CHAPTER 1

INTRODUCTION AND LITERATURE REVIEW

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1.1. Introduction

Reverse-running pumps as turbines (PATs) are considered one of the most interesting technologies used in developing countries. It is one of the outstanding solutions to the energy problem in such places. The need of a PAT appears significantly in micro-hydro plants which are widely available in remote and rural areas in order to generate electricity with a cheap process which is away from the national grid.

Reverse-running Pump as Turbine (PAT) refers to any standard pump which could be operated as turbine when the direction of the flow is reversed. The Major advantage of using a PAT over a conventional turbine in micro-hydro plants is that it is more economical as well as its wide availability in the market [1].

1.2. Using a Pump as Turbine

Using a pump in the reverse direction as turbine is one of the perfect solutions for isolated areas from the grid. Although the running cost of operating the PAT could be more than the running cost of conventional turbine due to the low efficiency. Using a PAT has many advantages over conventional turbines in micro-hydro plants. The main advantages are listed as follows:

- Pumps and their spare parts are significantly available all over the world as pump market is dominated over turbine one in most countries.
- From economical point of view, using integrated pump and motor unit as turbine and generator is cheaper than using a conventional small size turbine and generator to produce electricity.
- Using a PAT permits a wide range of heads and flow rates so the selection of a suitable PAT for any application is not complicated and available.
- Easier in installation and operation than conventional turbines.
- Short delivery time.
- Moderate cost as it's cheaper than the conventional turbine. The investment costs of PATs 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 [2].
- Wide range of sizes in the market
- Easier in maintenance and troubleshooting which is considered a major problem in conventional turbines.

Figure (1-1) shows a conceptual sketch of a PAT system in which an upper reservoir supplies water to a lower reservoir through piping system. The flow of water enters a pump as turbine or a turbine which is coupled by a generator or a motor as generator in order to generate electricity.

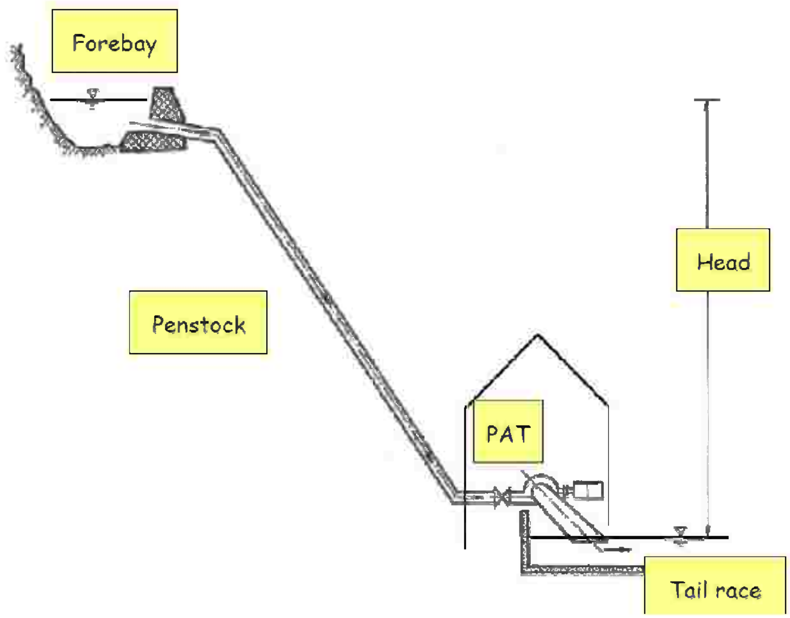

Figure (0-1) Conceptual sketch of a PAT system [1]

Rotational fluid machines are considered reversible and a pump can run in reverse mode as a turbine effectively. The best efficiency point flow and head in the reverse mode is greater than the pump mode because of the hydraulic losses of the machine [2]. When operating at the best efficiency point in the pump mode, the output head is reduced by the hydraulic losses compared to the ideal flow conditions.

Concerning the disadvantages of using a pump as turbine:

- 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 to seasonal variations of flow, the hydraulic losses of the installation will increase sharply.
- Lower efficiency at part load: a conventional turbine has an effective hydraulic control.
- If PATs are operated at other than the design flow, i.e. below their best efficiency point (bep), a relatively rapid drop of efficiency will occur (in addition to the hydraulic losses incurred by the flow regulation).

The disadvantages of PATs can be reduced to a minimum if the PAT is very carefully selected and only applied where justified. Poor performance due to an inappropriately selected machine or application will lead to a reduction of annual returns. Summed up over the entire lifetime of the machine, this reduced output might by far offset the cost advantage of the PAT (lower investment costs) in comparison to a conventional turbine.

1.3. Applications of a pump as turbine

Pumps as turbines have many applications in urban or rural areas [2]. These Applications are listed below as:

1.3.1 Micro-Hydro Schemes

These schemes are the direct application of using pumps as turbines. They are used mainly in rural areas for electrical needs.

The energy problem in rural areas is due to the isolation of these areas from the national grid. In some countries they depend mainly on diesel generating sets for power production. As a result the need for a source of electricity is urgent in such areas. In these countries isolated form the grid, the electricity generation with a good quality and moderate cost is one of the most important issues. So, the new trend of power generation is by using very small scale hydro power schemes, which was an old approach districted due to the widespread of electricity demand, which makes it cheaper. As a result, the micro-hydro plants appear significantly in these places in order to generate electricity from available resources. The Micro-hydro is a hydroelectric power that typically produces electricity up to 100 kW using the natural flow of water.

A micro-hydro consists mainly of an intake, a pipeline and generating equipment as shown in Figure (1-2). In the generating equipment a turbine is used to convert the hydraulic power to a mechanical one which in turn is converted to electrical power by a generator.

The cost of a micro-hydro system is inexpensive due to the usage of an electronic load controller which controls the frequency of the generator set by controlling the power consumption. The use of an electronic load governor is a great addition to the micro-hydro plants in order to control the speed of the generating set. This makes it easier to control than the speed governor used with conventional turbines. These installations can provide power to an isolated home or small community, or are sometimes connected to electric power networks. The PATs are economical, feasible and available. In addition, the spare parts of PATs are available as well which make PATs more convenient than conventional turbines. The lack of conventional turbines in the market is one of the greatest problems in such places so the need of PATs appears significantly. A PAT could be used in a grid-connected or stand-alone mode which is applicable in remote areas from the grid.

1.3.2 Industrial Applications

- Irrigation schemes usually involve the distribution of water to different topographical levels. For reasons of costs and complexity of systems, the total discharge required is usually pumped (or conveyed by gravity) to the highest point of the scheme and from there distributed to lower levels requiring dissipation of energy. By using a PAT instead of pressure reducing valves or stilling basins, significant amounts of energy can be recovered.

