COMPREHENSIVE REVIEW OF PUMP AS TURBINE (PAT)

Abdulbasit Nasir Jemal[1], Misrak Girma Haile[2]
Department of Mechanical Engineering, Addis Ababa Science and Technology University, Addis Ababa, Ethiopia.
abdulbasitnasir@ymail.com[1], misrakmgh@gmail.com[2]

***Abstract* -** The turbine is a heart of power generation in a hydro-electric power system. A variety of different turbines are available for that purpose. The common types of Hydraulic turbines are; Pelton, cross flow, Francis, Kaplan, and propeller turbine. However, using conventional turbines for low head and flow rate (i.e. micro hydropower) applications are not economically feasible. A low-cost alternative is to use the pump as a turbine. In this paper, existing Peer-reviewed articles from (Scopus, google scholar, umbrella, etc.) that are directly related to pump running as a turbine are collected and reviewed. Theoretical, numerical, and experimental investigations are considered. Performance improvement techniques for PAT are summarized and research gaps in related works are identified.

***Keywords* -** Turbines, Pumps, Review, Pump as Turbine, PAT.

List of Abbreviations and symbols -

BEP	Best Efficiency Point
b2	impeller inlet width (mm)
b3	volute outlet width (mm)
C	absolute velocity of the fluid
CFD	Computational Fluid Dynamics
C_H	coefficient of head
C_Q	coefficient of flow rate
D1	impeller outlet diameter (mm)
D2	impeller inlet diameter (mm)
D3	volute base circle diameter (mm)
D4	volute inlet diameter (mm)
g	gravitational acceleration (m/s2)
H_{bep}	head at best efficiency point (m)
H_p	pump head (m)
$H_p^{"}$	theoretical head (m)
I	turbulence intensity
ICEM	Integrated Computer Engineering and Manufacturing
k	turbulent kinetic energy (J/kg)
N-S	Navier-stock
n_{SP}	specific speed of pump (m, m3/s)
n_{ST}	specific speed of turbine (m, m3/s)
PAT	Pump as Turbine
PDE	Partial Differential Equation
PISO	Pressure-Implicit with Splitting of Operators
P_h	hydraulic power (W)
P_m	mechanical power (W)
Q_{bep}	flow rate at best efficiency point (L/s)
Q_p	pump flow rate (L/s)
rpm	revolution per minute
Re	Reynold's number
RNG	Re-normalization group
SST	shear stress transport
U	peripheral/tangential velocity of wheel (m/s)
V	relative velocity of fluid (m/s)
y+	y plus
Z	number of blades
α	flow angle (0)
β1	blade outlet angle (0)
β2	blade inlet angle (0)
ηbep	efficiency at best efficiency point
η_p	pump efficiency (%)
µ	slip factor for pump operation
λ	slip factor for turbine operation
ε	turbulence dissipation
ω	specific dissipation rate (rad/s)
ϕ	discharge number
Ψ	head number
π	power number

I. INTRODUCTION

The turbine is a heart of power generation in a hydro-electric system. A variety of different turbines are available for that purpose. However, using conventional turbines for low head and flow rate (micro hydro power) applications is not economically feasible [1]. A low-cost alternative is to use a Pump as Turbine (PAT).

Pumps are widely used for irrigation, industrial and domestic applications, transportation of liquid, industrial processes, as well as heating and cooling systems [2]. Flow directions of the pump are shown in Fig. 1.

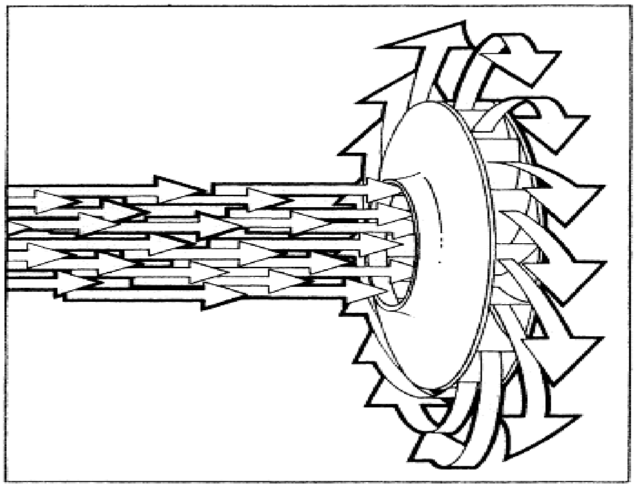

Fig .1 Flow direction of the pump [2].

