURE REVIEW

1.1 GENERAL

Energy is the basic requirement for economic development. Every sector of a country's economy (industry, agricultural, transport, commercial and domestic) needs input of energy. The installed electricity generating capacity in India at present is nearly 128 GW which is generated 66% through thermal, 26% through hydro, 3% through nuclear and 5% through other renewable source of energy. According to International Energy Agency (IEA), a threefold rise in India's generation capacity is expected by 2020[1]. As the nonrenewable fossil energy sources continues to deplete and realizing the summits held at Brazil , Kyoto, Copenhagen to reduce the greenhouse gas emissions, hydropower has moved towards the top power development option to meet the increasing energy demand.

Hydro power is probably the most oldest and yet the most reliable source of all renewable energy, with bulk of its potential yet to be harnessed in many countries. Though hydro power development started with small units of 100 kW in the beginning, attention was diverted to harnessing medium and major hydro because of their comparative economics. India has a large potential of hydro energy in medium and big projects. After independence many large hydroelectric projects have been executed, with some of them still under construction and some have been planned for future. The inherent drawbacks associated with large hydro are; large gestation period, large area along with vegetation has to be submerged, shifting people etc from the sites. Political and environmental implications have made planners to think for some other alternative to this large hydro and thus come the concept of small hydro. On a small scale hydro power involves exploitation of rivers hydro potential without significant damming and the negative social/environmental effects that can arise. Whether for the powering of a house or a small community, profileration of small hydro on a mass scale could provide one of the most significant and benign energy options available. Small Hydro is thus getting increasingly recognized as a serious player for future decentralized energy supply.

1.2 SMALL HYDRO SCENARIO IN INDIA

There is general tendency all over the world to define small hydro by power output. The plant capacity limit range for different countries is given in Table 1.1. Small hydro projects are economically viable and have relatively short gestation period. The major constraints associated with large hydro projects are usually not encountered in small hydro projects. The World estimated potential of small hydro is of around 180,000 MW. India has as an estimated potential of about 15,000 MW with perennial flow rivers, streams and a large irrigation canal network with dams & barrages. Of this, 5415 potential sites with an aggregate capacity of 14305.47 MW have been identified. A detailed breakup of this distribution is given in Table 1.2.

Table 1.1: Plant capacity limit range of Small Hydro for different countries

COUNTRY	PLANT CAPACITY
U.K.	UP TO 5 MW
UNIDO	UP TO 10 MW
INDIA	UP TO 25 MW
SWEDEN	UP TO 15 MW
COLUMBIA	UP TO 20 MW
AUSTRALIA	UP TO 20 MW
CHINA	UP TO 25 MW
PHILIPPINES	UP TO 50 MW
NEW ZEALAND	UP TO 50 MW

India is blessed with immense amount of hydro potential and ranks 5th in terms of exploitable hydro-potential on global scenario. As per assessment made by Central

Electricity Authority, India is endowed with economically exploitable hydro-power potential to the tune of 1, 48,700 MW of installed capacity.

Table 1.2: Small Hydro Power Scenario in India [2]

Overall potential	15,000 MW
Identified potential	14305.47 MW (5415 sites)
Installed capacity	2429.77 MW (674 projects)
Under construction	483.23 MW (188 projects)
Capacity addition during 2002-2007	536.7 MW
Target capacity addition – 11th Plan (2007-2012)	1400 MW

1.3 ADVANTAGES OF MICRO HYDRO POWER

The current concern on the global environment has imposed restraints on the production of electricity. The emphasis is put on the development of environmental friendly methods to promote the sustainable social development. It is in these circumstances, that micro hydro power is drawing more attention. The isolated micro-hydro power plants are usually the least-cost option for rural population which is scattered, and poor. This is mainly because other options for supply of energy such as grid extension, diesel power, etc are more expensive and difficult to install or operate in the long run. Since small water streams are usually available in the most of the region, micro hydro power plants can easily meet the energy needs of small village or cluster of settlements. These needs may be in the form of electricity or motive power to be used for agro-processing, wood working and for other small scale industries. This use of electricity in the form of heat can contribute significantly towards reducing the burning of wood and other biomass, which has many derogatory implications in terms of environment and health. In addition to meeting the needs of an area, a properly designed, installed and managed micro-hydro power plant can also contribute significantly towards employment generation, improved living conditions and improved education facilities.

They are reliable and within the limits of the water resources available, can be tailored to the needs of the end use market. The civil works for micro hydro power stations are simple. They do not need elaborate construction works in reinforced concrete, no expensive powerhouse and highly optimized electromechanical equipment. Operation is simple and maintenance is easy. Under favorable circumstances, micro-hydro represents one of the cheapest methods of electricity generation. There is no need for a long transmission lines because output is consumed near the source itself.

1.4 DISADVANTAGES AND CONSTRAINTS OF MICRO HYDRO POWER

Hydro project have an important characteristic that no two potential sites are alike. Topography, flow regime and volume of the river concerned together with the geological condition of the site are variables that make each installation unique. Long term hydrological data are seldom accurate and to some extent, topographical maps have insufficient details and place a severe constraint on hydropower development. The high capital cost of hydropower development has been a constraint to the development of Micro Hydro Power plants. The most severe constraint on the technical side is the dissemination of low cost, local hydropower technology. This is especially true regarding the problems of regulating speed of the generating set. The identification in these schemes requires physical involvement and reconnaissance at micro level of the state organization. Economic viability of Micro Hydro Power Plants is another aspect. In spite of the above constraints the major problem associated with micro hydro power scheme is the non-availability of standardized equipments like turbines because there are number of different micro hydropower sites having different head and discharge, therefore turbines need to be tailored for particular sites. This process is a costly one, which hinders development of micro hydropower sites.

1.5 TYPES OF HYDROTURBINE USED IN MICRO HYDRO SCHEME

In case of selection of turbines for a micro hydropower, as can be seen from Figure 1.1 the usual alternatives would be either cross flow turbine or Pump as Turbine. The Cross Flow Turbine used in the micro hydro power range suffers from poor efficiency though the construction is simple and can be carried out easily. The use of standard Pumps as turbines may often be an alternative with a considerable economic advantage and might therefore contribute to a broader application of micro-hydropower. Direct drives of machinery, electricity generation (in parallel to a large grid or isolated) or combinations of these are possible just as with a conventional turbine. The only difference is that a Pump as Turbine cannot make use of the available water as efficiently as a turbine due to its lack of hydraulic controls. Pump as Turbine has a property that it gives maximum efficiency at a particular discharge, which is greater than the rated discharge of pump.

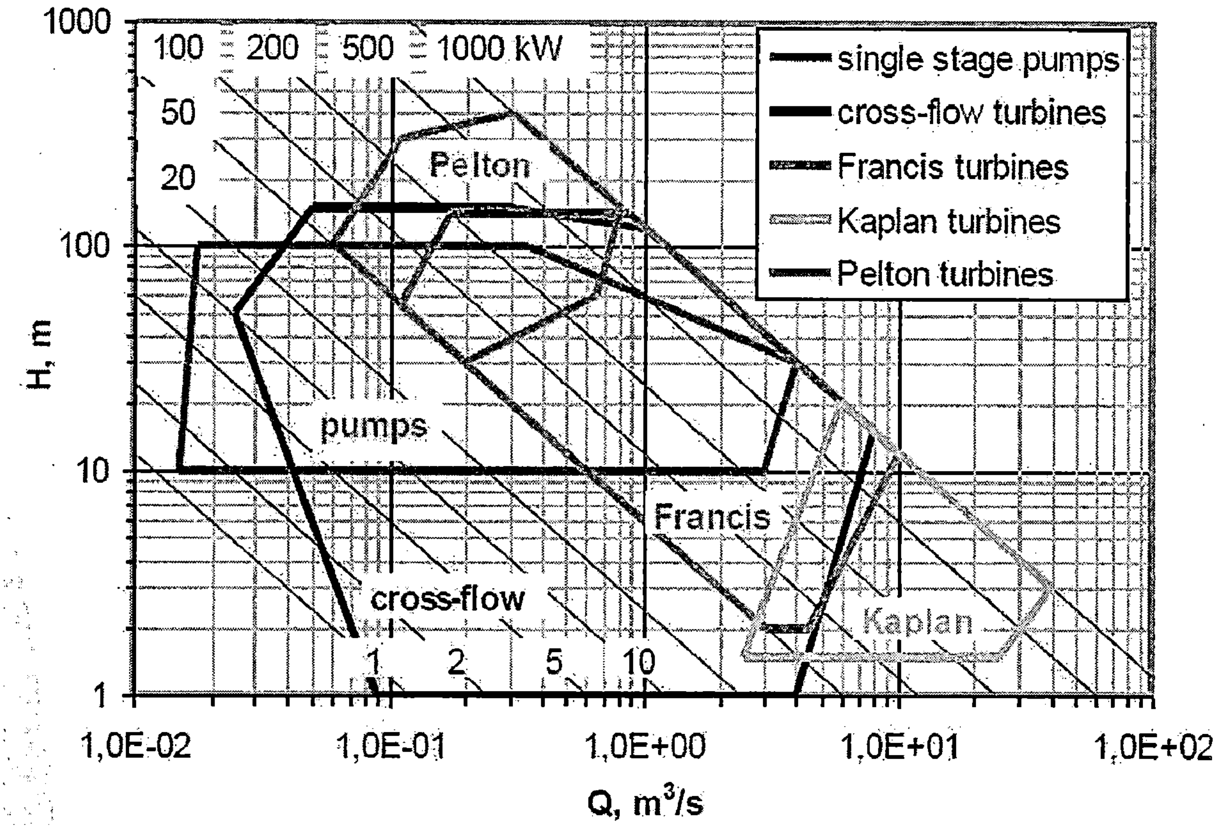

Figure 1.1: Types of Turbine used in Micro Hydro Range [3]

If PAT is operated at other than that particular discharge, rapid drop of efficiency occurs. Thus, pumps as turbines are best suited for constant flow applications. The problem with PAT arises when there are seasonal variations in flow rate. As there are no guide vanes in standard pump units, therefore when the discharge varies or the load varies there is loss of efficiency of PAT resulting in poor part load efficiency of PAT.

1.6 LITERATURE REVIEW

The research on using pumps as turbines started around 1930 [4]. The use of pump as turbine was first realized by the pump manufacturers. They made use of multistage radial flow units as power recovery units in the process industries. At the same time a few independent developers focused their attention on the use of pump as turbine for their small hydro electric plants as a means to cut down the initial cost. Also a few pump as turbine installations were installed in the water supply line to recover energy lost in the pressure reducing station. The details of the above work were published by the U S Department of Energy.

Hydraulic pump when operated as a turbine shows certain relations between the characteristics as a 'pump' and 'PAT'. These characteristics are operating flow, operating head, operating efficiency, shaft torque and shaft power. Different approaches [5-19] were proposed by different people to find out operating relations between pump and PAT but the methods had a large tolerance band in prediction curves and methods were mainly applicable to high head micro-hydro systems.

Similar relationships are provided by Williams [20] for centrifugal pumps but he warns that predicted results may deviate from the actual results by +/- 20 % and recommends testing of PAT before putting the selected pump into service wherever it is possible.

Saini and Ahmad [21] carried out experiments on pump having specific speed ranging from 10 to 300 and developed a nomogram to aid in the selection of a pump to be used a turbine for a particular site. This method reduced the calculation effort involved in the selection as was proposed by early investigators.

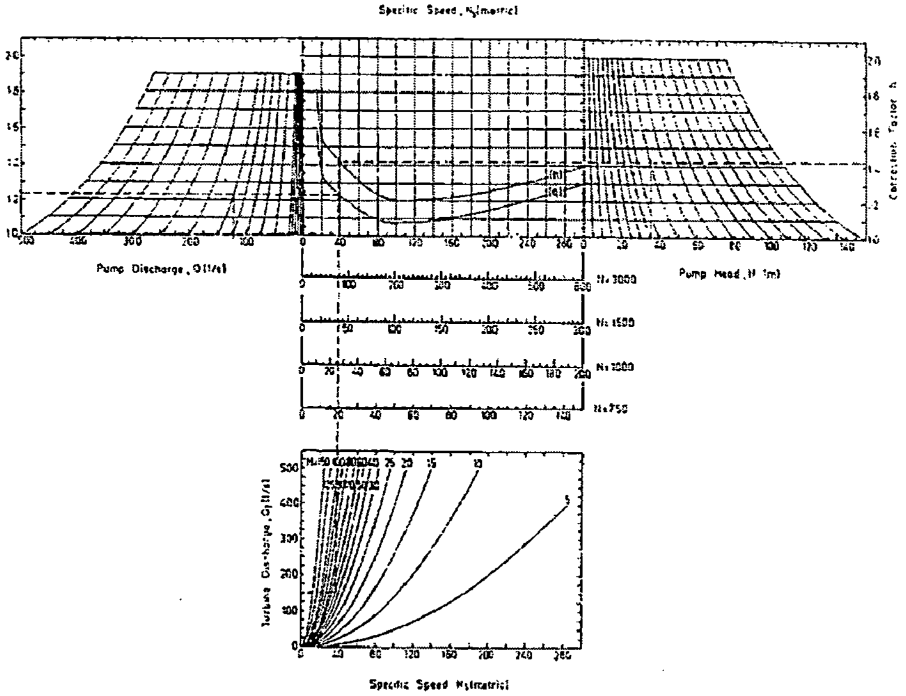

Figure 1.2: Monogram For Selection of PAT[21]

Arthur Williams [22] carried out experimental investigations on Pump as Turbine units and suggested certain design modification for increasing the performance of a pump operated as turbine.

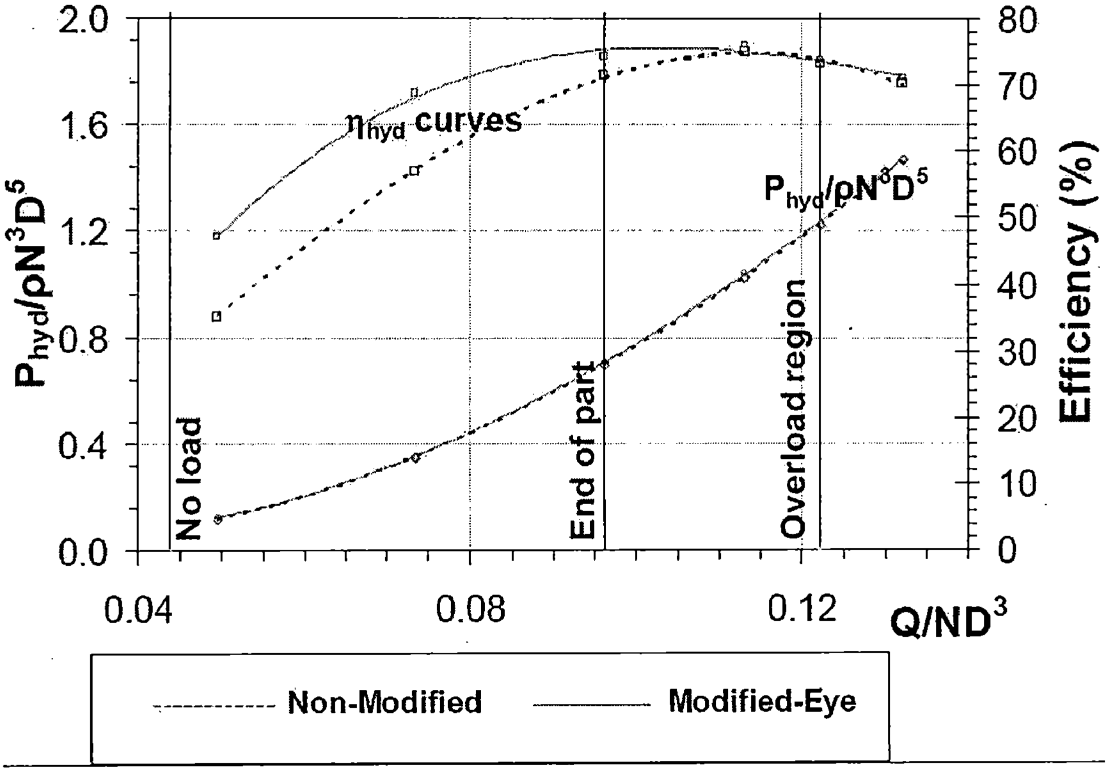

Figure 1.3 Efficiency curve with the design modification[22]

Ramos and Borga [23] provided information with regard to any water system which has excessive available energy, the application of a pump instead of a turbine, for energy production, seems to be an alternative solution with easy implementation and considerable cost savings. They arrived at this conclusion based on Suter parameters, the analysis of steady-state conditions will take place to prove that although pumps are widely used in industrial process they can also be used to produce renewable energy.

Joshi et al. [24] carried out investigations and based on their experimental data using three pumps presented a simple approach based on the specific speed of pump to select a pump that is best suitable to run as a hydro turbine for a small hydro power site.

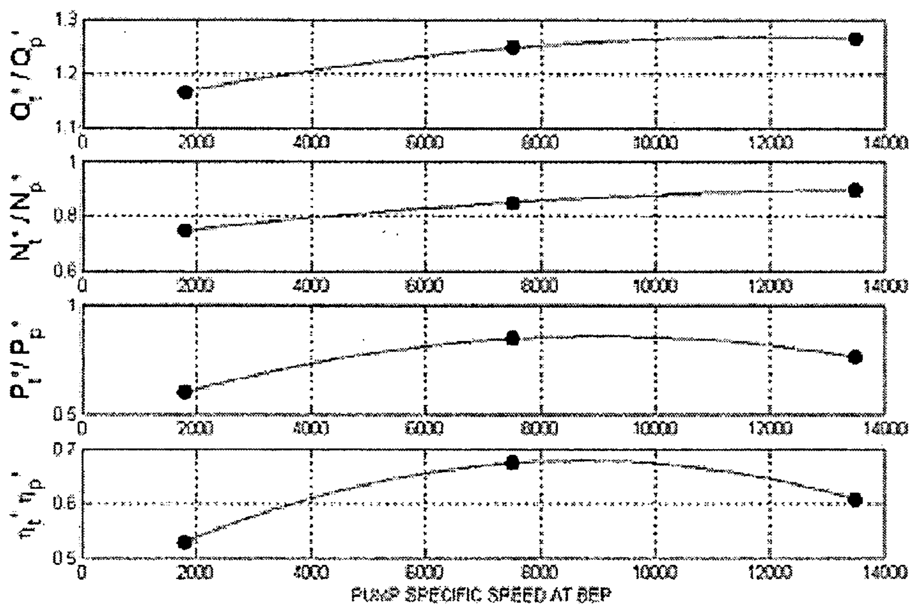

Figure 1.4: PAT characteristics BEP ratios plotted against pump specific speed at BEP in constant head mode[24]

Fernandez et al. [25] carried out investigations and presented functional characterization of a centrifugal pump as turbine. The paper described the influence of rotational speed on efficiency. Their paper also compared the forces actuating in the impeller in the pump and turbine mode. The research indicated that the turbine characteristics can be predicted to an appreciable degree of confidence from the pump characteristics. Also it was found out that radial stresses in the turbine mode were lower than that of pump mode.