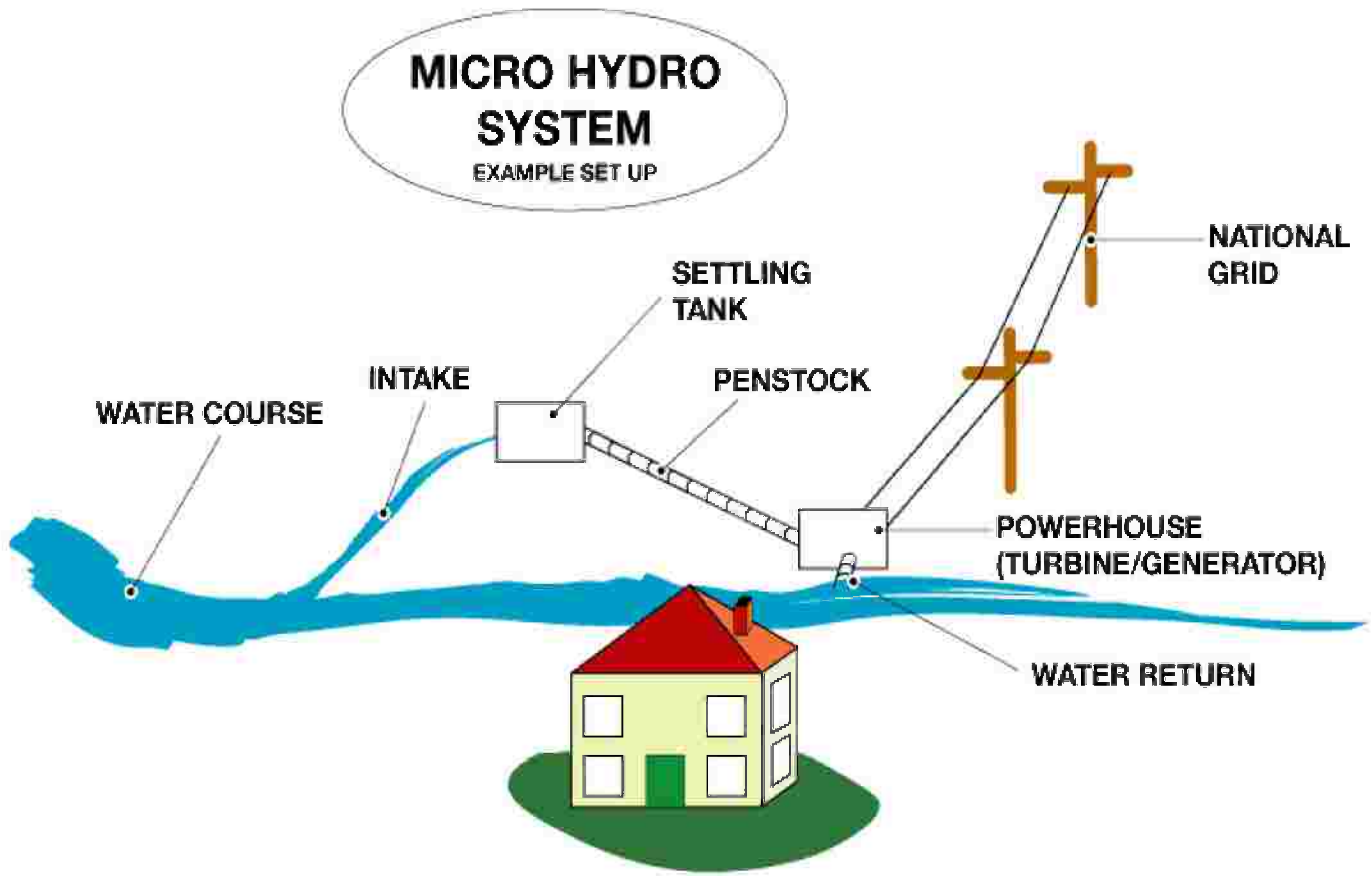

Figure (1-2) Micro-Hydro system

- A common feature of hydraulic power recovery systems is the need to reduce pressure of a fluid at the end of a process or before further application or treatment of the fluid. The conventional approach would dissipate this surplus pressure in any kind of throttling device (valve, orifice, stilling basin). Using a reverse running pump as a hydraulic power recovery turbine (HPRT) recovers this energy at almost no costs thereby improving overall plant efficiency. Interest on this kind of application has grown recently mainly as a result of increasing energy prices and energy conservation awareness.

  In many process applications, a fluid must be kept at a high pressure during the actual process but must be let down afterwards; this provides a vast field of applications for hydraulic power recovery turbine ”HPRT”:
  - Petro-chemical processes (e.g. discharge of hydrocracker residuals)
  - Gas scrubbing (unwanted gases in gas mixtures are dissolved in a fluid at high pressure).
  - Mining [to provide supportable working conditions in deep mines (1500 to 3000m below surface level) cooling water is delivered into the mine thereby attaining high pressures which have to be released.

1.3.3 Drinking Water Supply Systems

A PAT is used in water supply systems as there is a difference in head between elevated and low level tanks. In addition, it is used as a pressure control method in closed-loop systems [1].

1.3.4 Parallel and Series Operations

PATs could be used in parallel or in series with pumps as shown in Figure (1-3) to make variable duty pumping stations more efficient instead of throttling. A series PAT with a pump is used to reduce the head of the operation. On the other hand, it can be used in parallel as a bypass to control the flow rate.

1.3.5 Lightening Homes

In Small villages in developing countries where the only electric load is lightening in the evening, a pump as turbine is the best and most suitable choice [2]. The generator or the motor used as generator is used to lighten the houses in the evening and to power different equipment in the morning.

Figure (1-3) Four of eight parallel PATs in service at the water utility authority [1]

1.3.6 Charging Batteries

Charging batteries and intermittent load applications is a direct application of pumps as turbines. In some countries led acid batteries are used for electrical needs which need to be recharged continuously. The PATs are used to recharge the batteries instead of being recharged in another town.

1.3.7 Desalination

Reverse osmosis in desalination plants (pressing sea-water through a semi-permeable membrane to obtain potable water). It can be used as an energy recuperator in such application. Desalination applications, such that a pump as a turbine, is used to generate electricity to desalinate seawater via electro dialysis as shown in Figure (1-4).

1.3.8 Pumping Storage Systems

Pump-turbines have been used for several decades in large pumped storage schemes as shown in Figure (1-5) with power ratings of several megawatts. Pump-turbines are specifically designed to operate in both modes, pumping water into an elevated storage lake overnight at low tariff electricity and, during the day, generating peak demand electricity through the same machine operating in turbine mode [2].

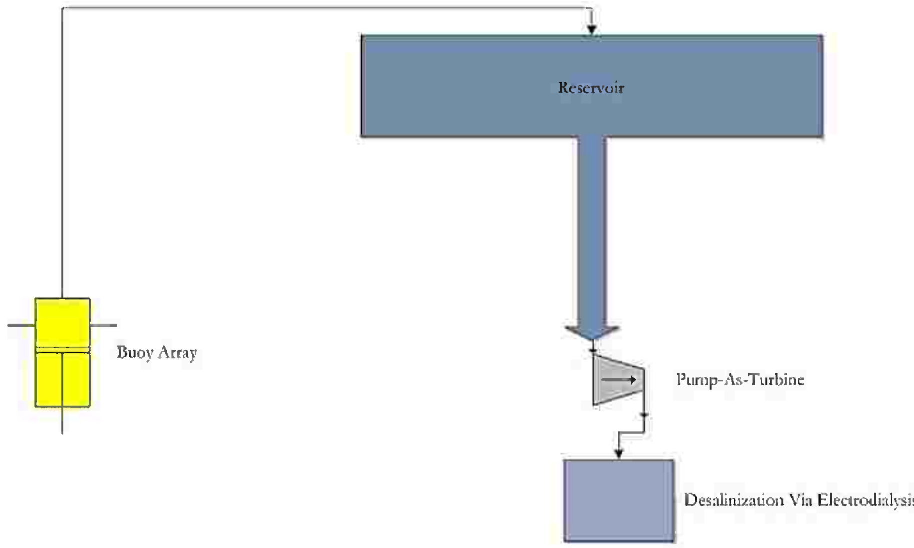

Figure (1-4) A schematic diagram of the desalination plant [3]