In addition to the basic functions, pumps can be used to generate electricity when operating in a reverse way. The basic hydraulic theory of both pump and turbine modes to be is the same, Fig. 2. However, the behavior of real fluid flow including friction and turbulence result is different [3].

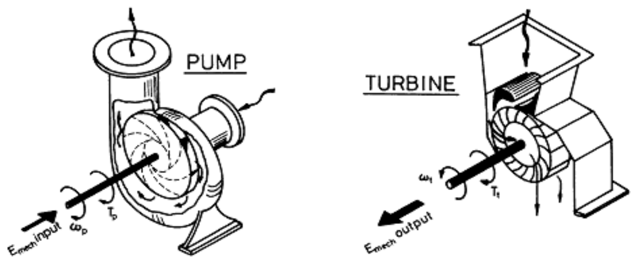

Fig .2 Flow direction in pump and turbine modes [4].

PAT can be applied to micro-hydropower plants as well as water supply piping and distribution systems [5-9], reverse osmosis systems [10], pressure reducing system [11, 12], energy recovery in irrigation networks and industries [13-22].

Pumps have various advantages compared to turbines, such as availability, proven technology, low initial installation and maintenance cost, available for a wide range of heads, and flows [1, 7]. Pump impellers have no significant disadvantages in turbine mode, but the efficiency coefficient of a pump in turbine operation is lower [16]. The pump manufacturers do not provide characteristic curves of their pumps working as turbines. This makes it difficult to select a suitable pump to run as a turbine for a specific application [23].

In this paper, existing peer-reviewed articles from (Scopus, google scholar, umbrella, etc.), that are directly related to pump running as a turbine, are collected and reviewed. Peer-reviewed articles other than review papers are considered. Theoretical, numerical, and experimental investigations are studied. Performance improvement techniques for PAT are summarized and research gaps in related works are identified.

Pump manufacturers only supply pump mode performance curves and that makes it difficult to predict the performance of the pump working as a turbine. Three main ways of conducting researches on the pump running as a turbine are: analytical/theoretical method, the numerical/computational method, and the experimental method.

II. THEORETICAL INVESTIGATION OF PAT

A theoretical investigation is the study of fluid flow problems analytically, using partial differential equations without any approximations. Many attempts have been made to predict pump in turbine mode performance theoretically. A mathematical model was applied to investigate the installation of PAT, by solving the system of partial differential equations into ordinary differential equations [7].

It was found that pump impellers have no significant disadvantages in turbine mode and the efficiency coefficient of a pump in turbine operation is hardly lower (in some cases even higher) than in pump operation [16]. The efficiency at the BEP in turbine mode corresponds approximately to the efficiency coefficient in pump mode.

$$\eta bep_T = \eta bep_P \pm 0.02 \tag{1}$$

If the rotational speeds are the same for both modes, there are equal and opposite heavy line velocities for pump mode and turbine modes based on the infinite-blade theory. Consequently, Euler head in pump and turbine mode are the same [19,24].

$$H_{Euler} = \frac{u_1 v_{u1} - u_2 v_{u2}}{g} \tag{2}$$

Where V_{u1} and V_{u2} represent the peripheral component of velocity at the high-pressure side and low-pressure side, respectively.

In equation (2), v_{u2} is negligibly small in general and, as a result, the Euler head can be as

$$H_{P\ Euler} = \frac{u_1 v_{u1}}{g} = H_{T\ Euler} \quad (3)$$

Due to slip of finite blade number, pump and turbine theoretical head is given by:

$$H_p^{"} = \mu H_{p\ Euler} \quad (4)$$

$$H_t^{"} = H_{t\ Euler}/\lambda \quad (5)$$

Where μ is a slip factor for pump operation $\mu < 1$, λ is a slip factor for turbine operation. The slip factor for reverse mode is approximately equal to 1.0 [4].