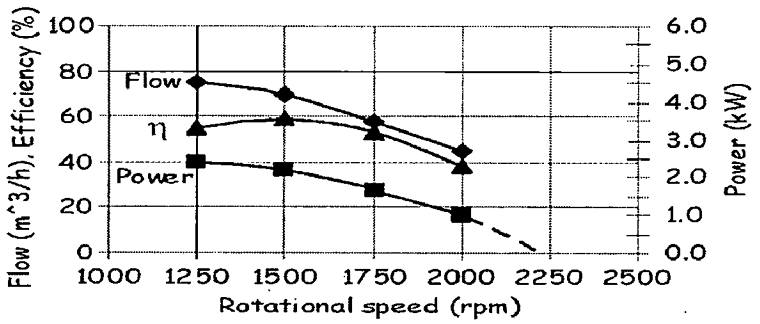

Figure 1.5: Performance curve of PAT unit[25]

Anees Ahmad [26] carried out investigations on a commercially available centrifugal pump with the addition of movable guide vanes. Based on his experimental results it was observed that efficiency of PAT after modification has been increased. It was also observed that maximum efficiency of PAT without modification can be achieved over a wider operating range with the modification.

Sonia and Kshirsagar [27] studied the experimental result of a pump with specific speed 4843 in U.S. units and carried out numerical investigations using CFD tools to stimulate the pump in turbine mode. They found satisfactory results near to best efficiency point. They found the numerical technique to be useful in finding out losses in components like the impeller, the draft tube and the casing.

Derakhshan and Nourbaksh [28] carried out investigations to predict the best efficiency point of a centrifugal pump working as turbine based on pump hydraulic characteristics. They experimented with several pumps having specific speed less than 60. Experiments showed that a low-specific-speed centrifugal pump can operate as turbine in different rotational speeds and various heads and flow rates without any mechanical problem. The results are shown in Figure1.6, Figure 1.7 and Figure 1.8. In these figures, ψ, ϕ, π are

$\psi = \frac{gH}{n^2 D^2}$, $\phi = \frac{Q}{nD^3}$, $\pi = \frac{P}{\rho n^3 D^5}$ where H (m), Q (m3/s) and P (W) are head, flow rate and power, respectively. n (rps) is rotational speed and D (m)

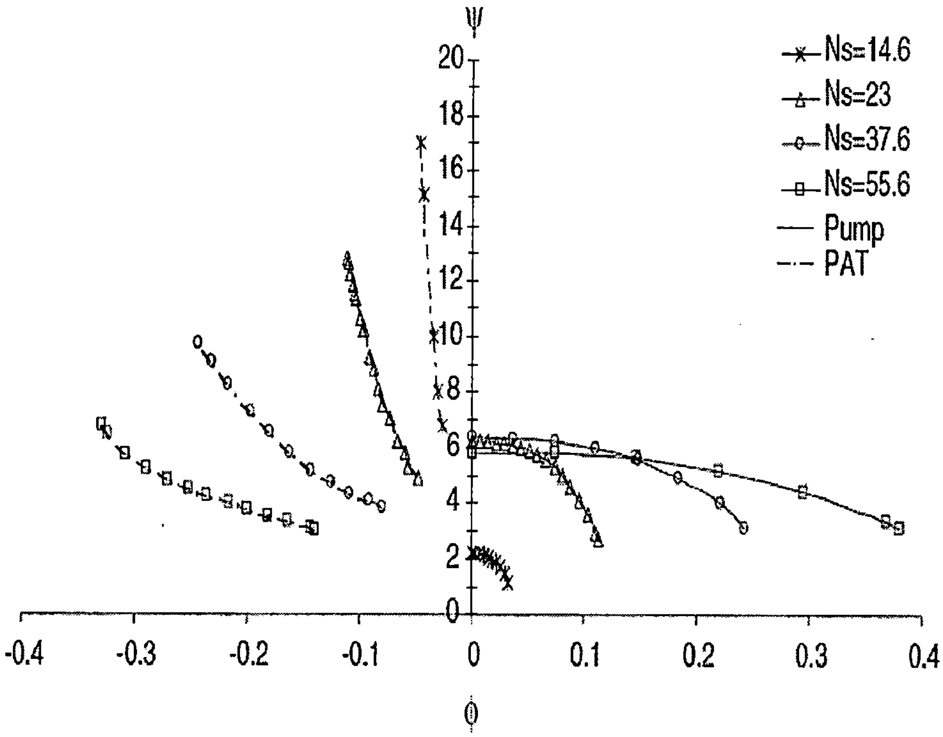

Figure 1.6: Dimensionless head curves of tested PATs in pump and turbine mode[28]

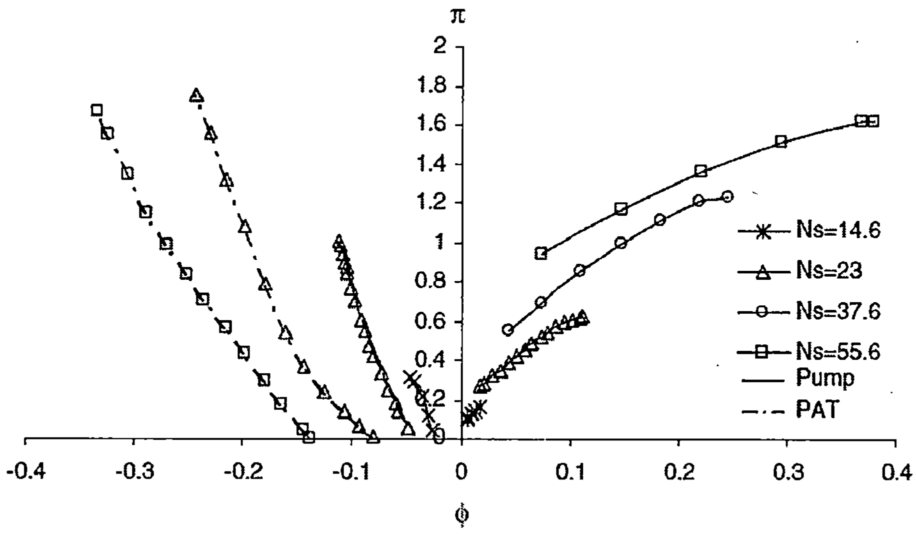

Figure 1.7: Dimensionless power curves of tested PATs in pump and turbine mode[28]

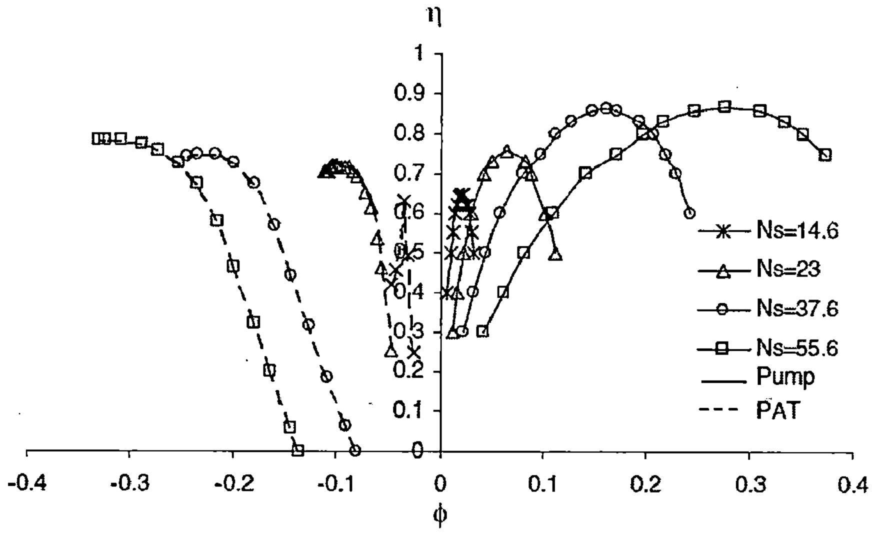

Figure 1.8: Dimensionless efficiency curves of tested PATs in pump and turbine mode[28]

Rahul Pandey [29] studied PAT performance with the use of guide vanes as a flow control mechanism and trimming of the runner blades to reduce the shock losses. With these modifications an increase in efficiency has been reported compared to an unmodified pump that was used as a turbine.

Derakhshan and Nourbaksh [30] carried out theoretical analysis using area ratio method developed by Anderson [31] to achieve the best efficiency point of an industrial centrifugal pump running in reverse. They were able to estimate turbine mode hydraulic characteristics based on pump characteristics. Numerical study was done using computational fluid dynamics to stimulate the machine in reverse and direct modes. Using numerical results, complete characteristic curves of the pump in the direct and reverse modes were obtained. They further tested the simulated pump as a turbine. The analysis showed that CFD results showed a good coincidence with the experimental data for pump mode in the best efficiency point, part load operation and over load zones. But the CFD results failed to achieve an acceptable coincidence with the experimental data from turbine mode.

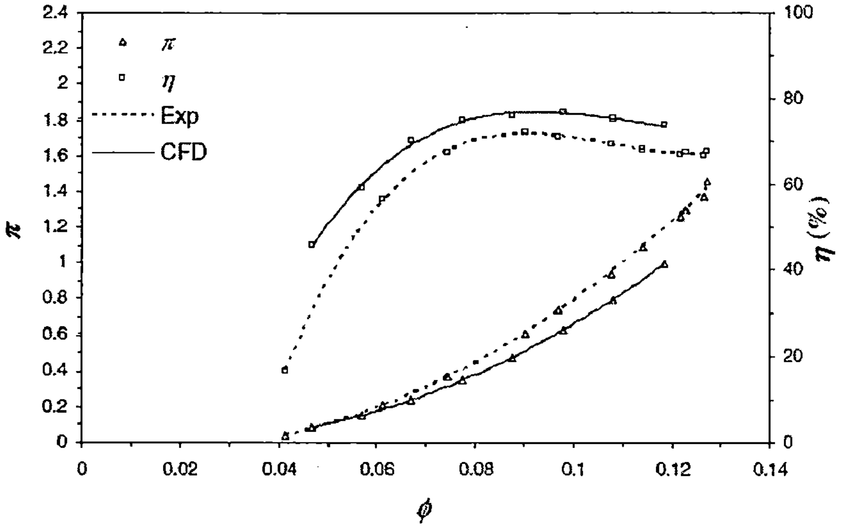

Figure 1.9: Measured and numerical head number and efficiency curves of turbine mode[30]

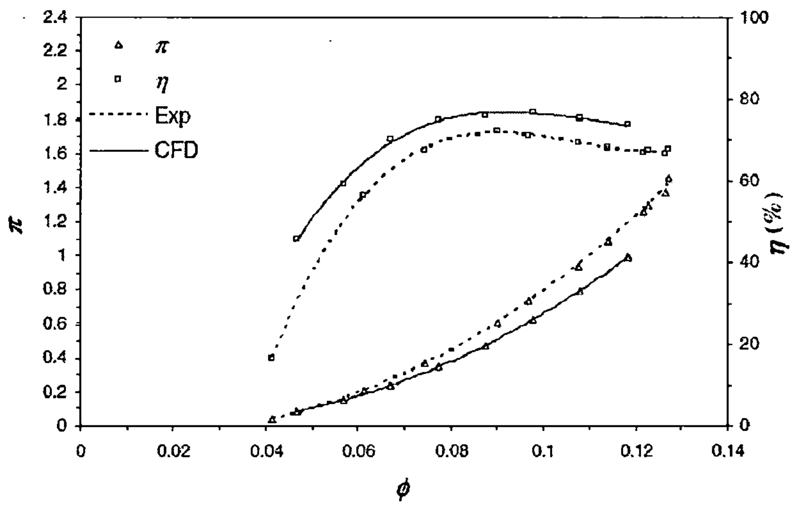

Figure 1.10: Measured and numerical power number and efficiency curves of turbine mode[30]

Derakhshan et al. [32] carried out investigations on impeller geometry of a pump to improve its turbine mode efficiency. The optimization was done using a gradient based optimization algorithm coupled by a 3D Navier-Stokes flow solver. They also studied the effect of turbine performance by other modifications like rounding of the leading edges and hub /shroud inlet edges.

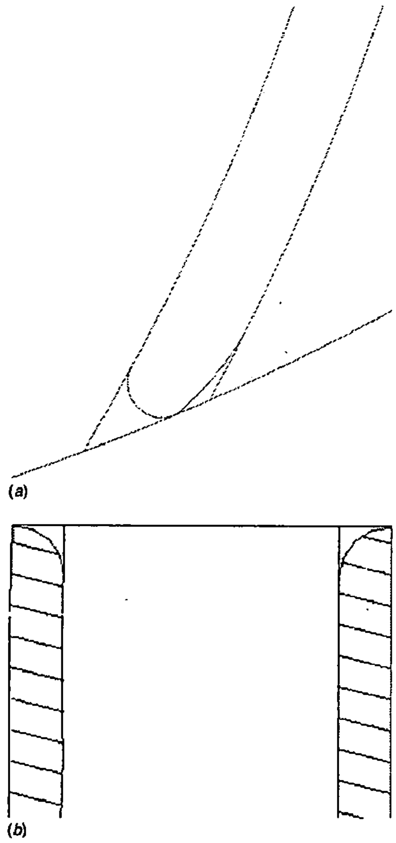

Figure 1.11: (a)Rounding of blades profile at impeller inlet in turbine mode.(b)Rounding of hub/shroud inlet edges[32]

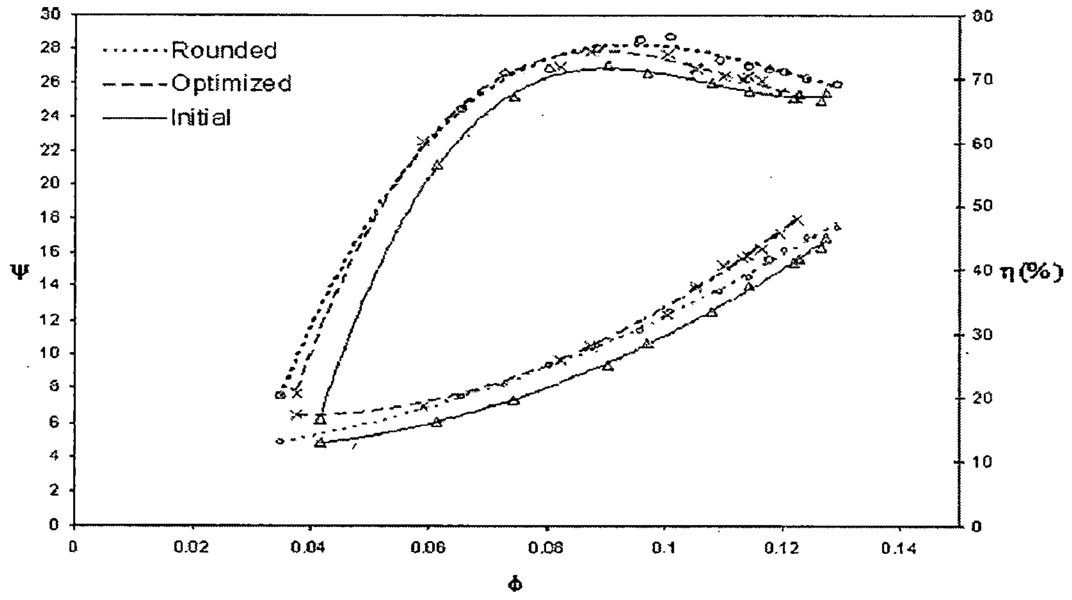

Figure 1.12: Experimental results for head number and efficiency[32]

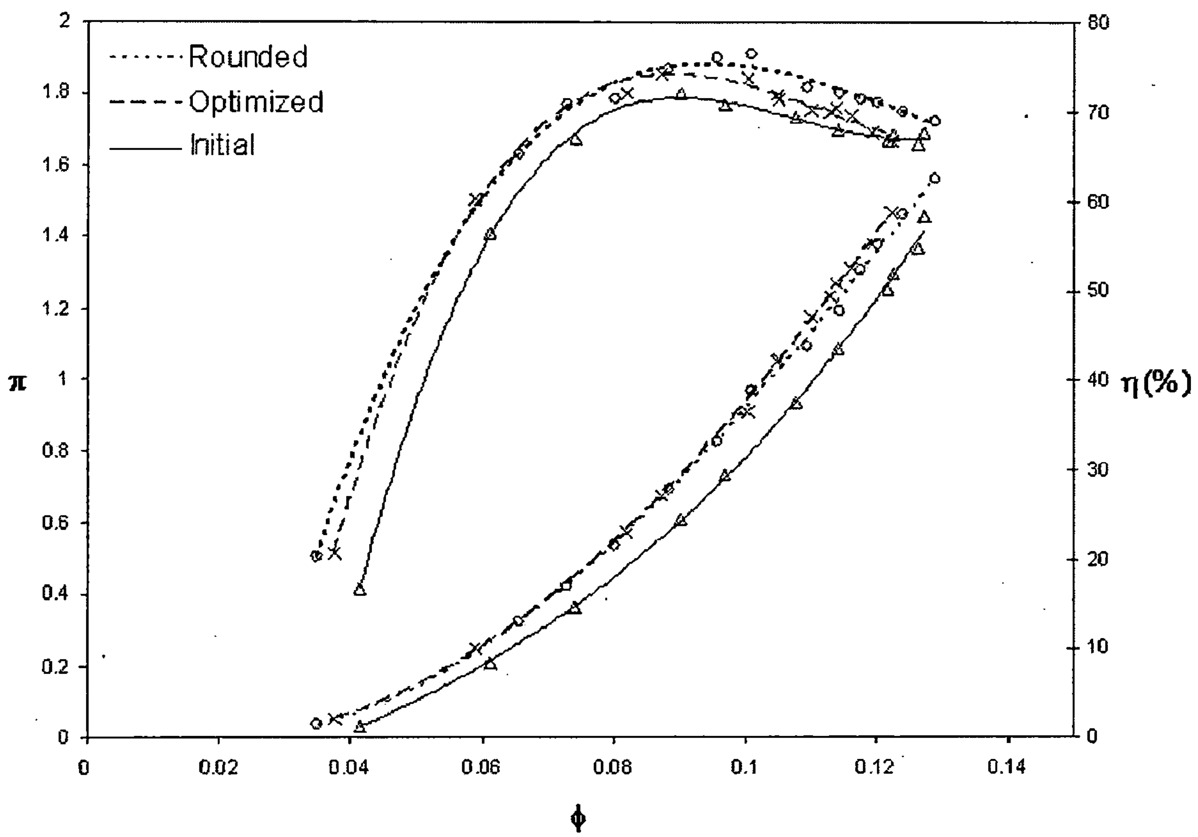

Figure 1.13:Experimental results for power number and efficiency[32]

Fernández et al. [33] carried out investigation on a conventional centrifugal pump working as a turbine. The numerical simulations were performed with the code Fluent by means of unsteady flow calculations and a sliding mesh technique to account for the impeller-volute interactions. Thus, it was possible to properly simulate the effect on the local flow of the passage of the impeller blades in front of the volute tongue. The numerical results were compared with the experimentally determined performance curves and additionally with the static pressure distribution measured around the impeller periphery. Once validated, the model was used to estimate the steady and unsteady radial forces on the impeller for a number of flow rates. The steady radial force was also experimentally estimated from the static pressure measurements around the periphery of the impeller. The numerical predictions showed that, for the flow interval considered in the present investigation, the unsteady radial force varied between 24% and 54.3% of the steady magnitude, and that its maximum amplitude was reached when the trailing edge of one of the blades was located 3 deg downstream the tip of the tongue

White et al. [34] studied turbine performance of a high specific speed pump using CFD.A pump with a specific speed of 12000 was chosen to operate as a turbine for a micro-hydro site having 5 m of head. Turbine performance of the pump was unavailable so it was simulated using CFD. The CFD model was first verified by comparison of simulated pump performance and manufacturer data. Simulated PAT performance covered a range of flow rates, from one to three times that of pump best efficiency point (BEP), for blade angles of 0 and ±4°. The PAT BEP was located at a flow rate of 1.4 times that of pump BEP and a head of 1.6 times. For the specific site this corresponded to a shaft power of 32 kW and a flow rate of 770 lps. The PAT was found to have an extended range of good efficiency, of more than 60%, for up to 3 times the pump BEP flow rate.

