MODIFICATION IN CENTRIFUGAL PUMP USED AS TURBINE

A DISSERTATION

Submitted in partial fulfillment of the requirements for the award of the degree

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ALTERNATE HYDRO ENERGY SYSTEMS

By

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CANDIDATE'S DECLARATION

I hereby declare that the report which is being presented in this dissertation work "**MODIFICATION IN CENTRIFUGAL PUMP USED 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 2005 to June 2006 under the supervision of **Dr. R.P. Saini**, Senior Scientific Officer, Alternate Hydro Energy Centre, Indian Institute of Technology, Roorkee.

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

Anees Ahmed

DATE: JUNE 30, 2006 **(ANEES AHMED)**

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

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It is with great pleasure that I take this opportunity to bow my head in respect and gratitude for all those who helped me in making this dissertation a great success. I am in dearth of words to express myself in such a joyous moment

I take this opportunity to grace myself from the benign self of my teacher and guide, Dr.R.P. Saini, for ushering me from theoriticality to practicality and from plutonic to pragmatic ideas. No rhapsody or rhetoric eloquence can replace of what he had done for me and the way he has helped me in bringing out this dissertation. I will always be indebted to him all our long life.

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ABSTRACT

The major problem associated with micro hydropower schemes is the non-availability of standardized equipments like turbines because micro hydropower sites are having different head and discharge. Since there is non-availability of standardized equipments, therefore particular type and dimension turbines need to be ordered for particular sites. This may not be economically viable.

An option for such sites is to use a standard pump used in reverse mode (PAT) instead of a conventional turbine. The use of standard pumps as turbines (PAT) may often be an alternative with a considerable economic advantage and might therefore contribute to a broader application of micro-hydropower. Technically there is no problem to use a pump as turbine in reverse mode, but the problem is its selection. The performance characteristics of pump as turbine will be different than pump as pump. Another problem with pump as turbine is that, pump has fixed flow control mechanism, which restricts its application at those sites, where there is variation in flow. Many investigators suggested the ways and means for selection of pump as turbine, however no work has been reported for the modification of pumps to facilitate the variable flow control mechanism.

In this dissertation an attempt has been made to modify a commercially available centrifugal pump by providing movable guide vanes. In order to compare the efficiency of modified pump, a centrifugal pump was tested for its efficiency in turbine mode. After designing suitable guide vanes and their regulating mechanism, six movable guide vanes have been provided around the pump impeller and it has again been tested for its efficiency after such modification.

After conducting the experiments and analyzing the results it has been found that after its modification, its efficiency has been found to be increased by 4%. Further, it has been found that the operating range of the modified pump as turbine is also increased substantially.

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NOMENCLATURE

SYMBOL	DEFINITION	UNITS
M_{toe}	Million Ton Oil Equivalent	-
$TREH$	Total Required Exhaust Head	m
$TAEH$	Total Available Exhaust Head	m
Q	Volumetric Flowrate	m^3/s
H	Head	m
η	Efficiency	%
η_h	Hydraulic Efficiency	%
u	Peripheral Velocity of Rotor	m/s
c_u	Tangential velocity of Flow	m/s
g	Acceleration due to gravity	m/s^2
α	Absolute Flow Angle	Degree
β	Blade Angle	Degree
c_r	Radial Velocity	m/s
N_s	Specific Speed	RPM
N	Shaft Rotational Speed	RPM
P	Shaft Power	kW
C_Q	Conversion Factor for Capacity: $Q_p = Q_t/C_Q$	-
C_E	Conversion Factor for Efficiency: $E_p = E_t/C_E$	-
C_H	Conversion factor for Head: $H_p = H_t/C_H$	-
σ_r	Cavitation Constant	-

D_I	Pump Impeller Diameter	mm
D_G	Diameter of Circle along which Guide Vanes have been provided	mm
C_G	Circumference of Circle along which Guide Vanes have been provided	mm
P_i	Pump Inlet Pressure	atm.
C_e	Coefficient of Discharge for Rectangular Weir	-
b	Notch Width of Rectangular Weir	ft
h	Level Rise of Water in Rectangular Weir	ft
K_b	Accounts for Effects of Viscosity in Rectangular Weir	ft
K_h	Accounts for effects of Surface Tension in Rectangular Weir	ft
PAT_i	Pump as Turbine Power Input	kW
PAT_o	Pump as Turbine Power Output	kW

SUBSCRIPTS

bep	At Best Efficiency Point	-
p	Refers to Pump Operation	-
t	Refers to Turbine Operation	-
1	Refers to Inlet Location of Rotor Blading	-
2	Refers to Outlet Location of Rotor Blading	-

INTRODUCTION AND LITERATURE RIVIEW

1.1 INTRODUCTION

1.1.1 General

Two hundred years ago, the world experienced an energy revolution that launched the industrial age. The catalyst to this epochal shift was ordinary black coal, an energy-rich hydrocarbon that supplanted wood as the primary fuel. The energy stored in coal gave inventors and industrialists the power they needed to process steel, propel steamships, and energize machines. A century later, the industrialized world's thirst for energy had increased tremendously. Petroleum and natural gas were exploited as versatile and high quality energy products, and soon joined coal as principal fuels. Fifty years later, scientists tapped uranium to fuel nuclear reactors and provide atomic energy.

There is a dangerous dark side to relying on non-renewable resources like coal, oil, natural gas, or uranium to supply our growing energy demands. Not only the use of non-renewable fuels threatens our health and environment but also the supply of these fuels is physically limited. World primary energy demand, as shown in Table 1.1 and Fig. 1.1, is to expand by almost 60% between 2002 and 2030, reaching 16.5 billion tonnes of oil equivalent [1]. Analyzing the rate of increase of future growing energy demands, it seems that these non-renewable energy sources alone would not be able to fulfill these demands and this would lead to an energy crisis.

One inexpensive solution to the above propagating energy crisis may be the use of abundant renewable energy sources. Renewable energy would play a major role in the energy industry of the twenty-first century and beyond. As shown in Table 1.2, renewable energy would have 14% share of total energy demand in future. Industry experts realize that these alternative energy systems not only help reduce greenhouse gas emissions, but they predict that over the next half century, renewables may grow to supply half the world's energy. The renewable energy sources are solar power, wind power, hydroelectric power, biomass materials, geothermal energy, tidal energy, wave energy and hydrogen as fuel cells, among which hydropower is the second largest renewable source. The media and industry claim that renewable energies are not yet economically competitive with fossil fuels. Perhaps not, but when one considers the health, environmental costs and future energy needs associated with burning coal and oil, the price of renewable energy becomes more attractive. No renewable energy system will single-handedly replace oil, but together they would share 14 % of total energy demand and thus become a very important part of the energy mix of the future in which hydroelectric power would play a major role and would have 2% share of total energy demand [1], coming second among renewables after biomass with least number of disadvantages, as shown in Table 1.2.

Hydroelectric power can be classified based on generation capacity [2] (power output) in the following ways:

(i) Large Hydropower: It provides more than 100 MW power output, usually feeding into a large electricity grid.

(ii) Medium Hydropower: It provides 15 MW to 100 MW power output, usually feeding into a grid.

(iii) Small Hydropower: It provides 1 MW to 25 MW power output, usually feeding into a grid.

(iv) Mini Hydropower: It has 100 kW to 1 MW power output, having either stand-alone schemes or more often feeding into a grid.

(v) Micro Hydropower: It has upto 100 kW power output, and usually provides power to a small community or rural industry in remote areas away from grid.

(vi) Pico Hydropower: It provides power out put from few Watts to 5 kW.

Over the last few decades, there has been a growing realization in developing countries that micro-hydro schemes have an important role to play in the economic development of remote rural areas, especially mountainous ones. Micro-hydro schemes can provide power for industrial, agricultural and domestic uses through direct mechanical power or by the coupling of the turbine to a generator to produce electricity.

1.1.2 Micro Hydropower

Micro-hydropower can be one of the most valuable answers to the question of how to offer isolated rural communities the benefits of electrification and the associated progress, as well as to improve the quality of life. These solutions can use natural or

Table 1.1 World Primary Energy Demand in M_{toe} (Million Ton Oil Equivalent) [1]

Source / Year	1971	2002	2010	2020	2030	2002-2030*
Coal	1407	2389	2763	3193	3601	1.5 %
Oil	2413	3676	4308	5074	5766	1.6 %
Gas	892	2190	2703	3451	4130	2.3%
Nuclear	29	692	778	776	764	0.4 %
Hydro	104	224	276	321	365	1.8 %
Biomass	687	1119	1264	1428	1605	1.3 %
Other Renewables	4	55	101	162	256	5.7 %
Total	5536	10345	12194	14401	16487	1.7 %

* Average Annual Growth Rate

Table 1.2 Worlds Renewable Energy Consumption [1]

Year	2002		2030	
Source	**Renewables use (M_{toe})**	**Share of total demand (%)**	**Renewables use (M_{toe})**	**Share of total demand (%)**
Biomass	1119	11 %	1605	10 %
Hydro	224	2 %	365	2 %
Other Renewables	55	1 %	256	2 %
Total	1398	14 %	2226	14 %

artificial falls of water and do not contribute to environment damage, offering decentralized electrification at a low running cost and with long life. High costs of equipment and civil works, or more generally, the capital-intensive nature of small hydropower plants, have long been a major constraint. However, in many situations it is necessary not only to achieve a better relation of costs compared to other energies, but also to reduce them in absolute terms. This is possible to some degree by standardizing equipment, but the scope for using such standardized equipment remains limited since no two sites are exactly the same.

In India, total potential including all the hydropowers having output up to 25 MW is 10324 MW, in which 1693 MW is the installed capacity [3]. Micro hydropower systems use the energy in flowing water to produce electricity or mechanical energy. As shown in Fig. 1.2, water from the river is channelled through a settling basin, which helps to remove sediment that could harm the turbine. The water then flows into the forebay tank where it is directed downhill through a pipe called a penstock. When the water reaches the bottom, it drives a specially designed turbine to produce the electricity. A small-scale hydroelectric facility requires that a sizable flow of water and a proper height of fall of water, called head, is obtained without building elaborate and expensive facilities. Micro hydroelectric plants that can be developed at existing dams have been constructed in connection with river and lake water-level control, and irrigation schemes. By using existing structures, only minor new civil engineering works are required, which reduces the cost of this component of a development.

There are two major problems associated with micro hydro power schemes. The first problem associated with micro hydropower is the non-availability of standardized equipments like turbines because there are number of different micro hydropower sites having different head and discharge. And since there is non-availability of standardized equipments, therefore particular dimension turbines need to be ordered for particular sites. And this process is a costly one, which is the second problem associated with micro hydropower.

When we talk about the selection of turbines for a micro hydropower, the usual alternatives would be either cross flow turbine or a multi-jet Pelton turbine. Both of these options run at relatively low speed and would require a 6-pole generator or a belt drive to

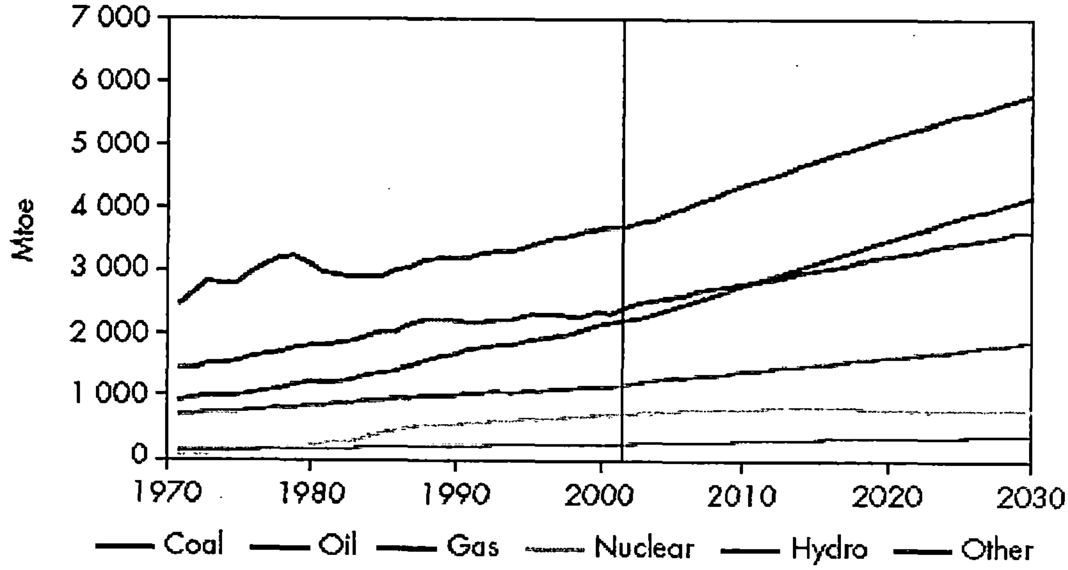


Fig 1.1 World Primary Energy Demand by Fuel [1]

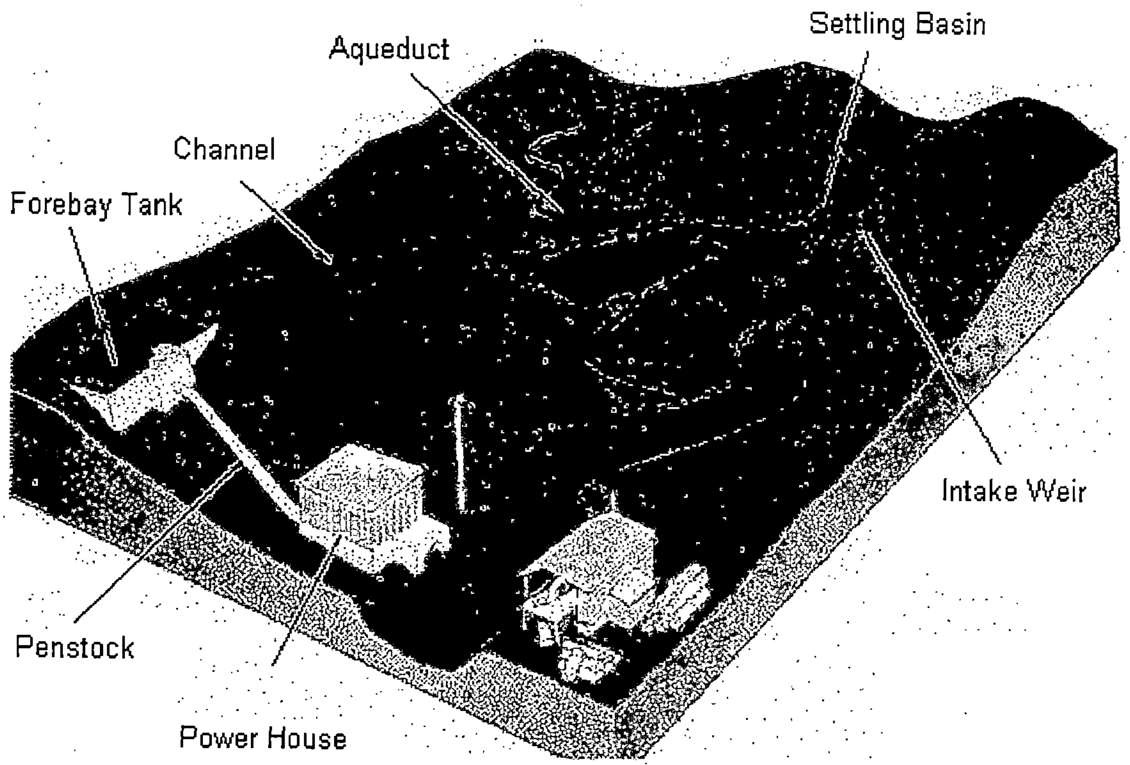


Fig 1.2 Micro Hydropower Scheme [2]

produce electric power. Another option is to use a standard pump unit in reverse mode (PAT with an induction generator) instead of a conventional turbine. The use of standard pumps as turbines (PAT) 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 PAT cannot make use of the available water as efficiently as a turbine due to its lack of hydraulic controls.

Efficiencies of pumps used as turbines may be the same as in pump mode but are more often several percent (3 - 5%) lower. PAT has a property that it gives maximum efficiency at a particular discharge, which is greater than the rated discharge of pump. If PAT is operated at other than that particular discharge, rapid drop of efficiency occurs. Thus, pumps as turbines are best suited to constant flow applications. The problem with PAT arises when there are seasonal variations in flow rate. As there are no guide vanes in standard centrifugal pumps, therefore when the discharge varies or the load varies there is loss of efficiency of PAT resulting in poor part load efficiency of PAT.

1.2 LITERATURE RIVIEW

The research on using pump as turbine started already around 1930. Numerous papers have been published on the subject, but there has been comparably limited use of this knowledge. Some of these papers are discussed below.

Fernandez, J. et al. [4] presented in their paper about the functional characterization of a centrifugal pump used as a turbine. It shown the characteristics of the machine involved at several rotational speeds, comparing the respective flows and heads. In this way, it is possible to observe the influence of the rotational speed on efficiency, as well as obtaining the characteristics at constant head and runaway speed. Also, the forces actuating on the impeller were studied. An uncertainty analysis was made to assess the accuracy of the results. The results indicated that the turbine characteristics can be predicted to some extent from the pump characteristics, that water flows out of the runner free of swirl flow at the best efficiency point, and that radial stresses are lower than in pump mode.

Fucfaipoug, T., et al. [5] modified and researched a 2" diameter, centrifugal pump to run as a Francis turbine because of many similar parts such as volute and impeller. The impeller is modified so that the blade is applicable for the operation. A guide vane is installed at the inlet of the impeller for higher efficiency. This turbine is tested in constant speed and constant head situation and it has been noticed that the efficiency is higher by 6% in case when guide vanes are installed. This turbine can use as a topic for industrial power technology experiment laboratory subject.

Williams, A., & Piggot, H. [6] presented the method of calculating the pump performance in turbine mode. It gave many examples of pump as turbine schemes. It discussed the main problem associated with pump used as turbines.

Ramos, H., & Borga, A. [7] presented in their paper in which in any water system, which has excessive available energy (e.g. natural falls, irrigation systems, water supply, sewage or rain systems), the application of a pump instead of a turbine, for energy production, seems to be an alternative solution with easy implementation and considerable equipment cost savings. Micro-hydropower corresponds to a typical renewable energy source without any relevant impacts, and has multiple advantages, as a decentralized, low-cost and reliable form of energy. Unconventional solutions are in the forefront of many developing countries to achieve energy self-sufficiency.

Tamm, A., et al. [8] presented a paper containing results on CFD simulations of a spiral casing test pump having a specific speed of N_S of 620 rpm (in US Units). The investigation in a turbine mode has been carried out for three rotational speeds with the aim to determine the head and flow rate corresponding to the optimum efficiency.

Varma, G., et al. [9] conducted extensive studies on the feasibility of small, mini and micro hydel stations in Kerala and prepared a few Detailed Project Reports also. A control system was developed for Induction Motors, used as Generators (IMAG) and the use of Pumps as Turbines (PAT). However, the methodology for choosing the correct specifications for the most suitable machine is not clearly established. Nor is the loss in performance properly evaluated under field conditions.

Hulse, R., & Seckel, K.W. [10] presented the study made by the Municipal Water District of Orange Country (MWDOC) of hydroelectric installations ranging in size from 50 kW to 350 kW. The first design project to develop from the MWDOC study was the city of La Habra. The hydroelectric power generation station utilizes a reverse-running centrifugal pump as a turbine. The use of a pump and motor for the turbine and the generator unit is discussed.

In his paper, Mike, H.P.E. [11] found about a number of sites being equipped with standard pumps arranged to run backwards as fixed geometry turbines. It makes us aware of the care that should be taken in the selection and modification of such equipment and it contains notes based on writer's experience at a number of sites that are intended to serve as an outline of the points that must be kept into consideration to ensure the success of the project.

Buse, F. [12] investigated on how to select a pump that will efficiently convert fluid energy into useful work.

Kittredge, C.P. [13] suggested that centrifugal pumps may be used to advantage as small hydraulic turbines where low initial cost is imperative. Pump and turbine characteristics are presented for typical unit and procedure for pump selection is discussed.

Knapp, R.T. [14] described the technique of determining complete operating characteristics of a hydraulic machine such as centrifugal pump or a turbine, together with a method of presenting these characteristics in a convenient manner on a single diagram. The characteristics of a modern, high head, high efficiency pump were analyzed and presented in the manner proposed. The use of these complete characteristics for the prediction of the behavior of the machine during operating transients is discussed and the analytical background was presented. The assumptions involved were investigated and experimental checks of their validity were offered. The inter relationships between the hydraulic characteristics of the machine and the pipelines were also indicated.

Cooper, R., Cormic, M.M., & Worthen, R. [15] discussed on the feasibility of using a large vertical pump as a turbine for small-scale hydropower based on run-of-river operations. The study indicated that circulating pumps operating as turbines provide some distinct advantages. In cases where impoundment can be used and the flow matched to the fixed geometry reverse pump, the further advantage of full water utilization can be attained.

Lawrence, J.D., & Pereira, L. [16] presented a paper based on the results of a study undertaken by Acres American incorporated for the department of Energy. The essential purpose of that study was to investigate the feasibility of using off-the-shelf pumps as turbines, with induction motors as generators, and other appropriate combinations of available equipment such as speed increasers, inlet valves, and gates for typical small-scale hydro installations. In this paper turbine performance of pumps was estimated by using a computer simulation program based on Kittredg's formulation, with test data mainly from various references.

Ingersoll-Rand Research, inc., under contract to the U.S. Department of Energy, completed a feasibility study [17] of using large vertical pumps as turbines for small-scale hydropower projects. The objective of the study was to establish the economic and technical feasibility of operating pumps as turbines. The economics of using a pump as a turbine were shown to be competitive with hydraulic turbines, and efficiencies of 87% were obtained in actual tests.