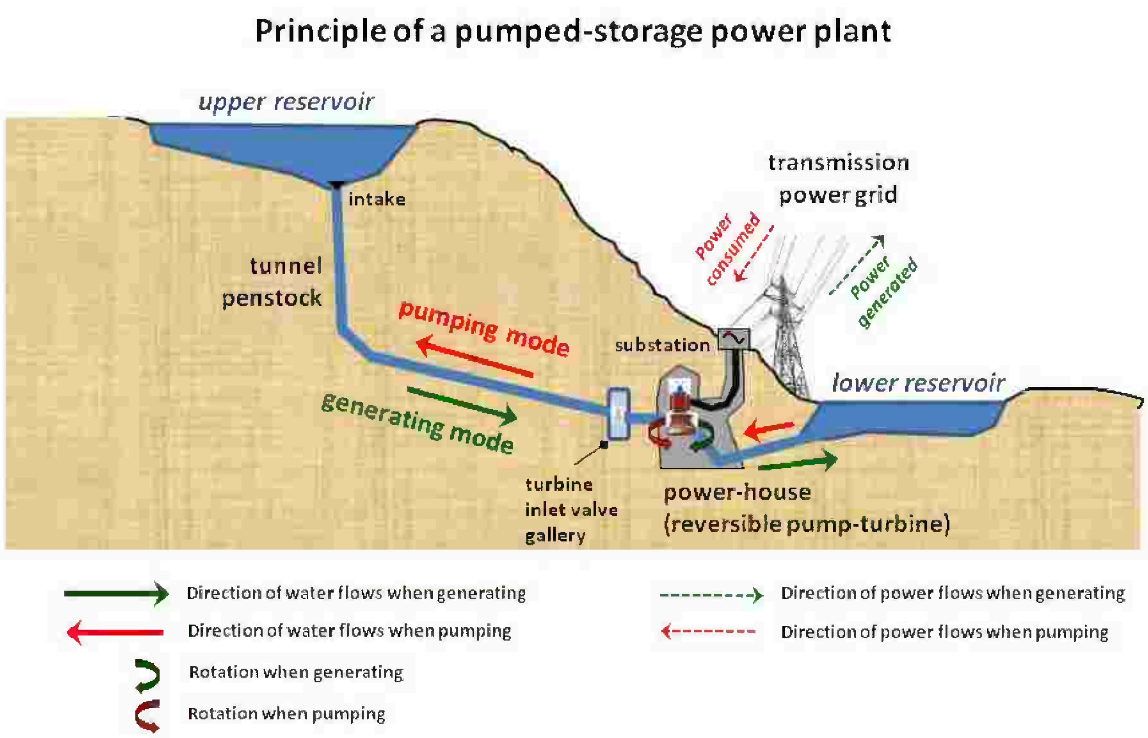

Figure (1-5) Pumped storage plant [4]

1.4. Limitation of Using PATs and Ways of Overcoming

The limitation of using traditional PATs without controller is that a fixed flow rate is required which means that it is only suitable for sites with a constant water supply throughout the year. On the other hand, it is not possible to sustain a fixed flow rate throughout the year but instead the PAT is designed according to the minimum flow rate around the whole year. If the flow rate decreases accidently the power developed will be lower than the targeted one. So as a result of these variable conditions several solutions are arisen to overcome these difficulties. These solutions are listed as:

- Electronic load controllers were found to be effective in governing with both synchronous and induction motors as generators to regulate the speed of rotation of a PAT so as to control its operation.
- When the flow rate increases a parallel PATs could be a very suitable solution to get the maximum power.

Concerning the electronic load controllers, new developments have been made in this area. A control circuit should be used to accommodate the flow rate changes throughout the year. The control circuit consists of an electronic load controller (ELC) for a synchronous generator or induction generator controller (IGC) for induction generator. Electronic load controllers are based on keeping the load of a PAT constant by switching in ballast loads whenever the electricity demand drops.

In order the pump to be stable, the curvature of a PAT is reversed in shape at outlet as shown in Figure (1-6). Although a pump can be used as turbine in reverse direction, the turbine cannot be used as pump in an efficient way due to short channels and large angles of the runner.

The shape of the impeller (outlet angle) affects greatly the energy transfer between the fluid flow and the impeller. On the other hand, in the reverse mode the share of the pump casing (volute) determines the energy transfer. Different pump designs and manufacturing details will therefore affect performance. Machines may have similar performance in pump mode [2].

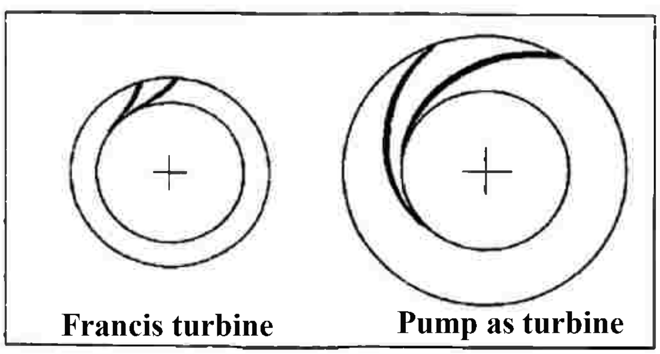

Figure (1-6) Comparison between the runner of a Francis turbine and the runner of a PAT for a similar performance based on a drawing by Meier [5]

The upper limit of PATs application was suggested to be according to their availability in the market off the shelf which means around 100 kW or 250 kW [5]. Even so, other researchers suggested that it is up to a much higher value such: 1.5 MW [5]; 2 MW [6], according to [5], Grant increases the power limit of PATs to several MW by using multi 500 kW machines in parallel, and other researcher describes a Swiss hydropower station with seven 931 kW PATs in parallel, using "a pump model, cheap and solid, frequently used in South African gold mines and in the Gulf countries”[5].The application range of using pumps as turbines varies with the flow rate range. The ring-section pumps are preferable between 10 to 100 l/s and volute casing pumps are used in higher flow rates as illustrated in Figure (1-7).

On the other hand, the use of integrated pump and motor units is recommended only for the production of electricity in micro-hydro plants. Albeit, the use of such integral units is limited due to some reasons:

- The Turbine’s rotational speed is fixed (no guide vanes attached) to the generator speed which means that the speed is limited to site conditions.
- The generator selection for a specific PAT is limited.
- Mechanical loads cannot be connected directly to the PAT (a belt drive or a control circuit is needed).