Javed and Mohammad [35] carried out investigations on an Axial Flow Pump as turbine. The objective of the study was to explore cheap alternate sources of energy production in remote locations of Pakistan. The site conditions for micro-hydro power station usually find axial flow pumps to be more appropriate compared cross flow and Pelton turbines. A commercially available axial flow pump was selected and test rig was designed and constructed in order to determine the performance characteristics of using the pump as a turbine. The test bed has a provision of simulating various head and flow rate conditions and dynamometer to measure the power output in order to determine the performance of the turbine. The simulated head and flow rates were varied for various typical conditions. Some minor modifications in the basic pump unit were made to accomplish these tests.

Mario and Silvio [36] presented a simple method based on a one-dimensional numerical code for deriving the turbine efficiency of commercially available centrifugal pumps. The code estimated a sizing of the component using information such as impeller diameter, specific speed, head and flow rate at pump BEP, machine overall dimension which are provided in manufacturer catalogues, to deduce geometrical parameters of the machine, calculating the losses and thus determining PAT performances. The method was validated by a comparison of the predicted characteristic curves with some experimental measurements available on PATs working in a range of specific speed.

Anurag Shukla [54] conducted CFD analysis of different components of pump to predict the Best Efficiency Point of pump in turbine mode of operation.

1.7 OBJECTIVE OF DISSERTATION

From the literature review it is found that lot of work has been carried out in the area of Pump as Turbine especially for analyzing the performance of pumps in turbine mode. Based on these analyses it was concluded that the operating point of a pump in turbine mode are shifted. Accordingly many researchers have suggested the methods for selection of pumps in turbine mode. It is also found from literature survey that very little work is available on axial flow pump of lower capacity.

These axial flow pumps are used for low head and high discharge. Further it has been found that in the Himalayan region a number of water mill sites exist with a head range of 3-5 m and a capacity up to 5 kW. Potential of these sites can be exploited efficiently by installing efficient equipment.

Keeping this in view, the present study has been carried out with the following objectives

i. Selection ,design and fabrication of an axial flow pump to be used in the micro hydro range

ii. To carry out Computational Fluid Dynamic analysis using the commercially available FLUENT package to simulate the flow in turbine mode of operation of the axial flow pump.

CHAPTER 2: CONCEPT OF PUMP AS TURBINE

2.1 INTRODUCTION

A liquid pump is very similar to hydraulic turbine except that it operates in reverse mode. Mechanical work usually supplied by an electric motor or any other prime mover is used to add energy to the fluid being handled. The mechanism for transferring the energy is usually increases the velocity of fluid as it passes through the impeller flow passages. After passing through the impeller, the fluid is moving at a high velocity so that the energy is in the form of kinetic energy. A diffuser is use to convert a portion of kinetic energy to pressure head by increasing the flow passage area.

Pumps are designed as fluid movers .The fluid enters at suction side of pump at low pressure and get energized by the shaft energy of impeller which is rotated by some external means and leaves the water at discharge side of the pump at high pressure. While in case of pump-as-turbine the pump operates in reverse mode, water enters in the pump at very high pressure and moves through the impeller blades and releases its pressure and kinetic energy to the impeller shaft as mechanical energy and fluid comes out from the eye of pump at low pressure. In the reverse operation of pump it may be less efficient because the direction of flow is reverse and hydraulic and frictional losses increase sharply. The main difference in the pump and turbine design is that, conventional turbines are employed with one or two flow control mechanism to increase its part load efficiency but the standard pumps are not having any flow control mechanism to increase its part load efficiency.

Centrifugal pumps when operated in reverse are hydraulically similar to the Francis turbine without having flow control mechanism. But the performances in both modes are not same. The head and discharge in case of pump is less than in the turbine mode .This is due to the involvement of hydraulic losses of the machine.

The behavior of real fluid flow including friction and turbulence results for the PAT. But the theory of ideal fluid would predict the same performance in both the modes. Figure 2.1 and Figure 2.2 shows the concept of pump in pumping mode and pump in turbine mode respectively.

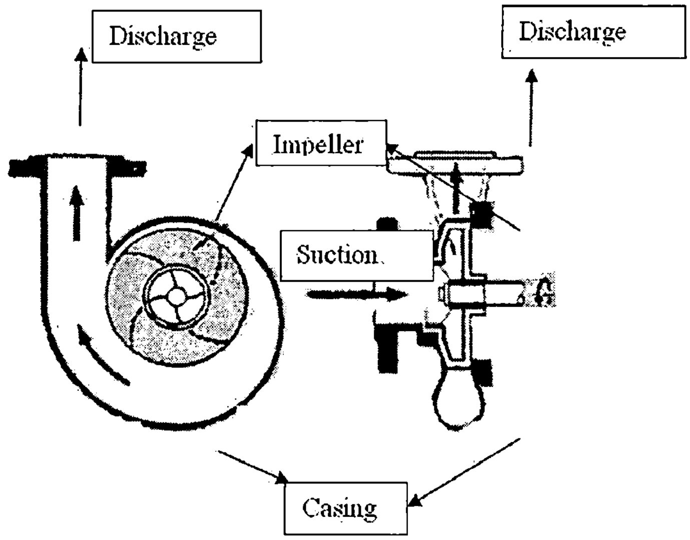

Figure 2.1: Radial Flow pump as fluid mover[16]

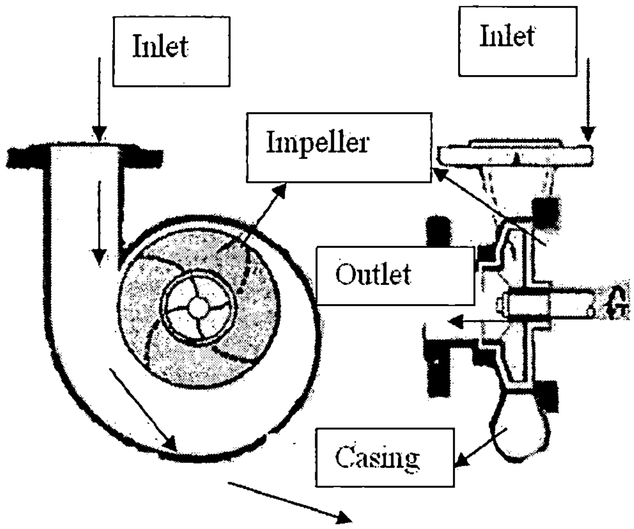

Figure 2.2: Radial flow pump as hydraulic turbine[16]

A hydraulic turbine is used to extract energy (usually in form of pressure head) from the fluid. The pressure head of the fluid is partially converted to kinetic energy by allowing the fluid to flow through a scroll case, which some time have stay vanes and moveable vanes to direct and control the flow, and which act as a nozzle to convert some of the pressure energy into velocity, usually swirling about the unit axis. The turbine runner extracts kinetic energy from the fluid so that the reaction produces torque on the shaft.

The velocity diagram for the flow at the entrance and exit of a pump runner are essentially identical to those for when the unit is acting as a turbine, except the sense of each vector is reversed which is shown in Figure 2.1 and Figure.2.2.

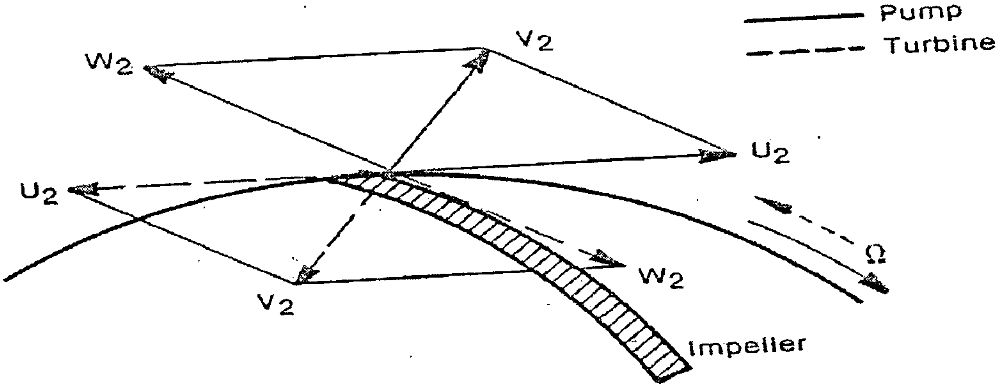

Figure 2.3:Pump inlet velocity vector

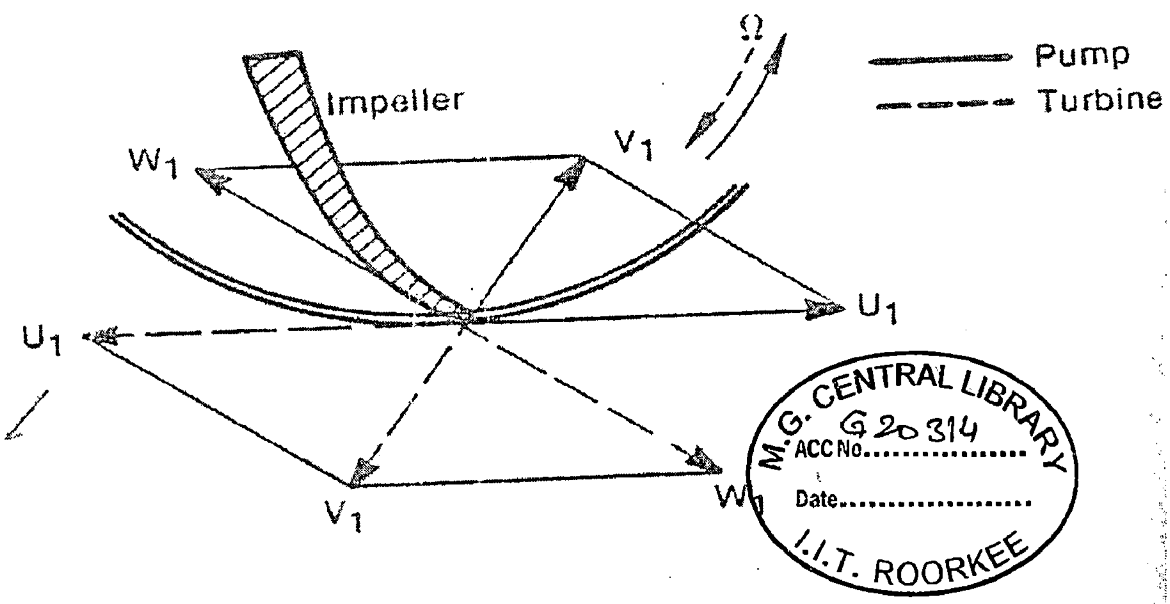

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Figure 2.4:Pump outlet velocity vector

2.2 SCOPE OF PUMP AS TURBINE

Previously there was major scope of electricity production from large hydro power plants where conventional turbines were used. But now there has been emphasis on electricity generation from small, mini and micro hydro power plants with a view to develop non renewable and non conventional energy resources. Since these micro hydro power plants are in remotely located areas, where communities rarely have access to grid connected electricity and have little prospects of obtaining it, use of pump in reverse mode has been advocated for these micro hydro power plants with an eye on bringing down the costs.

A pump can be used as turbine for these micro or mini hydropower plants because it was found that its use is more cost effective with respect to conventional turbines. Pumps are also found much suitable in remote areas in terms of its readymade availability, spare parts availability and its easy operation known by the local people.

Approach towards the use of PAT is helping to lower down the capital cost of micro hydro schemes, which is a major constraint towards its wide spread deployment. PAT are useful for developing micro hydro projects in developing countries where there is no manufacturer of conventional hydro turbine, since they can be purchased in virtually all cities and any major towns throughout the world. It is much easier to purchase a pump than to manufacture a turbine.

Standard pump unit when operated in reverse mode have number of advantages over conventional turbine for micro hydropower generation .Pumps are mass produced and have many advantages during its deployment in micro hydro compared with the custom made turbine .The main advantages are as follows:

i. The availability of pumps and their spare parts are far better than that of turbines, especially in developing countries.
ii. Standard pumps are simple and sturdy and do not require highly qualified mechanic for maintenance. This makes the use of PAT more appropriate in developing countries than conventional turbine.
iii. Custom made turbines are more expensive than standard pumps.
iv. The investment cost of PAT is less than of a comparable turbine(less then 50kW). This might be an important issue for project with limited budget and loan possibilities.
v. Absence of flow control mechanism felt as a drawback, is at the same time an advantage since the pump construction is simple.
vi. Due to the wide application, standard pumps are readily available in the market.
vii. No special skill is required for installation.

The following are the constraints of using PAT

i. There is no hydraulic control device, so control valve must be incorporated in the penstock to start and stop of PAT.

ii. The efficiency of PAT is inferior to the sophisticated turbines of medium to high output range.

iii. There is no flow control, so the fluctuation in the flow results in poor part load efficiency.

iv. Limited choice of generator available for a particular PAT.

2.3 HYDRAULIC PERFORMANCE OF PUMP AS TURBINE

2.3.1 Axial Flow Pump Theory

The axial flow or propeller pump is the converse of axial flow turbine and is very similar to it an appearance. The impeller consists of a central boss with a number of blades mounted on it. The impeller rotates within a cylindrical casing with fine clearance between the blade tips and the casing walls. Fluid particles, in course of their flow through the pump, do not change their radial locations. The inlet guide vanes are provided to properly direct the fluid to the rotor. The outlet guide vanes are provided to eliminate the whirling component of velocity at discharge. The usual number of impeller blades lies between 2 and 8, with a hub diameter to impeller diameter ratio of 0.3 to 0.6.The Figure 2.5 shows an axial flow pump. The flow is the same at inlet and outlet. An axial flow pumps develops low head but have high capacity. the maximum head for such pump is of the order of 20m.The section through the blade at X-X (Figure 2.5) is shown with inlet and outlet velocity triangles in Figure 2.6.

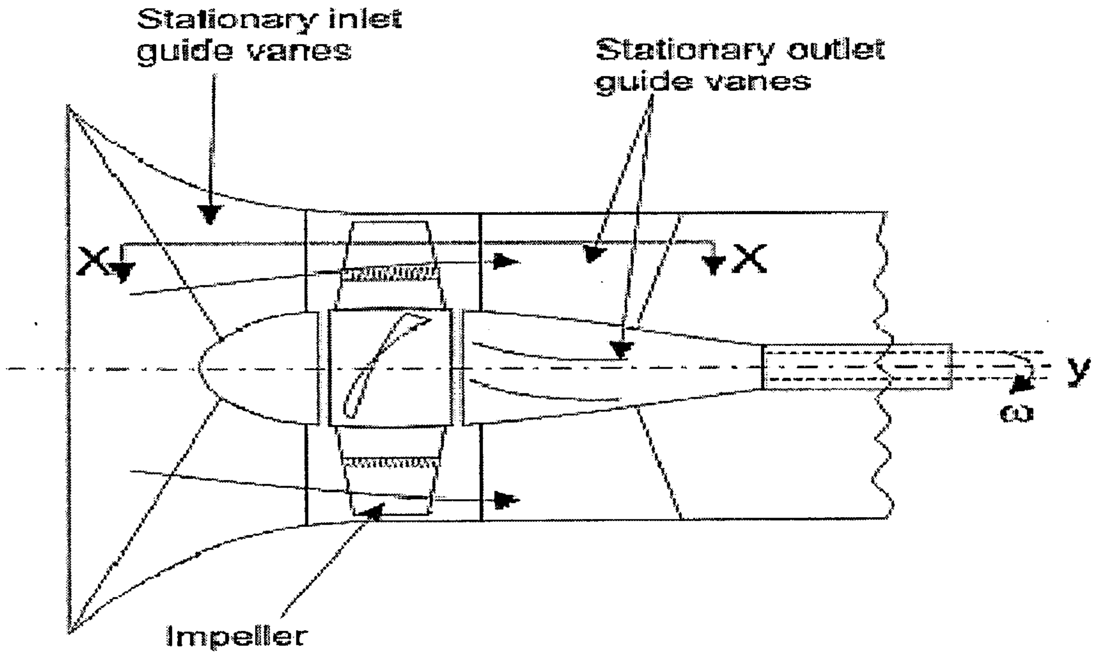


Figure 2.5:Axial Flow Pump[53]

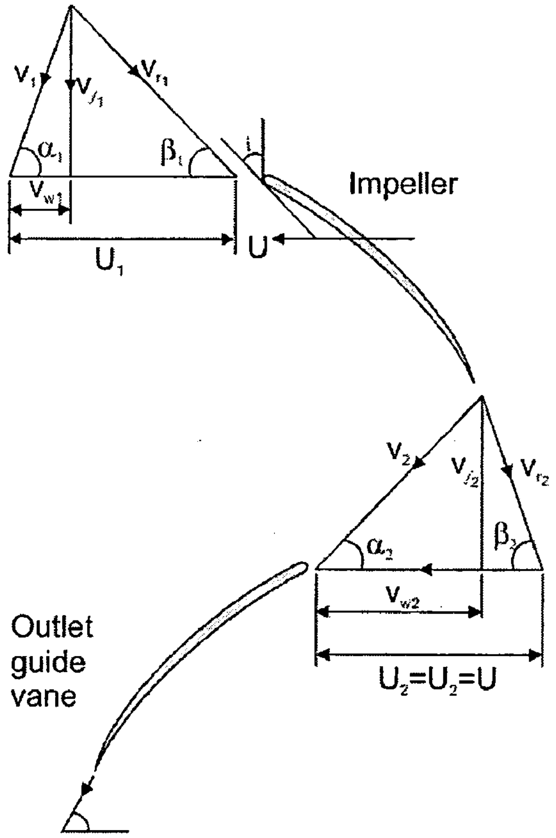


Figure 2.6 Velocity Triangle of Axial Flow Pump[53]

2.3.2 Analysis

The blade of the axial flow pump has an aerofoil section. The fluid does not impinge tangentially to the blade at inlet; rather the blade is inclined at an angle of incidence (i) to the relative velocity at the inlet V_1. If we consider the conditions at a mean radius r_m then

$$u_2 = u_1 = u = wr_m \qquad (2.1)$$

where w is the angular velocity of the impeller.

Work done on the fluid per unit weight = $u(V_{w2} - V_{w1})/g$ (2.2)

For maximum energy transfer, $V_{w1} = 0$, i.e $\alpha_1 = 90^0$ .Again , from the outlet velocity triangle,

$$V_{w2} = u - V_{f2} \cot \beta_2 \tag{2.3}$$

Assuming a constant flow from inlet to outlet

$$V_{f1} = V_{f2} = V_f \tag{2.4}$$

Then, we can write

Maximum energy transfer to the fluid per unit weight

$$= u(u - V_f \cot \beta_2) / g \tag{2.5}$$

For constant energy transfer over the entire span of the blade from hub to tip , the right hand side of above equation has to be same for all values of r. It is obvious that u^2 increases with radius r, therefore an equal increase in $uV_f \cot \beta_2$ must take place, and since V_f is constant then $\cot \beta_2$ must increase. Therefore, the blade must be twisted as the radius changes.