Williams, A.A. [18] investigated that small centrifugal pumps are suitable for use as hydraulic turbines and have the advantage of being mass-produced in many countries throughout the world. When used with an integral induction motor, they can be installed as a combined turbine and generator unit. Recent research and development work carried out at Nottingham Trent University in collaboration with the Intermediate Technology Development Group has concentrated on two aspects that had previously held back the wider application of this technology. A standard design of Induction Generator Controller (IGC), enabling these units to be used for isolated micro hydro schemes, has been proven, and is now being manufactured in five countries worldwide. Progress has also been made on the application of performance prediction methods, which facilitate the selection of a pump unit for particular site conditions.

Cattley, R., Widden, M.B., & Aggidis, G.A. [19] illustrated some of the current machine designs available for recovering power from low head hydraulic sources. Current opportunities and future research needs have been discussed in the paper. Modifications of some existing machines and use of pump as turbines to better utilize the available energy were discussed.

Waheed, A.R., & Robert, W.P. [20] discussed about the different recovery systems, which are becoming more and more useful and economical. On the reverse osmosis systems about 70% of energy for pumping is wasted at pressures of 800-1000 psi. This paper discussed the application of reverse running centrifugal pumps as power recovery turbines to recover upto 80% of that energy. The economics of Hydraulic Recovery Power Turbine was explored with respect to the equipment cost, operating costs and payouts for the different sizes of reverse osmosis systems.

Garay, P. N. [21] described about the feasibility of operating pump as turbine offering cost savings with operation and control differences, which should be thoroughly explored before installing a pump as turbine.

Engeda, A., & Rautenberg, M. [22] described about the design features of reverse running pumps.

Saini, R.P. [23] described about the selection procedure of a pump to be used as turbine for micro hydro power stations. A nomogram for selection of pumps used as turbine was established.

Williams, A.A. [24] described about various methods of predicting performance of centrifugal pump to be used as turbine.

In a detailed report, Singh, P. [25] described about the performance of a micro hydro system using pump as a turbine.

1.3 OBJECTIVE OF DISSERTATION

From the literature survey it is observed that much work has been done regarding pump as turbine testing, its concept, performance, selection and economic consideration, however very less work on the modification of pump to be used as turbine is available.

It is well known that turbines used at micro hydro sites can easily be replaced by pump as turbine (PAT) with a number of advantages. But the main disadvantage of PAT over conventional turbines is that they don't have any flow control mechanism, part load efficiency of PAT is very poor and it gives maximum efficiency at a particular flow rate, therefore the variation in the seasonal discharge of the stream affects the efficiency of the PAT. In order to use PAT with its maximum efficiency over a wider range of the varying discharge, adjustable or movable guide mechanism is proposed to be fitted around the pump impeller.

The objective of the present study is to modify the commercially available pump by providing, guide vanes, around the pump impeller. By selecting a suitable pump, pump was tested in turbine mode to find out its operating characteristics as turbine. The guide vanes were designed, fabricated and fitted with the pump. The modified pump was tested and then the characteristic curves were compared.

PUMP USED AS TURBINE

2.1 GENERAL

Pumps are designed as fluid movers. However, pumps have also been used for many years to generate electrical power or to provide rotating shaft energy. For example, a centrifugal pump can serve as a prime mover to drive a fan, another pump, a compressor, or a generator where high-pressure liquid is available as a source of power.

In an application of a pump used as a fluid mover, the liquid enters the suction part at low pressure, absorbs shaft work in the impeller, and leaves the discharge nozzle at higher pressure. When a pump is used as a turbine, the liquid enters at high pressure, drives the impeller in reverse, and leaves at a lower pressure.

A basic difference between conventional pumps and hydraulic turbines is that pumps generally do not have a flow control device. Pumps are usually designed for a specific total dynamic head and discharge requirement. Thus, pumps as turbines are best suited to constant flow applications. This can include sites where flow variations can be minimized by scheduling water releases in predefined increments. A sufficient number of pumps are installed to provide required discharge increment (in steps) and added reliability. Fig. 2.1 and Fig. 2.2 show a sketch of a pump being used as a fluid mover and as hydraulic turbine respectively.

Because pumps are designed as fluid movers, they may be less efficient as hydraulic turbines than equipment expressly designed for that purpose. However, using a pump as a turbine can be cost effective in many situations. Among the more obvious reasons to use a pump as a turbine are when a pump installation is significantly cheaper than a turbine installation or when a pump is immediately available but a turbine is not.

There are two major problems associated with the use of pump as turbine. The first problem is that pump as turbine gives maximum efficiency at a particular discharge and if the discharge varies from the best efficiency point discharge, then its efficiency falls rapidly. This is due to the absence of a flow control device in a centrifugal pump. This leads to poor part load efficiency of pump used as turbine. The second problem associated with the use of pump as turbine is the difficulty of finding the turbine head and

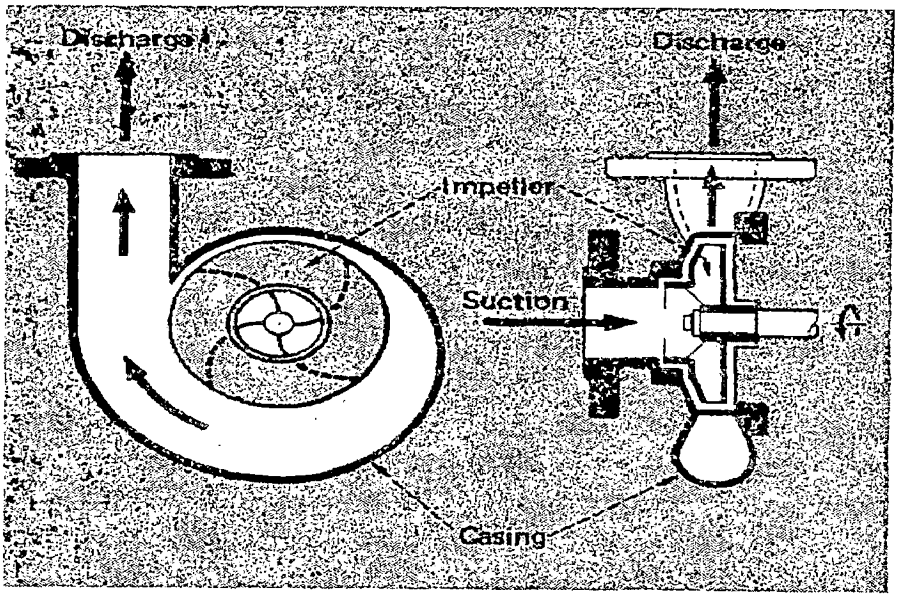


Fig. 2.1 Radial Flow Pump as Fluid Mover [12]

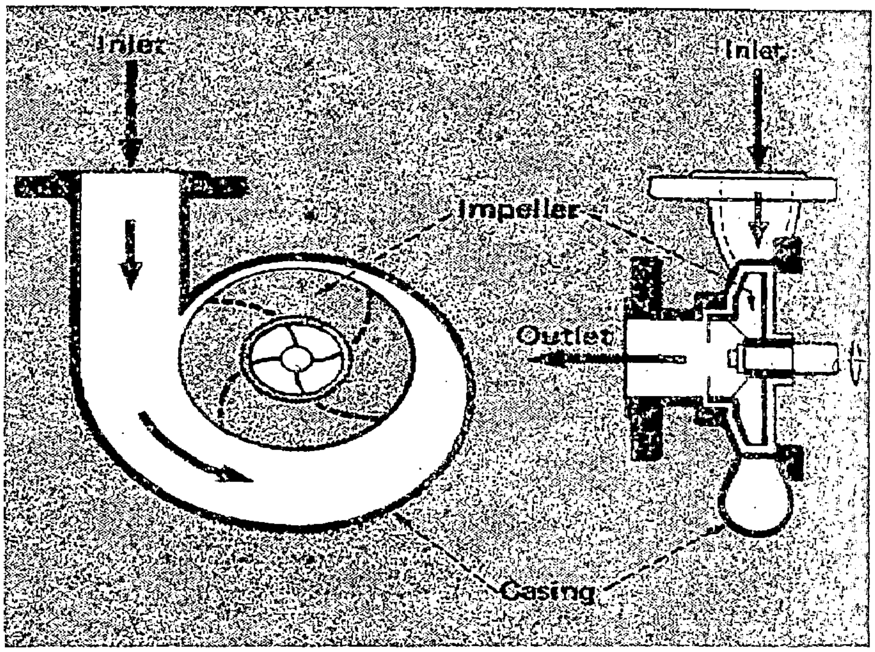


Fig 2.2 Radial Flow Pump as Hydraulic Turbine [12]

discharge, which are different than the rated pump head and flow, and are needed to select the correct pump for a particular site parameters.

There is no question as to whether a pump can operate as a power-generating unit. The concerns are more related to equipment application, efficiencies, operating range, cavitation characteristics, costs, etc. This chapter will discuss these and other factors.

2.2 CONCEPT OF PUMP AS TURBINE

The concept of PAT lies in the fact that apart of not having a flow control device, centrifugal pumps are physically and hydraulically similar to Francis turbines. Only the difference lies in their function. Centrifugal pump converts the mechanical energy of impeller into pressure energy and some kinetic energy of water whereas Francis turbine converts pressure energy and kinetic energy of water into mechanical energy of runner. Therefore if a centrifugal pump is operated in reverse mode, i.e. if the water outlet section of a centrifugal pump is used for entering water at high pressure and if the water inlet section of a pump is used for exiting water, it can function as a Francis turbine. The water, at high pressure entering in the centrifugal pump passes through the impeller and rotates the impeller and exits from the eye of the impeller.

2.3 HISTORICAL BACKGROUND OF PUMP AS TURBINE

In the late 1800s and early 1900s, hydraulic turbines were manufactured to standard designs, and a single manufacturer built several hundred each year. However, as the economics of size prevailed, a large hydropower plant became the rule, and as low cost hydrocarbon fuels became readily available, the number of turbine manufacturers was gradually reduced [26].

Pump manufacturers have used pumps running in reverse as hydraulic turbines for many years. Most of these applications have been multistage radial-flow units operating as power recovery turbines in the process industries. These manufacturers have developed empirical methods for determining performance in the normal turbine mode by reference to the known performance of the unit as a pump [26].

The first installation of a pump as turbine with an induction generator was at remote farm in the Yorkshire Dales of North England in 1991. This scheme, funded mainly by the UK Overseas development Administration, has been operating as a demonstration project for 5 years [18].

A 4.5 kW scheme was installed in 1992 as part of the programme of German Technology Exchange (GTZ) in West Java, Indonesia [27]. The pump is of Chinese manufacture, for which spare parts are easily available in the nearest large town. Assistance on the selection of the PAT for this site was also given from UK. The performance was close to that predicted, with an overall efficiency of 48% from a head of 19m and flow of 50 l/s.

An example of a small PAT scheme used for energy recovery is at Barnacre, which was installed in 1996, in the northwest of England. This uses a 30m (3 bar) pressure drop in the drinking water supply to Blackpool to generate 3.5 kW of electric power at a remote water treatment plant [18].

2.4 ADVANTAGES OF PUMP AS TURBINE

Pumps are mass-produced, and have the following advantages for micro hydro compared with purpose-made turbines:

- (i) Integral pump and motor can be used as a turbine generator set.
- (ii) Pumps are available for a range of heads and flows.
- (iii) Pumps are available in a number of standard sizes.
- (iv) Pumps have a short delivery time.
- (v) Spare parts of pumps such as seals and bearings are easily available.
- (vi) Pump has an easy installation and it uses standard pipe and fittings.
- (vii) The investment costs of PAT may be less than 50% of those of a comparable turbine (especially for small units below 50 kW). This might be an important issue for projects with limited budgets and loan possibilities.
- (viii) Pump requires nearly no special equipment and skills for its maintenance.

Further using PAT with an induction generator makes it possible to avoid a belt drive, with further following advantages:

- (ix) There are fewer losses in drive, which saves up to 5% of output power.

(x) Installation becomes easier as PAT and generator come as one unit.

(xi) Cost is saved as no pulleys are required and smaller base plate is required.

(xii) Again cost is saved as in this case of a monoblock design, the construction is simpler and fewer bearings are required.

(xiii) As there are no sideways forces on bearings therefore bearing life becomes longer.

2.5 DISADVANTAGES OF PUMP AS TURBINE

The main disadvantages of PAT are as follows:

(i) Unlike turbines, pumps have no hydraulic control device, therefore a control valve must be incorporated in the penstock line (additional costs) to start and stop the PAT. If the valve is used to accommodate the seasonal variations of flow, the hydraulic losses of the installation will increase sharply.

(ii) PAT has lower efficiency at part load. A conventional turbine has an effective hydraulic control (adjustable guide vanes, nozzles or runner blades) to adjust the machine to the available flow or the required output.

(iii) The major problem associated with the PAT is the difficulty of finding the turbine head and discharge, which are different than the rated pump head and flow, and are needed to select the correct pump for a particular site.

2.6 TURBINE USED AS PUMP

All types of centrifugal pumps, from radial-flow to axial-flow geometry, can be operated in a reverse mode as a turbine. In the turbine mode, the directions of through flow and rotation are opposite to those in the pumping mode, and the electric motor can be used as a generator. In fact, a well-designed radial pump can serve as a reasonably good turbine, but a highly efficient turbine doesn't necessarily show equally good performance and efficiency when operated as a pump. There are two main reasons for that:

(i) The blade angle at the inlet of an optimally designed turbine runner is larger than at the outlet of an optimally designed pump impeller (Fig. 2.3).

(ii) The outer diameter of an optimally designed pump impeller is larger than that one of an optimally designed turbine runner for the same head and rotational speed (Fig. 2.3).

This results from the choice of a higher blade number and blade loading in the case of turbines as compared to pumps, which corresponds to the mean acceleration vs. deceleration of the flow in the rotor, respectively. There from, a turbine will also normally show a higher efficiency than a pump, which is designed for the same operating conditions [8].

2.7 APPLICATIONS OF PUMP AS TURBINE

The chart in Fig. 2.4 shows the range of heads and flows over which various turbine options may be used. The range of Pelton and cross flow turbines shown is based on information from the range of turbines manufactured in Nepal, and is compared with the range of standard centrifugal pumps running with a four-pole (approx. 1500 rpm) generator. The range of pumps as turbines (PATS) can be extended by using either a two-pole (approx. 3000 rpm) or a six-pole (approx. 1000 rpm) generator [18].

The use of a pump as turbine has greatest advantage, in terms of cost and simplicity, for sites where the alternative would be either a cross flow turbine, running at relatively low flow, or a multi-jet Pelton turbine. For these applications, shown by the hatched area on Fig. 2.4, a cross flow turbine would normally be very large compared with an equivalent PAT. Very small cross flow turbines are not normally made due to the difficulty of fabricating the runner. A cross flow installation would require a large turbine running at slower speed than an equivalent PAT, resulting in the need for a belt drive to power a standard generator [18].

A pump as turbine requires a fixed flow rate and is therefore suitable for sites where there is a sufficient supply of water throughout the year. Long-term water storage is not generally an option for a micro hydro scheme because of the high cost of constructing a reservoir. In many village schemes in developing countries, where the main electric load is evening lighting, a PAT is suitable. During the daytime the generator can also be used to power equipment that will benefit the economy of the village, e.g. a circular saw, crop dryer or sewing machines. Sometimes, where only lighting is needed,

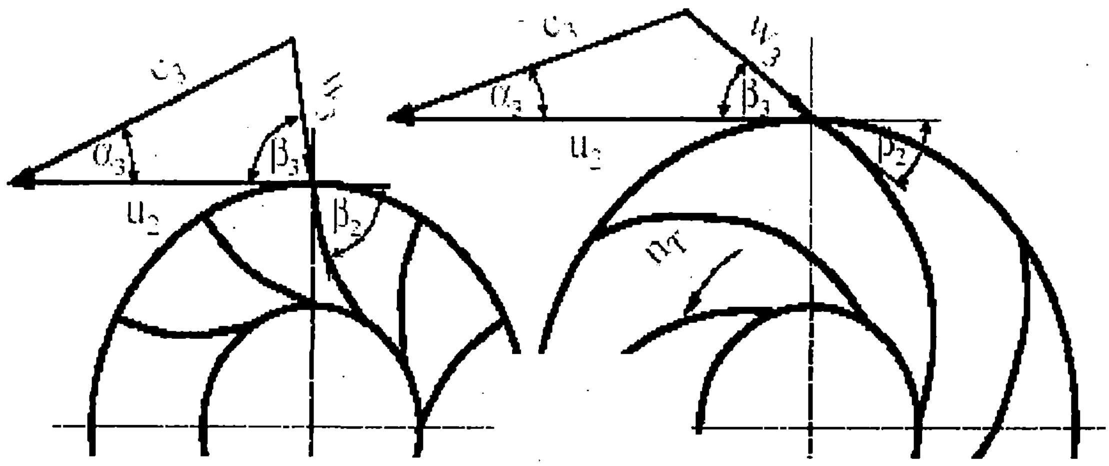


Fig. 2.3 Comparison of Turbine Runner (Left) and Pump Impeller (Right) in Turbine Mode [8]

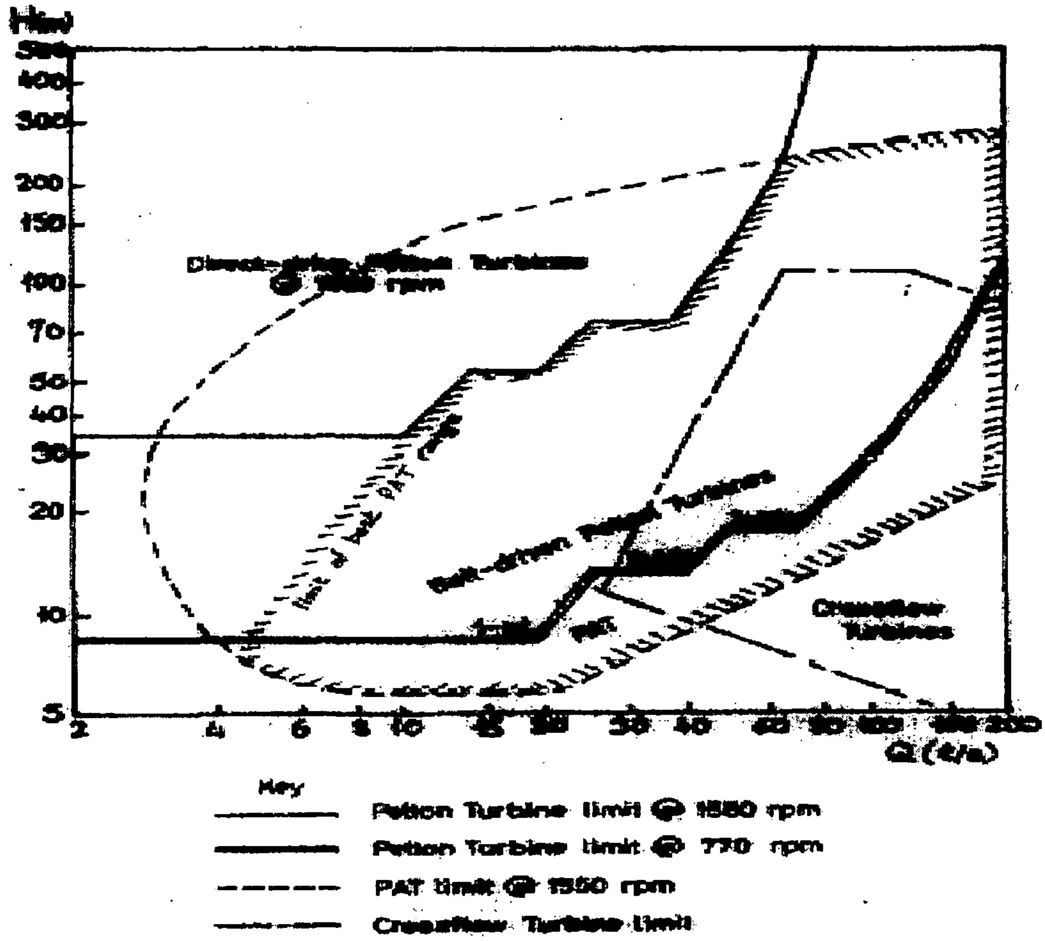


Fig. 2.4 Head -Flow Ranges for PAT and Other Turbine Options [18]

the water supplying the micro-hydro scheme can be used during the daytime for irrigation or for running a water mill for grinding corn [18].

Another application for pumps as turbines is in fluid supply lines, where pressure-reducing valves are often used to control supply pressure. In this case, installing a small turbine can recover energy that would otherwise be dissipated at the control valve. However, since the exit pressure of the system must be maintained above atmospheric pressure, an impulse turbine such as a cross flow or Pelton turbine cannot be used. These types of scheme are suitable for the application of a pump as turbine, particularly if the flow rate in the pipeline is more or less constant [18].

Pump as turbine schemes for slightly larger 'mini-hydro' plants have also been installed in Peru and Guatemala [28]. The advantages of low-cost and simplicity of installation also apply for these projects which are in the range 20 - 200 kW. An advantage of using a pump as turbine in this range, rather than a Francis turbine, is that the runaway speed (in case of electrical load loss) for a PAT is much lower, which reduces the need for protection against water hammer.

2.8 PUMP AS TURBINE PERFORMANCE CURVES

Pump manufacturers generally provide curves that describe the relationships between speed, capacity, head, efficiency and brake horsepower (BHP) for a pump used as a fluid mover. But these curves do not describe the performance of the same pump used as a turbine since peak turbine efficiency is several percent lower than peak pump efficiency, and occurs at a higher capacity, the head absorbed in turbine duty is greater than the head developed in pump duty and maximum BHP is higher in turbine duty [12].

Fig. 2.5 shows examples of constant-speed performance curves for a radial- flow centrifugal pump used as a pump and as a turbine. The curves show capacity, head, efficiency and BHP relative to the pump performance at its best efficiency point (BEP).