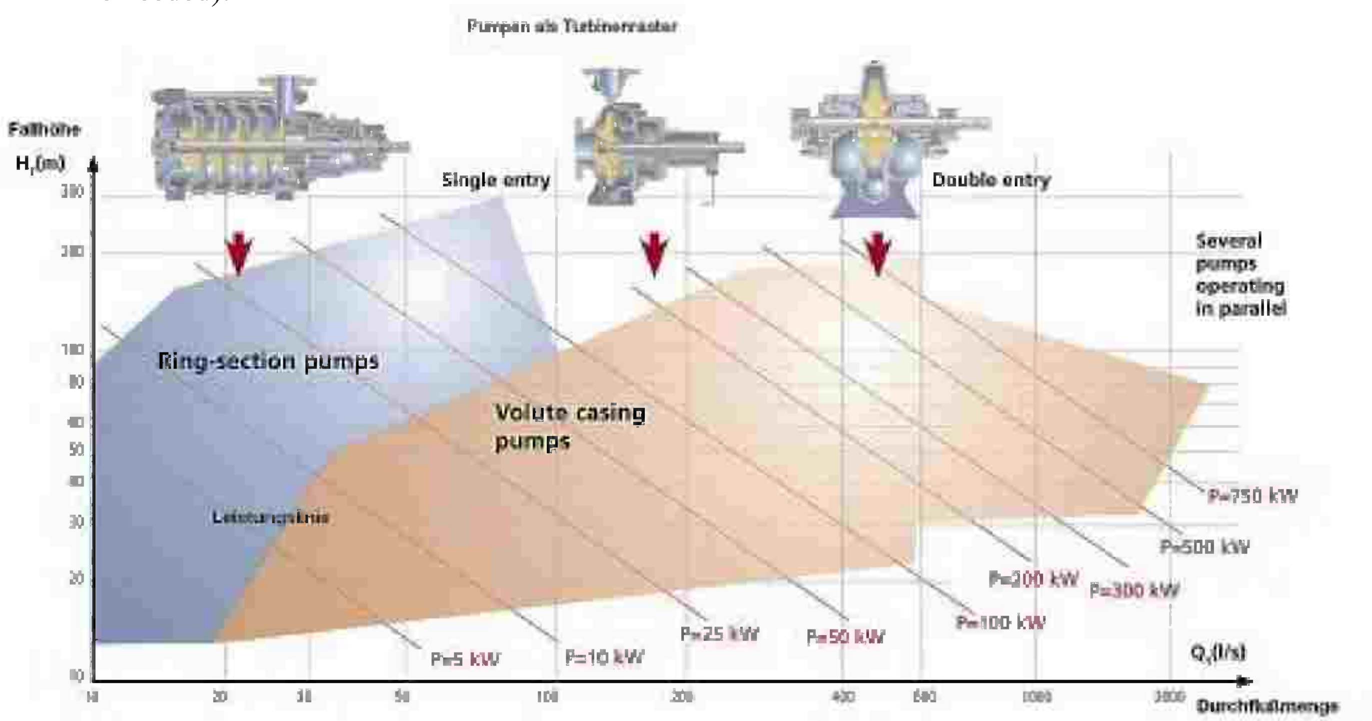

Figure (1-7) The application range for using section and volute casing pumps operating as turbines [7]

1.5. PAT Specific Speed Limits

- **Lower Boundary**

  The specific speed (N_s) is considered a major factor which affects the applicability of PATs. Concerning the low specific speed area, PATs have to compete with impulse pelton turbines. Below a certain threshold, reaction (Francis) turbines are less efficient than impulse turbines, and the same applies to PATs. This is due to the narrow flow passages characteristic of reaction machines of low specific speed [5].

In the domain of pumps, where the impulse principle does not work, very low specific speeds can only be achieved by using multistage pumps. Multistage pumps can be used as turbines, but their cost advantage over impulse turbines is offset by the high cost of multistage pumps (that are more expensive than single-stage pumps) and the low cost of small Pelton, Turgo and cross-flow turbines (that can be fabricated in local workshops) [2].According to [4], Ventrone thought that Pelton turbines are unbeatable for $N_{sT}<1.96$ (m, m^3/s). The dimensionless specific speed ranges for different kinds of turbines and its relation with efficiency is illustrated in Figure (1-8).

- **Upper Boundary**

In this range axial pumps have to be used which is not very efficient if it is used as PAT. There are a lot of disadvantages as an axial PAT. They are more expensive than conventional pumps as well as there is a little contribution in the turbine-mode researches of such type. Unlike other turbines, small axial turbines (with $N_{sT} > 19.6$) are suitable for standardization, on account of the possibility to adjust their performance to a range of conditions, just by changing the pitch of the blades.

For the specific disadvantages of axial PATs, some researchers proposed the use of several mixed-flow PATs in parallel instead of an axial machine. In the context of the USA, a set of up to five mixed-flow PATs was found to be cheaper than a single conventional hydraulic turbine of comparable capacity. This approach has a very important additional advantage: it permits part-load operation during low-flow seasons [5].

As a result, it is not possible to draw a general rule regarding the limits of applicability of PATs according to their specific speed, since they will vary from one country to another, depending on the availability and cost of pumps and turbines.

In countries like Peru or Nepal, where there is expertise in the manufacture of impulse turbines, the lower boundary may be near $N_{st}=1.96$ which was the limit proposed by Ventrone according to [5]. In other countries, this boundary will be shifted to a lower specific speed, especially if multistage pumps are locally available. Lastly, the standardized axial turbines proposed by Ventrone and Navarro require building up an industrial infrastructure that is unjustifiable in most countries. Under these circumstances, the upper boundary of specific speed is defined only by the availability of pumps, and axial-flow PATs or mixed-flow PATs sets may be the best solutions for low head sites, which have most of the small hydro potential in the world.

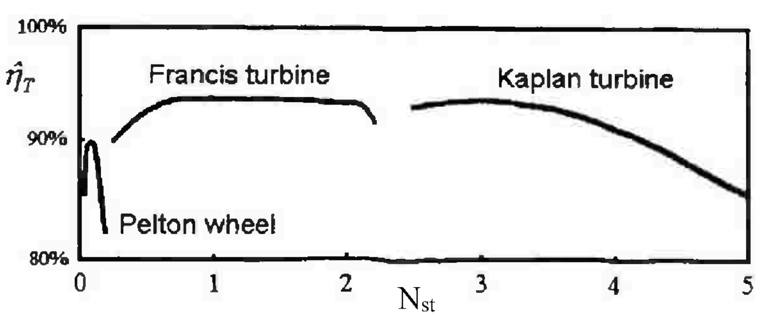

Figure (1-8) Relation between dimensionless specific speed $N_{st}$and η_T [5]

1.6. Pump and Turbine Performance Curves

When the flow of a pump is reverse by applying head to the discharge pipe instead of inlet pipe, in the case of the normal pump operation becomes a hydraulic turbine. In centrifugal pump operation this may happen if the motor is stopped and there is no Non-return valve to prevent the flow in the reverse direction.

PATs are operated in a specific range of heads and flow rates which is illustrated in the next Figure (1-9) [8]. The PAT range is wider than cross flow turbine ranges and are more applicable in low flow rates as illustrated in Figure (1-9).

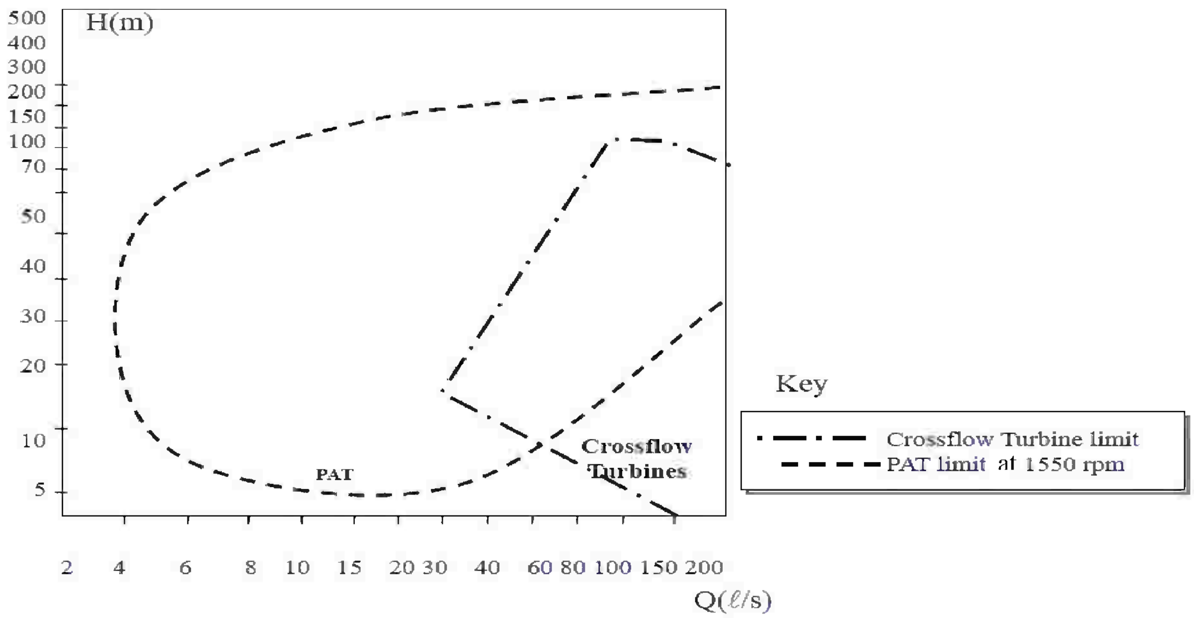

Figure (1-9) Head and flow ranges for various turbines [8]

1.7 Characteristics of PATs

Pump characteristics are readily available from the pump manufacturers but conversely the PAT characteristics are not included in datasheets of a certain pump. As a result, a correlation is needed to relate the pump and PAT performance in different speeds and conditions and to select the appropriate pump to work as a turbine. So, a test for a pump as turbine is performed to determine the whole characteristics of a PAT.