From the fundamentals of energy transfer in turbines, the output mechanical shaft power and Euler turbine head can be represented by [25]:

$$P = \rho g Q H_t - P_{mech} - P_{leak} \quad (6)$$

$$H_t = \sigma H_{Euler} \quad (7)$$

$$H_{Euler} = \frac{(U_2 C_{u2} - U_1 C_{u1})}{g} = \frac{(U_2 C_{m2} cot\alpha_2 - U_1(U_1 - C_{m1} cot\beta_1))}{g} = \frac{(U_2 \left(\frac{Q}{A_2}\right) cot\alpha_2 - U_1\left(U_1 - \left(\frac{Q}{A_1}\right) cot\beta_1\right))}{g} \quad (8)$$

PAT's P-Q curve is in inverse proportion to the inlet area A_2. PAT's required pressure head can be represented as the sum of the theoretical head and the losses. PAT's hydraulic efficiency can be represented by:

$$\eta_h = \frac{H_t}{H_t + h_{total}} \quad (9)$$

Both theoretical head and total hydraulic loss increase with the increase of the blade thickness.

In the study of S. Barbarelli et al. [26], a statistical method combined with a numerical model for selecting a pump running as turbine in micro-hydro plants is applied. The information of the site (flowrate and head) allow calculating two coefficients, $C_Q$and C_H, respectively.

For searching of energy conversion characteristics of PAT in detail, theoretical analysis and empirical prediction can only outline the energy conversion characteristics in the macroscopic point of view [27].

By considering the PAT's flow, i.e. the reverse of pump, the theoretical head transferred from the fluid to the runner is smaller than actual head HT between the inlet and exhaust nozzles because of the hydraulic losses Z [28]. Because of power losses, the power PT available at the coupling of the turbine is smaller than the hydraulic power $\rho g H Q$. The PAT's overall efficiency η_T is:

$$\eta_T = \frac{P}{\rho g H Q} \quad (10)$$

The basic parameters of pump and turbine mode are specific and deal with (i) geometric characteristics; (ii) flow and geometrical angles, and (iii) hydraulic and power losses. These parameters may be known from pump geometry or can be estimated through an optimization procedure [28].

The effect of variable guide vane numbers on the performance of pump as turbine was analyzed theoretically, having the turbulence kinetic energy under variable working conditions. The asymmetry of the volute and rotor-stator interaction causes turbulence kinetic energy concentration to appear in pump as turbine. Theoretically, the turbulence kinetic energy equals half of the product of turbulent velocity fluctuation variance and fluid mass, which generally is expressed by the physical quantity k and can be calculated through the turbulence intensity I [29].

$$k = \frac{3}{2}(uI)^2 \quad (11)$$

$$I = 0.16 Re^{\frac{-1}{8}} \quad (12)$$

Many attempts have been made to predict turbine mode performance by using analytical/theoretical model but the percentage of deviation is comparatively large compared to the actual performance. It can be concluded that the theoretical method is used to study flow problems with a few variables, while it is difficult to analyze pumps running as a turbine.

III. NUMERICAL INVESTIGATION OF PAT

A numerical investigation is the study of fluid dynamics problem using computer software, in this case approximating partial differential equations into system algebraic equitation. Computational Fluid Dynamics (CFD) is an active design tool for predicting the performance of centrifugal pumps running in turbine mode.

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Flow Conditions for PATs Operating in Parallel was performed using the CFD model [1]. The k- ε turbulent model is adopted, the domain has 713,954 cells. Variables, such as mass flow rate, outlet pressure, and rotational speed, and rotating zone are specified in Fig. (3). When the flow is different from the normal rated conditions, two PATs in parallel can better cover it.

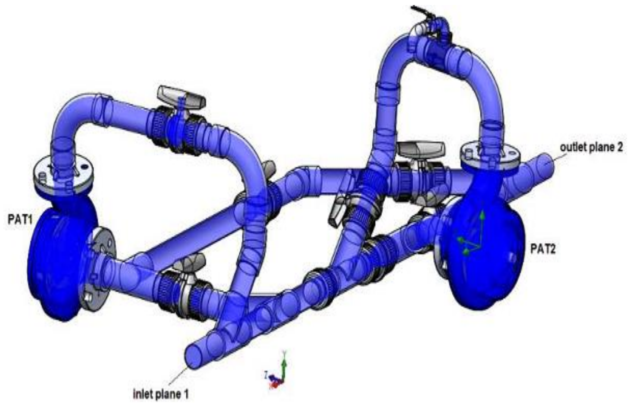

Fig .3 PAT's system set-up [1].