2.9 PUMP AS TURBINE PERFORMANCE PREDICTION

The performance of a standard centrifugal pump in turbine mode can be found from test results (which are rarely available) or by a mathematical prediction. There are many methods for PAT performance prediction. Some of these methods are based on pump best efficiency point (BEP), some are based on pump specific speed and some on pump geometry.

In many cases, pump manufacturers treat conversion factors as proprietary information. When these factors are unavailable, the manufacturer's application engineers can select an appropriate pump if given the head, capacity, rpm and total required exhaust head (TREH) of the pump in turbine duty. To predict the performance of the turbine under different operating conditions, one can use the general curves [12] presented in Fig. 2.6 to Fig. 2.9.

Note that we can get more horsepower out of a given unit by reducing the speed and increasing the flow rate. When speed is fixed, we cannot get more BHP out of the unit, but it may be possible to select a smaller pump for the same application. Fig. 2.10, a modified version of Fig. 2.5, shows the relationships between BHP, head, efficiency and capacity for a specific pump. Whereas Fig. 2.5 shows values for turbine duty relative to pump BEP, Fig. 2.8 shows values relative to turbine BEP. If we choose a pump such that its turbine design condition is to the right of BEP, we can get nearly the same power out of a smaller and less expensive pump [12].

2.9.1 Pump As Turbine Performance Prediction Using BEP

Several researchers recommended the use of pumps as turbines in which the turbine performance is predicted using the values of head and flow at pump best efficiency point, scaled to produce turbine head and flow values using a factor that is a function of the efficiency. The different methods with their formulae, which have been proposed by the various authors, are considered in the following sections.

2.9.1.1 Childs Method

Childs [29] stated that the turbine best efficiency and pump best efficiency for the same machine are approximately equal. He further assumed that the turbine output power for best efficiency operation was equal to the pump input power. Hence:

$$\frac{Q_t}{Q_p} = \frac{1}{\eta_p} \tag{2.1}$$

$$\frac{H_t}{H_p} = \frac{1}{\eta_p} \tag{2.2}$$

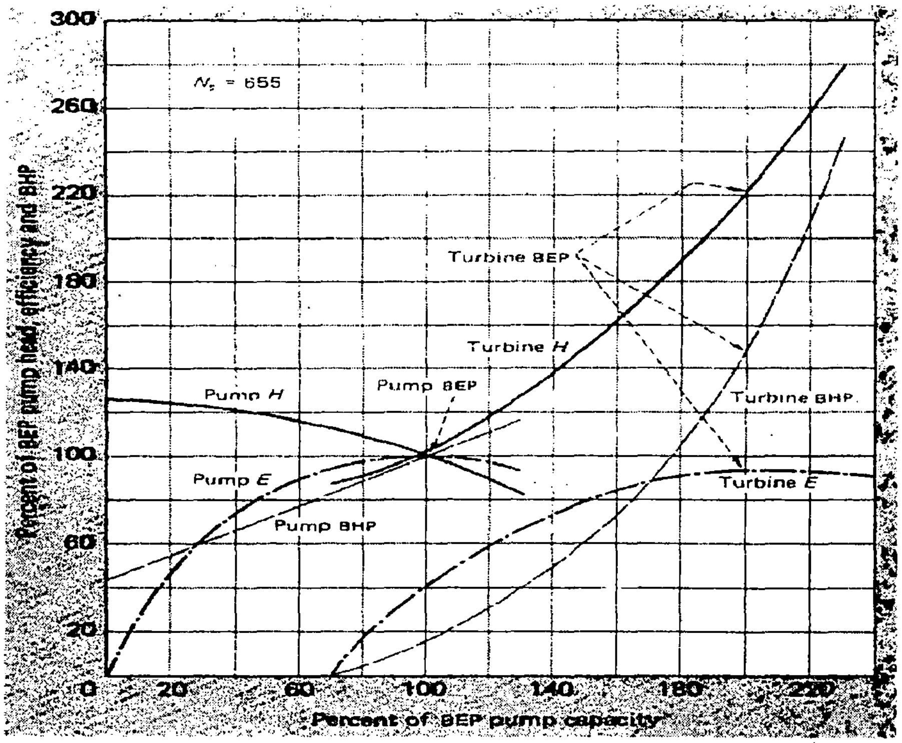


Fig. 2.5 Performance of a Pump at Constant Speed in Pump Duty & Turbine Duty [12]

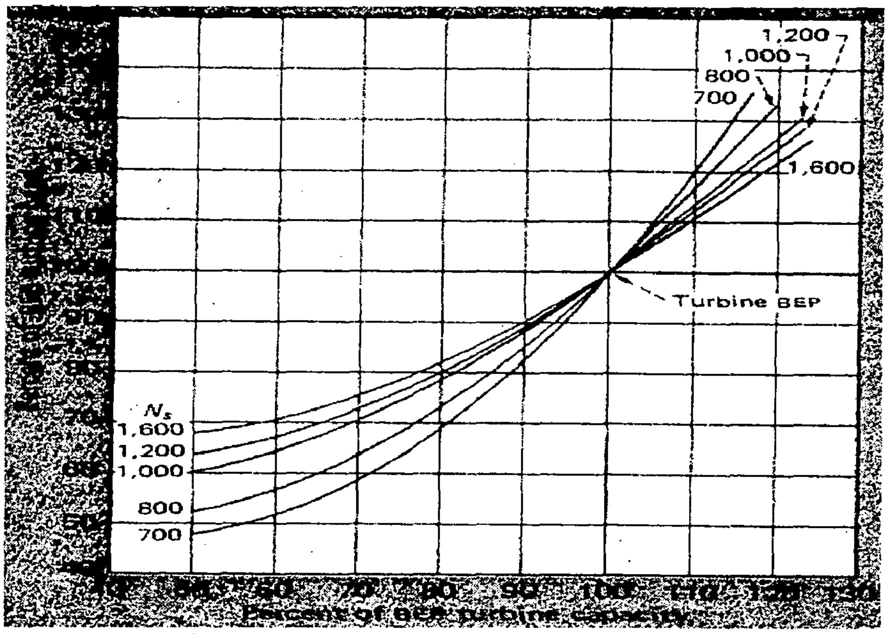


Fig. 2.6 Constant-Speed Curves for Turbine Duty [12]

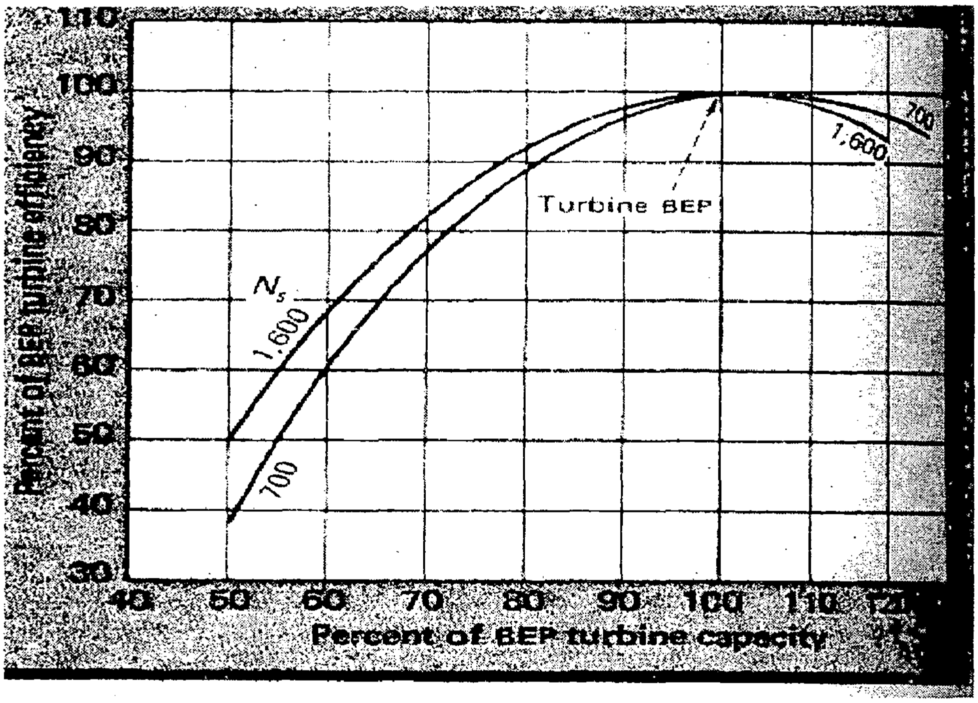


Fig. 2.7 Constant Speed Curves for Turbine Duty [12]

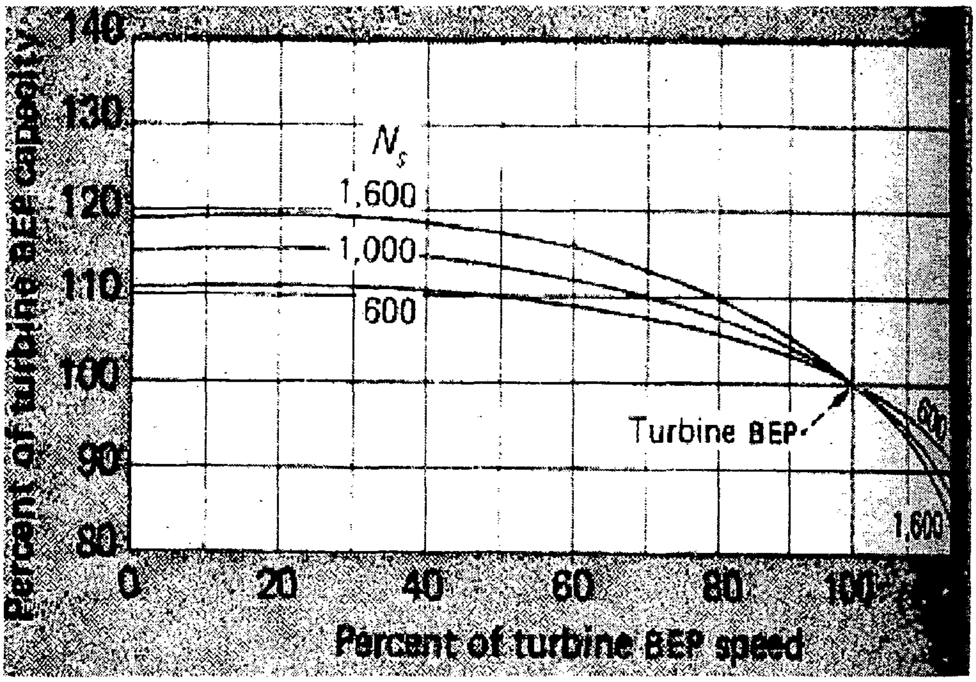


Fig. 2.8 Constant Head Curves for Turbine Duty [12]

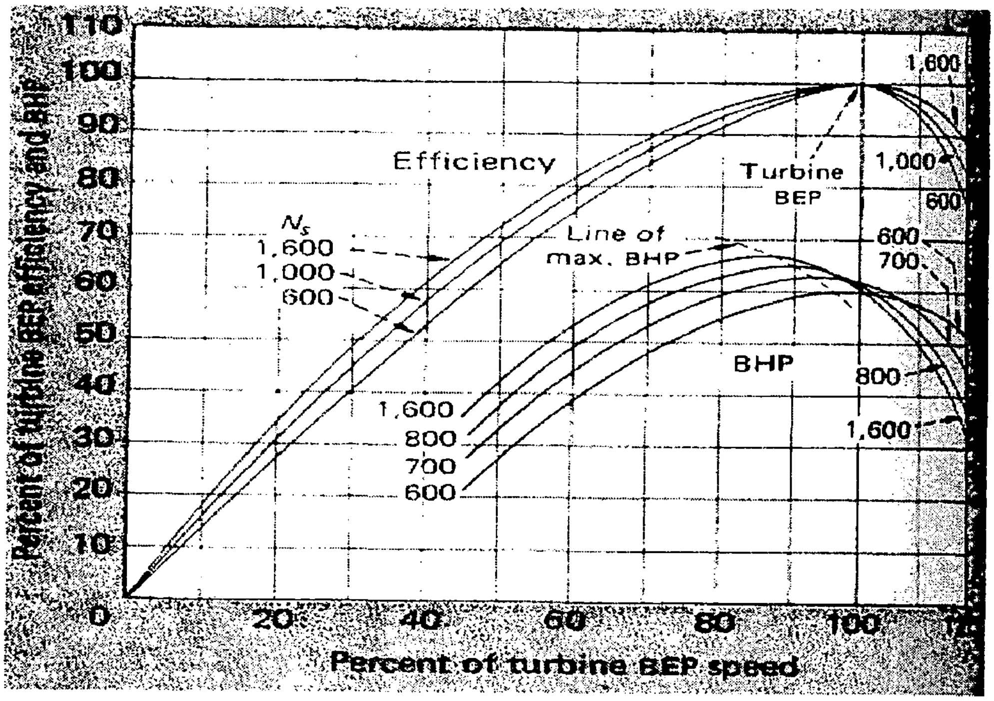


Fig. 2.9 Constant Head Curves for Turbine Duty [12]

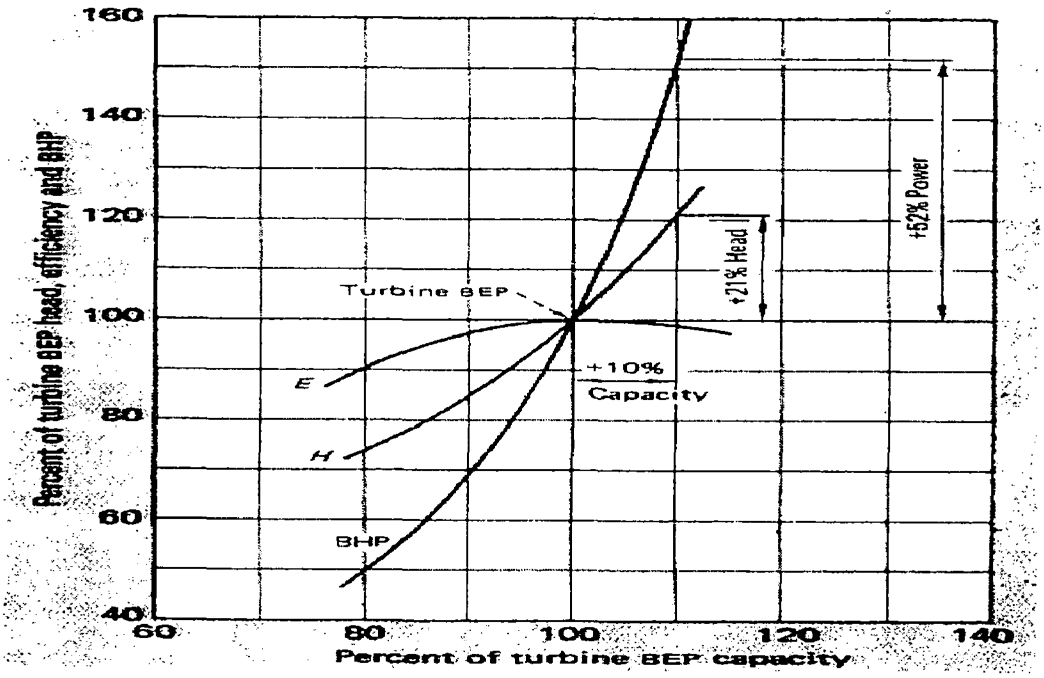


Fig. 2.10 Performance of Turbine Past BEP [12]

2.9.1.2 Engel Method

The theoretical work of Engel [24] relates the ratios of turbine & pump head and turbine & pump flow to the hydraulic efficiency of the pump, by the following equations:

$$\frac{Q_t}{Q_p} = \frac{1}{\eta_{hp}} \tag{2.3}$$

$$\frac{H_t}{H_p} = \frac{1}{\eta_{hp}^2}$$

2.9.1.3 Stepanoff Method

Stepanoff [30] also proposed a method that depends on the pump efficiency, which is based on the theoretical work of Engel. Stepanoff equations are given by:

$$\frac{Q_t}{Q_p} = \frac{1}{\sqrt{\eta_p}} \tag{2.4}$$

$$\frac{H_t}{H_p} = \frac{1}{\eta_p} \tag{2.5}$$

2.9.1.4 Sharma Method

Sharma [31] has developed a prediction method that also uses ratios dependent on pump efficiency. He uses initial assumptions, as in the method presented by Child and uses another of the equations developed by Engel. Incorporating both methods he has developed the following equations:

$$\frac{Q_t}{Q_p} = \frac{1}{\eta_p^{0.8}} \tag{2.6}$$

$$\frac{H_t}{H_p} = \frac{1}{\eta_p^{1.2}} \tag{2.7}$$

2.9.1.5 Alatorre-Frenk Method

The method presented by Alatorre-Frenk and Thomas [32] is based on fitting equations to a limited number of PAT data. The equations are again based on the pump efficiency, and can be expressed in the form:

$$\frac{H_t}{H_p} = \frac{1}{0.85\eta_p^5 + 0.385} \tag{2.8}$$

$$\frac{Q_t}{Q_p} = \frac{0.85\eta_p^5 + 0.385}{2\eta_p^{9.5} + 0.205} \tag{2.9}$$

2.9.1.6 Schmiedl Method

The turbine prediction method presented by Schmiedl [24] uses an estimated value of hydraulic efficiency. The equations for the turbine best efficiency point are:

$$\frac{Q_t}{Q_p} = -1.4 + \frac{2.5}{\eta_{hp}} \tag{2.10}$$

$$\frac{H_t}{H_p} = -1.5 + \frac{2.4}{\eta_{hp}^2} \tag{2.11}$$

2.9.2 Pump As Turbine Performance Prediction Using Specific Speed

Several authors have proposed equations that rate the head and flow ratios for pump and turbine operation to the pump or turbine specific speed. Some of them are discussed below.

2.9.2.1 Grover Method

The formula given by Grover [33] is as follows:

$$\frac{H_t}{H_p} = 2.693 - 0.0229\eta_{qt} \tag{2.12}$$

$$\frac{Q_t}{Q_p} = 2.379 - 0.0264\eta_{qt} \tag{2.13}$$

These equations are restricted in their application to specific speeds in the range $10 < \eta_{qt} < 50$.

2.9.2.2 Hergt Method

The formula given by Hergt [24] is as follows:

$$\frac{H_t}{H_p} = 1.3 - \frac{6}{\eta_{qt} - 3} \tag{2.14}$$

$$\frac{Q_t}{Q_p} = 1.3 - \frac{1.6}{\eta_{qt} - 5} \tag{2.15}$$

2.9.3 Pump As Turbine Performance Prediction Using Pump Geometry

A method, presented by Burton and Williams [34], uses detailed knowledge of the pump geometry to predict the turbine performance. The basis for this method is the area ratio analysis, in which the conditions for optimum operation (as a pump) are defined by the matching of the outlet flow from the impeller and the inlet flow to the volute. This method requires more information than is generally available from the manufacturer, and is best implemented through a computer programme. However, the method has been shown to give quite accurate predictions for pumps of different size and shape [35]. The area ratio method has also been used to predict differences in efficiency between pump and turbine operation. In some cases, turbine maximum efficiency may be greater than pump maximum efficiency. The method can also be used to determine adjustments or simple modifications to improve the match between turbine and site.

2.10 BEST EFFICIENCY POINT OF PUMP USED AS TURBINE

The head H_p of a pump is determined by the Euler equation for pump operation:

$$H_p = \frac{\eta_h(u_2c_{u2} - u_1c_{u1})}{g} \tag{2.16}$$

where u and c_u are the peripheral and tangential velocities of the rotor and the flow, respectively, and η_h is the hydraulic efficiency. The subscripts *1* and *2* denote the locations, which act as inlet and outlet of the rotor blading, respectively, in the pumping mode. In the pumping mode, there is normally no pre-rotation, i.e. $c_{u1} = 0$, for operation at or near the point of best efficiency. At the outlet of pump impellers in pumping mode, the finite number of blades and the vanishing local blade loading results in a deviation of the flow angle from the blade angle which reduces c_{u2} and, therefore, also the head. If the angle deviation at the outlet in pumping mode is neglected, it would be desirable to have in the turbine mode of a centrifugal pump the same velocity triangles (but reversed vector directions) at inlet and outlet as for the best efficiency point in pump operation. This flow condition would then lead to shock less inflow and no-swirl outflow and, thereby, to minimum losses and maximum hydraulic efficiency. To select a pump, which is suitable as a turbine for certain operating conditions, one must know the relationship between the values of head and flow rate in turbine mode and in pumping mode at the corresponding points of best efficiency. The respective values of the head, $H_{p,bep}$ and $H_{t,bep}$ result from the Euler equation:

$$H_{p,bep} = \frac{\eta_{h,p}(u_{2p}c_{u2p} - u_{1p}c_{u1p})}{g} \tag{2.17}$$

$$H_{t,bep} = \frac{(u_{2t}c_{u2t} - u_{1t}c_{u1t})}{\eta_{h,t}g} \tag{2.18}$$

If the velocity triangles are congruent, the equations can be combined as:

$$H_{p,bep} = H_{t,bep}\eta_{h,t}\eta_{h,p} \tag{2.19}$$

Obviously, there must be a higher nominal head in turbine mode than in pumping mode at the same rotational speed.

Another important value to determine is the ratio of flow rates at the respective points of best efficiency, $\frac{Q_{t,bep}}{Q_{p,bep}}$

In Fig. 2.11, the differing velocity triangles are shown for the best efficiency points in pumping mode and in turbine mode of operation. At the outer diameter (location 2), the absolute flow angle, α_3, is approximately the same for both modes of operation in respect to the shape of the volute. While the flow in pumping mode leaves the impeller with a relative flow angle, β_{3p}, which is smaller than the blade angle, β_2, (effect of angle deviation), the relative flow angle in turbine mode should be as close as possible to the blade angle to avoid shock losses, i.e. $\beta_2 = \beta_{3t}$. To accomplish this, the radial velocity c_{3r} must be higher in turbine mode than in pumping mode; thus also the flow rate.