As the most popular type of radial pumps, centrifugal pumps can operate as turbines effectively and efficiently. Centrifugal pumps are available in different sizes for different ranges of heads and flow rates. Figure (1-10) shows the whole characteristic curves of a standard centrifugal pump operation and the characteristics of the same pump at reverse mode is illustrated for the same rotational speed [2]. The main point in the characteristics is the best efficiency point which relates the pump in both modes with a simple form.

It is well observed that the best efficiency point (BEP) of pump operating in the reverse mode happens at a higher flow rate than the pump mode. A comparison of the characteristics of normal pump operation with the characteristics of the same pump operated as a turbine at the same speed is shown in Figure (1-11). The curves are normalized by the value of head, flow, efficiency and power at the pump BEP. The location of the turbine BEP is at higher flow and head than the pump BEP. The ratio of the turbine capacity to the pump capacity, at the BEP, and the turbine head to the pump head at the BEP have been observed to vary with specific speed [9].

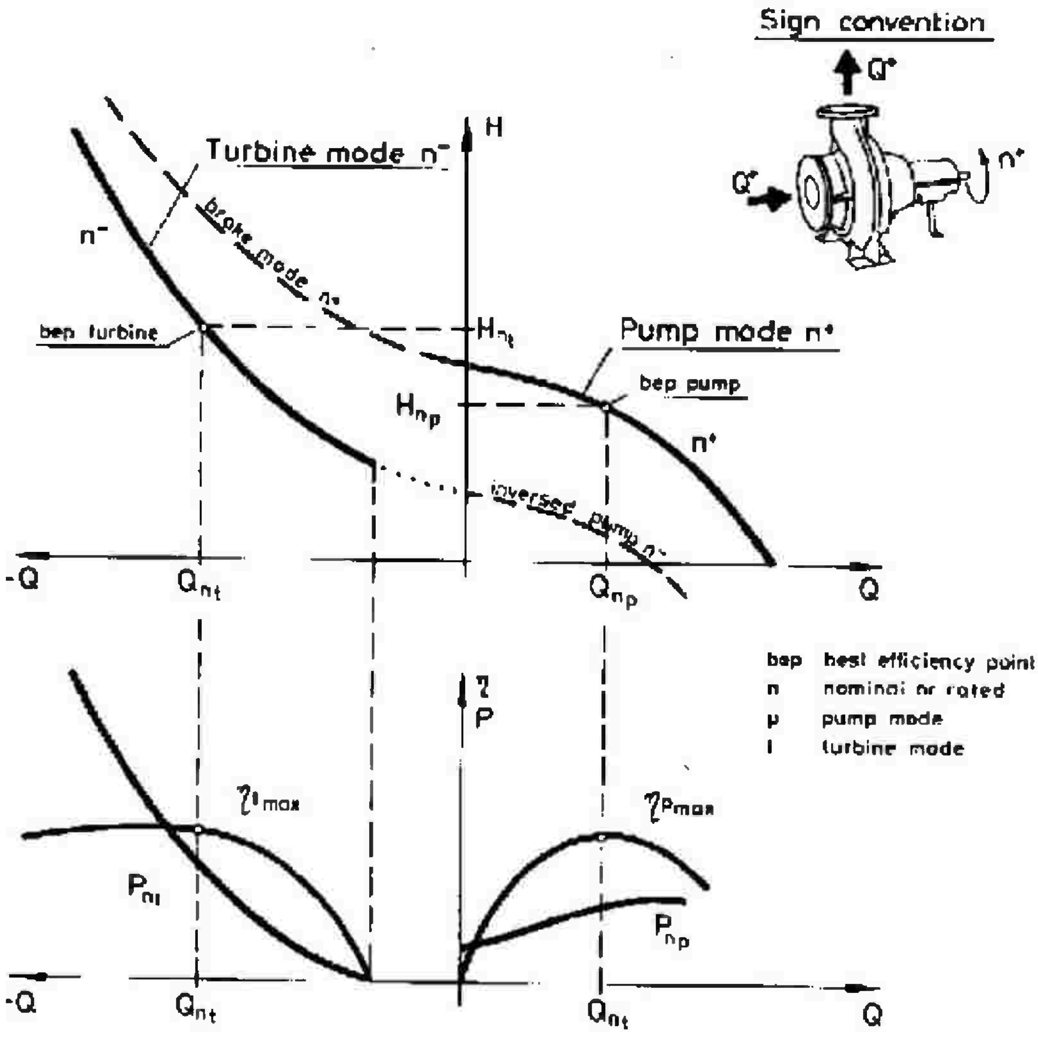

Figure (1-10) Complete characteristics of centrifugal pump for a constant speed [2]

The relationship between the pump and turbine mode is not the same for all types and sizes of pumps but depends on the flow pattern through the machine (expressed by the specific speed) and the losses incurred which are expressed by the efficiency of the machine. The problem is, that these losses in the pump mode are not exactly the same when flow is reversed in the turbine mode. Depending on the pump design (number of impeller vanes, vane angles, volute casing or guide vanes) the relation between turbine-mode and pump-mode performance may differ considerably from one machine to another although specific speed (and pump efficiency) may be the same.

Operating at the same speed in direct and reverse modes means that the head and flow rate is increased at the best efficiency point for the reverse mode. This rise in head and flow rate is due to the hydraulic losses as compared with the ideal fluid. Thus, the head in pumping mode is different from the turbine head with about twice the hydraulic losses [2].

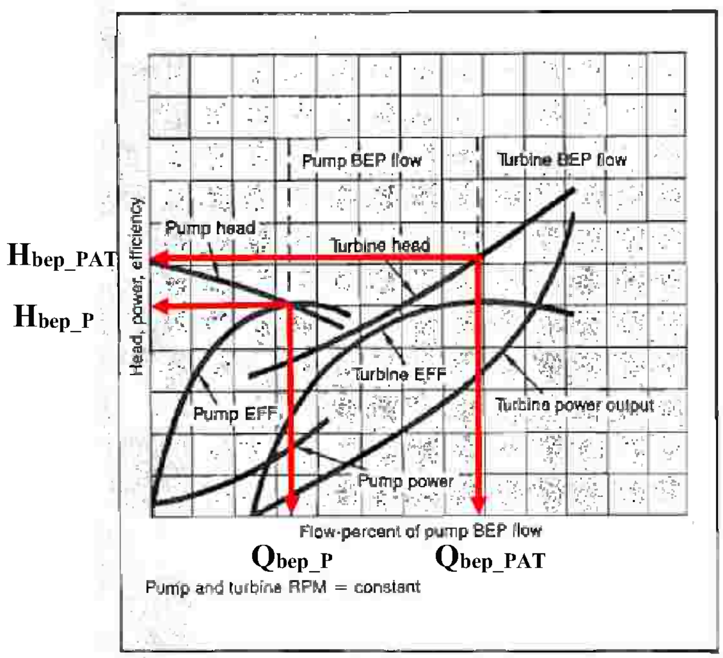

Figure (1-11) Normalized performance characteristics for a centrifugal pump operating in the normal pump mode and in turbine mode [7]

1.8. PATs versus Conventional Turbines

There are some differences between a PAT and a conventional turbine but the major one is that a PAT doesn't have a flow control device such that one used in conventional turbines which has the inlet guide vanes. This is an advantage and a disadvantage in the same time as it makes it cheaper but less versatile. On the other hand, an applicable solution to overcome this disadvantage appeared and used widely.

A PAT and a conventional turbine used for the same purpose were found to have major differences such as:

- Turbine was found to have a large runner, the blade curvature is reversed in the turbine.
- The runner of a turbine is different in shape with the impeller of a pump because the flow separation and diffusion losses should be minimized.