The behavior of the pressure distribution when PAT installing in a water network was analyzed [11]. The PAT model was built in SolidWorks, then simulated by using CFD, the boundary conditions are specified and the number of elements is about 100,000. The global variables are simulated in the CFD model and used to evaluate the overall PAT characteristics. Results of numerical and experimental values do relatively not agree.

The pressure fluctuation characteristic of the hydraulic turbine at a single rotational speed with guide vane was analyzed [13]. Unlike others in this work, the PISO algorithm is adopted to solve the Navier-Stokes equations. RNG k-ε turbulence model was used for this specific purpose. The grid independence is verified. the final grid number is 1,042,502. In the impeller blade region, the pressure fluctuation in the pressure surface is lighter than that of the suction surface.

The slip phenomenon was investigated and the effective value of slip factors for both direct and reverse modes is obtained. Pump and turbine head impeller can be predicted by computational fluid dynamics [19].

$$H_p = \frac{H_p}{\eta_{hp}} \tag{13}$$

$$H_t = H_t * \eta_{ht} \tag{14}$$

ANSYS-CFX was selected for the solution. A grid sensitivity analysis was performed. The final elements number of the fluid volume was 4,154,084. Testing different turbulence models, and RNG k- ε model confirms a good accuracy in the performance prediction of PAT. At BEP the effective value of slip is 0.28 for pump and 0.24 for turbine mode. Pump and PAT can be related

$$\frac{H_t}{H_p} = \frac{1-s_t}{1-s_p}\frac{1}{\eta_{ht}\eta_{hp}} \tag{15}$$

A Performance Prediction Method for three different Pumps as Turbines using a computational fluid dynamic modeling approach was presented [23]. Results have been first confirmed in pumping mode using data supplied by pump manufacturers. Then, the model results have been compared to experimental data for PAT. The analyzed pumps have three different specific speeds. The main characteristics are summarized in Table (1). From the CFD model results, the specific head, capacity, power, and efficiency have been evaluated and the best efficiency point of all the analyzed pumps was found.

Table 1 Characteristics of Different Pumps [23]

	Impeller diameter (mm)	Delivery outlet diameter (mm)	$H_{bep}(m)$	$Q_{bep}(m^3/h)$
$(N_s\ 37.6)$	190	80	39	148
$(N_s\ 20.5)$	200	70	60	45.4
$(N_s\ 64.0)$	120	80	3.9	54

For all cases, the maximum value is lower than the pump mode value. For low specific speed pumps, this maximum value is roughly equal to the pump mode, while for high specific speed pumps, it is different. Model results for Pump 1 at 2900 rpm are shown in Fig. 4 and the pressure distribution at the BEP in the 0–9 bar pressure range is presented.

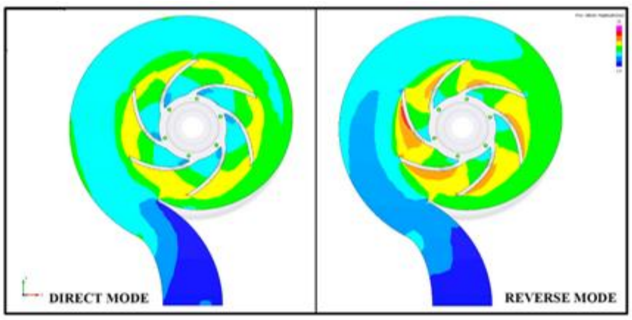

Fig .4 Pressure distribution of model results for pump 1 (Ns = 37.6) at 2900 rpm [23].

It is clear that maximum pressure is applied in reverse mode. This indicates that blade modification is required when using the pump as a turbine.