2.3.3 Efficiency

All the head in the pump is generated by the impeller. The rest of the part contribute nothing to the head but incur inevitable loss-hydraulic, mechanical and leakage. All losses of head which take place between the point where the suction and discharge pressure are measured constitute hydraulic loss. These includes skin friction loss along the liquid path from the suction to discharge, losses due to sudden change in area or direction of flow, and all the losses due to eddies. Hydraulic efficiency is defined as the ratio of the available total dynamic head to the input head.

$$e_h = \frac{H}{H_i} = \frac{H_i - hydraulicloss}{H_i} \tag{2.6}$$

Besides the losses of head there is loss of capacity in each pump known as leakage loss. The capacity available at the pump discharge is smaller than that passed through the impeller by the amount of leakage. The ratio of two is called volumetric efficiency which is represented as:

$$e_v = \frac{Q}{Q_i} = \frac{Q}{Q + Q_L} \tag{2.7}$$

Q_L = Amount of leakage

Mechanical losses include the loss of power in bearing and stuffing box and disk friction. The mechanical efficiency is the power actually absorbed by the impeller and converted into head and power applied to the pump shaft and given as;

$$e_m = \frac{Breakhorsepower - Mechanicallosses}{Breakhorsepower} \tag{2.8}$$

2.4 HYDRAULIC LOSSES

These are least known of all the losses in pumps and it can be said that hydraulic losses are caused by:

I. Skin friction.

II. Eddy and separation losses due to the change in the direction and magnitude of the velocity of flow. This includes the shock loss and diffusion loss.

2.4.1 Friction and diffusion losses

Actual measurement of length of fluid passage through the Impeller and the hydraulic radius may present difficulty in many cases. The selection of suitable friction coefficient is a problem in itself. For these reasons several investigators combine all the frictional losses in one term and expressed as

$$h_f = k_1' \frac{v_1^2}{2g} = k_1 Q^2 \tag{2.9}$$

Where K_1 is a constant for a given pump and includes all lengths, areas, and area ratios, and friction coefficients. Thus K_1 covers all the unknown factors and error also. Similarly an expression can be setup for the diffusion loss in the impeller channel or discharge nozzle as stated by:

$$h_d = f_2 \frac{v_2^2}{2g} \quad (2.10)$$

Again selection of coefficient f_2 for the impeller channel presents difficulty. Therefore for simplicity it is customary to express all diffusion losses by:

$$h_d = K_2' \frac{v_2^2}{2g} = K_2 Q^2 \quad (2.11)$$

Where K_2 is the constant for a given pump. Since the losses expressed by both the equations vary as the square of the capacity they can be combined into one equation:

$$h_{fd} = h_f + h_d = K_3 Q^2 \quad (2.12)$$

Hydraulic losses are shown in Figure 2.7, which shows a square parabola with its axis on the axis of head.

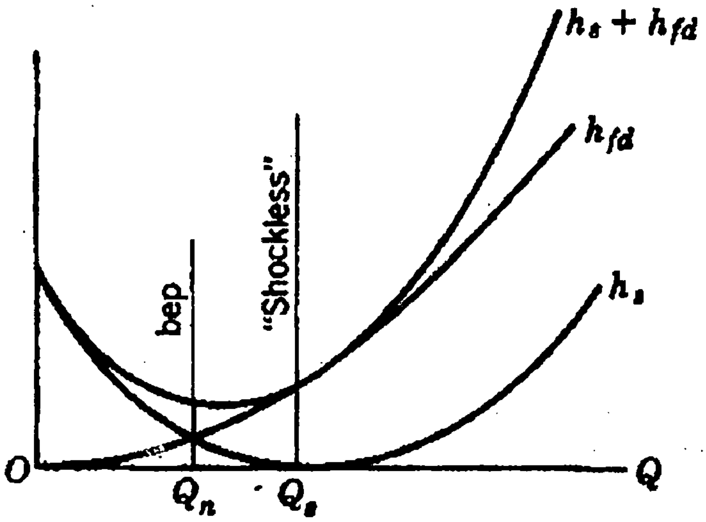


Figure 2.7 Hydraulic Loss

2.4.2 Eddy and Separation Losses

Losses at the impeller entrance and exit are usually called shock losses. Liquid flow in a pump tends to avoid shock by acquiring pre-rotation at the impeller inlet and by establishing a velocity gradient in the volute casing at the impeller discharge, thus cushioning the shock. The nature of the hydraulic loss at the impeller entrance, when liquid approaches at a high angle of attack, is that caused by a sudden expansion or diffusion after separation. At the impeller discharge the loss is mostly caused by a high rate of shear due to low average velocity in the volute and high velocity at the impeller discharge. At the best efficiency point, the average volute velocity is considerably lower than the tangential component of the absolute velocity at the impeller discharge. Figure 2.8 shows the schematic of shock loss occurrence

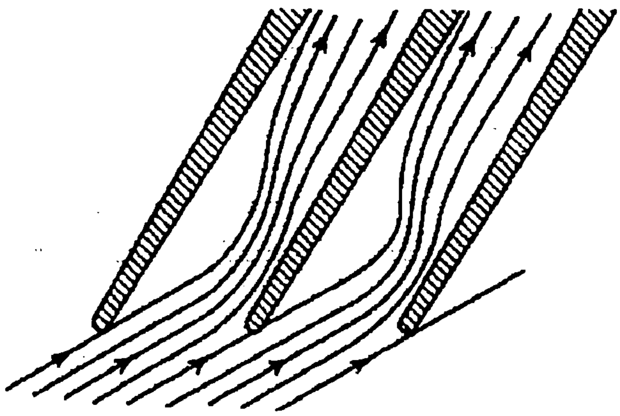


Figure 2.8 Shock Loss

Assume at a capacity Q_s (shock less flow) the direction of flow agrees with the vane angles at both entrance and discharge, thus incurring no additional losses at these points, then at capacity above and below Q_s there will be a sudden change in the direction and magnitude of velocity of flow. Shock loss at the entrance can be expressed as;

$$h_{s1} = K_4 \frac{\Delta c_{u1}^2}{2g} \qquad (2.13)$$

For the exit of impeller

$$h_{s2} = K_5 \frac{\Delta c_{u2}^2}{2g} \qquad (2.14)$$

Combine both above equation in one expression

$$h_s = K_6(Q - Q_s)^2 \qquad (2.15)$$

2.4.3 Leakage Losses

Leakage loss is a loss of capacity through the running clearance between the rotating element and the stationary casing parts. Leakage can take place in following places according to the type of pump.

- Disk Friction Losses Between the casing and the impeller at the impeller eye.
- Through the stuffing box.
- Through axial thrust balancing device
- Through bleed-off bushing when used to reduce the pressure on the stuffing box.

The capacity through impeller is greater than the measured capacity of pump by the amount of leakage, and the ratio of measured capacity to the impeller capacity is the volumetric efficiency.

$$e_v = \frac{Q}{Q_i} = \frac{Q}{Q + Q_L} \qquad (2.16)$$

2.4.4 Disk Friction Losses

Considerable test data are available on the disk friction loss for cold water and several formulas are in use. All of them arise from one fundamental equation given below;

$$(hp)_d = Kn^3D^5 \qquad (2.17)$$

2.5 TOTAL HEAD CAPACITY CURVE

The head capacity curve of an idealized pump is a straight line. For a given discharge vane angle, a single line will represent the characteristic of pumps. Hydraulic losses for the selected proportions of essential passage will determine the head capacity curve of actual pump. For a constant speed, the head capacity curve can be obtained by subtracting losses from the input head of an idealized pump. The best efficiency point will always occur at a capacity lower than the shock less capacity Q_s, because the sum of the friction and shock losses determines the location of peak efficiency. For a given capacity the actual head may be expressed by;

$$H = H_i - K_3Q^2 - K_6(Q - Q_s)^2 \qquad (2.18)$$

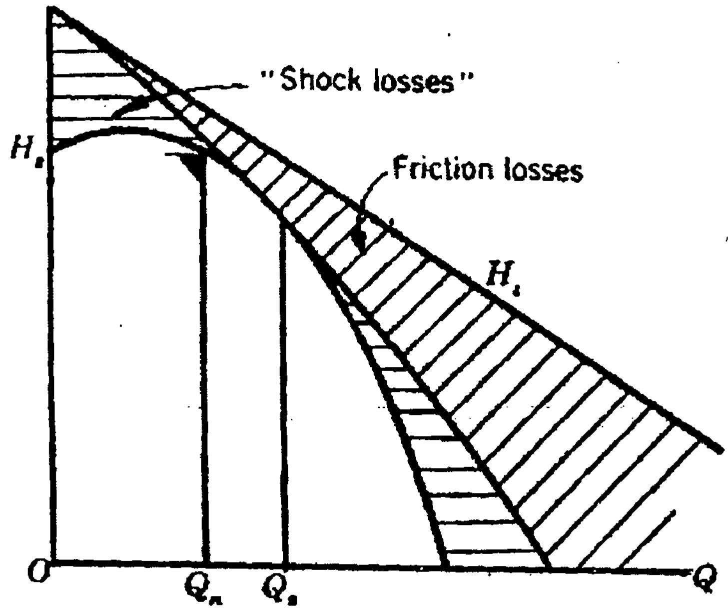


Figure 2.9 Flow Head curve by subtracting friction and shock loss from input head

2.6 TYPICAL PERFORMANCE CHARACTERISTICS OF PUMP AS TURBINE

Figure 2.10 shows typical performance characteristics of a centrifugal pump with the superimposed performance of the same pump operating as turbine. The head applied at the best efficiency point of the turbine is considerably higher than at pump b.e.p, and the turbine head continues to increase past the b.e.p. In many instances, the standard pressure rating of the pump casing is not suitable for these increased heads, and special materials may be required o strengthen the casing. Many reverse operating pumps are applied in pipelines as pressure-reducing devices. The downstream pressure on the exhaust side of these turbines may be very high, further increasing the pressure on the turbine casing.

Also it can be seen that the flow is considerably higher for the turbine than for comparable pump. This increased flow, combined with increased head, results in considerably increased power output compared to the power input required by the pump. Most pump shafts are not designed to accommodate the torque associated with this increased power and, as a result, must be redesigned with special materials or larger shafts.

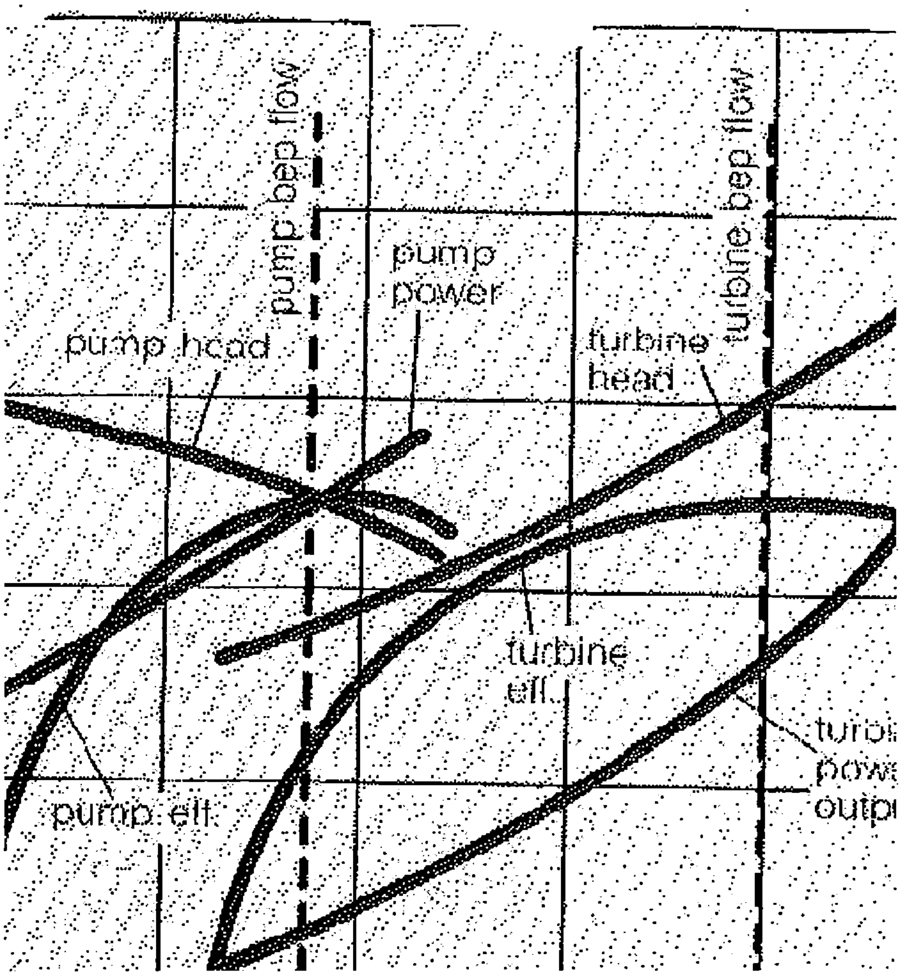


Figure 2.10 Performance of a machine as a pump and turbine[51]

Run away speed is also an important design consideration when applying a pump as a turbine. In Figure 2.11 it can be seen that the speed at runaway may be as much as twice the speed at the b.e.p of the turbine, which is normally the same as the operating speed of a pump. In some cases the bearings used in the pump may not be suitable for these high speeds and may have to be redesigned.

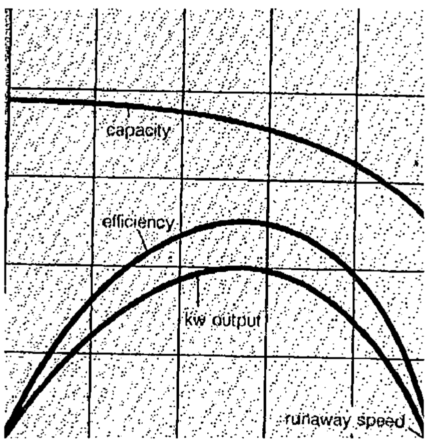


Figure 2.11 Runaway speed characteristics.[51]

These considerations make it very important that pumps selected to operate as turbines are given a careful design review by the manufactures to ensure that they are suitable for the application. Figure 2.12 represents typical efficiency characteristics of typical hydraulic turbines along with the pump used as a turbine.

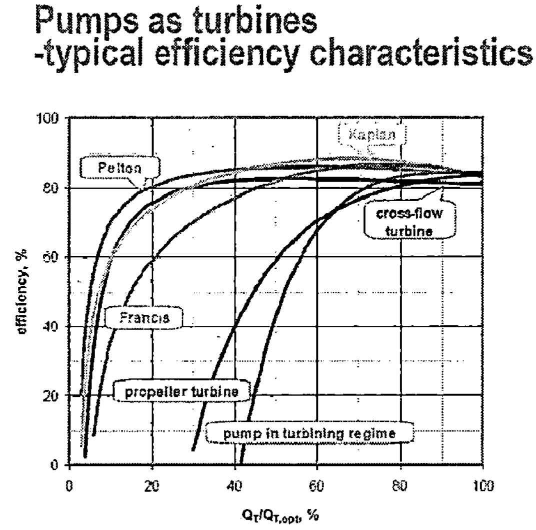


Figure 2.12 Typical Efficiency Characteristics [3]

Figure 2.13 shows a useful chart which can be used for the selection of a particular turbine for micro hydro application.

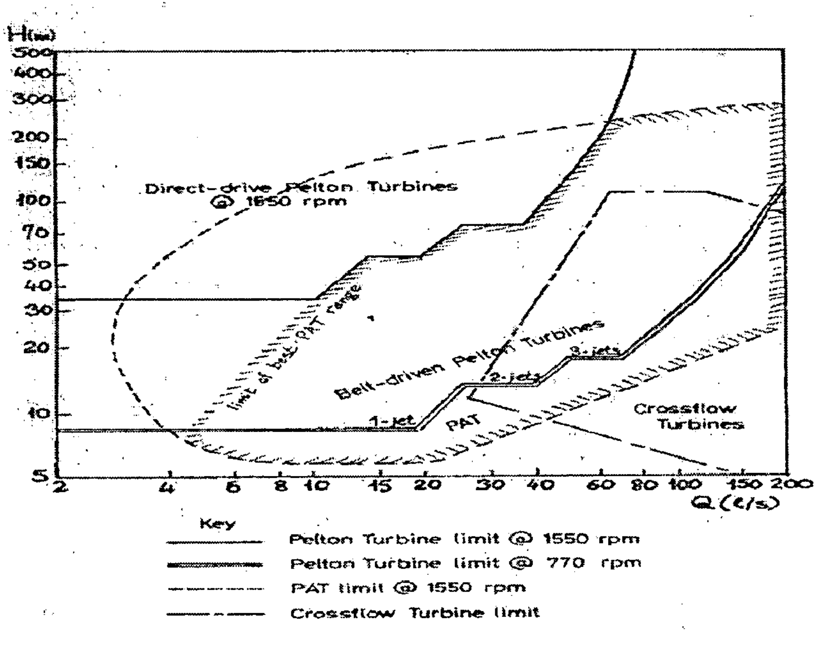


Figure 2.13 Selection Chart for PAT application[20]

CHAPTER 3: DESIGN DEVELOPMENT AND FABRICATION OF AXIAL FLOW PUMP

3.1 GENERAL

As was evident from the literature survey that, though the concept of PAT has been explored in micro hydro sector for power generation, the pumps that have been used were only centrifugal pump and mixed flow pump. So it was felt the need to investigate the performance of axial flow pump in reverse mode. This factor gain importance from the fact that there are many micro hydro power sites that are of high discharge and low head and these sites are not developed because of the lack of hydro turbine that suit this site conditions. Some of these sites have been fitted with Cross Flow turbine but the efficiency of these cross flow turbine has been low. It is in this context that use of axial flow pump in reverse operation gain significance. Axial flow pump cater low head high discharge application. So in the reverse mode it has to suit the site conditions which are now equipped with cross flow turbine. So an attempt has been made to design develop fabricate and investigate the performance of axial flow in reverse mode. This chapter describe in detail the steps involved in the design development and fabrication of the axial flow pump used for investigation purpose.

The first constraint that was encountered during the design stage was the limitation of the service pump available in the testing laboratory to provide flow to the PAT unit. The two service pump could together provide only 300 lps at a head of 10 m. As such we would we able to test only a low capacity pump in the laboratory. Since low capacity axial flow pump are not widely used in the industry the pump manufactures do not stock these pumps in their ware house. This called for design of a custom made axial flow pump to test in the laboratory. As the first step in design a computer program was written to arrive at the various combinations of head and discharge in the axial flow pump range. From the available combinations a particular combination of head and discharge was selected. The next step in design stage was to work out the preliminary dimensions of the axial flow pump to meet the above operating conditions. Once these dimensions were found out the details were provided to the pump manufacturer M/S“GITA FLOPUMPS SAHARANPUR “who then provided with the detailed dimensions of the pump and the pump was fabricated in their factory.

3.2 SIZING OF PUMP

The first step in the design stage was to work out the various combinations of head and discharge which fall in the axial flow pump range. The criteria that was used here is specific speed of the pump. The specific speed of axial flow pump falls between 150 and 400 metric units. Based on the above condition a computer program was written to simulate the head and discharge which falls in the axial flow pump range. The discharge was varied from 0.05 cumecs to 0.3 cumecs with an interval of .01 cumecs. The head was varied from 2 m to 10 m. Based on the speed of the unit the program will display all the combination of head and discharge which satisfy the above conditions. Program was made to run at 3000 rpm, 1500 rpm and 1000 rpm. It was then decided that results obtained from 1500 rpm would be used further because then the direct coupling of turbine unit with generator is possible which will avoid the use of speed increaser/decreaser though other options can also be tried out. Results obtained from the program when it was made to run at 1500 rpm are given in Table 3.1. The table shows discharge, head, specific speed, and power input required for each case.