- From an economical point of view the use of a PAT is better than conventional turbine because of mass production as the fixed costs for a pump is lower than a turbine.
- The efficiency of a PAT is less than a conventional turbine due to the change in fluid path shape between a pump and a turbine. Efficiencies of pumps used as turbines are more often lower by (3 - 5%) [2].

1.9. Literature Review

1.9.1. Historical Overview

Using a pump to operate in the reverse direction as a turbine was arisen in the late nineteenth century in pumped storage plants. Pump-turbines have been used for several decades in large pumped storage schemes with power ratings of several megawatts. Pump-turbines are specifically designed to operate in both modes, pumping water into an elevated storage lake overnight at low tariff electricity and, during the day, generating peak demand electricity through the same machine operating in turbine mode [2].

In recent years, there has been a growing realization of the importance of electricity in bringing social and economic development to rural communities in the less developed countries of the third world, and in the rural areas of the developed countries with relative isolation from the national distribution system. The need of standard urban quality electrical supply at not much greater cost reverse the idea back to the old approach of using very small scale hydro power schemes [5].

Operating as a pump and as a turbine as well with the same machine has difficulties such that at the same rotational speed the pump will only receive a portion of the turbine rated capacity at a lower efficiency when the head is the same in both cases. For that reason when the same hydraulic machine is used as a pump-turbine, a two speed electric machine is used as a pump operating at a higher speed.

1.9.2. Pump as Turbine Review

The oldest pump as a turbine was installed in Orchard Mesa, USA, in 1926 [5]. Boothe and Lewis [10] stated the abnormal conditions of pumps such as operating pumps as turbines while reversing there direction. On the other hand, an early research aiming to the analysis of water hammer in large pumping stations was the milestone of using a pump as a turbine. This research results in the 'four-quadrant' characteristics of pumps. According to [5], Four-quadrant performance graphs of six representative pumps with different specific speeds were presented. The graph describes the relation between discharge, head, speed, and gate opening in four quadrants or modes of pump-turbine operation The shape of the turbine performance curves is a function of N_{sp}(pump specific speed)as mentioned by D. Thoma and Clifford, referring to [5]. Kittredge and Sprecter noticed by coincidence from experiments that a reverse running pump was an efficient turbine and discovered the possibility of using the same machine as pump and turbine as well according to [5].

Alatorre-Frenk [2] mentioned that there are two options for accommodating seasonal flow variations using PATs compared with conventional turbine results. PATs were found to be more economic than conventional turbines. A heuristic one was developed techniques for predicting the turbine-mode performance of a pump. The economics of different ways of accommodating water-hammer were explored as well.

Maher et. al. [11] discussed off-grid electrification options for low-income households in rural Kenya, where less than 2% were grid connected.

Nautiyal and Kumar [12] reviewed the work done in the area of pump working as turbine has been explained. Based upon the literature survey, analytical, experimental and computational works on pump as turbine have been discussed. Several methods for predicting the behavior of pumps in turbine mode have been developed but no method was appropriate for the entire range of specific speeds.

Motwania et. al. [13] performed a cost analysis of 3 kW capacity Pico hydropower plant in India by considering PAT and Francis turbine as a prime mover. Based on the analysis, the Annual life cycle cost (ALCC) and the cost of electricity generated per unit were found to be very less for PAT than that of Francis turbine.

1.9.3. Correlations of PATs:

The turbine-mode performance of a pump are available through many ways; the users of the pump, the manufacturers and prediction using tests are the main ways. The prediction based on the pump-mode performance is considered the easiest technique, and the only one that can be easily done by the users. This prediction requires finding the best efficiency point data for a pump (the head and the flow rate). According to several authors as Garay [6], such a technique can be used with reasonable accuracy. Some researchers claimed that this technique may produce small errors that can be corrected by adjusting the rotating speed of the turbine by changing the gear ratio [2] or the capacitance, when using an induction generator [14].

Other authors claim that the prediction which is only based on the pump mode performance is unreliable, because the pump manufacturers in some countries, such developing ones, do not follow the standards according to [5]. Another reason for the unreliability of such technique is that two pumps with the same best efficiency point may have different turbine performance, as there are differences in their geometry. The influence of geometry is more obvious with recently-built pumps in the radial flow regime, which exhibit more radical differences in impeller and casing designs [5]. In addition, in many developing countries there are no standards for the accuracy of manufacturers' data as noticed by Williams [15], so this data leads to unreliable solution if it is the only base for predicting the turbine performance.

Pelton and Wilson provided examples of wrong prediction of the turbine-mode when relying on the performance data from the manufacturers instead of tests referring to [5].

The method is based on relations found in the experimental testing of several centrifugal pumps in reverse.

Nepal Micro Hydro Power [16] predicted direct factors of 1.38 for the head and 1.25 for the flow rate of any pump operating as a turbine. These factors represented the relation between the BEP of a pump and a pump as turbine. However,According to [5], Smit did experiments on a PAT system the experimental data showed a factor of 2 for the head and 1.65 for the flow rate. This shows that while this method is simple to use, the factors vary considerably depending on pump make and even model.

An empirical method, based on curve fitting of experimental data, is presented in the BUTU(in Spanish) method (referring to the acronym of "Pump as Turbine") [2]. The method predicts turbine performance at both Best Efficiency Point (BEP) and values away from this point. This is very valuable as a selected PAT do not typically operate at exactly its BEP but somewhere close to it.

While the previous methods all determined turbine mode performance from pump curves, Derakhshan and Nourbakhsh [17] proposed another method to choose a pump for a PAT system based on the required turbine mode characteristics. Several correlations between pump and turbine best efficiency point were developed using experimental and theoretical analyses. The results of these correlations have some error up to $\pm 20\%$ [17].

1.9.3.1. Prediction based on pump geometry alone

The PAT geometry is not available as most manufacturers consider these materials as confidential. As a result, the prediction of the PAT performance using its geometry is complex without these geometrical data. Therefore, the available data for the prediction of a PAT is the pump-mode performance data. The pump data is available in the datasheets of any pump.

1.9.3.2. Prediction Methods Using Performance Alone

Most published methods concerning prediction of turbine-mode rely on proposing factors to convert the pump performance to the turbine performance depending on the best efficiency point of a pump. Three factors are presented. One is the head (H), the other is the flow rate (Q) and the last is the efficiency (η) assuming fixed speed for the pump and turbine modes. These factors may be constant or function of η_p (pump efficiency) or N_{sp} (pump specific speed).

Some methods took the pump-mode characteristics shape into consideration. On the other hand, few methods depended on prediction of turbine-mode performance off the BEP.

Naber and Hausch proposed the following factors based on theoretical background (and they imply that $\eta_{TE} = \eta_P$) based on [5]

$$\frac{Q_{TE}}{Q_P} = 1.3\,, \frac{H_{TE}}{H_P} = 1.35$$

Where E refers to Estimated

According to [5], Palgrave suggests that:

$$\frac{Q_{TE}}{Q_P} = 1.471\,, \frac{H_{TE}}{H_P} = 1.471, and\ \frac{\eta_{TE}}{\eta_P} = 1.1$$

and Sanchez suggests also that:

$$\frac{Q_{TE}}{Q_P} = 1.35\,, \frac{H_{TE}}{H_P} = 1.3$$

Stepanoff [18] stated that it is possible to describe the relation between the head and capacity of pump at the best efficiency point (BEP) at the same speed which is based on theoretical considerations as follows:

$$H_{TE} = \frac{H_p}{\eta_{ht}\eta_{hp}} = \frac{H_p}{\eta_h^2}$$

$$where\ \eta_{ht} = \eta_{hp}$$

$$Q_{TE} = \frac{Q_p}{\eta_h}$$

$$N_{st} = N_{sp}\eta_h$$

Where

H is the total head at BEP

Q is the capacity

N_s is the specific speed

η_h is the hydraulic efficiency, taken the same for the turbine and the pump.