A single-stage centrifugal pump was selected for energy conversion characteristic of pump as turbine [27]. The centrifugal pump design parameters are: flow rate Q = 12.5 m^3/h, head H= 30.7 m, rotating speed n = 2900 rev/min, specific speed ns = 48, and the shape of the blade is cylindrical. The impeller is divided into six regions as shown in Fig. (5) by the radius.

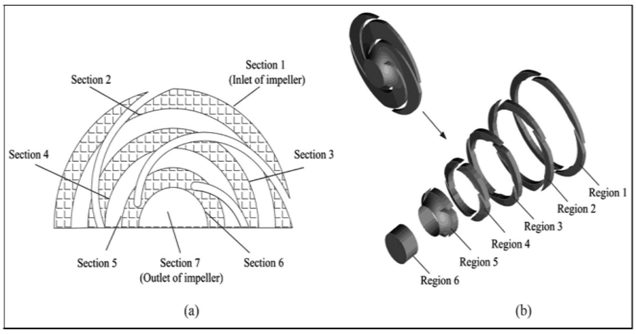

Fig .5 Schematic diagram of the impeller division of the PAT [27].

ICEM was used to generate a structured hexahedral mesh for each part. The grid number is about 1.1 million. The ANSYS-FLUENT was selected to calculate PDE. The standard k–e turbulence model was chosen. The results show that the front and middle part of the impeller i.e. (0.6–1.0) D2) are significant parts for energy conversion; in the rear area of the impeller. Thus, the PAT blades need to be optimized to improve the performance, especially in the impeller blade rear area.

Turbulence model has its own influence on the accuracy of the result. In the numerical method, a variety of turbulence models are available, among them RNG k-ε [13, 19], standard k-ω [22], and standard k- ε [1, 6, 11, 23, 25, 27, 29 and 30] are used to predict the effects of turbulence on the system. Because of the irregularity shape of impeller unstructured mesh found to be the best one, but the result of review indicates, many researchers used structured mesh.

Pressure fluctuations are very vital characteristics in pump turbine's operation. 3D numerical simulations using SST k-ω turbulence model was carried out to predict the pressure fluctuations distribution in a prototype pump-turbine at pump mode [31]. Three operating points with different flow rates and different guide vanes openings were simulated. The numerical results show how pressure fluctuations at blade passing frequency and its harmonics vary along with the whole flow path direction, as well as along the circumferential direction.

A complete numerical detail for a selection of centrifugal pump as turbine with a rotational speed of 2880rpm for micro-hydro power plants was provided [32]. The maximum power output is 17.78 KW. The BEP of centrifugal pump operated in pump mode was observed as 70% of that to turbine mode was 35.45%. It was clear that the pump relatively operated as a turbine is with lower efficiency. The future work focuses on the further development of PAT performance.

Hydraulic design and optimization of a modular pump-turbine runner were performed [33]. A mesh discretization study was performed and found that convergence was reached around a nine million cell mesh. The runner's performance was characterized in both the pump and turbine modes for its designed working conditions for both the initial and optimized design.

Numerical simulations were done to predict the distribution of pressure fluctuations with different numbers of runner blades in turbine mode using the k-ω turbulence model [34]. The two factors that influence the distribution of pressure fluctuations found to be the flow rate and a number of blades, especially at blade passing frequency along the circumferential direction.
The power loss and radial force characteristics of the pump as a hydraulic turbine under gas-liquid two-phase condition was studied [35]. Based on the N-S equation and standard k-e turbulence model, computational fluid dynamics technology was used to simulate the flow field in a hydraulic turbine. The result illustrates that the gas content has a serious effect on PAT performance. Under the two-phase condition, the fluid velocity distribution in turbine mode is uneven, and the power loss is not uniform enough when the gas content is lower.

Mesh refine [1, 6, 11, 19, 22, 23 and 25] especially at the boundary and the inlet of the pipe is used to:

- save memory capacity
- decrease computational time

In all cases, the value of the numerical result is higher than the experimental one. This indicates that numerical methodology is not enough to determine the exact solution of pump running as a turbine.

IV. EXPERIMENTAL INVESTIGATION OF PAT

The experimental investigation is the study of fluid flow problem using physical laboratory. Knapp (1941) published the complete pump characteristics for a few pump designs based on experimental investigations [36].