At the inner diameter (location *1*), the absolute flow angle, α_0, is approximately the same (90°) for pumping and turbine mode from the condition of no pre-rotation and no-swirl outflow, respectively. Pump impellers are normally designed to have a blade angle, β_1, which is somewhat larger than the relative flow angle, β_{0P}, at best efficiency condition to improve the suction behavior. In turbine mode, the relative flow angle, β_{0T}, is larger than the blade angle, β_1, caused by the effect of angle deviation at outlet. On the other hand, the flow rate in turbine mode should not be much higher than the flow rate in pumping mode, because a swirl will occur at the outlet, which will decrease the efficiency [8].

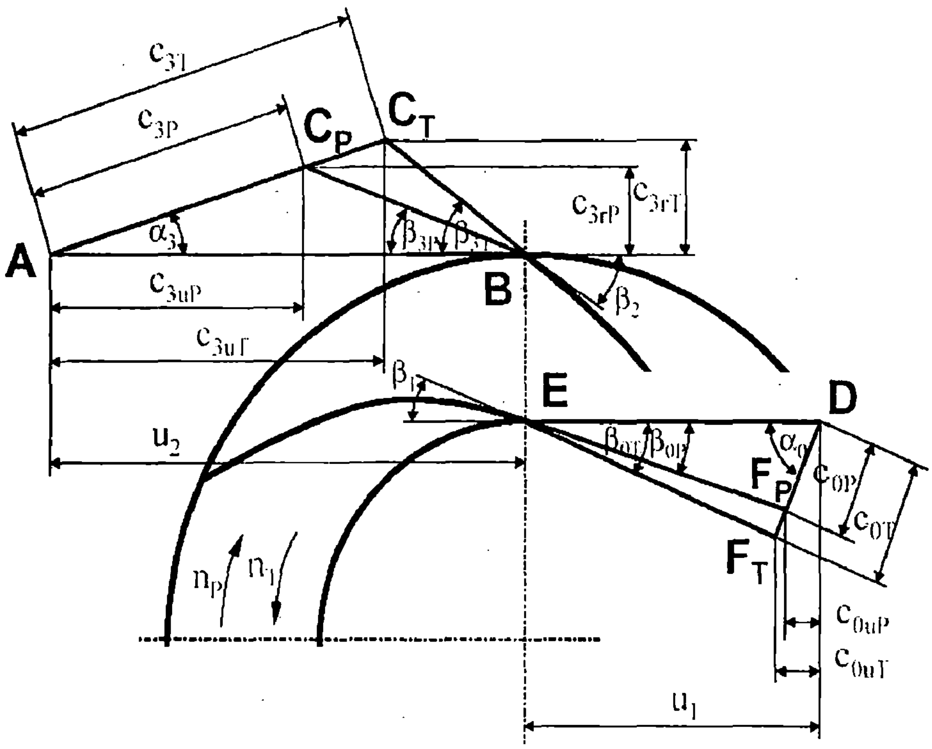


Fig. 2.11 Velocity Triangles for Best Efficiency Point in Pump & Turbine Mode [8]

From these reasons, a pump in reverse mode has its best efficiency point at higher flow rates compared to the pumping mode. This can also explain why the efficiency remains nearly constant or – especially for low specific speed pumps – is even higher in the turbine mode than in the pumping mode. The relative influence of the secondary losses (leakage and disc friction losses) on the total efficiency is lower due to the higher shaft power in turbine mode [8].

2.11 SELECTION OF PUMP AS TURBINE

In the most common situation, there is a liquid stream with fixed head and flow rate, and an application that requires a fixed rpm; these are the turbine design conditions. We want to select a pump with a turbine BEP at these conditions. With performance curves such as Fig. 2.5, we convert turbine design conditions to pump design conditions. Then, we select from a manufacturer's catalog a model that has its pump BEP at those values.

The most common error in pump selection is using the turbine design conditions in choosing a pump from a catalog. Because catalog performance curves describe pump duty, not turbine duty, the result is an oversized unit that fails to work properly. For the unit shown in Fig. 2.5, the turbine BEP capacity is 202% of the pump BEP capacity, so the pump would be twice the size needed for the job in this case [12].

Instead of performance curves, we can use a manufacturer's conversion factors that relate turbine BEP performance with pump BEP performance. As specific speed N_S varies from 509 to 2,800, these conversion factors generally vary as follows: from 2.2 to 1.1 for C_Q and C_H; from 0.92 to 0.99 for C_E. Applying these to the turbine design conditions yields the pump design conditions [12].

2.11.1 Method for Selection of Pump As Turbine

The following procedure [23] may be adopted for selection of a pump to be used as turbine for a given site data:

(i) Arrive at the values of rated head (H_t in m) and rated discharge (Q_t in m^3/s) from a given site data.

(ii) The above data yields the value of available hydraulic power $P_t = QgH$ in kW. (2.20)

Keeping this in view select the value of speed of generator and also the speed of pump.

(iii) Find the value of specific speed for turbine data by using the following formula:

$$N_{st} = \frac{N_t \sqrt{Q_t}}{H_t^{3/4}} \tag{2.21}$$

(iv) Examine whether it is desirable to use double suction pump. This should be done wherever possible because shaft of double suction pump experiences negligible thrust. Neglecting the effect of efficiency and assuming $\eta_t = \eta_p$ we get:

$$N_{sp} = N_{st} \text{ for single suction pump} \tag{2.22}$$

$$= \frac{N_{st}}{\sqrt{2}} \text{ for double suction pump} \tag{2.23}$$

(v) Find the value of conversion factors from test theoretical curves, as shown in Fig. 2.12, Fig. 2.13 & Fig. 2.14 and test curves, as shown in Fig. 2.15 and find the values of Q_p and H_p.

(vi) Find from pump catalogues whether pump for chosen values of Q_p, H_p and η_p is available. If not select suitable pump speed for which a pump is available to match desired Q_p and H_p. Find new specific speed and check the values of conversion factors for the same. With little trial and error, it will be possible to select a pump.

(vii) In all cases recheck all your values for selected pump and find out the out put power expected.

2.12 CAVITATION PREVENTION IN PUMP USED AS TURBINE

Just as pumping requires a minimum net positive suction head, turbine duty requires a net positive exhaust head. The relationship between total required exhaust head (*TREH*) and turbine head, H_t (per stage) is the cavitation constant σ_r :

$$\sigma_r = \frac{TREH}{H_t} \tag{2.24}$$

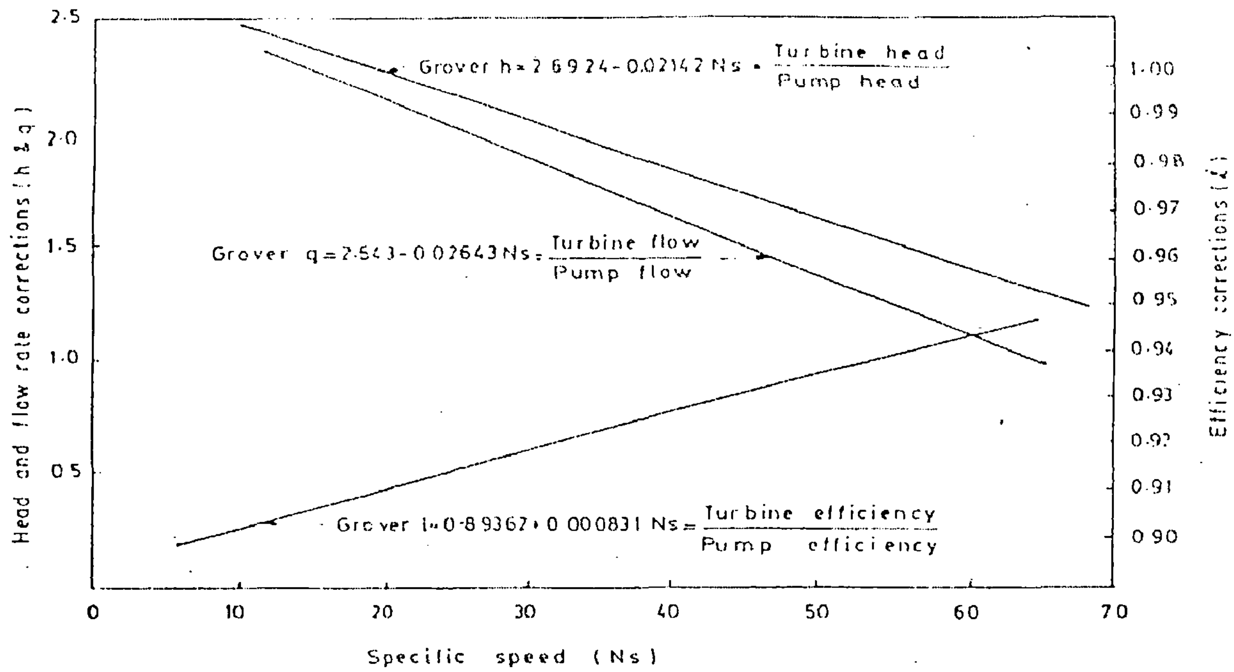


Fig. 2.12 Conversion Coefficients for Turbine Pump Parameter Transformations as a Function of Specific Speed [23]

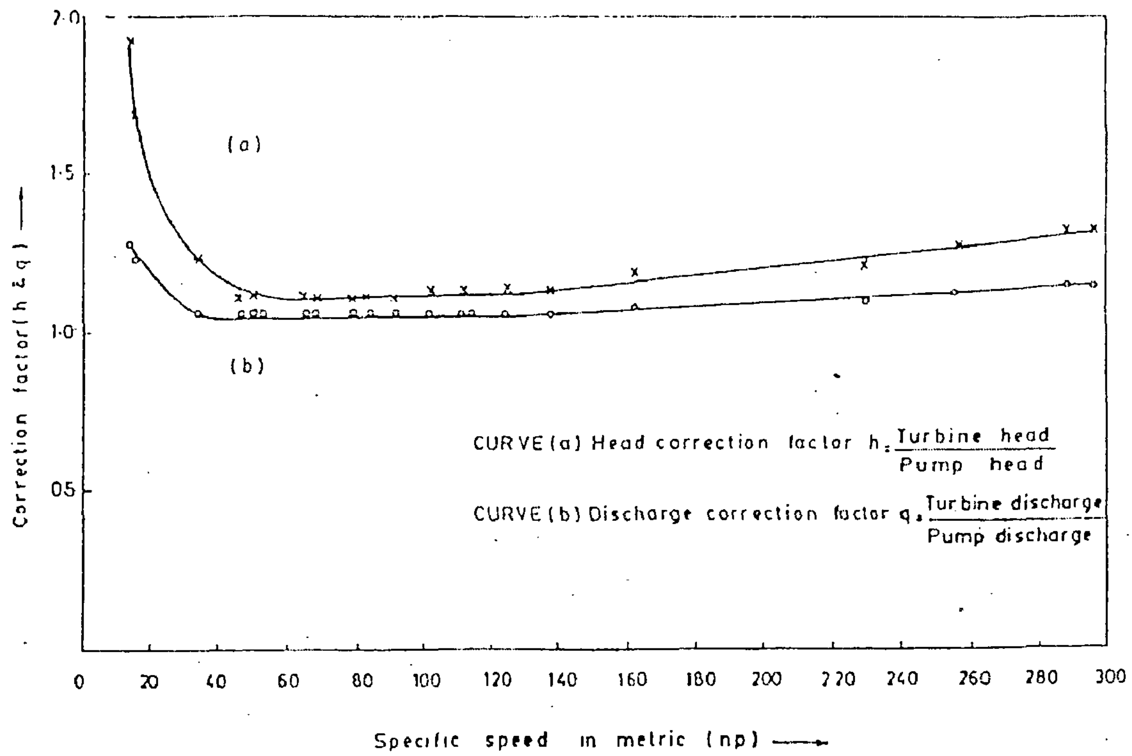


Fig. 2.13 Head and Discharge Correction Factor by Stepanoff Equations [23]

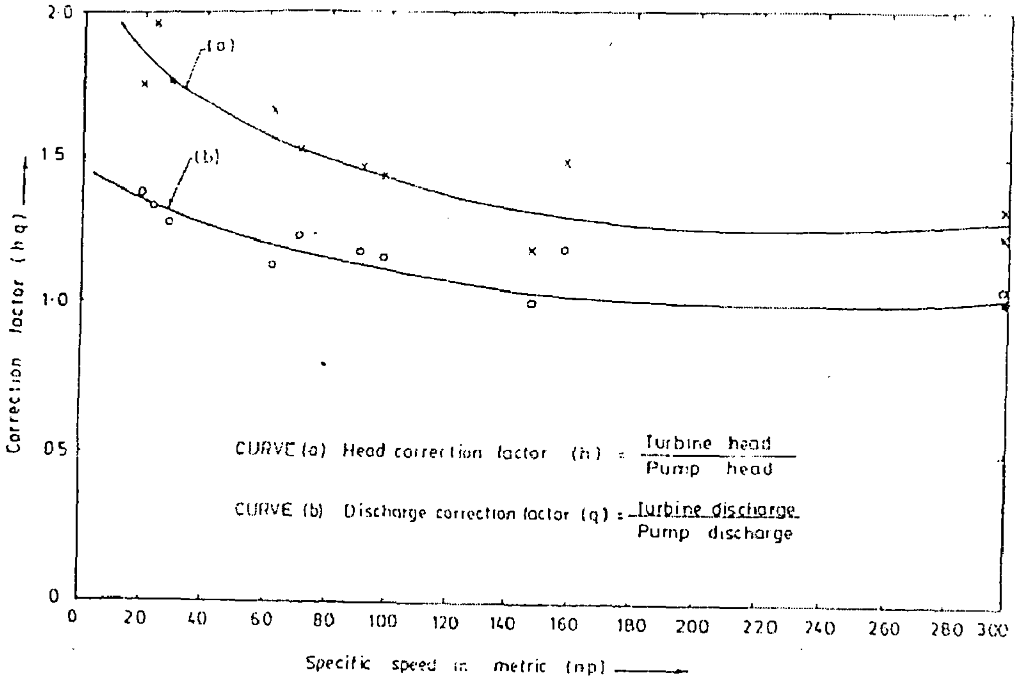


Fig. 2.14 Head and Discharge Correction Factor Theoretically Calculated [23]

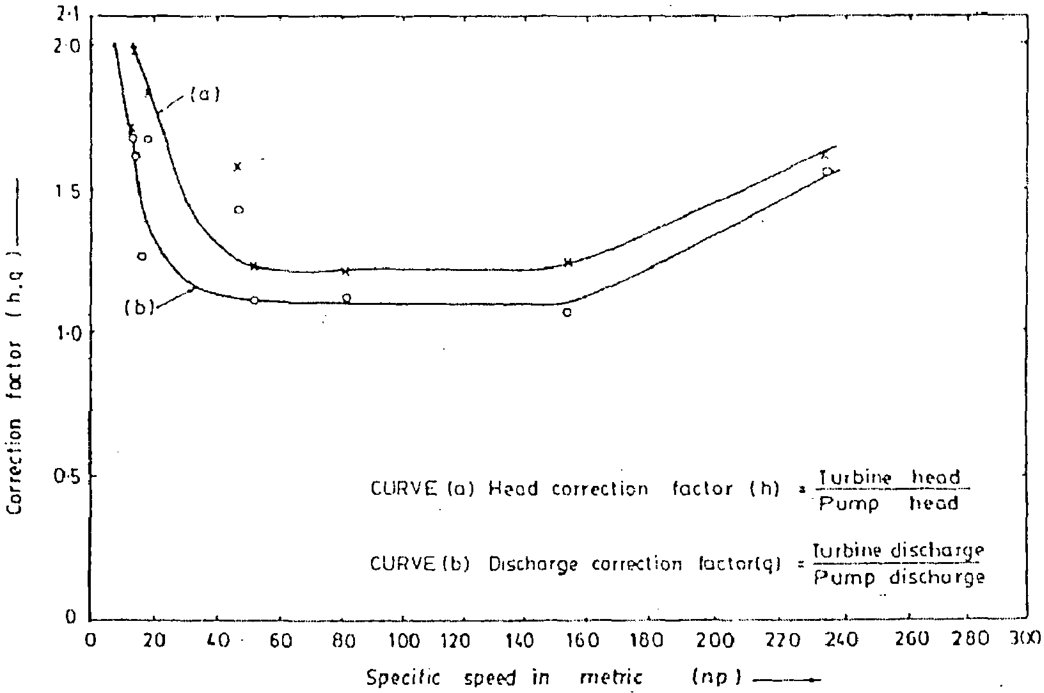


Fig. 2.15 Head and Discharge correction factor from Test Data [23]

Fig. 2.16 shows a graph of cavitation constant (σ_r) vs. specific speed (N_s) for hydraulic turbines. Though a pump used, as a turbine will not have exactly the same relationship, this curve provides a good estimate of σ_r for turbine duty. To prevent cavitation, the total available exhaust head (*TAEH*) must be greater than the *TREH* [12].

2.13 RUNAWAY PREVENTION IN PUMP USED AS TURBINE

When the speed of the turbine is beyond the normal operating speed, the turbine can become "unloaded" and continue to speed up uncontrollably until it destroys itself. For radial-flow centrifugal pumps operating as turbines, the runaway speed is generally between 120 and 140% of the normal speed. The best way to prevent runaway is to avoid excessive turbine speeds; but the pump used as a turbine can be equipped with an over speed trip that will stop flow when the speed hits a predetermined danger level [12].

In addition to efficiency, cavitation and runaway, we should, of course, also consider pump limitations such as shaft stress, deflection, bearing load and maximum working pressure. If the pump has threaded parts, the threading should be checked-to prevent unscrewing in reverse flow.

Because radial-flow centrifugal pumps are not often used as turbines, application experience is limited. But pumps applied using the principles of this article have performed well as hydraulic turbines. As costs for other forms of power continue to rise, pumps will find more applications in recovering fluid power.

2.14 MODIFICATIONS DONE SO FAR IN PUMP AS TURBINE

Few modifications were reported in adapting the pumps as turbines. The most frequent modification was the addition of a draft tube on the pump/turbine discharge. Other reported modifications to the pumps are as follows: epoxy coating added to the inside surface of the volute to increase the head capacity, rotor strengthened to handle over speed conditions, motor cooling fan motor replaced to allow bi-directional cooling flow to the motor/generator, discharge nozzle epoxy lined, speed increasers added, radical thrust bearings added, and threaded couplings on the shaft pinned for reverse rotation.

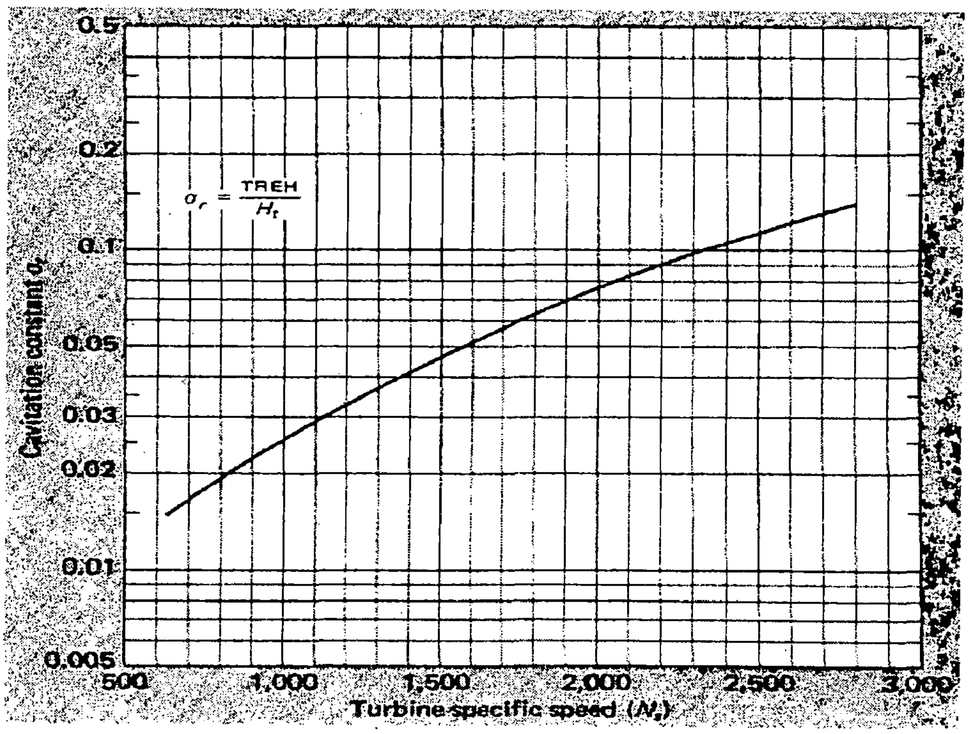


Fig. 2.16 Cavitation Constant for Hydraulic Turbines [12]

2.15 CFD ANALYSIS OF PUMP USED AS TURBINE

CFD simulation has been one of the finest methods to investigate the characteristics of the turbo machines. There have been several experimental investigations of a pump being used as a turbine. But there have been a less number of CFD simulations on PAT and then their comparison with the experimental results. Among them, one of CFD simulation [8] results and their comparison with the experimental ones have been presented in this topic in the form of graphs as shown in Fig. 2.17 to Fig. 2.25.

The investigated pump is a special test pump constructed for studying the maximum attainable efficiency. Its main geometrical and hydraulic data are given in the Table 2.1. The hydraulic data were determined experimentally.