As the exact value of the hydraulic efficiency was never known, $\sqrt{\eta_p}$ can be taken as an approximate value. Subscripts t and p refers to the operation as a turbine and as a pump respectively.

Childs proposed a simple method for prediction in terms of efficiency alone and this method was mentioned also by Garay [6], Hancock, McClaskey, Lundquist and Thome according to [5], (supposing $\eta_{TE} = \eta_P$)

$$\frac{Q_{TE}}{Q_P} = \eta_p^{-1} \ , \qquad \frac{H_{TE}}{H_P} = \eta_p^{-1}$$

According to [5], Ventrone and Navarro proposed that:

$$\frac{H_{TE}}{H_P} = \eta_p^{-2} \ , \qquad \frac{P_{TE}}{P_P} = \eta_p^{-1}$$

Therefore $\eta_{TE} = \eta_P$

$$\frac{Q_{TE}}{Q_P} = \eta_p^{-1} \ , \qquad \frac{H_{TE}}{H_P} = \eta_p^{-2}$$

On the other hand, Sharma stated that the relation between pump and turbine heads and flow rates are as follows where the flow rate and head at best efficiency point for the pump is related to the turbine flow rate and head by the maximum efficiency (η_{max}) of the pump referring to [5]:

$$Q_{TE} = \frac{Q_p}{\eta_{max,p}^{0.8}}$$

$$H_{TE} = \frac{H_p}{\eta_{max,p}^{1.2}}$$

Meanwhile, McClaskey and Lundqvist used the following equation to obtain Q_t according to [5]:

$$Q_{TE} = \frac{Q_p}{\eta_{\max}}$$

Williams [15] suggests some corrections to Stepanoff formula as follows:

$$\frac{Q_{TE}}{Q_P} = 1.1\,\eta_p^{-0.8} \ , \qquad \frac{H_{TE}}{H_P} = 1.1\,\eta_p^{-1.2}$$

Referring to [5], Alatorre-Frenk made curve fitting for a small set of pump and PAT data, and obtained the following formula:

$$\frac{Q_{TE}}{Q_P} = \frac{0.85\eta_p^5 \ + 0.385}{2\eta_p^{9.5} + 0.205},$$

$$\frac{H_{TE}}{H_P} = \frac{1}{0.85\eta_p^5 \ + 0.385}$$

$$and\ \eta_{TE} = \eta_P - 0.03$$

S. Derakhshan and A. Nourbakhsh [17] deduce a correlation from experiments relating the best efficiency point of the turbine mode using the best efficiency point in the pump mode using six pumps with different specific speeds. They used dimensionless parameters to express the head, flowrate, power and effeciency; $\psi, \phi\ and\ \pi.$

Where:

$$\psi = \frac{gH}{n^2D^2}\ ,\phi = \frac{Q}{nD^3}\ and\ \pi = \frac{p}{\rho n^3D^5}$$

Where n is the rotational speed in (rps)

D is the diameter of the impeller.

P is the power in (W).

The PAT works in higher flow rate and head in comparison with the pump mode. Fig. 6 shows the BEP of PATs, based on the following dimensionless parameters, which were also used by other researchers:

$$h = \frac{H_T}{H_p}\ ,q = \frac{Q_T}{Q_p}, p = \frac{P_T}{P_p}\ and\ \lambda = \frac{\eta_{T,max}}{\eta_{p,max}}$$

These dimensionless parameters are obtained for various pumps with different specific speeds as shown in Figure (1-12).

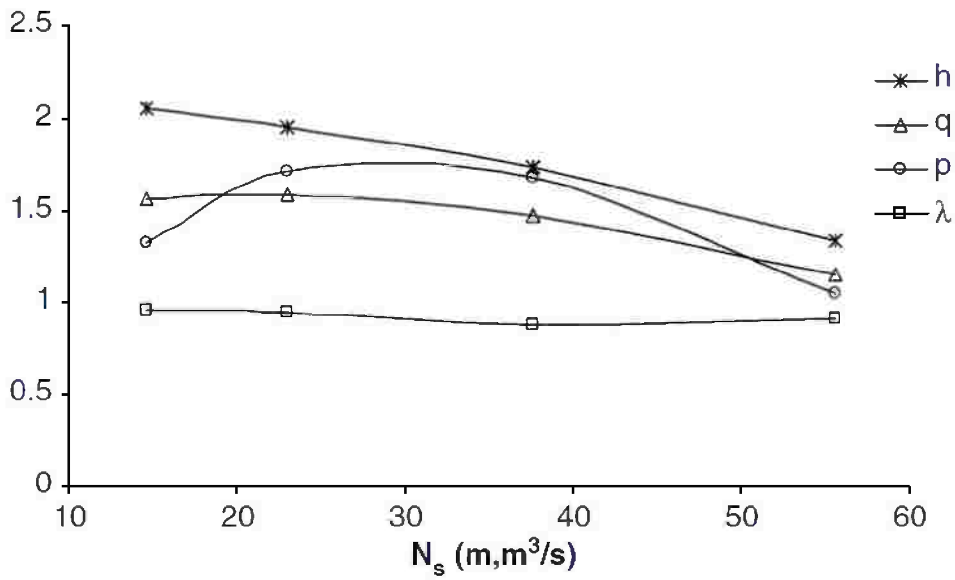

Figure (1-12) Dimensionless BEP of tested PATs [17]

Using experimental data, some relations were obtained to calculate the BEP of the PAT based on the BEP of the pump mode. These relations are only valid for low-specific-speed centrifugal pumps:

$$\gamma = 0.0233\alpha_p + 0.6464$$

$$\alpha_t = 0.9413\alpha_p + 0.6045$$

$$\beta_t = 0.849\beta_p - 1.2376$$

Where γ, α_t and β_t are dimensionless parameters.

$$\alpha_p = N_p.\frac{Q_{pb}^{0.5}}{\left(g.H_{pb}\right)^{0.75}}$$

$$\gamma = (h)^{-0.5}.\frac{N_t}{N_p}$$

$$\alpha_t = N_t.\frac{Q_{tb}^{0.5}}{(g.H_{tb})^{0.75}}$$

$$\beta_p = \frac{N_p.P_{pb}^{0.5}}{\rho^{0.5}.\left(g.H_{pb}\right)^{1.25}}$$

Where α_p (m,m3/s) and β_t (m,W) are the pump and turbine dimensionless specific speeds. In the above equations, H (m), Q (m3/s), P (W) and N (rpm) are head, flow rate, power and rotational speed, respectively.

The rated errors of the dimensionless parameters h and q is illustrated in Figure (1-13) for several methods and it showed that the new approach obtained by S. Derakhshan and A. Nourbakhsh [17] are applicable for a wide range of flow rates and heads.