Two equal PATs working in parallel and single-mode were performed [1]. Several experimental tests at a different flow rate (200 to 1150) were carried out for the two configurations by regulating the flow rate. During each test, the data were recorded. The performance in parallel design conditions illustrates a peak efficiency with less shock losses within the impeller.

The hydraulic facility, composed of one closed pipe, an air-vessel tank (allows to regulate the flow and pressure in order to reach the steady flow conditions) a recirculating pump, an open free surface tank, ball valves, an electromagnetic flowmeter, two pressure transducers were used to determine the behavior of the pressure distribution along the PAT [11].

The pressure fluctuation characteristic of a hydraulic turbine with guide vane using the test bench Fig. 6 was studied [13], with different flow rate. The pressure fluctuation is also different, the greater the flow rate, the more serious the pressure fluctuation.

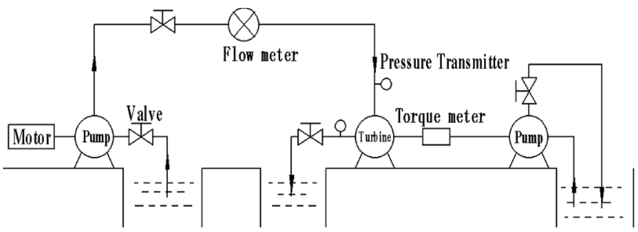

Fig .6 Test bench for hydraulic turbine [13]

Three centrifugal pumps with different heads and flow rates have been modeled and tested in the test bench [13], unitless parameters are calculated.

Head number $\Psi = \frac{gh}{n^2 D^2}$ (16)

discharge number $\phi = \frac{Q}{nD^3}$ (17)

power number $\pi = \frac{p}{\rho n^3 D^5}$ (18)

efficiency $\eta = \frac{P}{\rho Q H}$ (19)

Experimental investigations and laboratory measurements on the hydraulic machine are conducted at a turbine test rig to validate the theoretical work that is used to predict the behavior of a centrifugal stainless-steel pump in turbine operation [16].

A centrifugal multistage end-suction pump chosen for experimental investigation [17]. The test rig consists of the pump with a synchronous motor, two pressure transducers, a magnetic flow meter, a watt meter, and an optical speedometer. The focus of the study was comparing direct and indirect water supply a network and direct pumping found to be considered to be more efficient than indirect pumping.

Equation 16 through 19 used to determine the characteristics of the pump running as a turbine. A laboratory model of PAT test rig was used to conduct research on energy conversion characteristics of the pump as a turbine [27]. The main equipment composed of an electric motor, a feed pump, a control valve, an electromagnetic flow meter, a differential pressure transducer, a PAT, a torque meter, and an energy dissipation pump.

The characteristics of energy transformation, especially within impeller, plays significant roles for further optimum design of the pump as turbine, the area of (0.6–1.0) D2) are important parts for energy conversion; in the rear area of the impeller, this at least shows that the PAT blades need to be optimized to improve the performance, especially in the impeller blade rear area.

The feasibility study of using pumps in turbine mode in small hydroelectric stations was presented [37]. To regulate the power applying variations in the turning speed of the turbine-generator set.

The study of A. Carravetta et al. [38] affinity law for the evaluation of the behavior of a single machine under variable speed. The study shows that the use of performance curves calculated using affinity law and Suter parameters produces a limited error in the evaluation of the head drop, granting the satisfaction of the correct hydraulic constraint (pressure level within the network). Meanwhile, the error in terms of mechanical efficiency is greater but still acceptable in a limited range of velocity difference between a prototype and simulated machine.

An end suction centrifugal pump with a specific speed of 15.36 (m, m3/s) was tested experimentally, to determine the performance characteristic of the pump in reverse mode. The result showed that a centrifugal pump can satisfactorily be operated as a turbine without any mechanical problems. The best efficiency point (BEP) for PAT was found to be lower than BEP in pump mode [39].

There are some variations between numerical and experimental results in the case of PAT. As recommended by many researchers the variation can be minimized through development by using a fine grid and introducing appropriate turbulence models.