A 3-dimensional model of the pump described in Table 2.1 was generated where the model includes the whole impeller and the spiral casing because a single blade channel model of the impeller alone does not enable to study the circumferential variation of flow caused by the spiral casing. The model does not include the flow field in the spaces between impeller hub/shroud and casing nor the sealing gap. The simulations were performed with the impeller fixed in one position ("frozen rotor"). The grid size was approximately 700.000 cells. The CFD code used was FLUENT 5.0, and the grid was generated by using the code GAMBIT [8].

Table 2.1 Main Data of Centrifugal Pump under CFD Simulation [8]

S.No.	**Parameter**	**Value**
1	Specific Speed (n_s)	12 min^{-1}
2	Flow Rate at best Point ($Q_{b,bep}$)	994 m^3/h
3	Head at Best Efficiency Point ($H_{p,bep}$)	824 m
4	Maximum Pump Efficiency (η_p)	78 %
5	Maximum Inner Efficiency (η_{ip})	79 %
6	Maximum Hydraulic Efficiency (η_{hp})	89 %
7	Nominal Rotational Speed (n)	2000 min^{-1}
8	Impeller Diameter, Inlet (D_1)	100 mm
9	Impeller Diameter, Outlet (D_2)	350 mm
10	Impeller width, Outlet (b_1)	15 mm
11	Blade Angle, Outlet (β_2)	28°
12	Blade Angle, Outlet (β_1)	32.3° – 17.9°

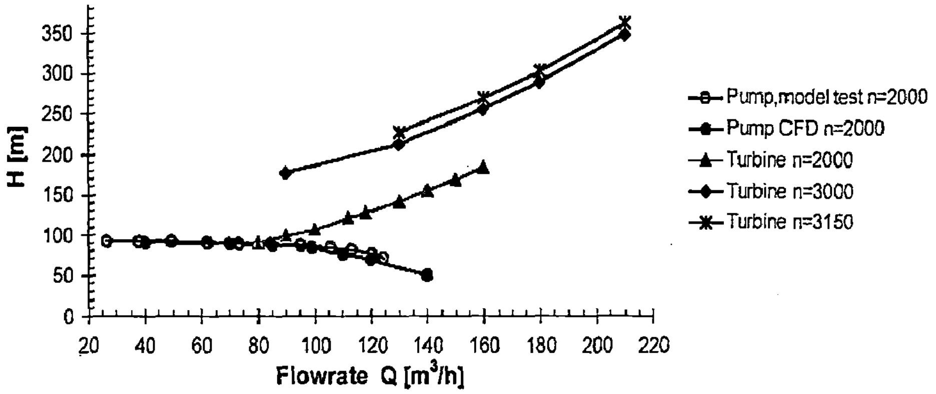


Fig. 2.17 Head (H) vs. Flow Rate (Q) for Pumping and Turbine Mode of Operation using CFD Simulation [8]

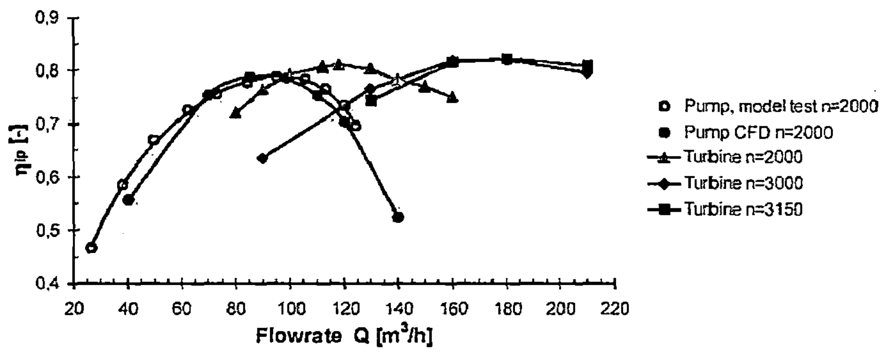


Fig. 2.18 Maximum Inner Efficiency (η_{Ip}) vs. Flow Rate (Q) for Pumping and Turbine Mode of Operation using CFD Simulation [8]

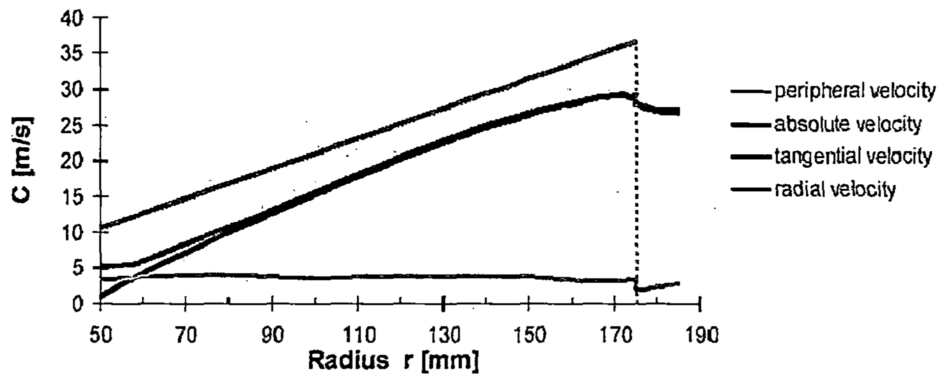


Fig. 2.19 Absolute Flow Velocity (C) in the Rotor vs. Radial Position (R) of PAT using CFD Simulation [8]

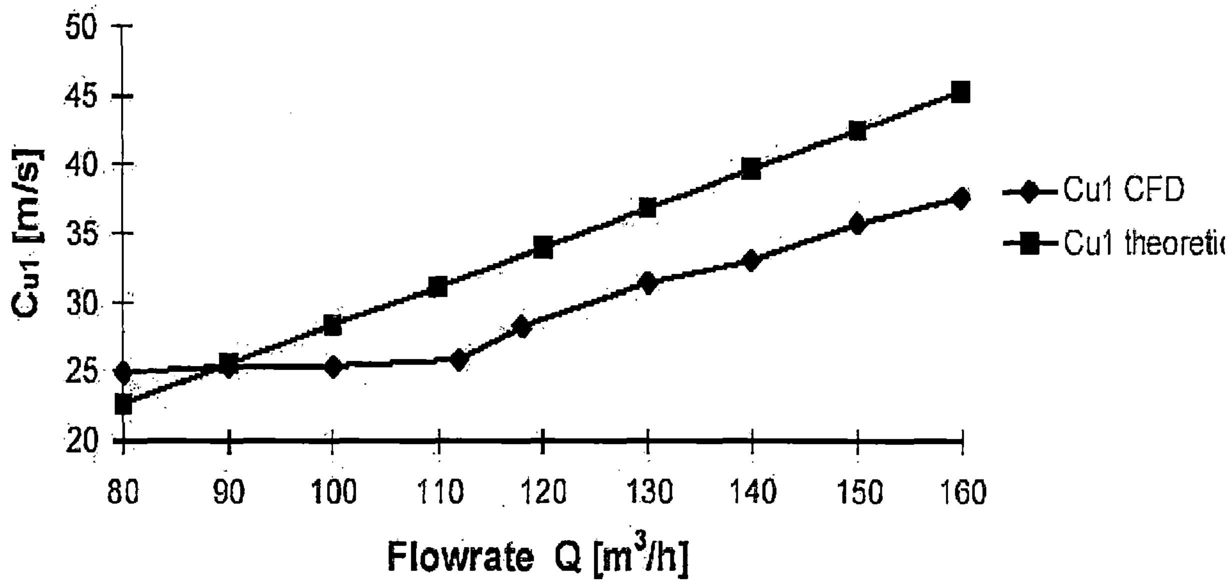


Fig. 2.20 Tangential Velocity at Inlet (C_{u1}) of the Rotor vs. Flow Rate (Q) of PAT using CFD Simulation [8]

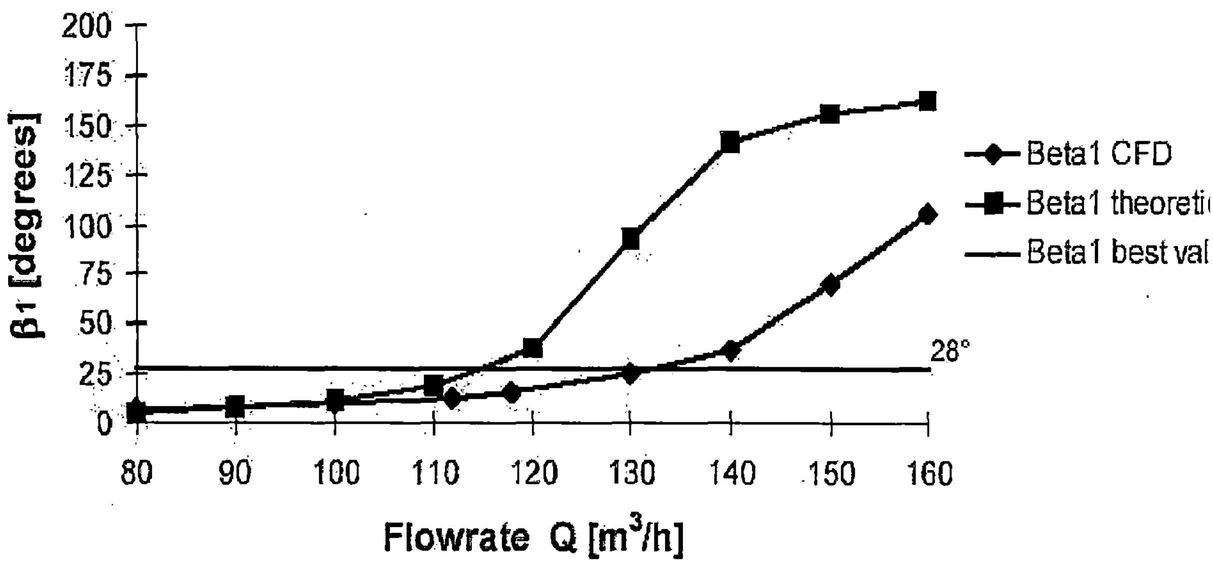


Fig. 2.21 Blade Inlet Angle (β_1) vs. Flow Rate (Q) of PAT using CFD Simulation [8]

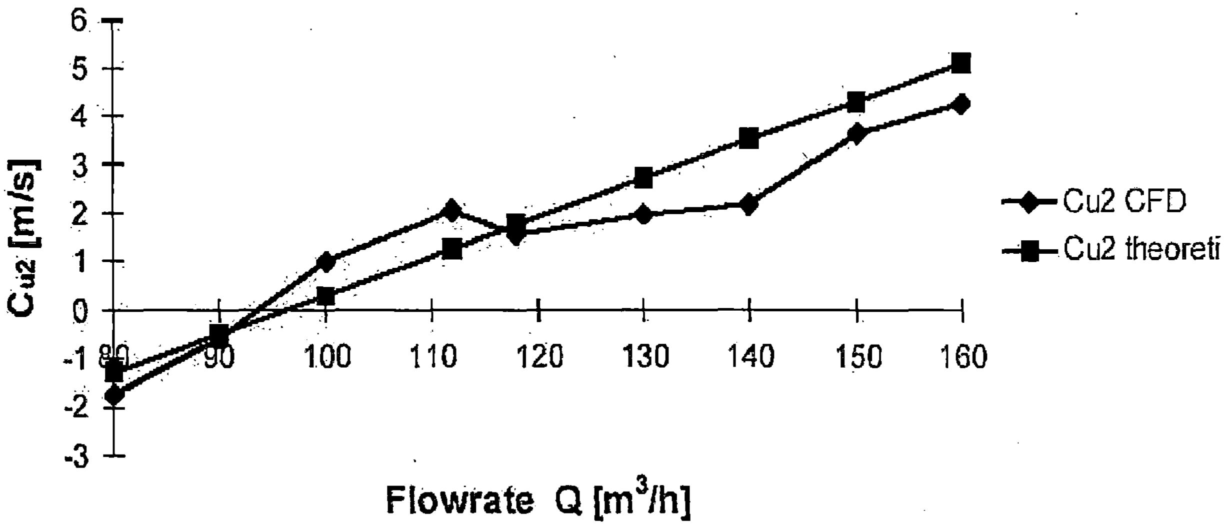


Fig. 2.22 Tangential Velocity at Outlet (C_{u2}) of the Rotor vs. Flow Rate (Q) of PAT using CFD Simulation [8]

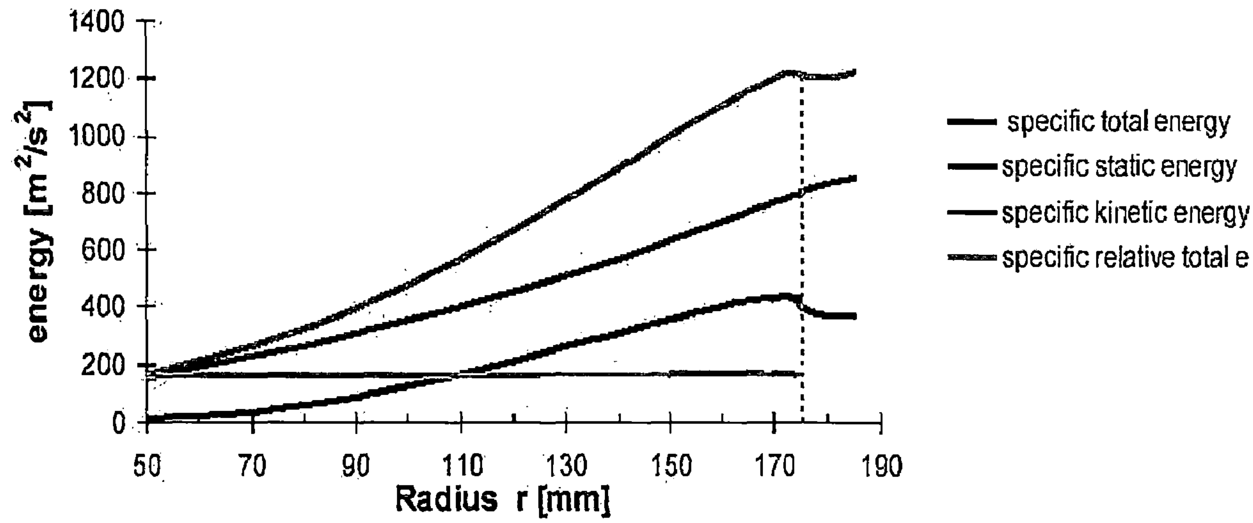


Fig. 2.23 Specific Energy within Rotor vs. Radial Position of PAT using CFD Simulation [8]

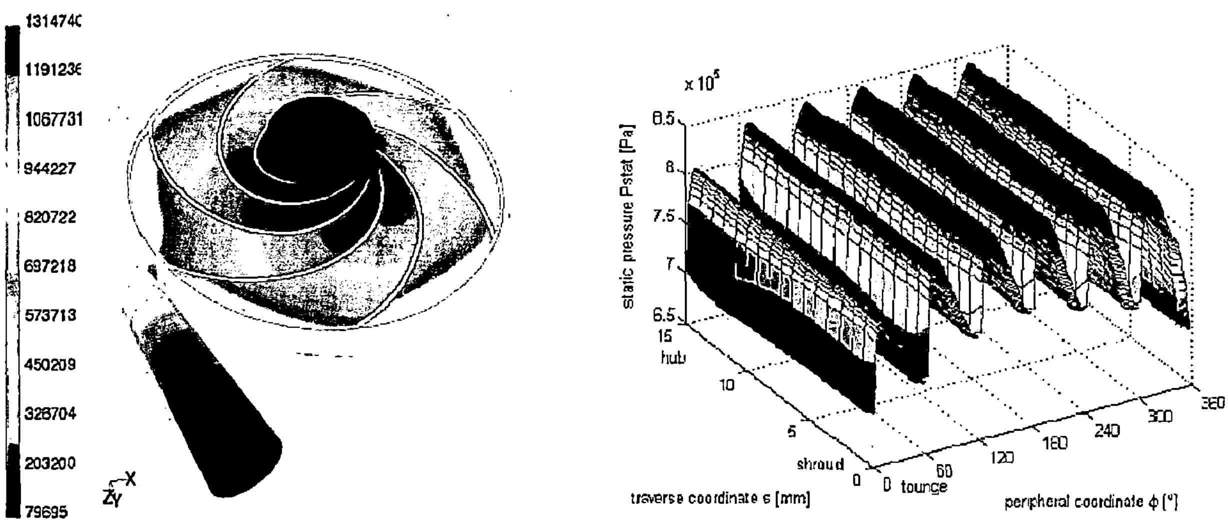


Fig. 2.24 Static Pressure in the Spiral Casing (Left) and Static Pressure Distribution at the Rotor Inlet (Right) of PAT using CFD Simulation [8]

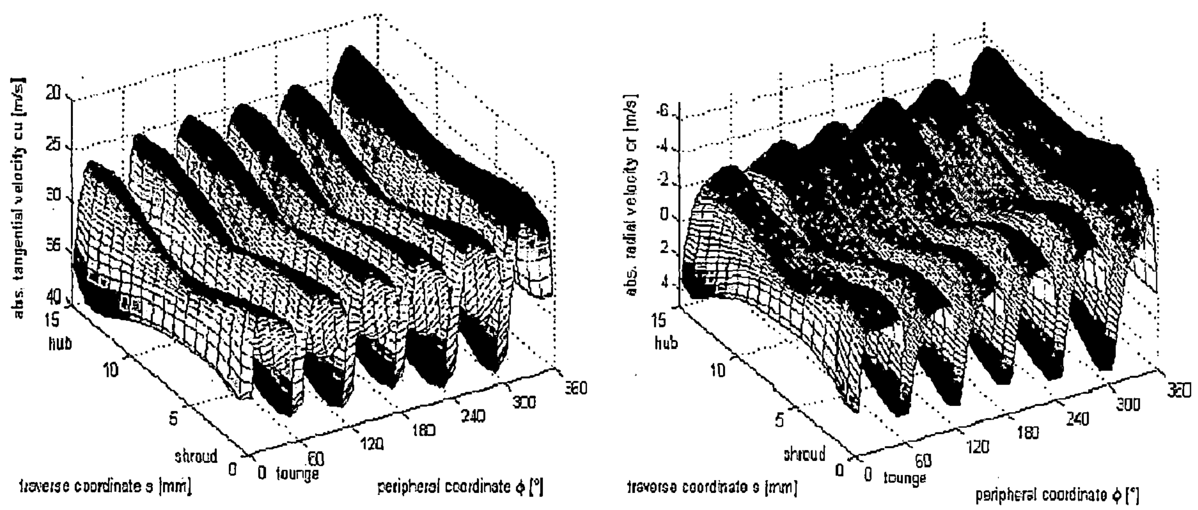


Fig. 2.25 Absolute Tangential Velocity Component at Rotor Inlet (Left) and Absolute Radial Velocity Component at Rotor Inlet (Right) of PAT using CFD Simulation [8]

MODIFICATION IN PUMP FOR TURBINE

3.1 GENERAL

It has already been discussed that there is no question whether a pump can be used as turbine. However, its selection to match the site condition is very difficult. Further, its part load efficiency is very poor. Use of PAT can be recommended for fixed flow conditions as it has no flow regulating mechanism. Thus the major problem of PAT is its poor part load efficiency which can be improved to a greater extend by modifying the centrifugal pump. The modification involves providing movable guide vanes around the pump impeller which can be moved according to the flow/load condition and thus a constant efficiency can be maintained over a wider range of discharge.

Under this chapter of the dissertation, details of guide vane design, its fabrication and fitting the same with the selected pump are given.

3.2 CENTRIFUGAL PUMP FOR MODIFICATION

A commercially available pump was selected for the modification, whose specifications are given below in Table 3.1.

Table 3.1 Technical Specification of Centrifugal Pump Considered for Modification

S.No.	Parameters	Details
1	Make	KBL, Dewas
2	Size	125 × 125
3	Impeller Diameter	260 mm
4	Head	21.5 m
5	Discharge	42.5 l/s
6	BHP	15.7
7	RPM	1500
8	Peak Efficiency	70 %
9	Head Range	14 – 24 m
10	Capacity Range	56.5 – 32 l/s

3.3 DESIGN OF GUIDE VANES

The diameter along which the guide vanes are to be provided should be greater than impeller diameter because the guide vanes need some space for their movement.

Since pump impeller diameter, D_I is equal to 260 mm, therefore the diameter of circle along which guide vane are to be provided, D_G is taken as 300 mm and circumference of circle along which guide vane are to be provided, ($C_G = \pi D_G$) calculated as 942 mm.

The location for fixing the guide vanes is calculated as follows. First of all a heptagon inscribed in a circle of diameter equal to D_G is drawn. After then the seven edges of the heptagon are marked as the location for fixing the guide vanes. The edges of the heptagon are adjusted (by rotating the heptagon about its centre) so as to adjust maximum number of guide vanes. As the number of slots (vanes) in pump impeller are provided as 7, therefore a polygon having seven sides (heptagon) has been drawn and hence the number of guide vanes which should be provided along the pump impeller should be equal to 7. But since there is no space for the movement of the 7th guide vane, therefore 6 guide vanes are to be provided at their respective locations and the 7th location is left as it is.

The length of one guide vane is determined as 130 mm and since pump impeller width is equal to 25 mm, therefore width of the guide vane is taken as 25 mm.

For obtaining the surface profile of the guide vane, software Design Foil (R 5.32) has been used, as shown in Fig. 3.1.

The detailed drawing of the designed guide vane has been shown in Fig. 3.2 and the inner view of the six guide vanes fixed around the pump impeller has been shown as a detailed drawing in Fig. 3.3.

The guide vanes can be adjusted through a rod, liver and link construction. The rod is connected to a handle. By rotating the handle by hands, all the six guide vanes move through the same angle simultaneously. The view of the regulating mechanism of the guide vane has been shown in a detailed drawing in Fig. 3.4.

Fig. 3.5, Fig. 3.6 & Fig. 3.7 respectively show photographs of the designed guide vane assembly in fully closed position, fully open position and its regulating mechanism.