Table 3.1: Operating conditions

Sl.No	Head range(metre)	Discharge range(cumecs)	Power I/P range (kW)	Specific speed (metric units)
1	0.50-2.00	0.01-0.04	0.05-0.80	252-178
2	2.50-4.00	0.04-0.09	1.00-3.50	150-159
3	1.50-4.50	0.10-0.11	1.50-5.00	349-157
4	1.50-6.50	0.12-0.17	1.75-10.0	383-151
5	2.00-8.50	0.18-0.25	3.50-20.0	378-150
6	2.50-9.50	0.26-0.30	6.00-30	385-151

From the above table the combination of a discharge of 0.105 cumecs and head rise of 4 metre was taken up for further design. The specific speed for this combination is 166 metric units.

3.3 DESIGN OF AXIAL FLOW PUMP

The flow characteristics calculated inside the rotating blades of an axial pump depend on a number of parameters, as well as the measurements taken at each section of the axial surface. The choice of rotational speed, for instance, is interlocked with other parameters, but there are empirical limits as given, for example, by the American Hydraulic Institute Standards [46].

The first authoritative reference on complete hydraulic design of a pump was produced by Stepanoff [11]. His design method is based on velocity and geometry analysis charts plotted against specific speed. These charts assume consistency of design so that only pumps of similar type, construction, number of impeller vanes, etc., are plotted on one chart. To predict the hydraulic performance of a mixed flow pump of the bowl type using graphical and empirical methods [47] it was recognized that there is a mismatch between the impeller and volute casing.

Pump dimensions, connected with frequency of rotation of runner, give the general picture inside the blade passage. Empirical data suggest the desirable number of blades. If there are few blades, and the gap–chord ratio at the hub section remains above 1, isolated aerofoil data may be used but, if blades are closer, corrections are needed [48].

The original energy derived from the motor is transferred nearly in total into useful work. There are various methods which are used to measure the brake horse power. Transfer of kinetic energy into pressure occurs during the flow of liquid through the propeller, and in the remaining parts of the pump. In some instances, changes take place to part of the energy in the form of heat, which is then posted as losses within the pump, e.g. the friction caused by the impeller revolving in the water, bearing losses, stuffing box friction and leakage flow.

Calculation of rotating blades requires several accurate tests in the sequence of approximation. Cavitation may exist during these tests when operating at high capacities.

The calculation of rotating blades would also depend on both the stream flow and the blade base. In general, the momentum in both of these components represents the characteristics for the whole runner and allows the possibility to examine the estimated design. Many engineers have attempted to apply a universal model to a cascade of blades, but without great success when applied to pump design. It was found that the hub–tip diameter ratios, the restrictions to blade number and adjustments to vane shape from hub to tip (e.g. for mechanical strength) all tend to increase the empirical input into the design method.

Turton [50] suggested that the methods employed for mixed and axial flow pumps tend to fall into two categories: those heavily reliant on empirical data and those based on highly advanced fluid flow analysis, which demand very powerful mathematical and computing capabilities. As long as the ratio of inner to outer diameters of the propeller is given, this will lead to a definite examination for possible values of variables.

To design a new impeller for which no model is available designers use "design factors" established experimentally from successful designs that give direct relationship between the impeller total head and capacity at the design point and several elements of Euler's velocity triangles. These are dimensionless velocity ratios, independent of the impeller size and speed, which are correlated on the basis of specific speed for different impeller discharge angles. In addition number of ratios of important linear dimensions, not directly related to velocities, is found helpful in perfecting hydraulic design of impellers. These ratios, too, are entirely experimental and do not lend themselves to theoretical treatment. The degree of perfection of a design is measured by the value of the pump hydraulic efficiency.

For the design problem in question the power equation used is $P = \frac{\rho \times g \times Q \times H}{\eta}$.

Considering the following data

Power (P) = 5 kW.

Density (ρ) = 1000 Kg/m^3.

Gravity (g) = 9.81 m/s^2.

Head (H) = 4 m.

Efficiency (η = 0.8 (assumed)

Therefore discharge (Q) = 0.102 m^3/s

Specific Speed = $N_S = \frac{N\sqrt{Q}}{H^{3/4}}$ = 163 units.

The impeller profile and vane layout is possible if the following elements are known

1. Meridonial velocities at inlet and outlet
2. Impeller outside diameter
3. Impeller vane inlet and outlet angles

The entrance velocity: In order to complete the impeller profile, the meridonial velocity at entrance should also be known. This is given by a ratio

$$K_{m1} = \frac{c_{m1}}{\sqrt{2gH}}$$

This is calculated for the area at the vane entrance tips, again omitting the leakage. The vane thickness can be disregarded as the vane tips are usually tapered and C_{m1} can be assumed to be the velocity just ahead of the vanes. Value of K_{m1} plotted against specific speed is given by Stepanoff [11]

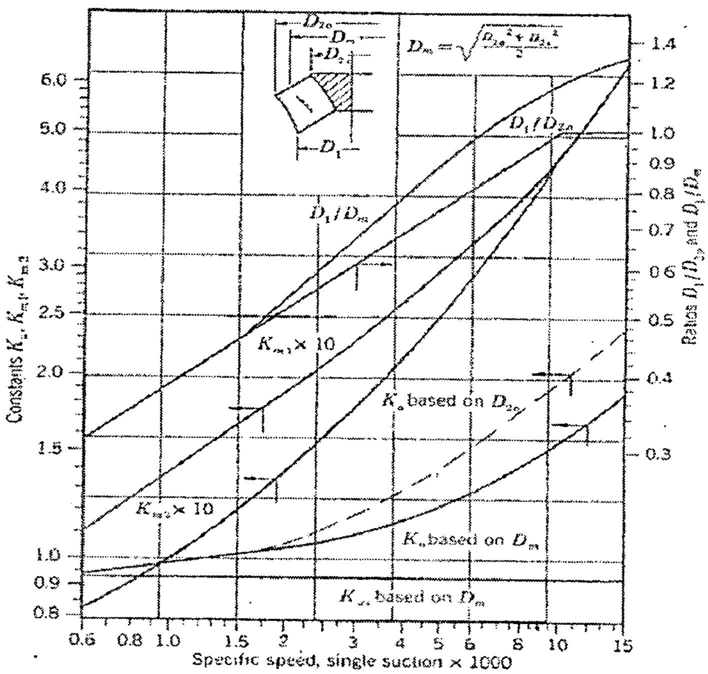


Figure 3.1 Impeller constants[11]

For the design problem in question we get the value of K_{m1} as 0.44 from the above chart. Therefore the meridonial velocity at exit becomes

$$K_{m1} = \frac{c_{m1}}{\sqrt{2gH}}$$

$$C_{m1} = K_{M1}\sqrt{2gH} = 3.9 \text{ m/s}$$

Impeller hub ratio: The ratio impeller hub diameter to the impeller outside diameter is directly connected with specific speed of axial flow pumps. This ratio is determined experimentally. Higher specific speed pump have smaller hubs, which give rise to a greater free area for the flow and a smaller diameter to the average stream line, resulting in greater capacity and a lower head.

For the design problem in question we choose the impeller hub ratio (v) as 0.65 based on the specific speed as is obtained from Figure 3.2. The number of blades is fixed as 4.

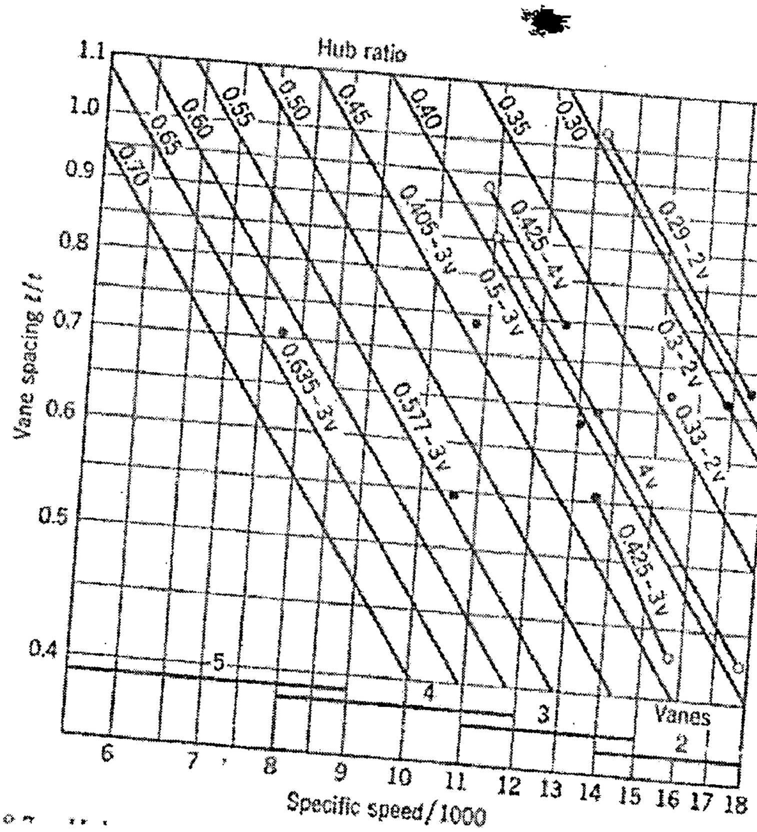


Figure 3.2 Hub ratio vs specific speed[11]

For a discharge of 0.1 cumecs it is found out from Figure 3.3 that

$$R_s = 0.11 \text{ m}$$

The rotational speed n = 1450 r/min and also

$$\omega R_s = 15 m/s$$

$$\phi = 0.2625$$

The real quantity of energy transferred per unit mass, by assuming an internal efficiency of 0.8 is

$$\Delta e = \frac{\Delta e_u}{\eta_i} = \frac{39.24}{0.8} = 49.05 \, J/kg$$

Again from Figure 3.3 it is obtained that

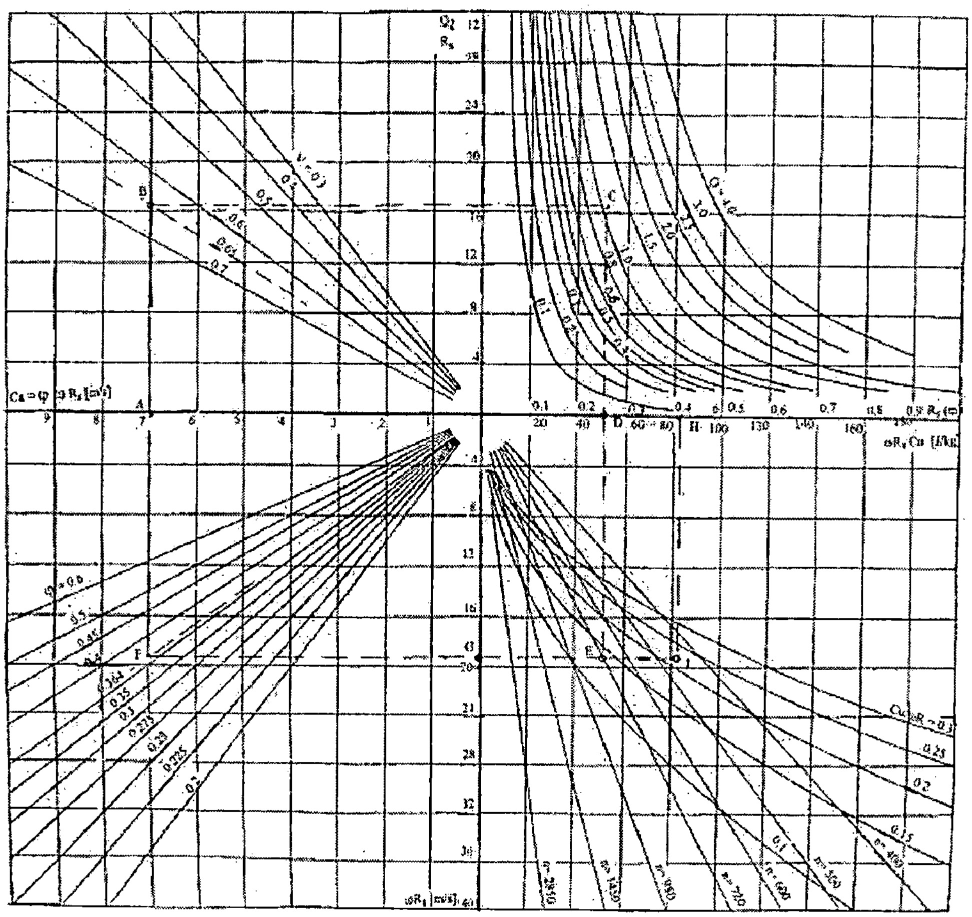


Figure 3.3 Chart used for dimension[52]

$$\frac{C_U}{\omega R} = 0.1$$

Next the value of angle β_{avg} is calculated

$$\tan\beta_{avg} = \frac{\phi}{1-\frac{C_U}{2\omega R}} = \frac{0.2625}{1-\frac{0.1}{2}} = 0..276$$

Therefore β_{avg} = 15.45°

Assuming a solidity of τ = 0.7, the value for coefficient of lift force is found out to be

$$C_y = 0.25$$

The outside diameter of impeller is given by [52]

$$D_z^{\ 2} = \frac{2}{1+\nu^2} D_S^{\ 2} = \frac{2}{1+0.65^2}(0..22^2) = 0.068m$$

D_z=0.26 m

D_w = 0.65 × 0.26 = 0.17 m

For calculating value of β_1 a case which holds true is given by the following equation [52]

$$\cot\beta_1 = \frac{1}{\phi} = \frac{1}{0.2625} = 3.809$$

β_1 = 14.7°

Using the relation $2\cot\beta_{avg} = \cot\beta_1 + \cot\beta_2$

β_2 = 16.28°

3.4 DESIGN DRAWING OF THE AXIAL FLOW PUMP

Based on the above preliminary dimensions of the axial flow pump, a pump manufacturer was approached to fabricate the pump. The following figures of the different parts of the pump will give better understanding of the dimensions of the axial flow pump.

3.4.1 HUB

The hub ot the pump is conical in shape to which blades are attached. Major dimensions of the hub are shown through Figure 3.4, Figure 3.5 and Figure 3.6. As can be seen from

Figure 3.4, the hub is tapered. The diameter reduces from 80 mm to 70 mm .The width of the hub is 80 mm.

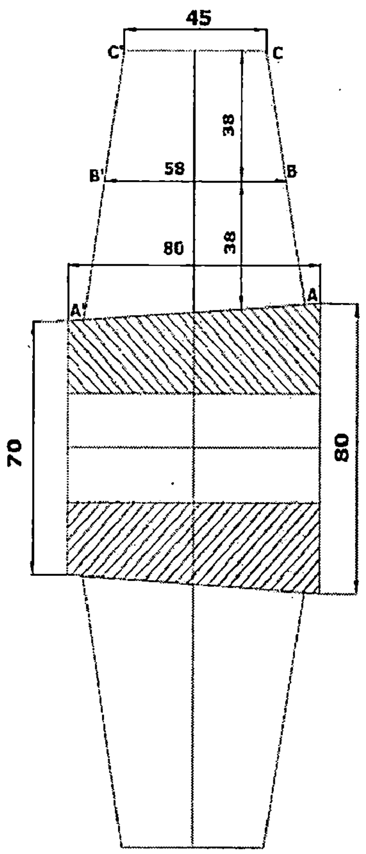


Figure 3.4 Section of HUB

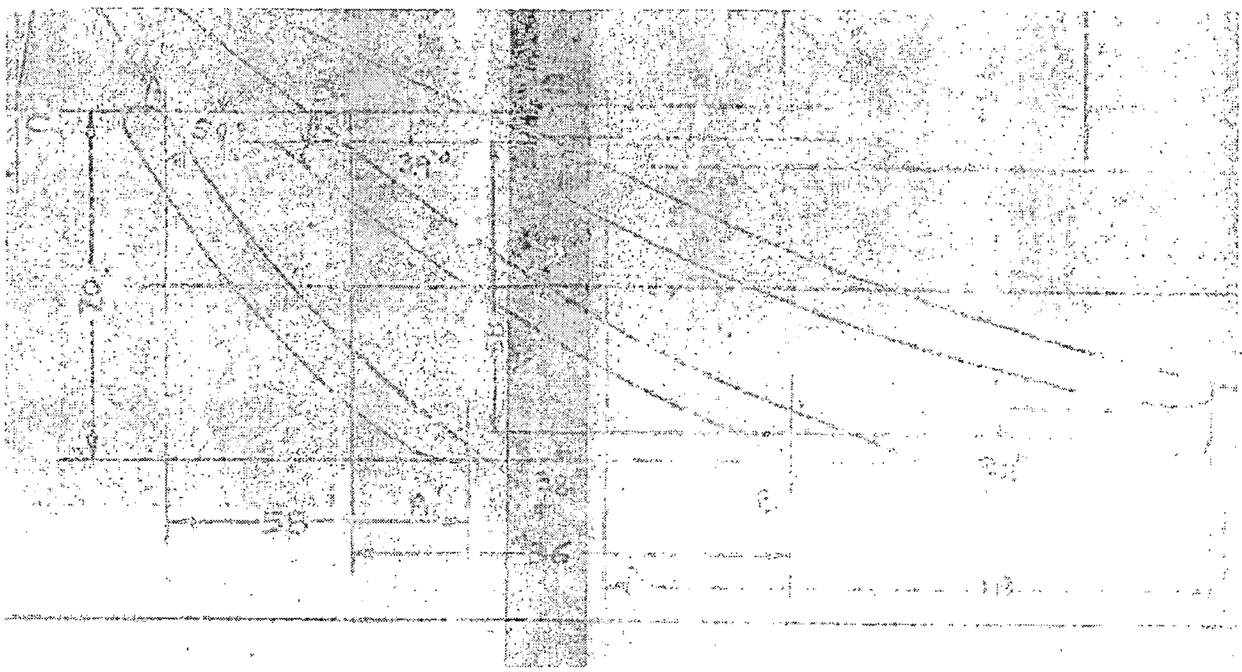


Figure 3.5 Section through ABC

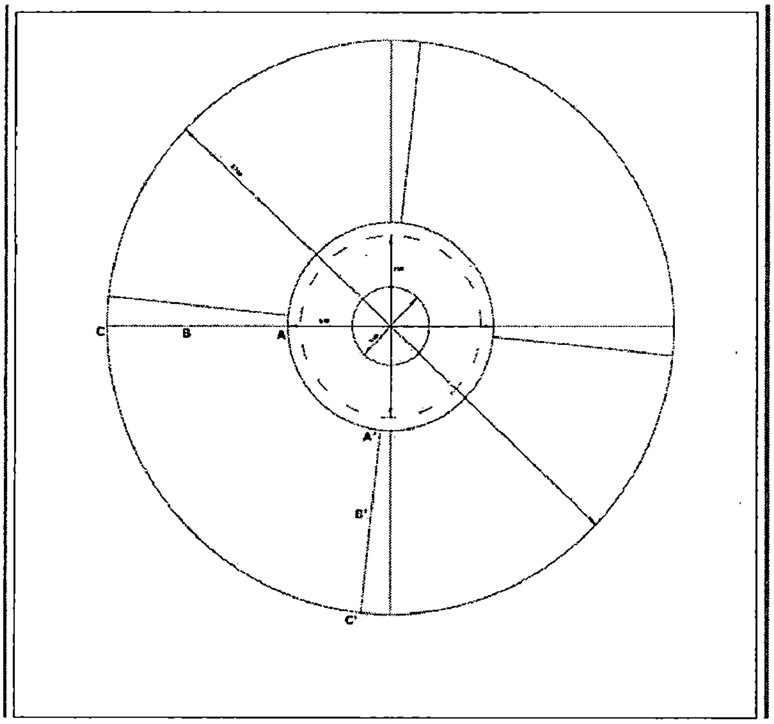


Figure 3.6 Plan view

3.4.2 BLADES

Four blades are attached to the hub as shown in Figure 3.4, Figure 3.5 and Figure 3.6. The thickness of the blade is 9 mm. The cross section view of the blades is given in Figure 3.5

3.4.3 GUIDE VANES

Six guide vanes each of thickness 9 mm are attached to the diffuser casing of axial flow pump.