The solution of the pump running as a turbine highly depends on different conditions like flow rate, head, impeller diameter, and rotational speed. Tasting and measuring PAT output with various parameters as done by many investigators is very important to predict a relatively more accurate solution.

V. MODIFICATION IN PAT

The result of the study shows the efficiency of PAT is 12.4 % [12], 4% [21], 34.55% [32], 19 % [39] lower than direct pump mode. The result of all investigations shows that the pump has low efficiency in turbine mode, geometric modification is required to improve the efficiency pump working as a turbine.

The influence of the different number of blades (10-13) with guide vane on the performance of PAT was investigated numerically and experimentally [22]. To perform the numerical simulation, applying ANSYS CFX and k-ω the turbulence model was used. Meshing is performed by ICEM. The grid independence is verified. The total number of grids is 15.287 million.

The PAT test bench consists of a model PAT, an overrunning clutch, an electric motor, a centrifugal pump, throttle valves, and bypass valves, and a pool is used to validate relationship curves among the head, efficiency, power, and flow rate of the PAT, which are drawn and compared with the simulation result. Results show that when the number of blades is 10 at the same flow rate, the highest efficiency is achieved and the internal flow becomes stable.

PAT covering different specific speeds was designed to explore the effects of blade thickness on the performance [25]. Numerical and experimental methodologies are applied. ICEM-CFD was used to generate the structured hexahedral grid for the components. A grid-independent test was performed. The final mesh number is over 1 million and the standard k-ε model was applied. After modification, a new impeller was manufactured and verified in the test rig. The efficiency decreases with increasing blade thickness. Using the thinner blades with sufficient strength to obtain higher efficiency was recommended. Based on the founding of research appropriate material selection is an important factor for the improvement of PAT.

Energy conversion characteristic of pump as turbine by considering a single-stage and single-suction centrifugal pump running in the reverse model as the turbine was selected [27]. The centrifugal pump design parameters are: flow rate Q = 12.5 m^3/h, head H= 30.7 m, rotating speed n = 2900 rev/min, specific speed ns = 48, and the shape of the blade is cylindrical. The result of the study shows that PAT blades need optimization to enhance the performance, especially in the impeller blade rear area.

Blade profile optimization by using a numerical approach was performed [30]. Coordinate values of the control point 1-8 in Fig. 7 were selected as the optimization design variables. The control point 9 remains unchanged.

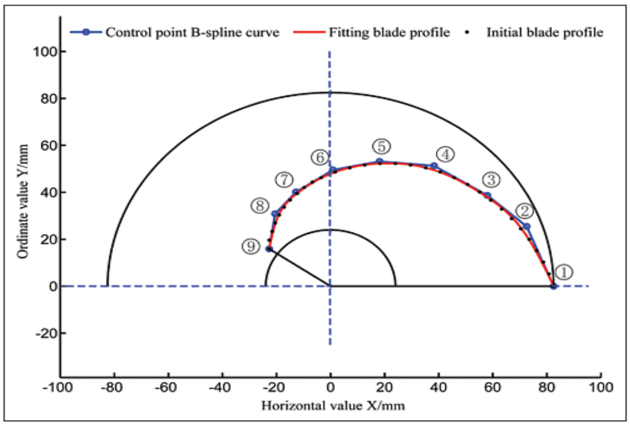

Fig .7 Blade parameterization (30).

The ANSYS-FLUENT software is used to calculate the selected model in the numerical method. N-S equations to describe the inner flow of the PAT, the standard k–e turbulence model is used. ICEM was used to generate the structured grid of computational domain. The final mesh number is 1,178,560.

The result shows that the efficiency of the optimized pump as turbine under the optimum operating condition increased by 2.91%. Through optimization of the blade, the hydraulic loss in the impeller decreased, the hydraulic loss in the volute and outlet pipe has a certain increase, whereas the total hydraulic loss decreased. Further investigation is required to control the hydraulic loss in the volute and outlet pipe.