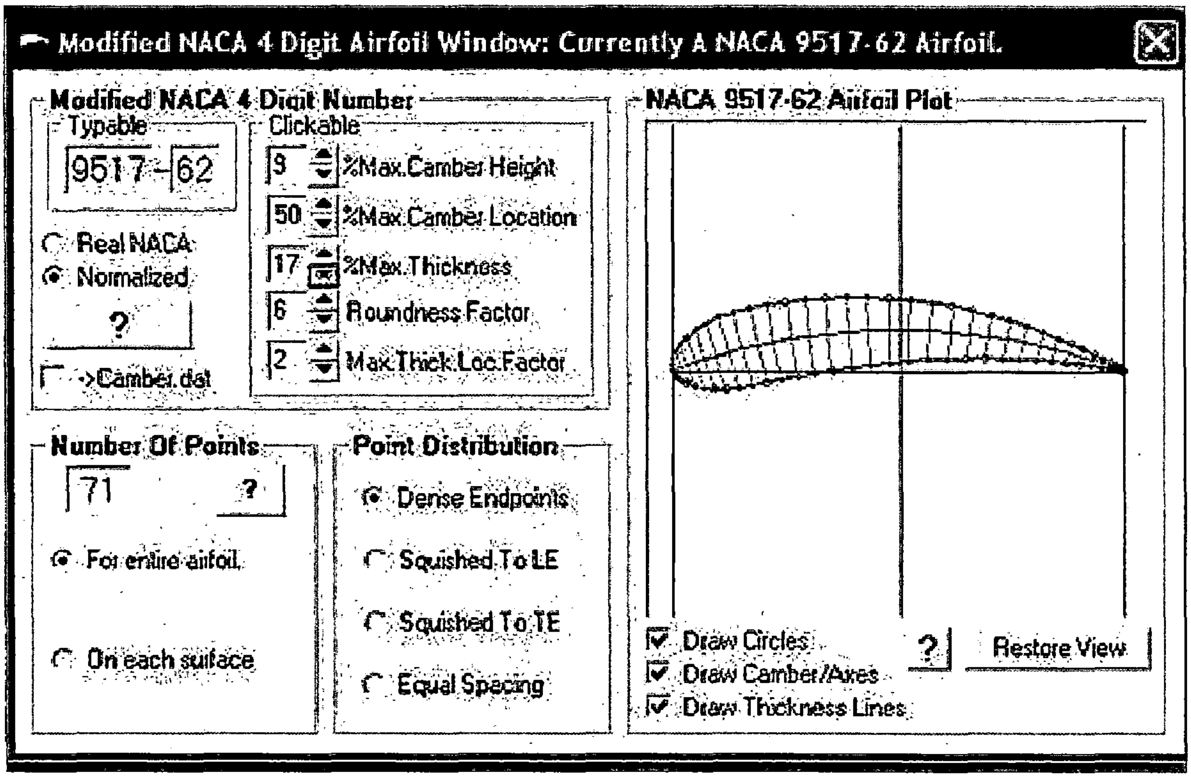
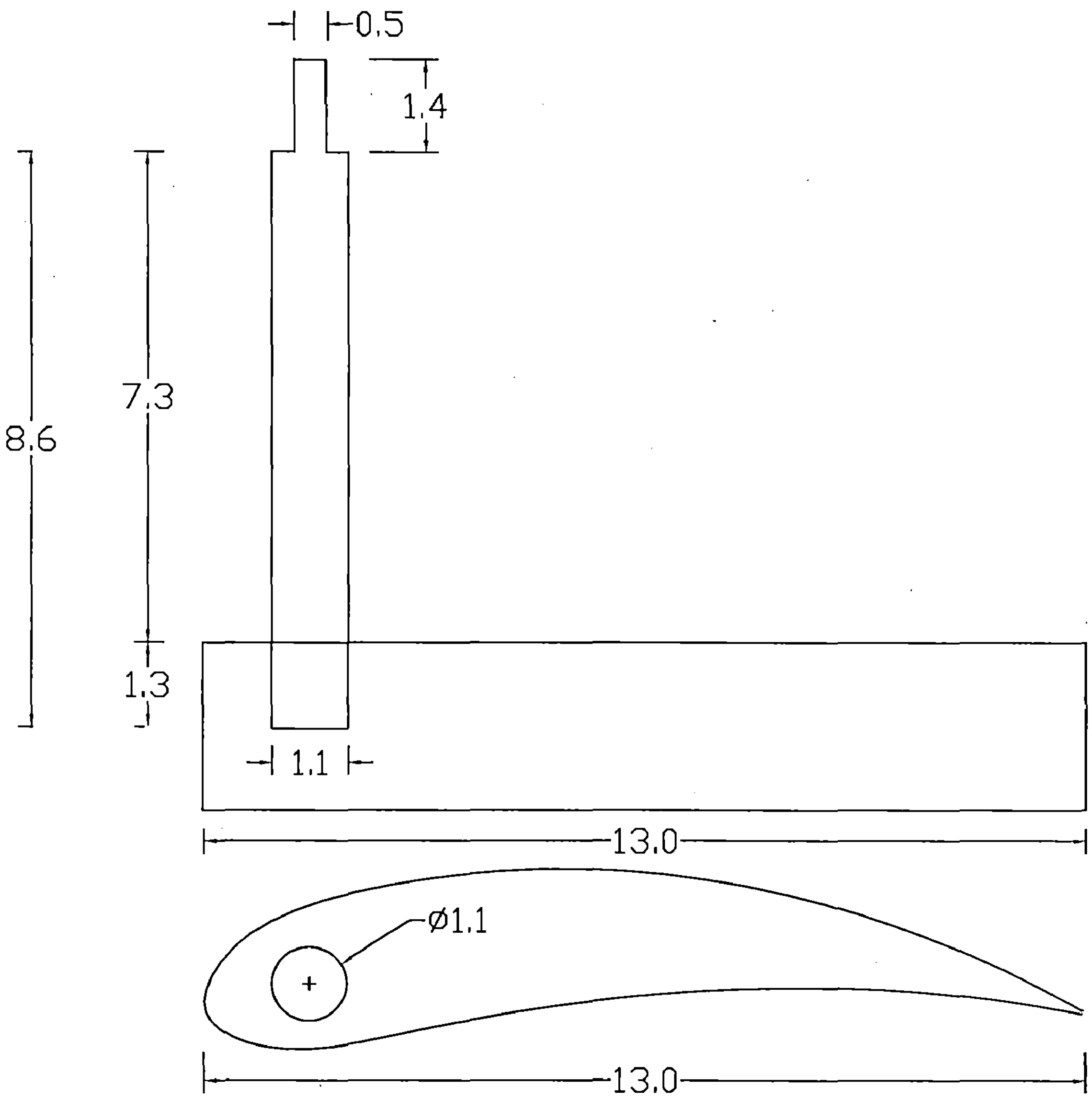


Fig. 3.1 Generation of Guide Vane Surface Profile using Design Foil Software



(all dimensions in cm

Fig. 3.2 Drawing of Designed Guide Vanes .

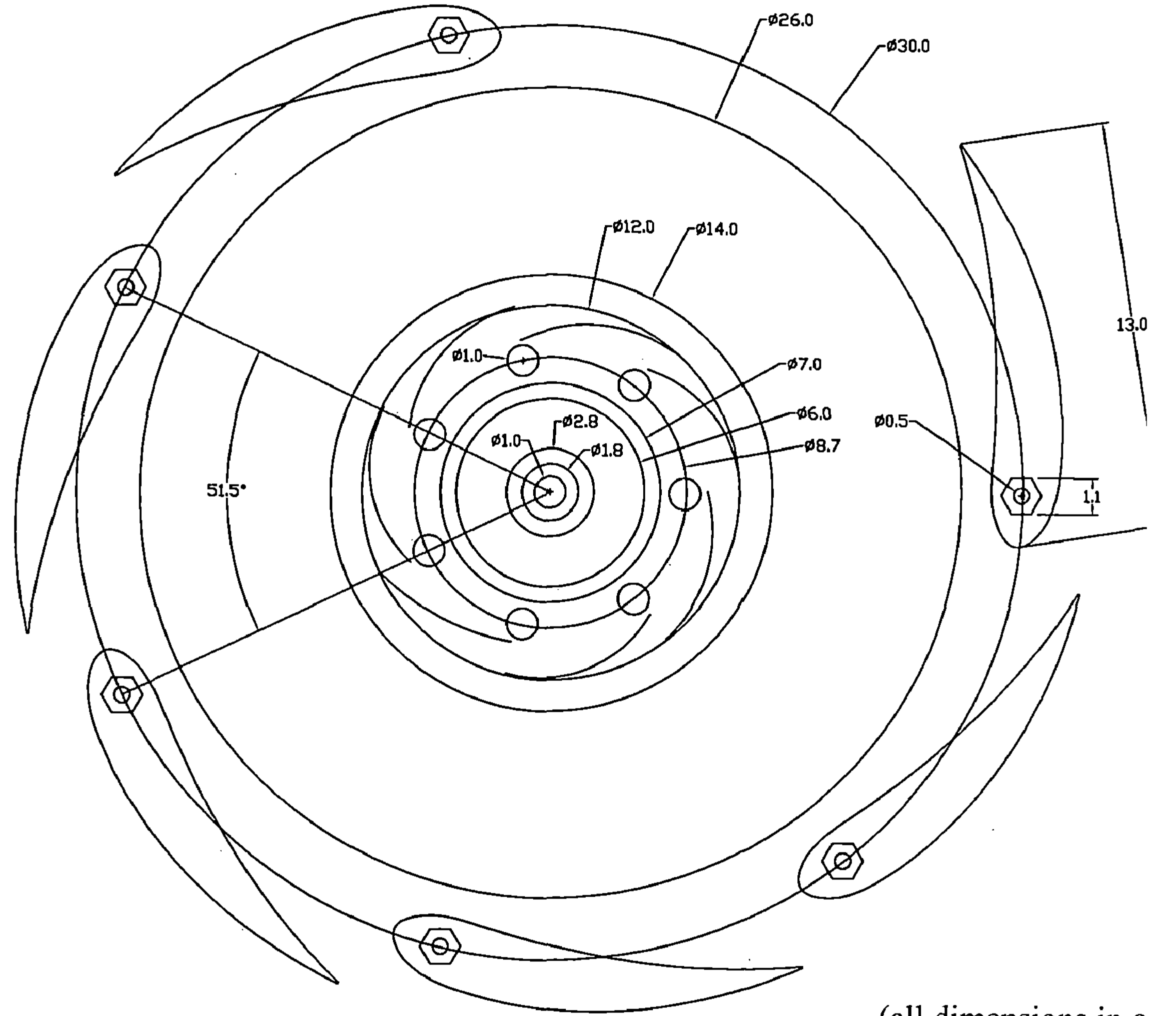


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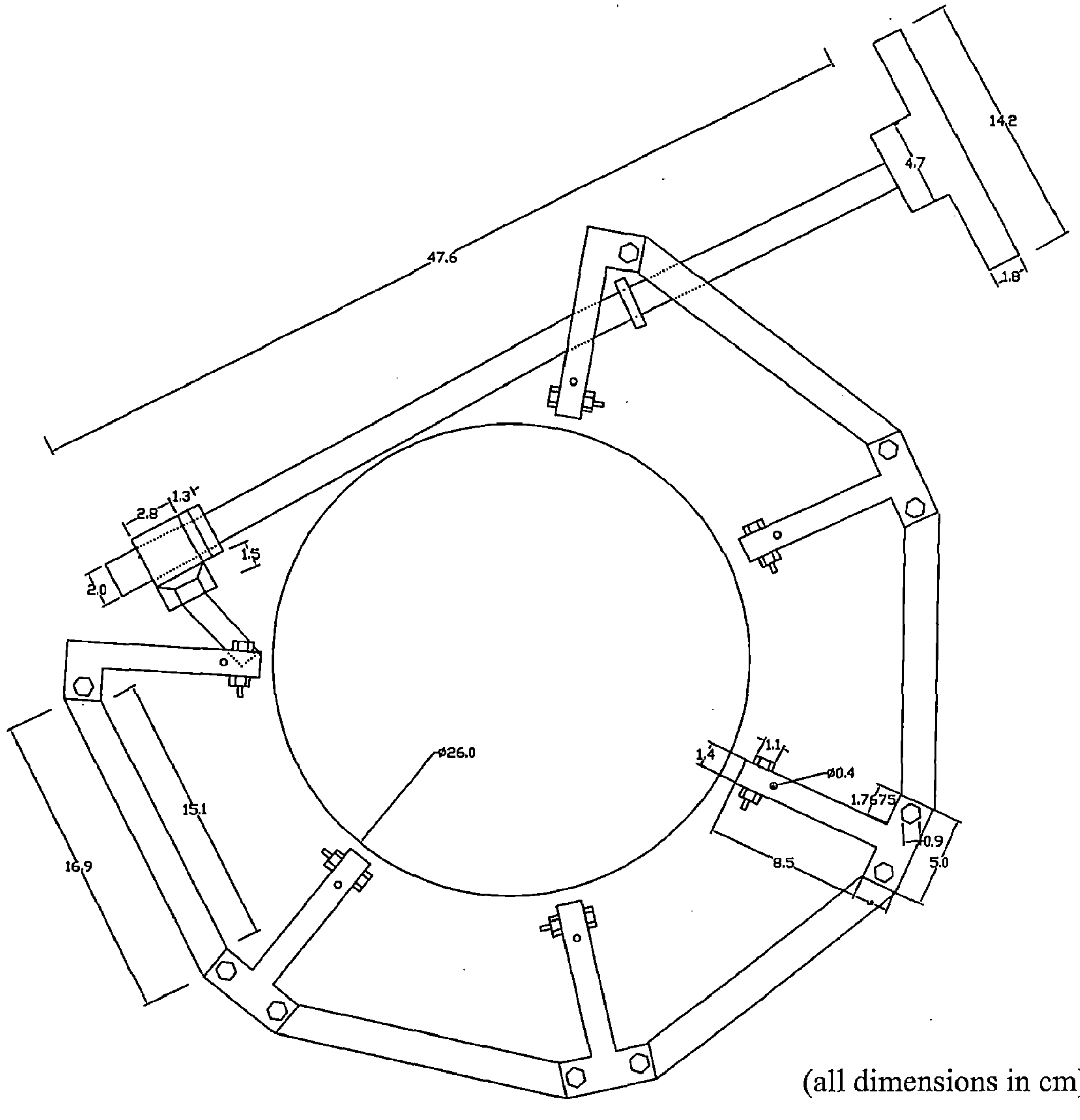
Fig. 3.3 Drawing of Guide Vane Assembly .



(all dimensions in cm)

Fig. 3.4 Drawing of Guide Vane Regulating Mechanism .

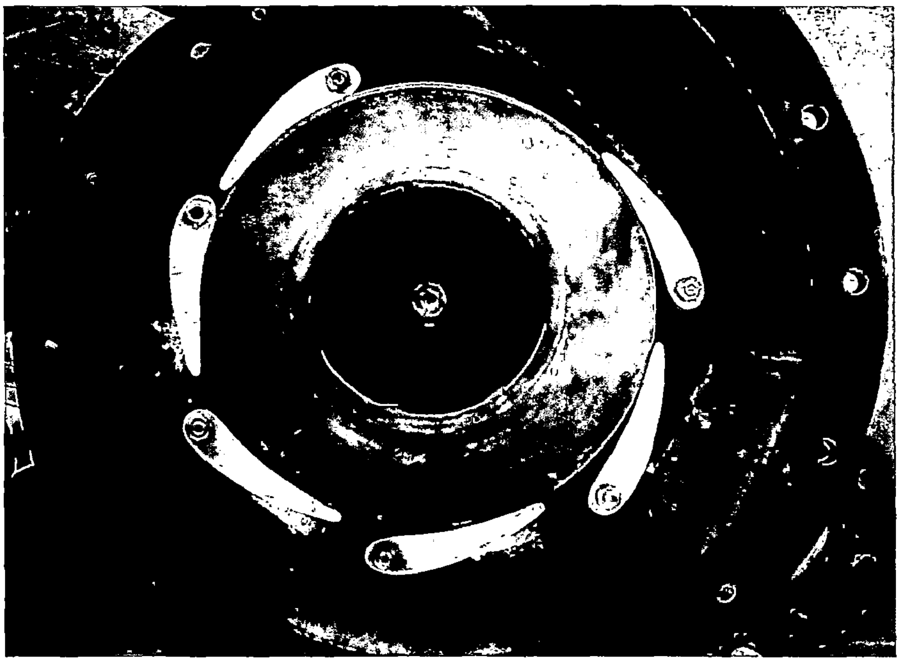


Fig. 3.5 Photograph Showing Guide Vanes at Closed Position

Fig 3.6 Photograph Showing Guide Vanes at Open Position

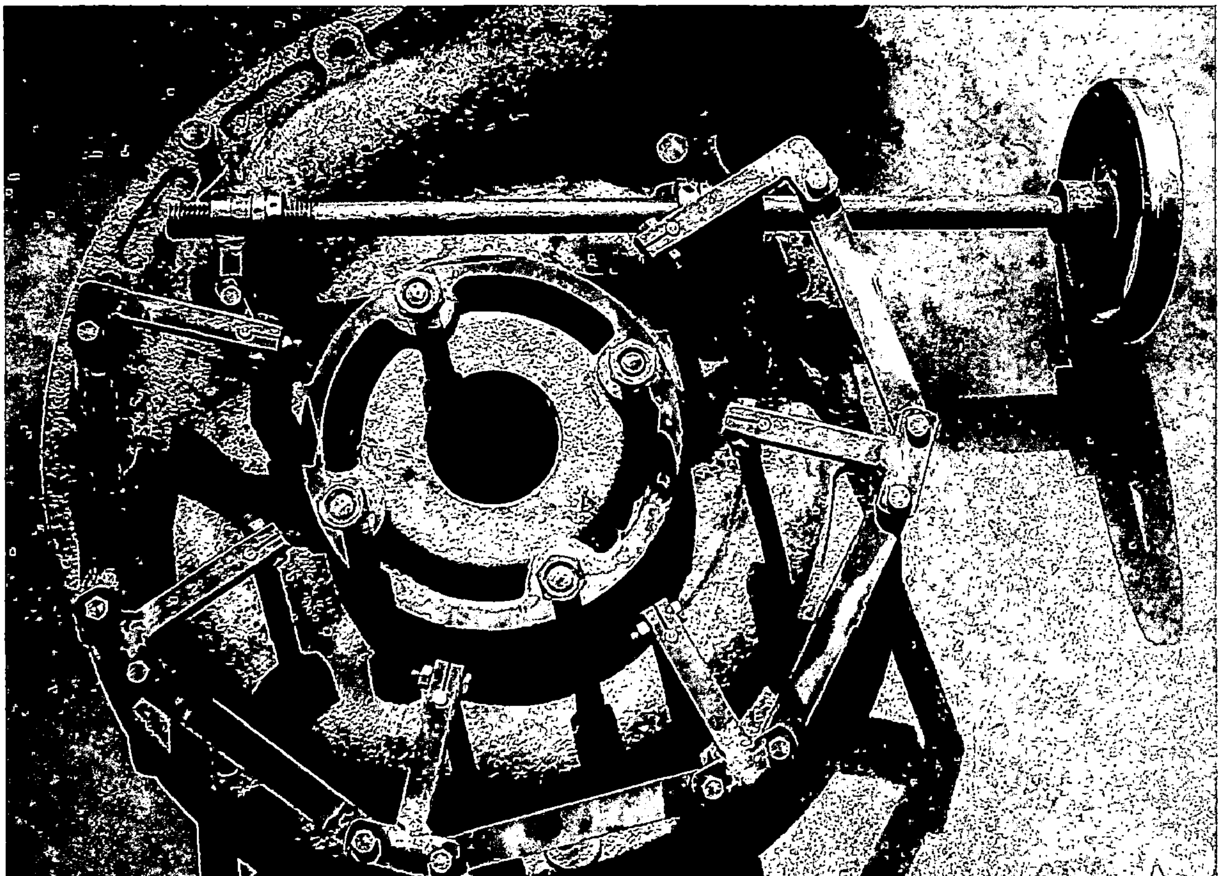


Fig. 3.7 Photograph Showing Regulating Mechanism of Guide Vanes for PAT

EXPERIMENTAL STUDY

4.1 GENERAL

In order to compare the performances of actual PAT with modified PAT, experimental study has been carried out. First of all the actual pump is tested in turbine mode on a testing rig at Alternate Hydro Energy Centre (AHEC) IIT ROORKEE. The details of the pump modified to facilitate the variable flow conditions have been discussed in chapter 3. Further the modified pump has been tested for its operating performance. The results of the pump used as turbine before modification and after modification have been presented in this chapter.

4.2 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. The specification of the mixed flow pump has been shown in Table 4.1. The two mixed flow pumps have been connected to the sump. From the sump the water reaches with high pressure in the centrifugal 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. From there water goes back again to the reservoir from which the two mixed flow pumps again pump the same water back to the PAT. An intelligent pressure transmitter (pressure gauge) has been connected to the PAT inlet so as to measure head. A rectangular weir provided in the channel was used to measure head.

A schematic diagram of the testing rig has been shown in Fig. 4.1. Respectively Fig.4.2 and Fig. 4.3 show the photograph of the testing rig and the experimental setup.

4.3 RANGE OF PARAMETERS MEASURED

Parameters, which have been measured, are given in Table 4.2 with their operating ranges for the pump used as turbine before modification.