3.4.4 DIFFUSER CASING

A diffuser is provided enclosing the guide vanes. The thickness of the shell of diffuser casing is 10 mm. The diffuser is conical with a larger diameter of 250 mm and a smaller dimension of 220 mm. Other major dimension of diffuser casing can be found out from Figure 3.7 and Figure 3.8

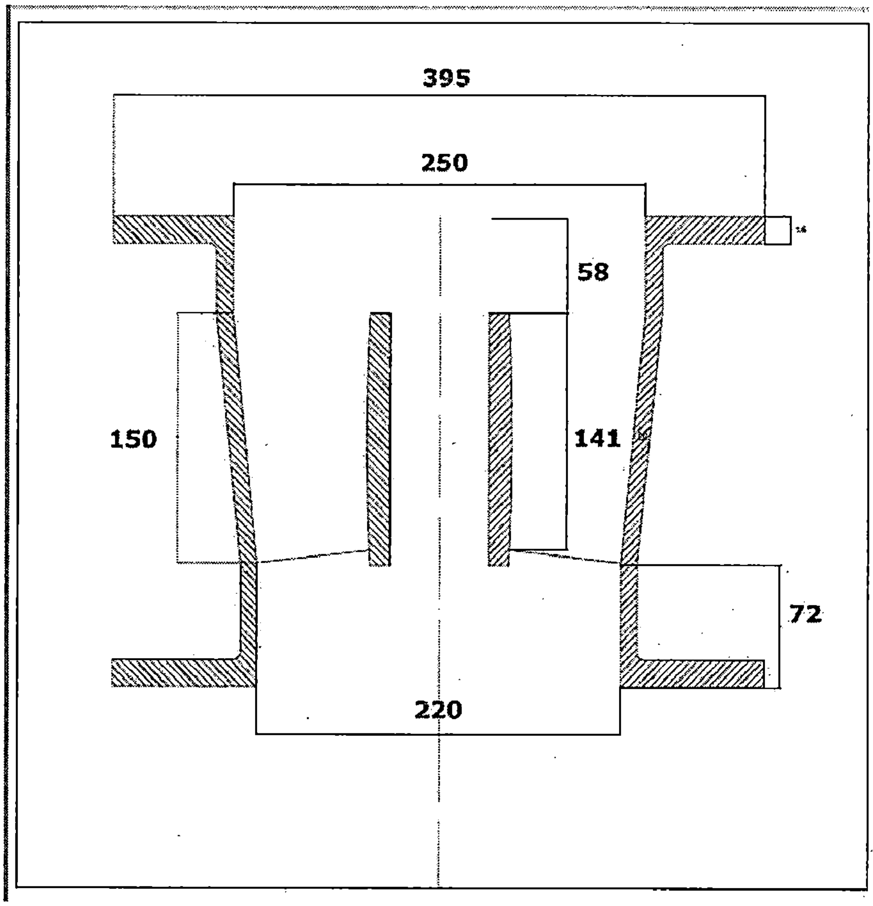


Figure 3.7 Diffuser casing

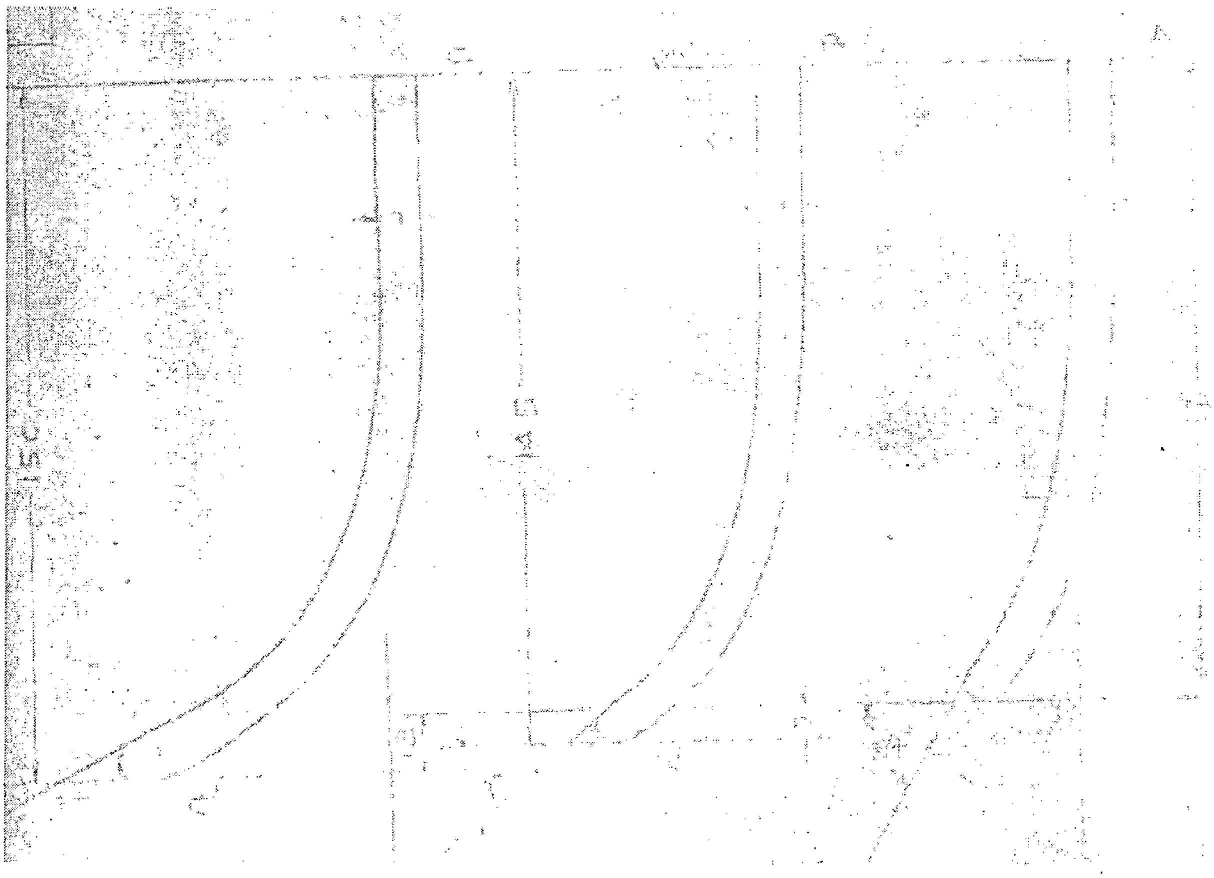


Figure 3.8 Section through guide blades

3.5 FABRICATION OF PUMP

3.5.1 HUB AND BLADES

Based on the above drawings first a pattern was made and then casting was done to make the final product. The pattern of the hub with blades attached is shown in Figure 3.9 and the final cast product is shown in Figure 3.10.

Figure 3.9 Wooden pattern of hub and blade

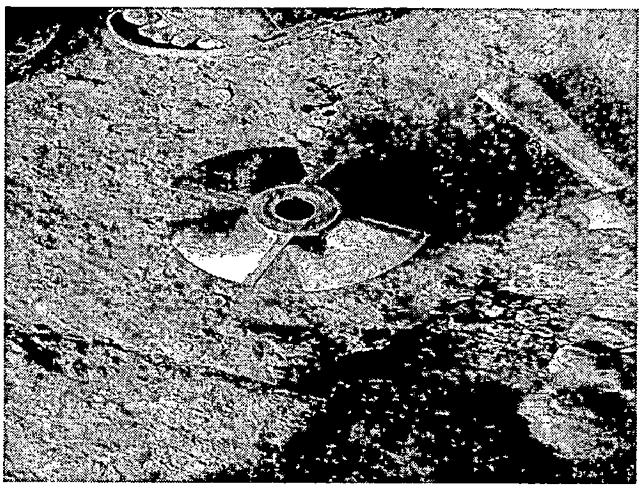


Figure 3.10 Final casted hub and blades

3.5.2 DIFFUSER CASING AND GUIDE BLADES

Pattern of the diffuser casing and the guide blades is shown in Figure 3.11.

Figure 3.11 Pattern of diffuser casing and guide blade

Figure 3.12 shows a worker giving the finishing touches to the casted diffuser casing body with the guide vanes.

Figure 3.12 Finishing touches being provided to diffuser casing

3.5.3 DELIVERY PIPE

Figure 3.13 shows the final machined delivery pipe of axial flow pump.

Figure 3.13 Final machined delivery pipe of axial flow pump

3.5.4 THE FINAL ASSEMBLY

The Figure 3.14 shows the final assembly of the axial flow pump in the shop floor .The Figure 3.14 clearly shows the delivery pipe, the diffuser casing and the suction pipe.

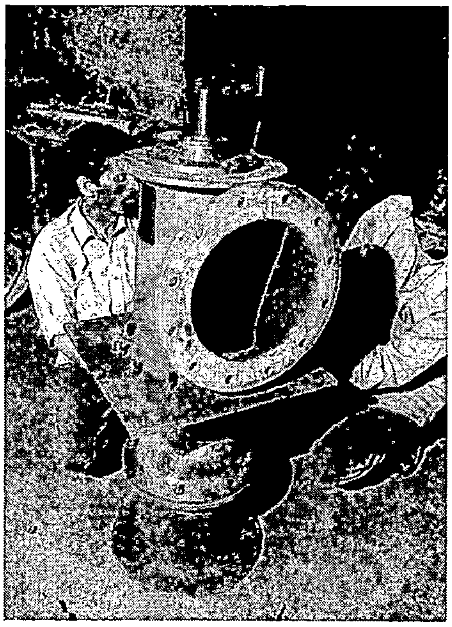


Figure 3.14 Final assembly

3.6 DRAFT TUBE

Since the axial flow pump is to be tested in turbine mode the draft tube is an essential part. A simple conical draft tube using M.S sheet of thickness 3 mm was fabricated in the Hydro Mechanical Laboratory of AHEC to fit to the outlet of the PAT unit. The draft-tube tapered from 10 inches to 12 inches with a draft angle of 2.5 deg. The final fabricated draft tube is shown in Figure 3.15.

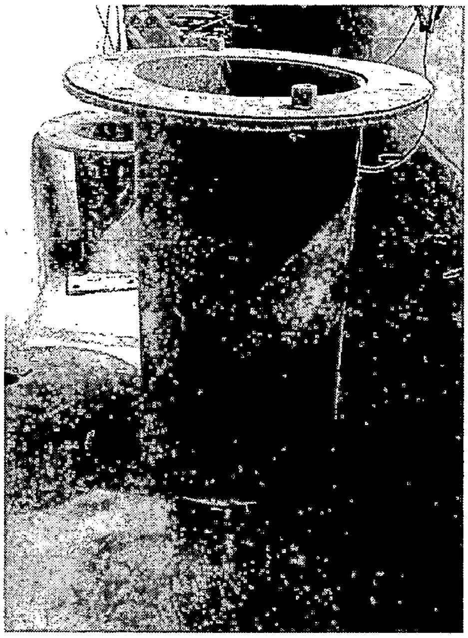


Figure 3.15 Draft tube

CHAPTER 4: CFD ANALYSIS OF PUMP AS TURBINE

4.1 GENERAL

In this chapter the attempt made using computational fluid dynamics (CFD) methodology to investigate the flow field in an axial flow pump as turbine unit is described. Axial flow pumps are widely used and are usually designed for pumping large volumes of water against low pressure head. The basic elements of the pump are impeller and guide vane. In an axial flow pump, mechanical energy is transformed into fluid energy through the interaction of water and impeller. Energy transfer takes place mainly through the tangential acceleration of the fluid due to the impact of the impeller, which can be seen by the fact that the fluid leaves the impeller in a vortex. The effect of a guide vane is to improve the overall pressure rise attainable by the pump. A reduction in kinetic energy must be compensated by a corresponding increase of pressure. The cross section of a blade is in an aerofoil form, and the relative flow pattern is in many ways similar to flow around the wing of an aircraft. The detailed pressure and velocity fields in the rotor are different for various mass flow rates and pressure head. At last the turbine characteristics are obtained; the efficiency is measured on several operating conditions and is compared.

4.2 GEOMETRIC AND MESH MODEL

A geometric model of the axial flow pump is shown in Figure 4.1. The rotor is equipped with 4 blades. The guide vane consists of 5 blades. The mesh model that is composed by the rotor domain and the fixed domain is built on the base of solid model.

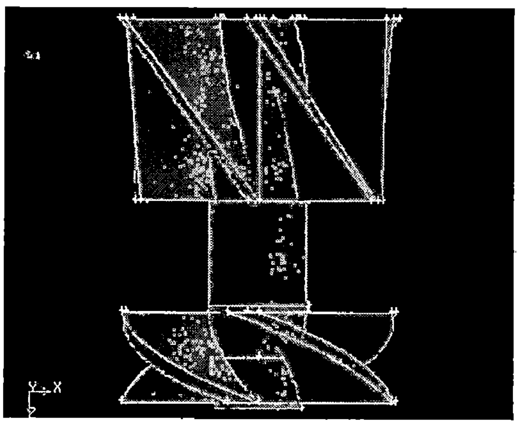


Figure 4.1 3D model of axial flow pump

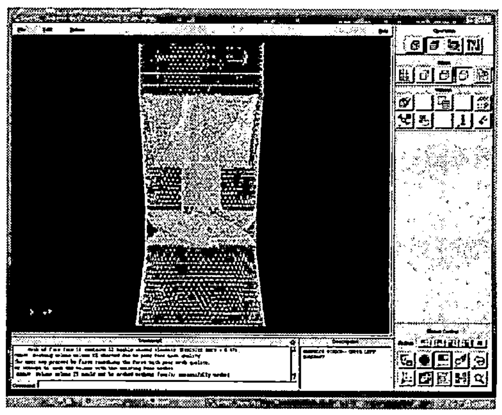


Figure 4.2 Mesh model

The diameter of impeller is 210 mm, the diameter of hub is 90 mm, and the diameter of pump as turbine inlet is 250 mm. The model's origin is the intersection of the axes and the rotor center. The rotor speed is 1500 r/min. The mesh model shown in Figure 2 is built after the blade surfaces are meshed subtly by means of tetrahedral mesh elements scheme. The rotor domain has 35, 467 nodes, and the fixed domain has 51,456 nodes. Then they are merged at last. The rotor's outlet is the exit of the pump as turbine unit.

4.3 GOVERNING EQUATIONS AND TURBLENCE MODEL

It is assumed in the present investigation that the flow can be treated as an incompressible fluid with constant physical properties. Furthermore, the flow is considered steady relative to moving reference system, the physical problem can be described by the Reynolds-averaged Navier-Stokes equations and the mass continuity equations, which can be expressed in general Cartesian tensor form as :

$$\frac{\partial U_j}{\partial x_j}=0$$

$$\rho U_j \frac{\partial U_i}{\partial x_j}=-\frac{\partial P}{\partial x_i}+\frac{\partial}{\partial x_j}\left[\mu\left(\frac{\partial U_i}{\partial x_j}+\frac{\partial U_j}{\partial x_i}\right)\right]-\frac{\partial}{\partial x_j}\left(\rho\, \overline{u_i u_j}\right)$$

The subscripts i and j can take values of 1, 2 and 3 in order to take into account the three dimensions in which the fluid revolves. These averaged equations contain the Reynolds-stress terms $\overline{u_i u_j}$, which characterize the influence of the turbulence on the mean flow field. This additional term must be modeled in order to solve these equations. Hence, the quality of the numerical results will depend on the choice of the turbulence model, which is a set of additional equations.

The standard $k-\varepsilon$ model is the most used formulation in engineering calculations. It provides a linear relationship between the Reynolds stresses and the mean rate of strain by using the eddy-viscosity hypothesis. Hence, the Reynolds stress terms can be expressed as :

$$\overline{u_i u_j}=-\nu_t\left(\frac{\partial U_i}{\partial x_j}+\frac{\partial U_j}{\partial x_i}\right)+\frac{2}{3}k\delta_{ij}$$

where δ_{ij} is unity for $i = j$ and zero otherwise. The turbulent viscosity v_t is directly related to the turbulent kinetic energy k and the dissipation rate ε :

$$v_t = \frac{\mu_t}{\rho} = C\mu \frac{k^2}{\varepsilon}$$

The two extra equations introduced by the $k - \varepsilon$ model take the following form:

$$U_j \frac{\partial \varepsilon}{\partial x_j} = \frac{1}{\rho} \frac{\partial}{\partial x_j} \left[\left(\mu + \frac{\mu_t}{\sigma_\varepsilon} \right) \frac{\partial \varepsilon}{\partial x_j} \right] + C_{\varepsilon 1} \frac{\varepsilon}{k} P_k - C_{\varepsilon 2} \frac{\varepsilon^2}{k}$$

$$U_j \frac{\partial k}{\partial x_j} = \frac{1}{\rho} \frac{\partial}{\partial x_j} \left[\left(\mu + \frac{\mu_t}{\sigma_k} \right) \frac{\partial k}{\partial x_j} \right] + P_k - \varepsilon$$

where the production of turbulence kinetic energy P_k can be written as:

$$P_k = v_t \left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right) \frac{\partial u_i}{\partial x_j}$$

The model constants $C_\mu, C_{\varepsilon 1}, C_{\varepsilon 2}, \sigma_k$ and σ_ε are determined experimentally and are assigned standard values of

$C_\mu = 0.09$, $C_{\varepsilon 1} = 1.44$, $C_{\varepsilon 2} = 1.92$, $\sigma_k = 1.0$, $\sigma_\varepsilon = 1.3$.

However, this type of model is based on the assumption of the isotropic distribution of turbulence production and its application is not recommended for the flow with strong curvatures and recirculation's.

4.4 BOUNDARY CONDITIONS

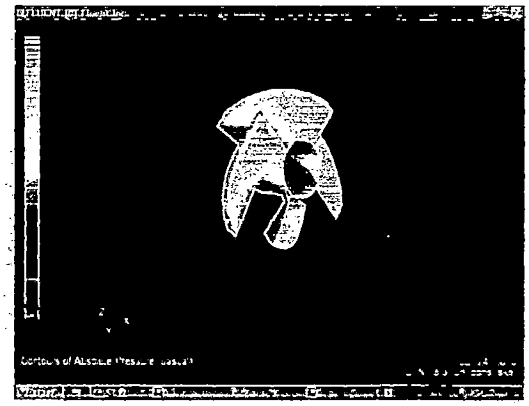
In the present study, numerical simulation of the axial flow PAT unit is aimed. It is a rotor-stator interaction problem and cannot be modeled by a simple coordinate transformation to a rotating reference frame. The multiple reference frames is used. The rotor domain is steady since the moving reference system, which rotors synchronously with the impeller, is adopted. The flow is dynamic .Considering that the evaluation is the time average result and the floating part could be ignored, the steady flow simulation is adopted, and it saves time compared with dynamic flow simulation. The fixed domain is the stationary obviously. Relevant boundary conditions have to be specified in order to complete the mathematical formulation. It is assumed that the upstream flow pattern is fully developed. For the solid wall, the no slip boundary condition, i.e., $v = 0$, is used. In this study, the flows at five different flow rates: 0.106 cumecs, 0.125 cumecs, 0.143 cumecs, 0.155 cumecs , 0.162 cumecs and corresponding pressure of 1.6 metre ,2.4 metre ,5.2 metre , 5.8 metre and 7.2 metre are simulated .