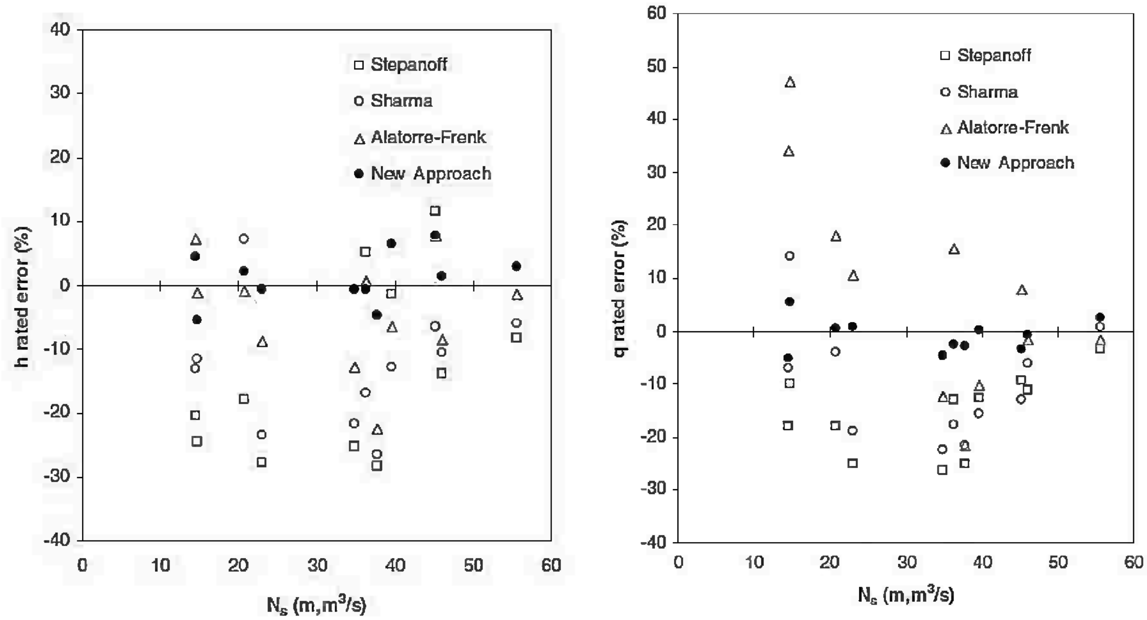

Figure (1-13) Rated errors in h and q from experiment and various methods [17]

The curves estimated by this method and experimental data were compared in Figure (1-14). The results were in good agreement with experimental data. However, it must be noticed that this method can only provide an approximate view of the characteristic curves of PAT.

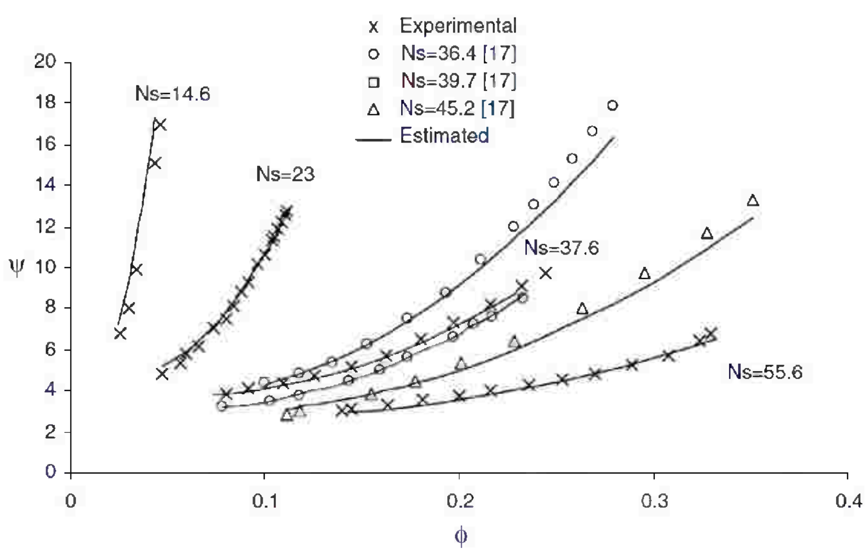

Figure (1-14) Measured and estimated PATs dimensionless head curves [17]

They also mentioned dimensionless head and power curves of a PAT can be estimated as below using second and third order polynomials, respectively:

$$\frac{H_t}{H_{tb}} = 1.0283\left(\frac{Q_t}{Q_{tb}}\right)^2 - 0.5468\left(\frac{Q_t}{Q_{tb}}\right) + 0.5314$$

$$\frac{P_t}{P_{tb}} = 0.3092\left(\frac{Q_t}{Q_{tb}}\right)^3 + 2.1472\left(\frac{Q_t}{Q_{tb}}\right)^2 - 0.8865\left(\frac{Q_t}{Q_{tb}}\right) + 0.0452$$

These equations will be used in chapter (3) to validate the experimental results.

1.9.3.3. <u>Conversion Factors in Terms of Specific Speed Only</u>

All methods based on the specific speed are empirical, ***i.e.*** based on test data.

Conversion factors were published in two graphs by Diederh which is valid for $\eta_{TE} \geq \eta_P$. The following formula is concluded from the graphs as mentioned in [5]:

$$\frac{Q_{TE}}{Q_P} = 1.402\ N_{sp}^{-0.171}\ , \frac{H_{TE}}{H_P} = 1.556\ N_{sp}^{-0.174}$$

Gopalakrishnan [19] presented two graphs, to show the trend only not for using it in the prediction (assuming $\eta_T = \eta_P$). The formula deduced from the curve fitting was as follows:

$$\frac{Q_{TE}}{Q_P} = 1.86 - 0.551\ln(5N_{sp}) + 0.11[\ln(5N_{sp})]^{2.2}$$

$$\frac{H_{TE}}{H_P} = 2.6 - 9.1\ln(5N_{sp}) + 7.96[\ln(5N_{sp})]^{1.1}$$

Grover presented three linear graphs valid for 1.96< N_{sp}<10.78, the equations, according to [5] are:

$$\frac{Q_{TE}}{Q_P} = 2.643 - 1.399\ N_{sp}\ , \qquad \frac{H_{TE}}{H_P} = 2.693 - 1.212\ N_{sp}$$

$$and\ \eta_{TE} = 0.893 + 0.0466\ N_{sp}$$

1.9.3.4. <u>Conversion Factors in Terms of Both Efficiency and Specific Speed</u>

According to [5], Schmiedl was the first to propose a prediction method based on both η_P and N_{sp} (valid for 1.96 < N_{sp}< 10.29):

$$\frac{Q_{TE}}{Q_P} = -1.378 + 2.455(\eta_P \eta_T)^{-0.25},$$

$$\frac{H_{TE}}{H_P} = -1.516 + 2.369(\eta_P \eta_T)^{-0.5}$$

$$and\ \frac{\eta_{TE}}{\eta_P} = 1.158 - 0.265N_{sp}$$

Chapallaz et. al. [2] displayed two graphs based on test data with specific speed as the abscissa and, respectively, a flow and a head conversion factors as the ordinates. A curve fitting, based on the model proposed by Anderson, although the correlation is not excellent (Chapallaz's curves were drawn by hand and do not follow a uniform trend)[2], the obtained equations are the following:

$$\frac{Q_{TE}}{Q_P} 1.12\ \eta_p^{-0.6}[1 + (0.4 + lnN_{sp})^2]^{0.15},$$

$$\frac{H_{TE}}{H_P} = 1.1\eta_p^{-0.8}[1 + (0.3 + lnN_{sp})^2]^{0.3},$$

$$\&\ \eta_{TE} = \eta_P - 0.03$$

1.10. Scope and Objectives

In the previous sections the pump as a turbine operation were discussed. In addition the applications of using pumps as turbines were illustrated concisely. Moreover, the limitations of using pumps as turbines were presented as well as the ways of overcoming such limitations.

The main objectives of the present work are as follows:

1- Running special type of pumps; inline pumps in direct (pump mode) and reverse (turbine mode) at the same rotational speeds and deduce the relation between the performance in both modes experimentally.
2- Simulating the real pump case in both modes numerically using CFD.
3- Verification and validation of the computational results with experimental ones.

The present thesis consists of six chapters. Chapter one is an introduction and literature review about using pumps as turbines. In addition, chapter two illustrates the experimental setup components and experimental procedures for pump mode and turbine mode. Moreover, in chapter three the experimental results are discussed and full characteristic curves for both modes are well defined. On the other hand, chapter four includes the governing equations and numerical model used in the numerical study using CFD. Furthermore, chapter five showed the computational results from ANSYS FLUENT and a comparison between the computational and the experimental results are illustrated with complete analysis of the differences between them. At last, a conclusion from the present study and recommendations for future work are discussed in chapter (6).