Table 4.1 Specification of a Service Pump of 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 l/s
5	Motor	22.5 kW

Table 4.2 Range of Parameters Considered for the Experiment at Study

Parameters	Range for Pump as Turbine
Head (m)	6.0 - 21.5
Discharge (l/s)	22.00 – 53.00
Pump RPM	1000 rpm
Generator RPM	1500 rpm
Power Output (kW)	0 – 4.0

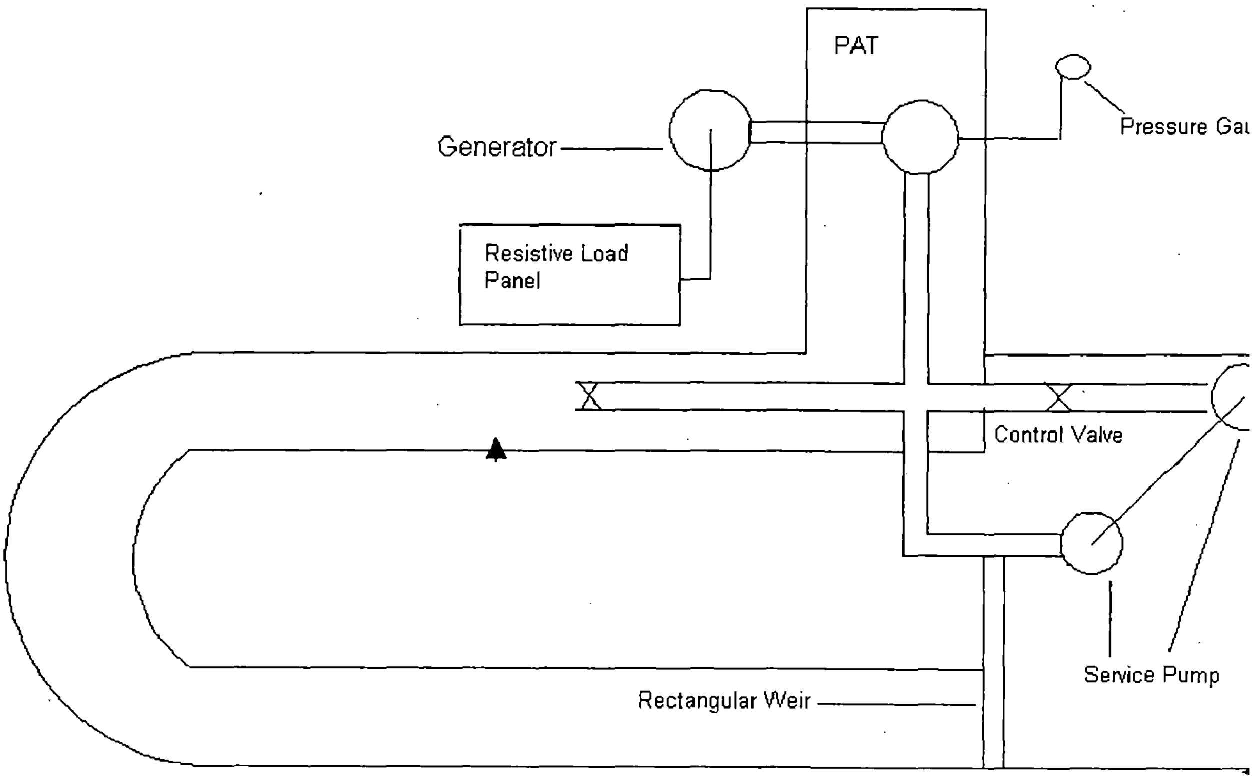


Fig. 4.1 Schematic Diagram of Test Rig and Experimental Setup

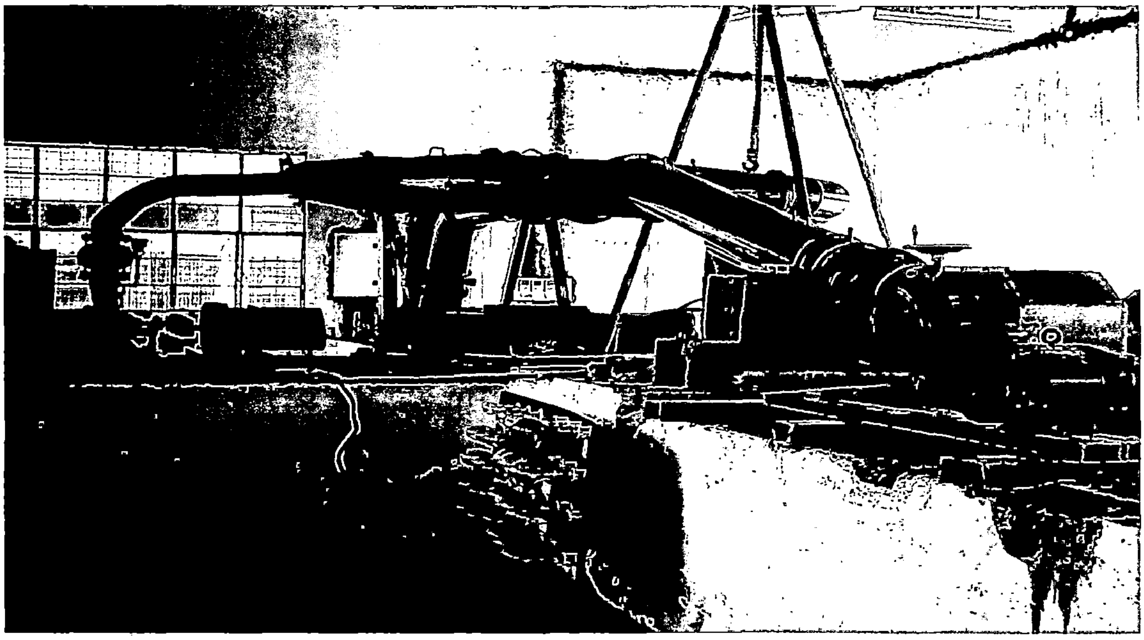


Fig 4.2 View of the Test Rig at AHEC IIT ROORKEE

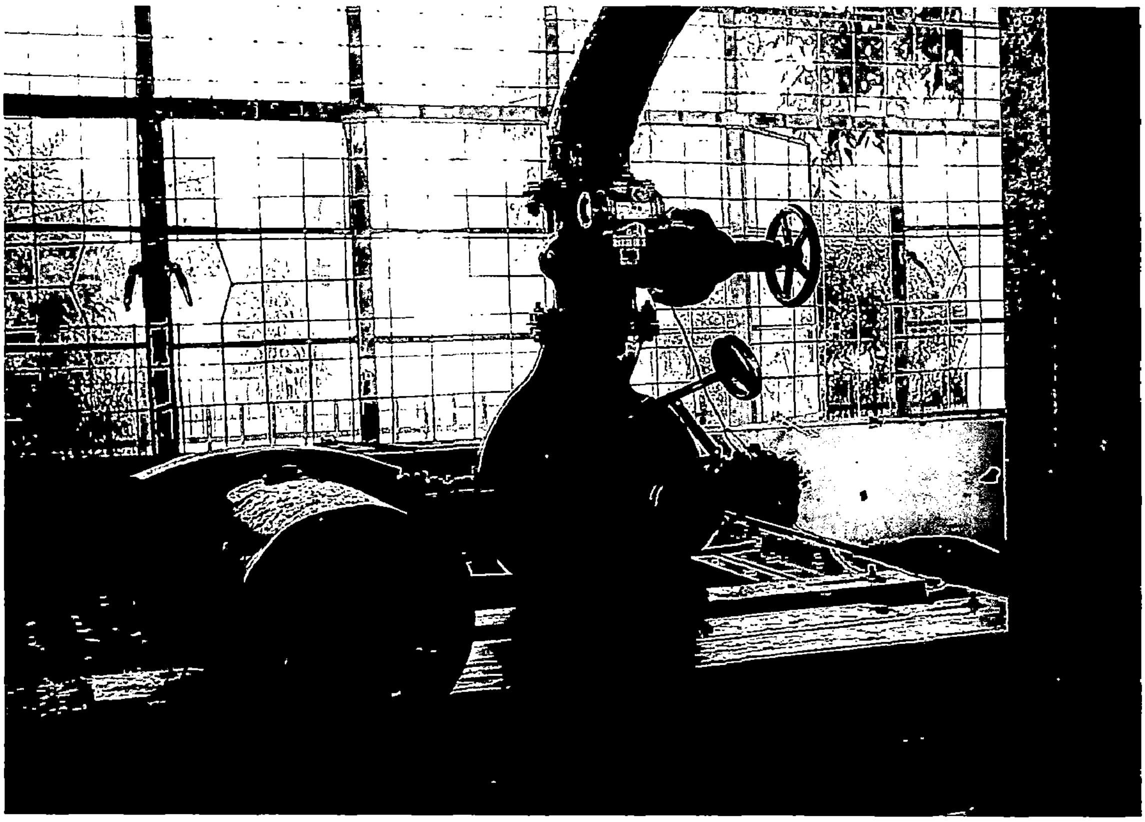


Fig 4.3 View of the Experimental Setup for Pump as Turbine at AHEC IIT ROORKEE

4.4 INSTRUMENTATIONS

Various parameters have been measured while conducting the experiment. Following parameters have been measured by using below mentioned instruments.

4.4.1 Head

Head has been measured using an Intelligent Pressure Transmitter, LD301 (pressure gauge). The measured head in atmospheres can easily be converted into meters by multiplying it with ten. The LD301 is a smart pressure transmitter for differential, absolute, gauge, level and flow measurements. It is based on the field proven capacitive sensor that provides reliable operation and high performance. The digital technology used in the LD301 enables the choice of several types of transfer functions, an easy interface between the field and the control room. Fig. 4.4 shows a view of LD301 while taking a reading while conducting the experiments.

4.4.2 Discharge

Discharge has been measured, by using a rectangular weir. Fig. 4.5 shows the view of the weir with pointer and scale.

4.4.3 Pump and Generator RPM

RPM of pump and generator were measured by using tachometer.

4.4.4 Resistive Load

The load on the generator was put as bulb load using 100 W, 200 W and 500W bulbs fitted in a panel, as shown in Fig. 4.6

4.4.5 Generator Voltage

Generator voltage has been measured in volts by using a voltmeter attached in the panel as shown in Fig. 4.6. A voltage of 420 V has to be maintained during the experiment.

4.4.6 Power Output

Power output of the generator has been measured using a wattmeter as shown in Fig. 4.6. The generator used in testing is a 3-phase synchronous generator, as shown in Fig.4.7.

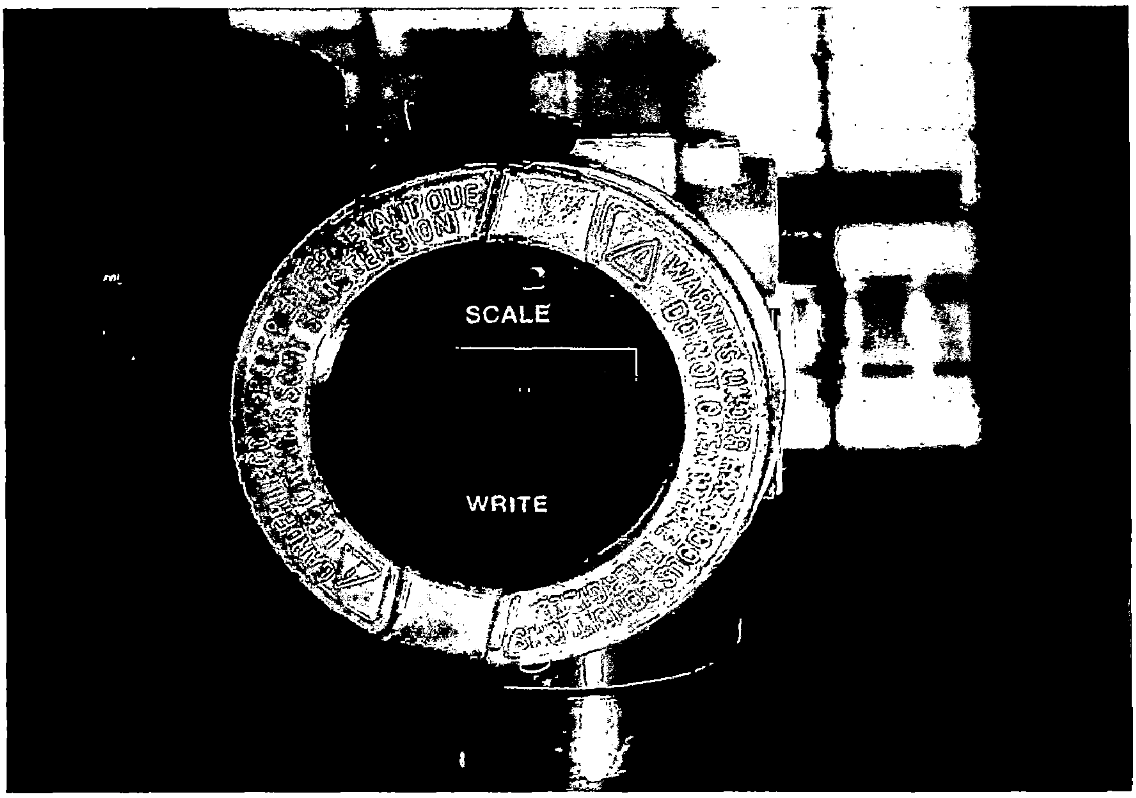


Fig. 4.4 Intelligent Pressure Transmitter, LD301 used for Measuring Head in atm.

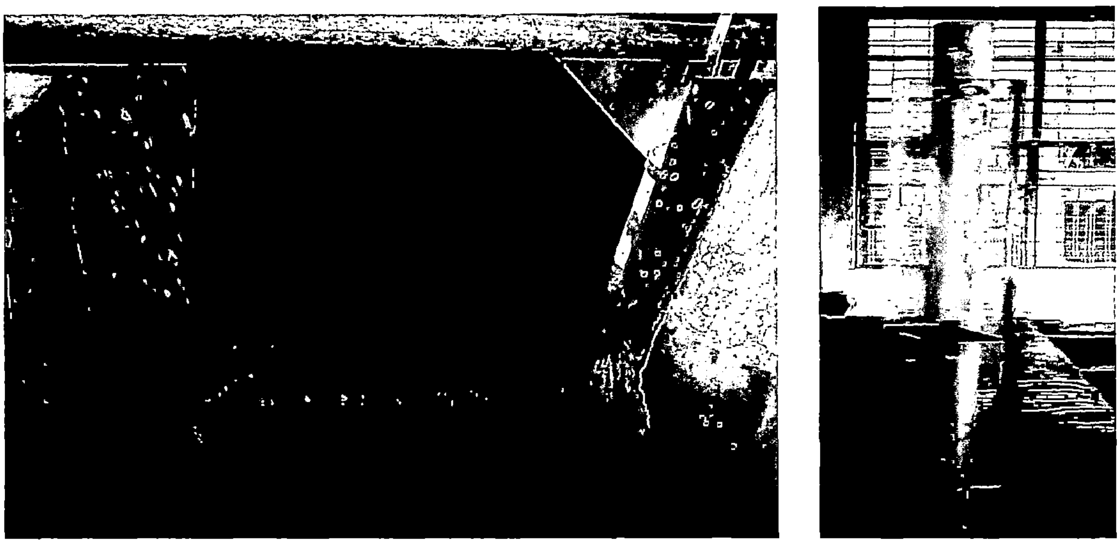


Fig 4.5 Rectangular Weir with Pointer and Scale for Measuring Discharge

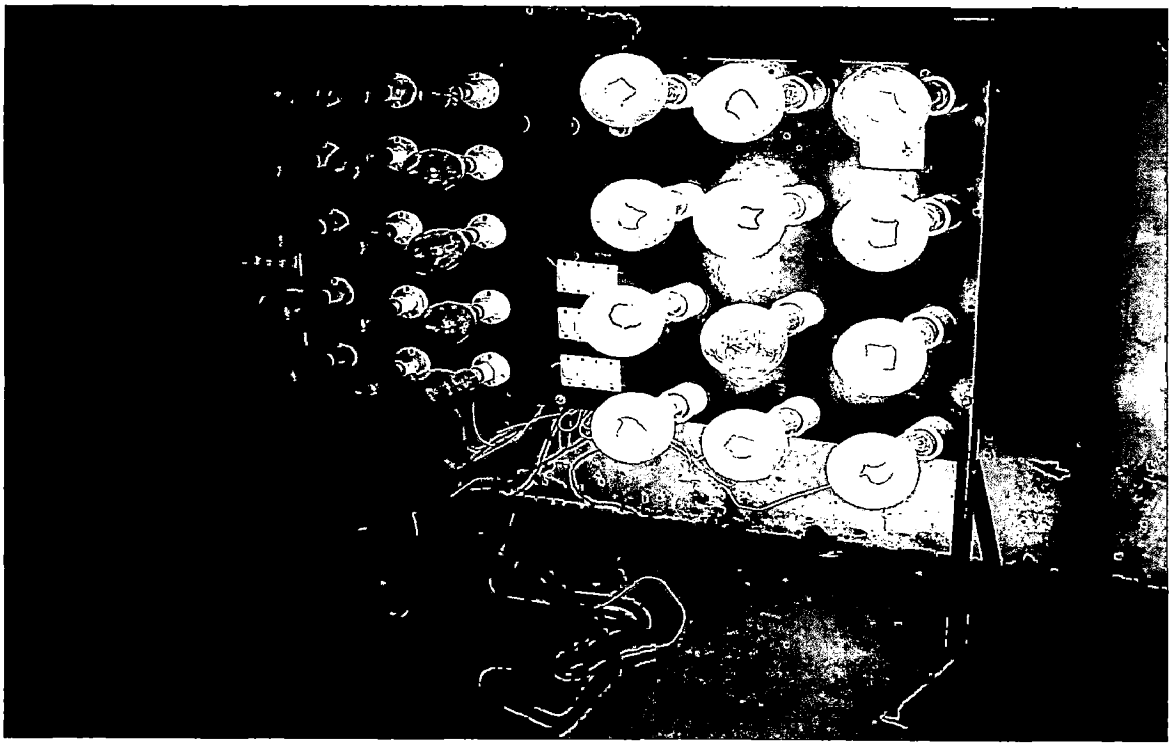


Fig. 4.6 View of the Panel Consisting of Bulb Load, Voltmeter and Wattmeter

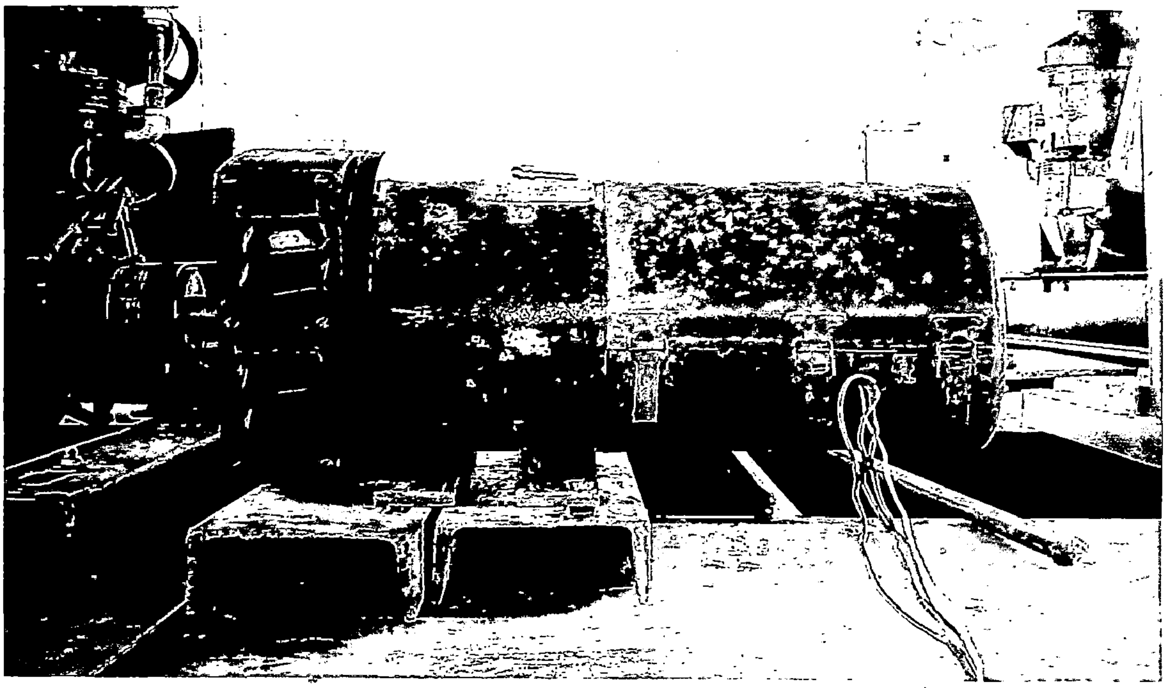


Fig 4.7 View of AC Generator used for the Experiment

4.5 EXPERIMENTAL PROCEDURE

First of all the pump was tested as turbine at the rig before its modification. An intelligent pressure transmitter was connected to the pump inlet for measuring head. A valve was connected at the pump inlet so as to vary the discharge entering in the pump. Discharge was measured using a rectangular weir. The pump was connected to an AC generator using a belt and pulley system. The generator was connected to bulb loads and a wattmeter for measuring output. Then the service pumps were made to run to pump the water to the PAT. Slowly the valve was open and the water was made to flow through the pump. When the pump impeller just started to rotate, reading of the pressure transmitter (head), weir column height (discharge) was noted. After that the valve was opened further and the pump started to rotate with more rpm. 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 420 volts in the voltmeter. The reading of the pressure transmitter (head), weir column height (discharge), bulb loads, and wattmeter reading were noted. After that valve was opened further more and again the procedure was repeated and 9 readings for varying discharge were taken.