4.5 RESULTS AND DISCUSSION ON CFD SIMULATIONS

In this section the pressure contours along the hub, blade and casing for the particular flow rate of 0.125 cumecs and a pressure of 2.4 metre is presented. Also presented is how the efficiency for that particular flow rate is found out. The efficiency for other operating conditions is found out in a similar fashion and is presented in Table 4.1.

4.5.1 PRESSURE CONTOURS

The variations of pressure along the hub and blades for a flow rate of 0.125 cumecs and a pressure of 2.4 metre is given in Figure 4.3. From the pressure contours it can be seen that there is a variation of pressure along the blade wall. The pressure at the top surface of the blade and the bottom surface of the blade are different as is evident from the figure.The variation in pressure along the casing for a flow rate of 0.125 cumecs and an operating pressure of 2.4 m of head is given in Figure 4.4.From the pressure contours the steady drop in pressure from the inlet to exit of draft tube from where it discharges out can be seen clearly.



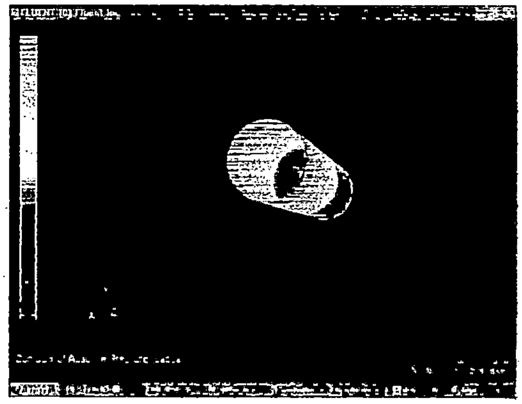


Figure 4.3 Pressure contours along blades   Figure 4.4 Pressure contours along casing

4.5.2 Measurement of torque

Output power is measured by means of the torque which the PAT unit developed when it was made to run at the above operating conditions and at a rotation speed of 1500 rpm. The torque developed for the flow rate of 0.125 cumecs and an operating pressure of 2.4 m is 8.9 NM. It is shown in Figure 4.5. Power output is then measured by the formula

$$\text{Power} = T \times 2\pi \times 1500/60$$

where T is the torque developed.

Therfore power output $= 8.93 \times 2 \times \pi \times 1500/60 \times 1000 = 1.39$ kW.

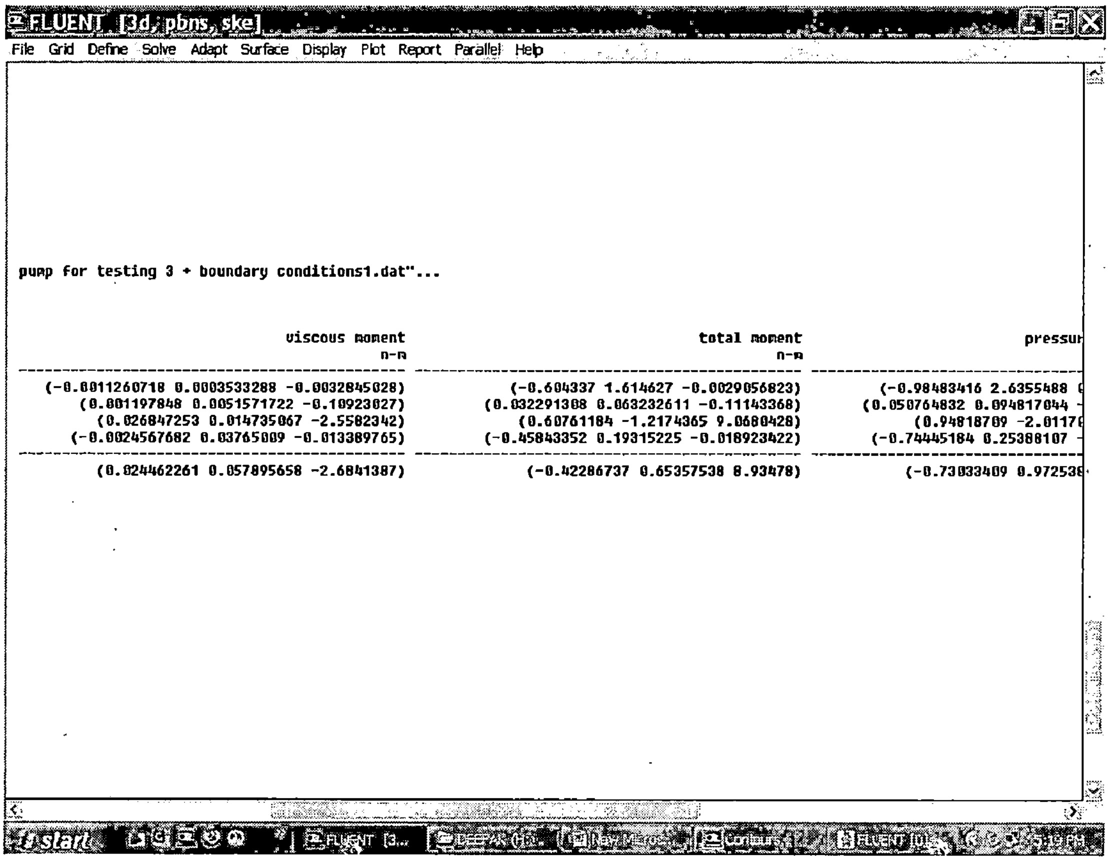


Figure 4.5 Estimation of moment

4.5.3 Calculation of power input

Power input to the PAT unit is given by the power equation

$$\text{Power input (kW)} = 9.81 \times Q \times H$$

where Q is the discharge in cumecs and H is the head in metre.

so power input = $9.81 \times 0.125 \times 2.4 = 2.94$ kW.

4.5.4 Calculation of efficiency

Efficiency of the PAT unit is given by

$$\text{Efficiency (\%)} = \frac{Poweroutput \times 100}{Powerinput} = \frac{1.39 \times 100}{2.94} = 47.2\ \%$$

4.6 TABULATION OF RESULT

The above steps are carried out for the other flow rates and inlet pressure combinations and the results are tabulated in Table 4.1.

Sl.No	Head(m)	Flow(cumecs)	Power I/P(kW)	Torque(NM)	Power O/P(kW)	Efficiency(%)
1	1.6	0.106	1.66	0.7	0.11	7.1
2	2.4	0.125	2.94	8.9	1.39	47.2
3	5.2	0.143	7.29	29.89	4.7	65
4	5.8	0.155	8.81	38.09	5.99	68
5	7.2	0.162	11.43	45.09	7.09	62

4.7 OPERATING CHARACTERISTIC CURVE

4.7.1 Efficiency versus discharge curve

The efficiency versus discharge curve is shown in Figure 4.6. The graph shows the variation of efficiency with discharge. The peak efficiency reached is 68 % at a flow rate of 0.155 cumecs.

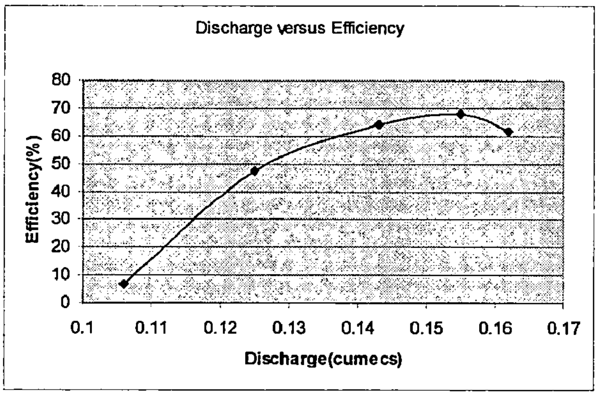


Figure 4.6 Discharge versus efficiency curve

4.7.2 Head versus discharge

The Figure 4.6 shows the variation of head with discharge. Head varies almost linearly with discharge as can seen from the graph.

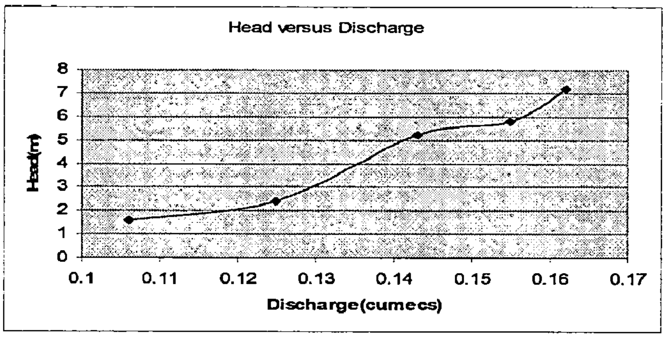


Figure 4.7 Head versus discharge curve

4.7.3 Discharge versus power

The Figure 4.6 shows the variation of power with discharge. As can be seen from the graph output power varies almost linearly with discharge.

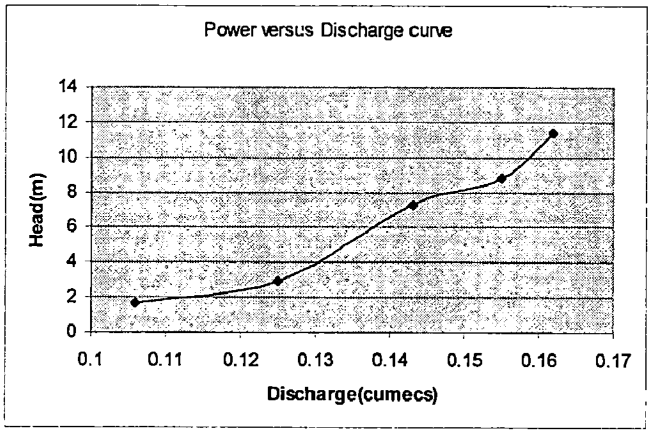


Figure 4.8 Power versus discharge curve

CHAPTER 5: EXPERIMENTAL STUDY OF PUMP AS TURBINE

5.1 GENERAL

In order to find the performance of an axial flow pump running in reverse mode as turbine experimental study has been carried out. The actual pump is tested in turbine mode on a test rig at Alternate Hydro Energy Centre (AHEC), IIT Roorkee. In this chapter experimental investigations of PAT has been carried out and the performance curves are prepared.

5.2 EXPERIMENTAL SETUP

A semi-closed loop-testing rig has been used for testing the pump in turbine mode. The testing rig consists of two service pumps of mixed flow type for pumping water at high pressure for providing the necessary head and flow. The specifications of the mixed flow pump used as turbine have been given in Table 4.1. The two mixed flow pumps have been connected to the sump tank. From the sump the water supplied with high pressure in the axial flow pump being used as a turbine. After imparting motion to the pump impeller, water is discharged through the conical draft tube connected to the pump outlet. Water goes back through the channel to the same tank, from which the two mixed flow pumps again pump the same water back to the PAT. A digital pressure gauge has been connected to the PAT inlet, to measure the head. An ultrasonic flow meter provided in the pipe line is used to measure flow. The test rig has been shown in Figure. 5.1 and Figure.5.2 shows the photograph of the test rig fitted with the pump and generator.

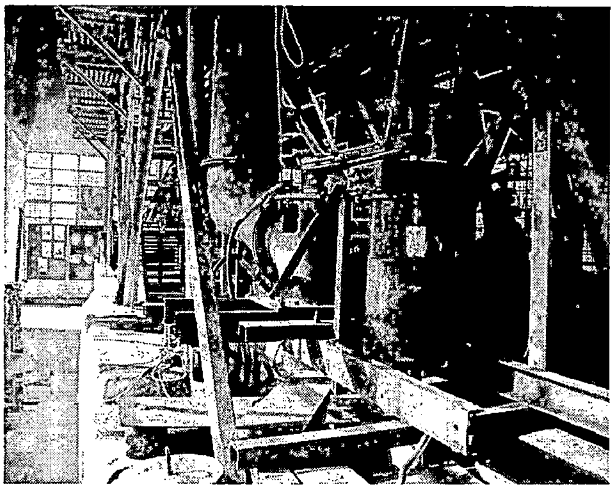


Figure 5.1 The test rig used for experiment

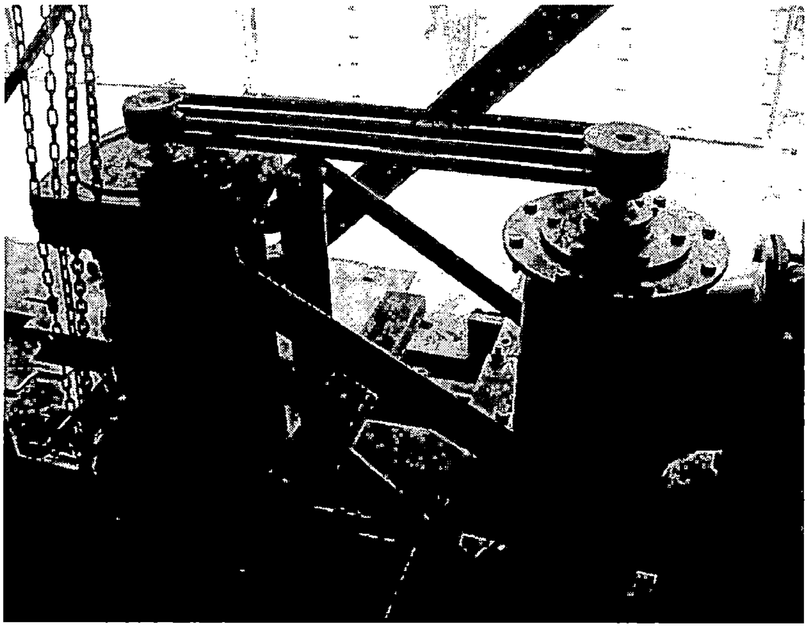


Figure 5.2 Experimental setup

5.3 RANGE OF PARAMETERS MEASURED

The specification of the mixed flow pump has been shown in Table 5.1. Parameters, which have been measured, are given in Table 5.2 with their operating ranges for the pump used as turbine before modification.

Table 5.1: Specifications of service pump in test rig

S.No.	Parameters	Details
1	Make	M/S HSMITC, Karnal
2	Type	Mixed Flow, Vertical Shaft
3	Head	10 m
4	Discharge	150 lps
5	Motor	22.5 kW

Table 5.2: Range of parameters considered for experimental study

Parameters	Range for Pump as Turbine
Head (m)	1.4-8.0
Discharge (cumecs)	0.109-0.175
Pump speed (rpm)	1500 rpm
Generator speed (rpm)	1500 rpm
Power Output (kW)	0 – 7.73

5.4 INSTRUMENTATIONS USED

During the investigations and experiment various parameters have been measured. Different instruments are used for measuring those parameters are discussed below.

5.4.1 Head

Head has been measured using a digital pressure gauge, LD301 (pressure gauge). Figure 5.2 shows a view of LD301 during the course of the experiment. Figure 5.3 shows the pressure probes being connected to the inlet of the PAT.

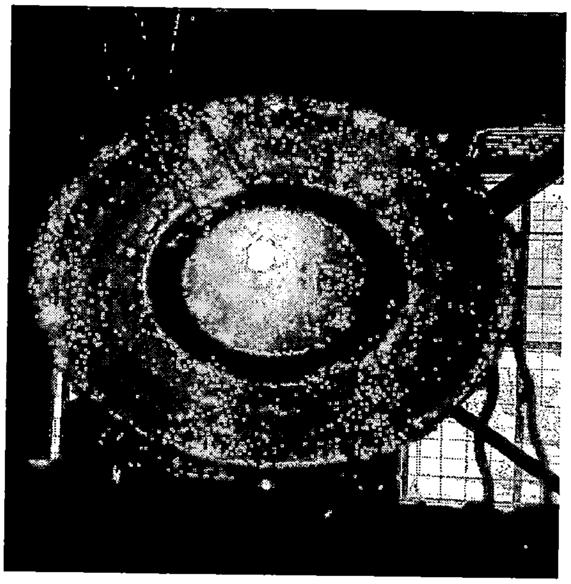


Figure 5.3 Pressure transmitter used for measuring head

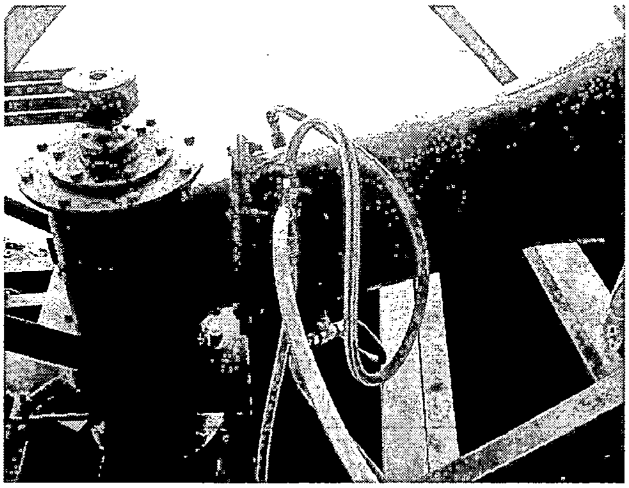


Figure 5.4 Pressure probes connected to inlet of PAT unit

5.4.2 Discharge

Discharge has been measured, by using an ultrasonic R.R flow meter of model F-133. Figure 5.5 shows the digital flow meter used for measurement of discharge

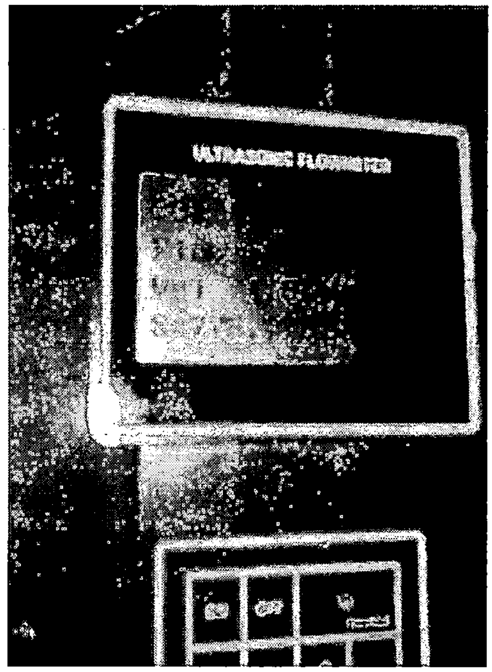


Figure 5.5 Digital flow meter for discharge measurement

5.4.3 Generator RPM

Speed of pump and generator was measured by using a tachometer and non contacting type digital speed recorder. A constant speed of 1500 rpm was maintained through out the experiment.