One original impeller and three modified impellers of an industrial centrifugal pump with a specific speed of 23.5 m, m3 /s were tested numerically and experimentally. In this work, the shape of blades was redesigned to reach a higher efficiency in turbine mode using a gradient-based optimization algorithm coupled by a 3D Navier–stokes flow solver. Also, another modification technique was done by rounding the leading edges of blades and hub/shroud interface in turbine mode [40]. The result of the study shows modifications on the impeller lead to achieving maximum efficiency in reverse mode.

A comparison between centrifugal impeller pumps mode [41] and turbine mode [42], with and without splitter blades in terms of suction performance, is presented by experimental tests and numerical analyses. The efficiency of PAT is improved when splitter blades are added to impeller flow passage.

Multi-objective optimization to improve the hydrodynamic performance of a counter-rotating type pump-turbine operated in pump and turbine modes was illustrated [43]. The inlet and outlet blade angles of impellers/runners with four blades, which were extracted through a sensitivity test, were optimized using a hybrid multi-objective genetic algorithm with a surrogate model based on Latin hypercube sampling. Three-dimensional steady incompressible Reynolds-averaged Navier-stokes equations with the shear stress transport turbulence model were discretized via finite volume approximations and solved on a hexahedral grid to analyze the flow in the pump-turbine domain. For the major hydrodynamic performance parameters, the pump and turbine efficiencies were considered as the objective functions. The result shows that the arbitrarily selected optimal designs in the Pareto-optimal solutions were increased as compared with the reference value.

Many researchers have reported that the efficiency of pump in turbine mode can be improved by simple modification such as rounding impeller tips, installation of splitter blades, reducing impeller thickness, increasing number of blades, applying of guide vane, and optimizing blade profile . Table 2 summarizes the outcome of the modifications different researchers made.

Table 2 Summary of modification result proposed by different researchers.

Author	Shi et al. [22]	Yang et al. [25]	Feng Xia Shi et al. [29]	S. Miao et al. [30]	S. Derakhshan et al. [40]		P. Singh and F. Nestmann [44]
Method	Number of blades	Blade thickness	Guide vane number	Blade profile optimization	Rounding of inlet	Optimization	Impeller rounding
Rise in efficiency (η%)	≈1.75	≈1.6	≈2	≈2.91	≈5.5	≈2.75	≈2

Among the various techniques attempted by different investigators for performance improvement of pump as turbine, rounding of impeller inlet and blade profile optimization is found to be the most promising technique. Testing of more than one modification techniques per system is important to further improvement.

VI. RESEARCH GAPS IN RELATED LITERATURE

When the pump is operating in the turbine mode, the direction of flow is reversed; therefore, the pattern of loss distribution is not the same as in the pump mode. To improve performance, one of the important factors is to identify the causes of losses that may occur in turbine mode. Many researchers have studied performance improvement of PAT focusing on shock loss, while other losses such as diffusion and hydraulic losses should be considered when using pump as turbine.

Guide vane is an important part of a turbine, but centrifugal pumps have no guide vane. An extra row of fixed blades called inlet guide vane are required to direct the water at the correct angle onto the PAT blade; therefore, testing and using different guide vane angle are important to improve the efficiency of the system.

There is a need for studies that focus on the multi-stage multi-flow pumps to increase the power output from pump as turbine (PAT). Furthermore, a comparative study is needed to provide information on the various turbulence models then finding out the best one.

The velocity triangle is one of the fundamental tools to analyse turbomachinery problems. After each modification, the velocity triangle of the modified blade should be specified.

Cavitation is caused by local vaporization of the liquid. It usually occurs in hydraulic machines and it is a cause of different potential problems. Optimum design is required to avoid the effect of Cavitation.

VII. CONCLUSION

In various parts of this review paper, it has been recognized that extensive studies have been carried out on pump as turbine. From the entire study, it can be concluded that Commercial centrifugal pump, i.e. PAT, will provide an attractive alternative for power generation in off-grid areas. The limitations of PAT can be further reduced by selecting a proper pump for a specific site. The characteristics of the pump running as a turbine can be predicted by a theoretical, numerical, and experimental approaches. The efficiency can be increased by using various modification techniques. Among the various techniques attempted by different researchers is rounding of impeller inlet and blade profile optimization, which were found to be the most promising techniques.

For future research, in addition to introducing new modifications, a study on the importance of applying existing techniques should be carried out.

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