After the modification of the pump by providing guide vanes around the pump impeller, the pump was again tested in turbine mode. The setup was same as above but the only difference was that the flow could be adjusted not only by using the valve but also by moving the guide vanes via a handle provided outside the pump. The PAT was connected to the rig as above and the service pumps were made to run. Then the valve was slowly open so as just to move the pump impeller and the reading of the pressure transmitter (head), weir column height (discharge) was noted. At that time the opening of the guide vanes was very less. After that valve was opened more and bulb loads were given on the generator. After that at a particular valve opening, the guide vanes were slowly opened further and the rpm of the pump was checked. At a particular opening of the guide vanes at which the pump gave the maximum rpm, the bulb loads were adjusted so as to maintain 1500 rpm and 420 volts of the generator. At this point the reading of the pressure transmitter (head), weir column height (discharge), bulb loads, and wattmeter reading were noted. Again the valve was opened more and the above procedure was repeated so as to take 14 readings for varying discharge.

4.6 EXPERIMENTAL RESULTS FOR PUMP AS TURBINE BEFORE MODIFICATION

The pump as turbine has been tested in the test rig. Table 4.3 shows the original readings, which have been taken while conducting the experiment on pump as turbine before its modification.

4.7 EXPERIMENTAL RESULTS PUMP AS TURBINE AFTER MODIFICATION

After implanting six guide vanes around the pump impeller, pump as turbine has been tested by adjusting the flow by moving the guide vanes. Table 4.4 shows the original readings, which have been taken while conducting the experiment on pump as turbine after its modification.

4.8 DATA REDUCTION

Data measured during the experimentation were reduced in the required form and are discussed below:

4.8.1 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.

$$H = 10P_i \tag{4.1}$$

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

4.8.2 Discharge

Discharge has been measured, by using a rectangular weir whose reference water level is 2.5cm.The Kindsvater-Carter rectangular weir equation (ISO 1980), as shown below has been used for calculating discharge in cusec (ft^3/s) and then converted into lps.

Table 4.3 Original Experimental Data Obtained during the Testing of Pump as Turbine before Modification

S.No.	Pump RPM	Generator RPM	Pump Inlet Pressure (Head) (atm)	Flow (cm) (weir datum level-2.5cm)	Voltage (V)	Wattmeter Readings (kW)
1	1000	1500	0.618	8.70	420	0.0
2	1000	1500	0.812	9.60	420	0.4
3	1000	1500	0.824	9.80	420	0.8
4	1000	1500	0.837	10.2	420	1.0
5	1000	1500	1.054	11.0	420	1.6
6	1000	1500	1.258	11.3	420	2.4
7	1000	1500	1.601	12.1	410	3.0
8	1000	1500	1.847	12.3	420	3.8
9	1000	1500	2.120	13.5	410	4.0

Table 4.4 Original Experimental Data Obtained during the Testing of Pump as Turbine after Modification

S.No.	Pump RPM	Generator RPM	Pump Inlet Pressure (Head)(atm)	Flow (cm) (weir datum level-2.5cm)	Voltage (V)	Wattmeter Readings (kW)
1	1000	1500	0.060	5.00	420	0.00
2	1000	1500	0.490	9.10	420	0.20
3	1000	1500	0.540	10.4	420	0.55
4	1000	1500	0.630	10.6	420	0.80
5	1000	1500	0.750	11.0	420	1.20
6	1000	1500	0.820	11.3	420	1.50
7	1000	1500	0.910	11.5	420	1.80
8	1000	1500	0.970	12.0	410	2.20
9	1000	1500	1.008	12.2	420	2.40
10	1000	1500	1.350	12.4	420	3.10
11	1000	1500	1.543	13.1	420	3.70
12	1000	1500	1.737	13.3	420	4.20
13	1000	1500	1.856	13.5	420	4.80
14	1000	1500	2.117	13.8	420	4.50

$$Q = C_e \frac{2}{3}\sqrt{2g}(b + K_b)(h + K_h)^{\frac{3}{2}} \tag{4.2}$$

where Q is discharge in cusec, C_e is discharge coefficient, g is acceleration due to gravity in ft/s^2, b is the notch width in ft, h is the rise in water level in ft taking weir reference level as 2.5cm, K_b accounts for effects of viscosity and is taken as 0.012 ft, K_h accounts for effects of surface tension and is taken as 0.003 ft.

4.8.3 Power Input

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

$$PAT_i = QgH \text{ kW} \tag{4.3}$$

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

4.8.4 Pump Output

Pump as Turbine Output, PAT_o has been obtained using the following formula which includes the efficiency of generator:

$$PAT_o = 0.6198 + \frac{77}{75000} \times G_o \text{ kW} \tag{4.4}$$

where G_o is the watt meter reading or the generator output.

4.8.5 Pump as Turbine Efficiency

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

$$\eta_{PAT} = \frac{PAT_o}{PAT_i} \tag{4.5}$$

Using above formulae, the measured data were converted into required parameters. Table 4.5 and Table 4.6 give the parameters obtained by data reduction of experimental data for pump as turbine before modification and after modification respectively.

Table 4.5 Parameters Obtained by data Reduction of Experimental Data for Pump as Turbine before Modification

S.No.	Discharge Q (l/s)	Head H (m)	Input (kW)	Output (kW)	Efficiency η (%)
1	22.61	6.2	1.371	0.000	0.00
2	27.65	8.1	2.200	1.030	46.79
3	28.81	8.2	2.329	1.441	61.87
4	31.19	8.4	2.561	1.646	64.29
5	36.13	10.5	3.735	2.260	60.56
6	38.05	12.6	4.695	3.084	65.68
7	43.31	16.0	6.802	3.700	54.39
8	44.67	18.5	8.092	4.521	55.87
9	53.07	21.2	11.015	4.730	42.90

Table 4.6 Parameters Obtained by data Reduction of Experimental Data for Pump as Turbine after Modification

S.No.	PAT Discharge Q (l/s)	PAT Head H (m)	PAT Input (kW)	PAT Output (kW)	PAT Efficiency η (%)
1	05.97	0.6	0.0363	0.00	00.00
1	24.81	4.9	1.1926	0.83	69.19
2	32.40	5.4	1.7165	1.18	69.00
3	33.63	6.3	2.0785	1.44	69.34
4	36.13	7.5	2.6583	1.85	69.66
5	38.05	8.2	3.0605	2.16	70.57
6	39.34	9.1	3.512	2.47	70.27
7	42.64	9.7	4.0577	2.88	70.94
8	43.99	10.08	4.3497	3.08	70.90
9	45.35	13.5	5.9939	3.80	64.44
10	50.21	15.43	7.5860	4.42	58.24
11	51.63	17.3	8.7814	4.93	56.16
12	53.07	18.5	9.6437	5.55	57.53
13	55.24	21.17	11.4509	5.75	50.24

RESULTS AND DISCUSSIONS

In order to compare the performance of PAT before and after modification an experimental study was carried out and reported in chapter 4. Characteristics curves have been prepared for the results obtained which are already listed in Table 4.5 and Table 4.8.

Fig. 5.1 shows the variation in head of PAT without modification with discharge. For the full opening of the valve maximum head upto 21.2 m is obtained for maximum discharge of 53 lps. Corresponding to these values, power output of PAT is represented by Fig. 5.2. Using these values operating characteristics curve has been drawn and shown in Fig. 5.3. It has been observed that the PAT without modification is having maximum efficiency of 66%. It is also observed that the operating range of PAT in this case for best efficiency is between the flow rate of 32 lps to 38 lps (95% to 105%), which is considered as narrow. This verifies the constraints of PAT.

Similarly characteristics curves have been drawn for the results obtained for PAT after modification. Fig. 5.4 shows the variation in head of PAT after modification with discharge. For the full opening of the valve maximum head upto 21.00 m is obtained for maximum discharge of 55 lps. Corresponding to these values, power output of PAT is represented by Fig. 5.5. Using these values operating characteristics curve has been drawn and shown in Fig. 5.6. It has been observed that the PAT after modification is having maximum efficiency of 70%. It is also observed that the operating range of PAT in this case for best efficiency is between the flow rate of 27 lps to 40 lps (85% to 120%).

Fig. 5.7 has been prepared to compare the operating characteristics of PAT before and after modification. Form the figure it is observed that the efficiency of PAT after modification has been improved by 4%. Further it is also observed that the maximum efficiency of 66% of PAT without modification can be achieved over even wider range i.e. from 23 lps to 45 lps (65% to 130%). Therefore, it is concluded that the efficiency as well as the operating range of PAT has been improved by providing movable guide vanes.

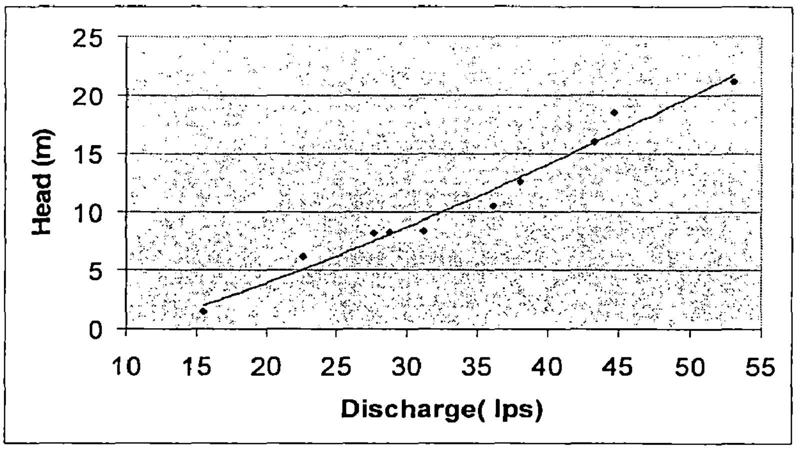


Fig. 5.1 Graph of Head vs. Discharge of PAT before Modification

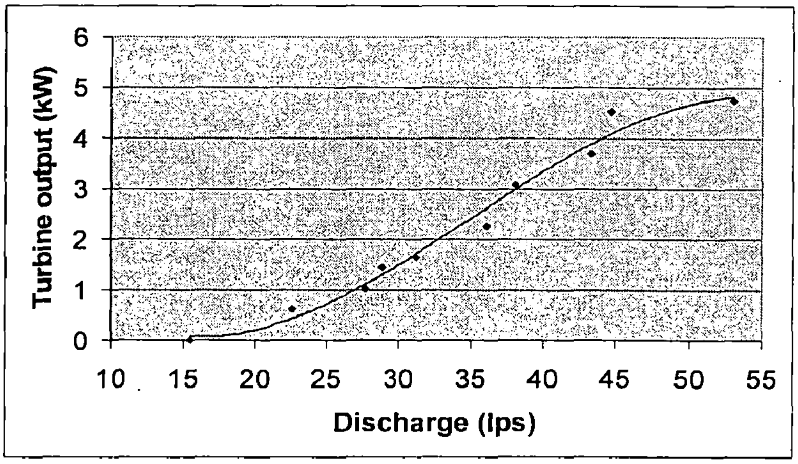


Fig. 5.2 Graph of Turbine Output vs. Discharge of PAT before Modification

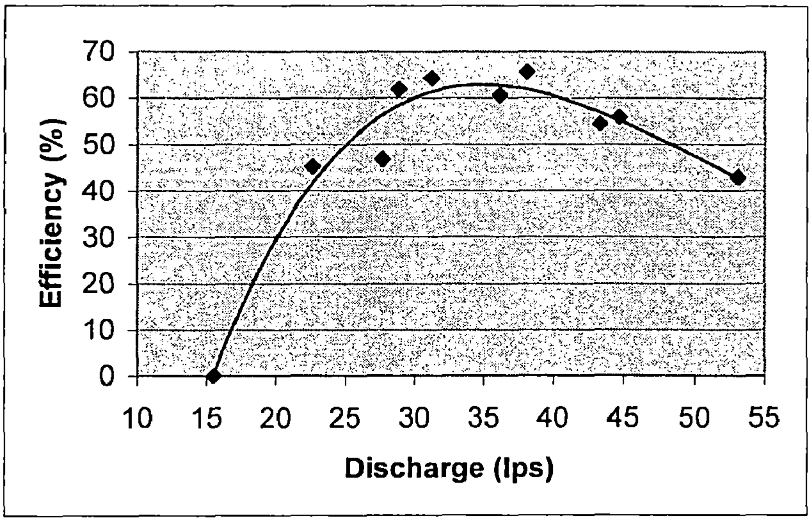


Fig. 5.3 Graph of Efficiency vs. Discharge of PAT before Modification

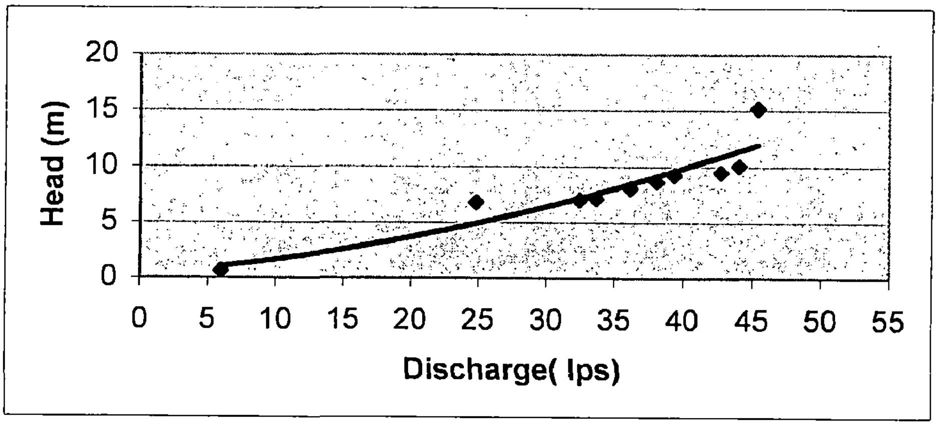


Fig. 5.4 Graph of Head vs. Discharge of PAT after Modification

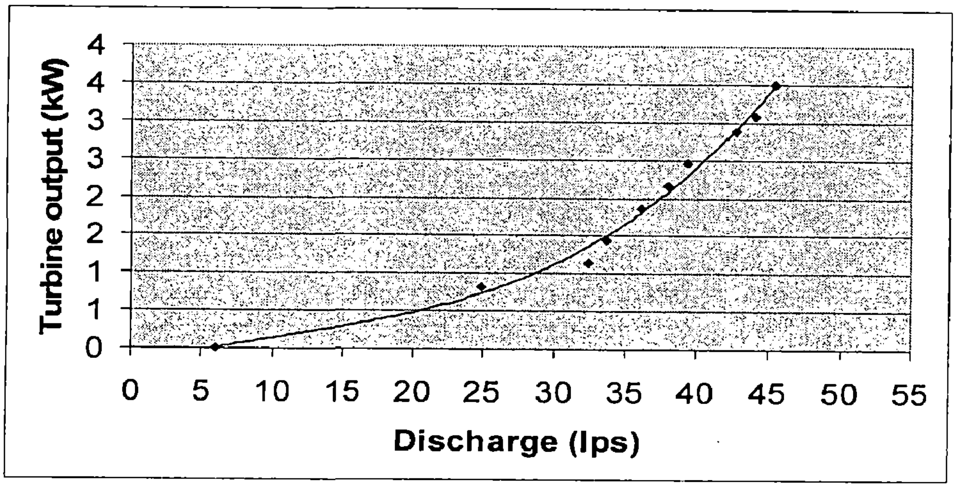


Fig. 5.5 Graph of Turbine Output vs. Discharge of PAT after Modification

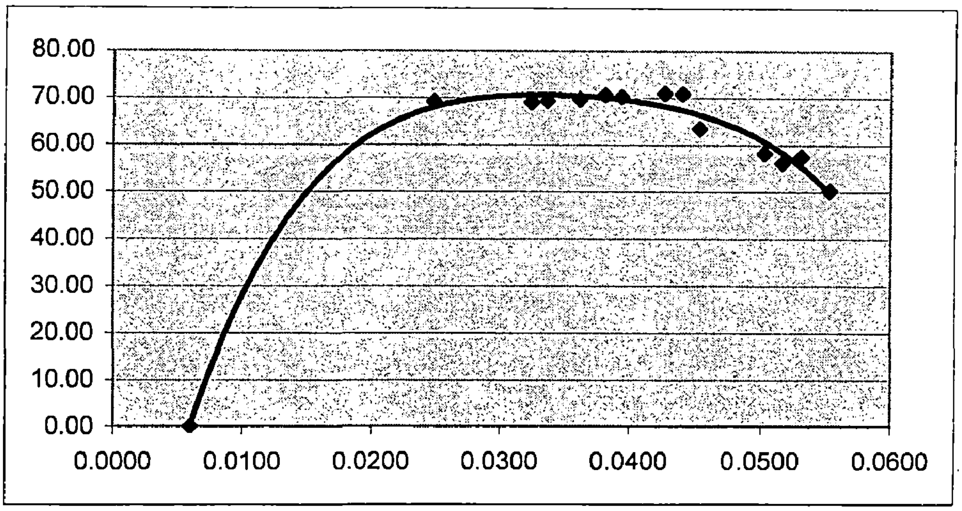


Fig. 5.6 Graph of Efficiency vs. Discharge of PAT after Modification

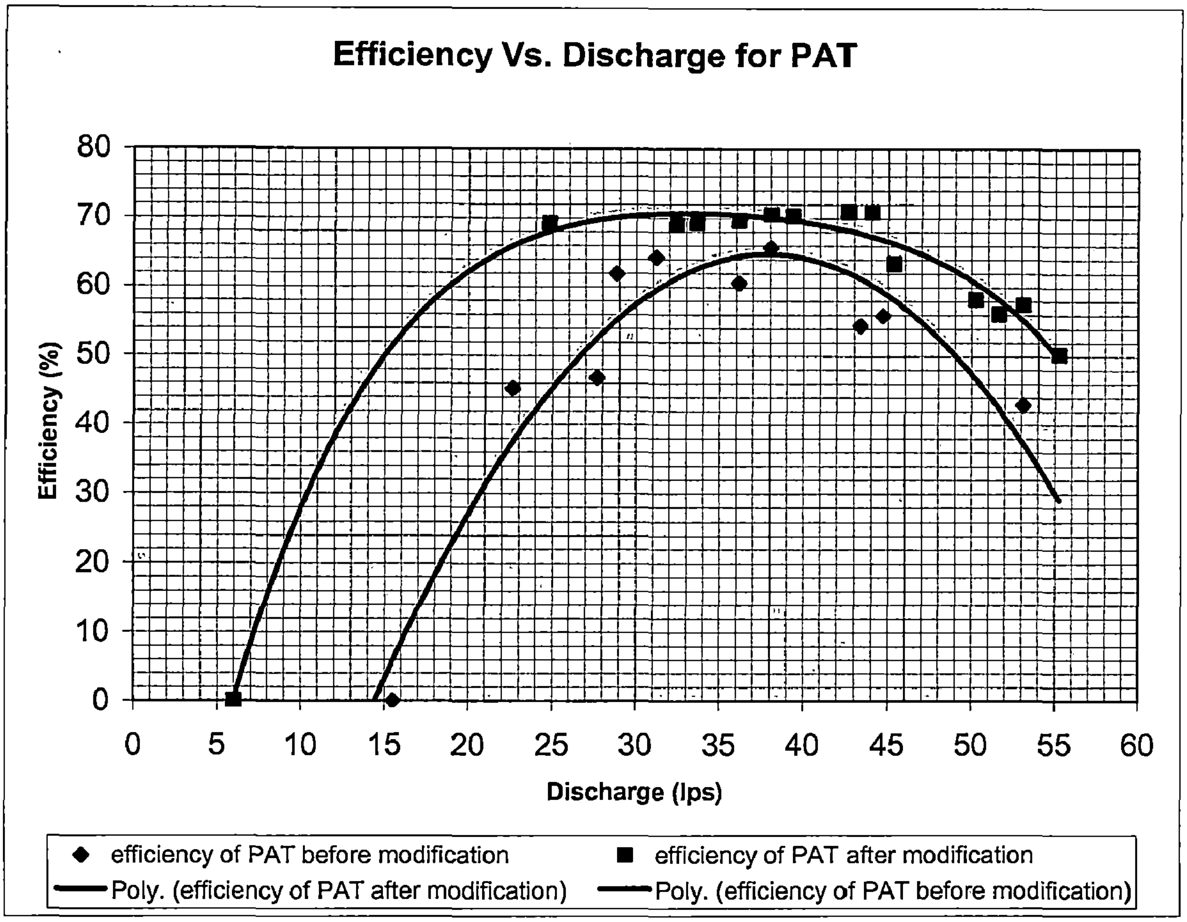


Fig. 5.7 Graph of Efficiency vs. Discharge of PAT before and after Modification

CONCLUSIONS AND RECOMMENDATIONS

6.1 CONCLUSIONS

It is very well known that Micro-hydropower can be one of the most valuable answers to the question of offering isolated rural communities the benefits of electrification and the associated progress, as well as to improve the quality of life.

It is also well known that turbines used at such micro hydro sites can easily be replaced by pump as turbine (PAT) with a number of advantages except the disadvantage of having poor part load efficiency due to absence of flow regulating mechanism in it.

Therefore in order to eliminate pump as turbine's disadvantage the centrifugal pump mentioned in the above chapters has been modified by providing movable guide vanes around its impeller.

The characteristics of the PAT before modification and after modification have been studied in the previous chapter and the following has been concluded:

(i) The efficiency of the PAT after modification has increased by 4% as compared to the PAT before modification.

(ii) The operating range of pump as turbine before modification has been observed from 32 lps to 38 lps (95% to 105%) discharge for the maximum efficiency of 66%.

(iii) The efficiency versus discharge curve for PAT after modification is quite flat, maintaining an average efficiency of 70 % over a discharge range of 27 lps to 40 lps (85% to 120%). It may be concluded that the operating range of modified pump as turbine has been increased substantially.

6.2 RECOMMENDATIONS

The major constraint of using pump as turbine of having poor part load efficiency can be eliminated upto an extent by providing movable guide vanes with the centrifugal pump. However, a techno-economic study is required which can be taken up as a future study.

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