5.4.4 Resistive load and generator voltage

The load on the generator was put as resistive load using 100 W, and 500W bulbs fitted in a panel, as shown in Figure 5.6.Watt meter is used to measure the power generated by the generator connected with PAT as shown in Figure 5.7

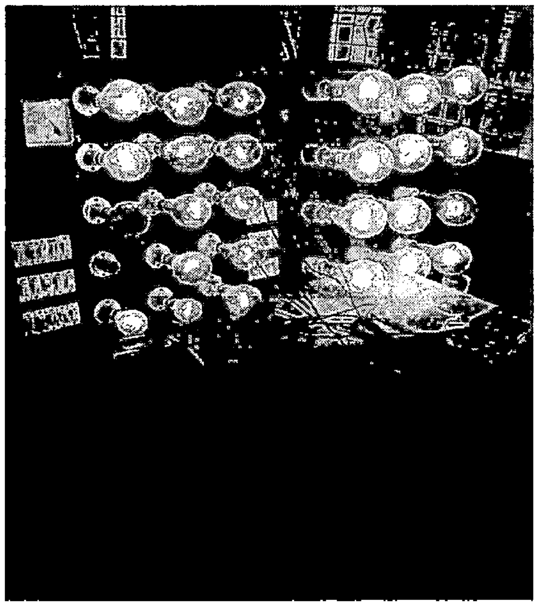


Figure 5.6 Resistive load panel

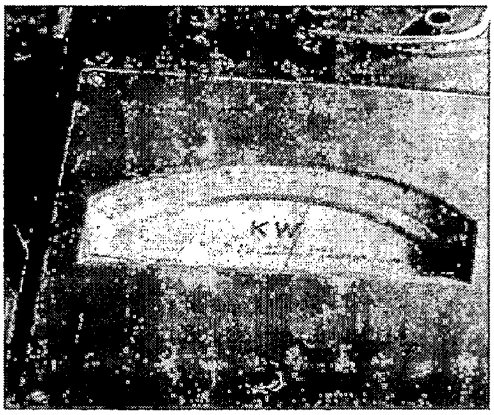


Figure 5.7 Wattmeter

5.4.5 Power output

A 3 phase synchronous generator used for testing is shown in Figure 5.8

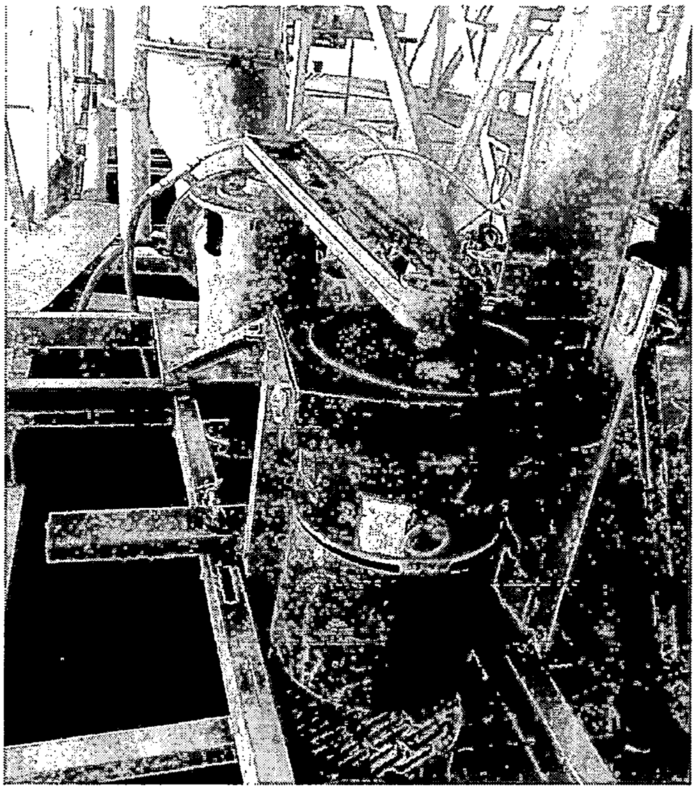


Figure 5.8 AC generator used for experiment

5.5 EXPERIMENTAL PROCEDURE

The pump was tested in reverse mode as turbine at the rig without modification. Two service pumps have provided in the sump tank which supply the required flow for the operation of PAT. A valve was connected at the pump inlet to vary the discharge supplying to the pump. A digital pressure gauge was connected to the pump inlet for measuring the pressure head. For the discharge measurement ultrasonic flow meter was used, which was fitted in the pipe line. The PAT was connected to a synchronous generator through a belt and pulley arrangement. The generator was connected to the panel having bulb loads and a wattmeter for measuring the generator output. Slowly the valve was open and the water was made to flow through the PAT impeller. When the pump impeller just started to rotate, readings of the pressure gauge (head), flow meter (discharge) were noted. After that the valve was opened further and the pump started to rotate with more speed. Then the load on the generator was given by switching on the bulbs. The flow and the bulb loads were so adjusted as to maintain 1500 rpm of the generator and 400 volts in the voltmeter. The reading of the pressure gauge (head), flow meter (discharge), bulb loads, and wattmeter were noted. After that valve was opened further more and again the procedure was repeated and several readings for varying discharge were taken as provided in the tables as given below.

5.6 PARAMETERS OBTAINED DURING TESTING

5.6.1 Head

The head, H acting on the pump as turbine has been calculated using the pump inlet pressure. The pump inlet pressure is obtained in terms of atmospheres. It has been converted into meters of head of water by multiplying it by ten.

$H = 10P_i$ where H is the head in meters and P_i is the pump inlet pressure in atmospheres.

5.6.2 Discharge

Discharge has been measured, by using an ultrasonic flow meter .the flow meter is capable of providing discharge in lps correct to two decimal places. Discharge obtained has been converted to cumecs by dividing it by 1000.

$$Q = q/1000$$

Where Q is the discharge in cumecs and q is the discharge in lps.

5.6.3 Power input

Pump as Turbine input, PAT_i has been obtained using the following formula:

$$PAT_i = QgH \quad \text{kW}$$

Where Q is the discharge in m^3/s, H is head in m and g is equal to 9.8 m/s^2.

5.6.4 Pump as turbine output

Pump as Turbine Output, PAT_o has been obtained using the following formula which includes the efficiency of generator and the transmission efficiency of belt pulley system

$$PAT_o = \frac{G_O}{\eta_g \times \eta_{t\ O}} \quad \text{kW}$$

Where G_o is the watt meter reading η_g and η_t are the efficiency of generator and transmission efficiency respectively

5.6.5 Pump as turbine efficiency

Efficiency, η_{PAT} of Pat has been calculated using the following formula:

$$\eta_{PAT} = \frac{PAT_o}{PAT_i}$$

5.7 EXPERIMENTAL INVESTIGATIONS ON PUMP AS TURBINE

Pump as turbine has been tested on the test rig. Different data are collected during the testing at different flows and constant speed, which defines the performance of PAT. These collected data during experiment are shown in table 5.3. Figure 5.9 shows the curve for efficiency versus flow of PAT. Maximum head available at turbine inlet is 7.0 m. The maximum efficiency has been obtained corresponding to 6.0 m head and 151 lps discharge. Figure 5.10 shows curve for power versus flow of PAT. Maximum power generated through the PAT 7.25 kW. Figure 5.11 shows head versus discharge curve for PAT. It has been found that the maximum efficiency obtained 70.9 % corresponding to 6 m head and 151 lps discharge. It has been observed from the investigation that range of operation of PAT at best efficiency point is distributed over a considerable range of flow.

Table 5.3: Data for pump as turbine

S.No.	R.P.M.	Head (m)	Flow (cumecs)	Power input (kW)	Power output (kW)	Efficiency (%)
1	1500	1.8	0.109	1.92	0.23	12.18
2	1500	2.3	0.121	2.72	1.17	43.04
3	1500	4.4	0.132	5.69	3.74	65.72
4	1500	4.8	0.140	6.63	4.68	70.58
5	1500	6.0	0.151	8.88	6.30	70.94
6	1500	7.0	0.160	11.04	7.25	65.67

5.8 SAMPLE CALULATION

Sample calculation for Sl.No 5 is provided below.

Head (H) = 6.0 metre

Discharge (Q) = 0.151 cumec

Power input = $PAT_i = QgH$

$= 0.151*9.81*6.0 = 8.88$ kW.

Pump as turbine output $PAT_o = 6.3$ kW

Pump as turbine efficiency = $\eta_{PAT} = \frac{PAT_o}{PAT_i}*100 = 6.3/8.88 = 70.94$ %

5.9 OPERATING CURVES

Figure 5.9 shows the efficiency versus discharge curve for the PAT unit. Figure 5.10 and Figure 5.11 shows Head versus discharge and Power versus discharge for the PAT unit respectively.

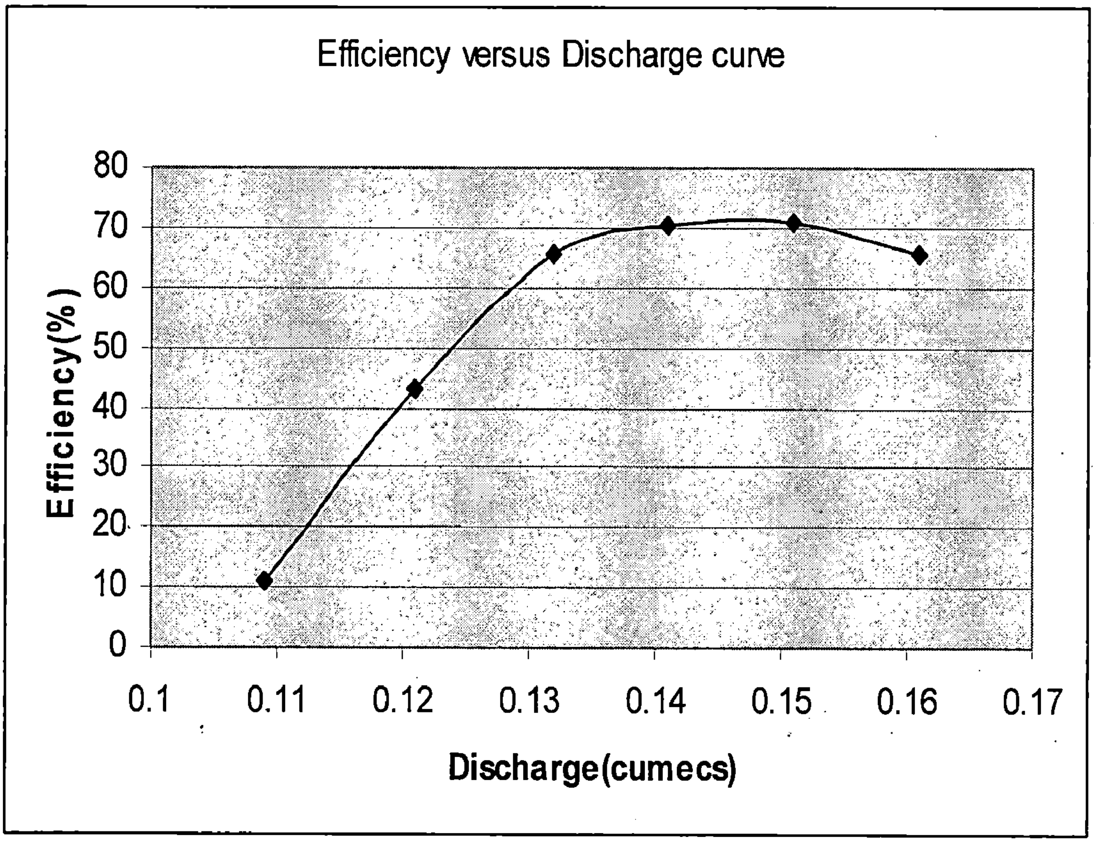


Figure 5.9 Efficiency versus discharge curve

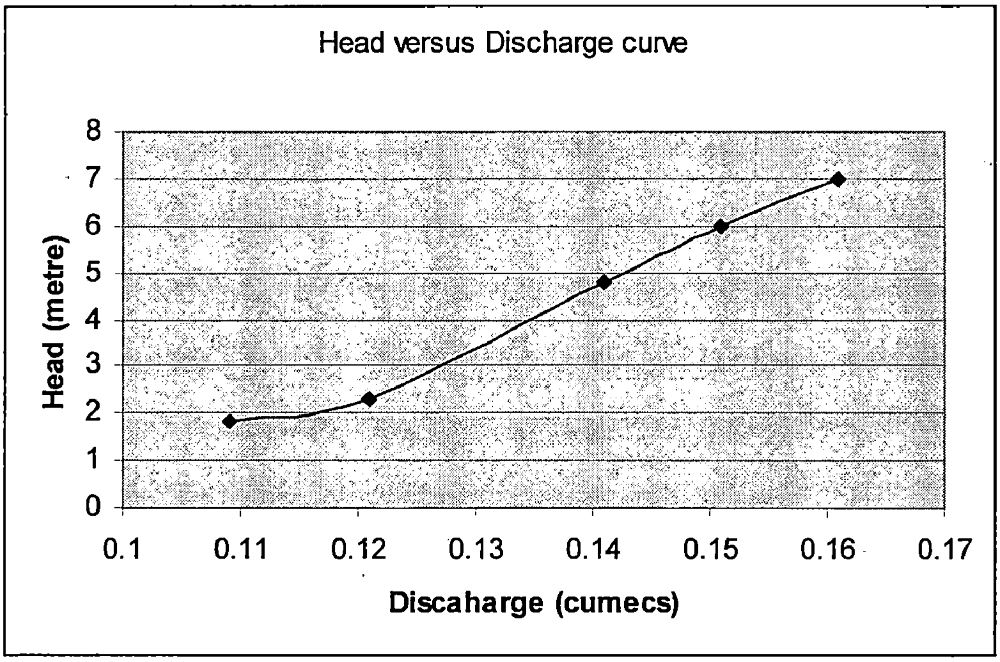


Figure 5.10 Head versus discharge curve

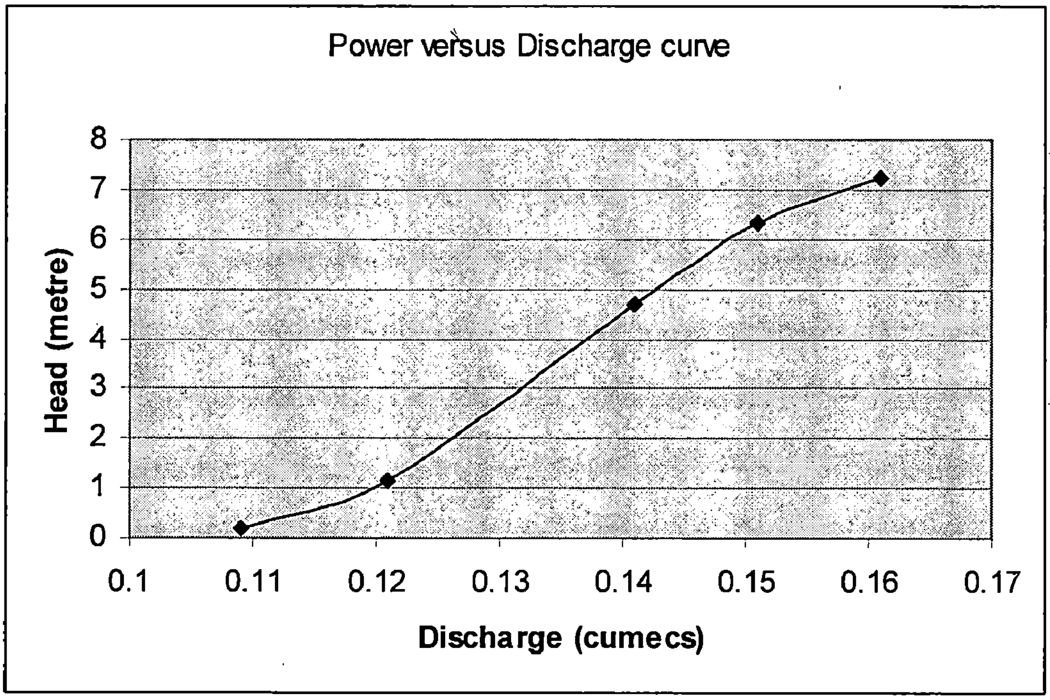


Figure 5.11 Power versus discharge curve

5.10 ASSESSMENT OF UNCERTAINITY IN EFFICIENCY MEASUREMENT

5.10.1 Uncertainty in discharge measurement

Uncertainty in velocity measurement in flow meter = $\pm 1\%$

Uncertainty in cross section area of pipe = $\pm 1\ \%$

Total uncertainty in discharge measurement $= \sqrt{1^2 + 1^2} = \pm 1.414\ \%$

5.10.2 Uncertainty in head measurement

Uncertainty in gauge pressure transmitter $= \pm\ 0.1\ \%$

5.10.3 Uncertainty in electrical power measurement

Uncertainty in wattmeter $= \pm\ 0.5\ \%$

Therefore total uncertainty in efficiency measurement $= \sqrt{1.414^2 + 0.1^2 + 0.5^2} = \pm 2.26\ \%$

5.11 COMPARISON OF EXPERIMENTAL RESULTS WITH SIMULATION

Figure 5.12 shows the comparison of experimental and simulation. During simulation the peak efficiency of 68 % was obtained at a flow rate of 0.155 cumecs and 5.8 m head and in experimentation it is found as 70.94 % at a flow rate of 0.151 cumecs and head of 6.0 m. The curves shows a good agreement between experimental and simulation results.

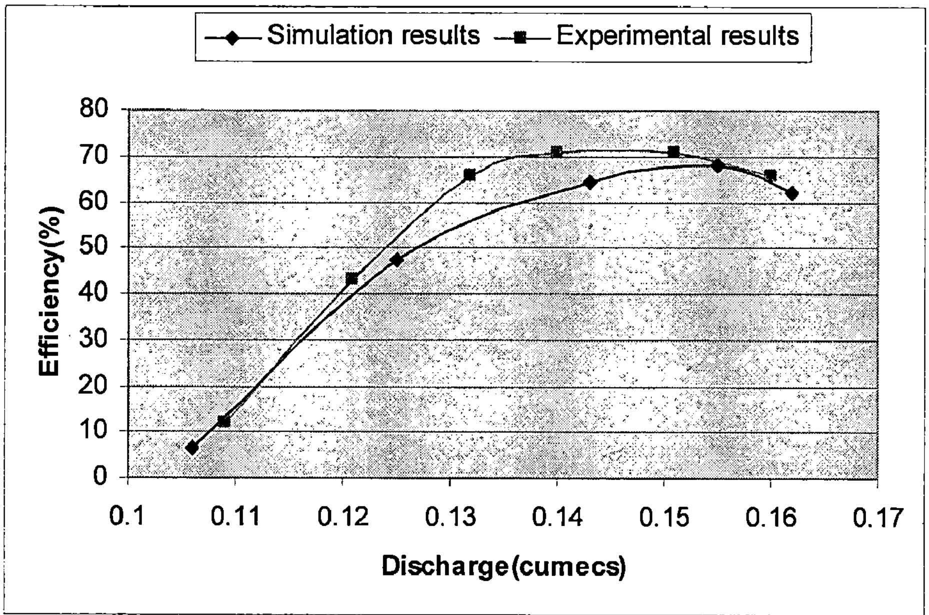


Figure 5.12 Comparison of experimental results with simulation

CHAPTER 6: CONCLUSIONS AND RECOMMENDATIONS

6.1 CONCLUSIONS

An Axial flow pump is designed, developed and fabricated .The pump was tested in reverse mode as turbine in order to determine its operating characteristics. The following conclusions are drawn from this study

i. Results obtained from the experimentation showed that efficiency of the axial flow pump when operated as turbine is obtained better than the turbine reported in the literature in the capacity range considered. A peak efficiency of 70.94 % is obtained at a flow rate of 0.150 cumecs and a head of 6.0 m

ii. These units offer a suitable replacement for the water mills due to their improved efficiency.

iii. It is also found that peak efficiency of the axial flow PAT falls cover a wider range of discharge in comparison to centrifugal pumps of radial flow as reported in the literature.. An average efficiency of 70 % was maintained for a flow rate from 0.135 cumecs to 0.152 cumecs (that is 128.6 % to 144.8 % of the designed discharge of the pump).

iv. The CFD analysis carried out for the reverse operation of axial flow PAT and compared with experimental results resulted in good agreement.

6.2 RECOMMENDATIONS

As regards the future scope, the work can be directed in the following areas

i. To test the performance of axial flow pump as turbine with adjustable flow control mechanisms.

ii. To carry out CFD analysis on reverse operation of axial flow pump with a finer mesh quality so that results can be matched to a greater extent and more accurate design analysis can be carried out.

iii. To carry out economical analysis between a cross flow turbine and an Axial flow PAT based on their part load efficiency and peak efficiency.

The above recommendations can be taken as a part of further study in the future.

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