Energy Conversion and Management 89 (2015) 808–824

Contents lists available at ScienceDirect

Energy Conversion and Management

journal homepage: www.elsevier.com/locate/enconman

Effects of impeller diameter and rotational speed on performance of pump running in turbine mode

Sanjay V. Jain [a], Abhishek Swarnkar [b], Karan H. Motwani [c], Rajesh N. Patel [a,*]

[a] Mechanical Engineering Department, Institute of Technology, Nirma University, Ahmedabad 382481, Gujarat, India
[b] Mechanical Engineering Department, Gandhinagar Institute of Technology, Gandhinagar 382721, Gujarat, India
[c] Mechanical Engineering Department, Faculty of Engineering, Marwadi Education Foundation, Rajkot 360003, Gujarat, India

ARTICLE INFO

Article history:
Received 30 May 2014
Accepted 15 October 2014
Available online 12 November 2014

Keywords:
Impeller diameter
Part load
Pump running in turbine mode
Rotational speed

ABSTRACT

The major limitations of mini/micro hydropower schemes is the higher cost of small capacity hydro turbines. Also, it is very cumbersome, time consuming and expensive to develop the site specific turbines corresponding to local site conditions in mini/micro hydro range. In such plants, small centrifugal pumps can be used in turbine mode by running in the reverse direction. The efficiency of pump as turbines (PATs) is usually lower than the conventional hydro turbines; however, there may be substantial decrease in the capital cost of the plant.

Hydropower plants usually runs at part load for several months in a year due to insufficient water availability for the power generation. The application range of PAT can be widened if its part load and/or maximum efficiency can be improved. In the present study, experimental investigations are carried out on centrifugal pump running in turbine mode to optimize its geometric and operational parameters e.g. impeller diameter and rotational speed. The experiments were performed in the wide range of rotational speeds varying from 900 to 1500 rpm with original (⌀ 250 mm), 10% trimmed (⌀ 225 mm) and 20% trimmed (⌀ 200 mm) impellers. Impeller trimming led to improvement in efficiency at part load operating conditions. The performance of PAT was found better at the lower speeds than that at the rated speed. The effects of blade rounding were studied in all the cases and it led to 3–4% rise in efficiency at rated speed with the original impeller. The empirical correlation is also developed for prediction of efficiency in terms of impeller diameter and rotational speed in non-dimensional form.

1. Introduction

Hydropower is probably the oldest and most reliable source of energy on the earth. Large hydropower plants require construction of large scale dams, long-term investments, extensive and time consuming construction processes. The reservoir behind the dam can flood enormous valuable agricultural lands and agricultural activities [1]. These factors make large hydropower plants unfavorable in many cases. Nevertheless, small hydropower plants (SHP) are free from such issues and provide option of decentralized power generation particularly in remote, rural and hilly areas.

In India, the plants having overall capacity up to 25 MW are called small hydropower plants. The SHP is further categorized as pico (below 5 kW), micro (5.1–100 kW) and mini (101 kW to 2 MW) hydropower [2]. Application of micro hydropower plants has gained worldwide attention during the last decades of the 20th century [3]. Although Micro hydropower plants are by no means comparable to large hydropower projects as far as their power generation capacity is concerned; however, their simple design and relatively simple manufacturing processes, low price per kilo watts, easy installation with simple construction, cheap and easy maintenance and their insignificant or rather non-riverine impacts have made them attractive particularly to the countries with abundant micro hydro potential [1].

The major hindrance in the development of mini/micro hydropower schemes is the higher initial cost of the conventional turbines in low capacity range. The cost of electro-mechanical components in typical large hydro schemes is around 20% but in micro-hydro it is relatively higher and varies from 35% to 70% of the total project cost depending on size, rating and local ecology [4,5]. One of the main objectives of the small hydro researchers worldwide is to decrease the cost of electro-mechanical equipments and its standardization in low capacity range. Many investigators have explored the possibility of using different turbines in

* Corresponding author. Tel.: +91 79 30642102; fax: +91 2717 241917.
E-mail address: rajeshnpatel1969@yahoo.com (R.N. Patel).

http://dx.doi.org/10.1016/j.enconman.2014.10.036
0196-8904/

Nomenclature

Amp	ampere
BEP	best efficiency point
BR	blade rounded
D	impeller diameter (m)
DC	direct current
g	acceleration due to gravity (m/s^2)
H	head (m)
I	current (A)
IS	Indian standard
kW	Kilowatt
m	meter
mm	milli meter
MNRE	ministry of new renewable energy
MW	megawatt
n	rotational speed (rps)
P	Power (W)
p	static pressure (N/m^2)
PAT	pump as turbine
psi	pound per square inch
Q	discharge (m^3/s)
R	radius of blade rounding (m)
rpm	revolutions per minute
rps	revolutions per second
SHP	small hydropower plants
t	blade thickness (m)
TW h	terawatt-hour
UR	unrounded
V	fluid velocity (m/s)
v	voltage (V)
z	datum head (m)

Greek symbols

Δ	change in parameter
η	efficiency
π	power number
ρ	density of water (kg/m^3)
ϕ	discharge number
$\varnothing$	impeller diameter
ψ	head number

Subscripts

1	at inlet of PAT
2	at outlet of PAT
g	generator
i	input
o	overall, output
r	rated value
s	shaft
t	turbine

mini/micro hydro range e.g. cross flow turbine, propeller turbine, Pelton turbine, small Francis turbine etc. [6–8]. However, these options are subjected to complicated arrangements like large size, necessity of belt drive, multi-jet arrangement and/or higher cost [9].

One of the cost-effective and attractive alternatives for mini/micro hydropower plants is to utilize the pump as turbine. Use of centrifugal pump in turbine mode facilitate various advantages associated with the pump e.g. mass production, low maintenance and installation costs, less complicated to operate than turbines, available for a wide range of heads and flows, short delivery time, available in a large number of standard sizes, easy installations, ease of availability of spare parts, etc. [10,11]. The performance of pump running as turbine is usually inferior to conventional hydro turbines. However, use of PATs can significantly reduce the initial cost of the hydropower plant [12] as well as of water distribution network [13]. For low capacity power plants (up to 100 kW) there is substantial reduction in the capital investment of the plant, typically of the order of 10–1 or even more, depending on the capacity of the plant [14]. Hence, the concept of utilizing PAT is favorable due to lower cost of pumps used in turbine mode; particularly for rural, remote and hilly areas.

Virtually any type of pump e.g. axial flow, mixed flow, radial flow, double suction as well as multistage pumps can be used in turbine mode for power generation. However, from techno-economic considerations use of single stage end suction centrifugal pump, working in the range of low to medium head, is recommended by many of the researchers in low capacity range [14–18]. The performance of pump in turbine mode very closely depends on its performance in pump mode.

Many investigators have suggested the relations for prediction of PAT efficiency either based on efficiency in pump mode [19–26] or based on specific speed in pump mode [27–31]. Few researchers have also derived the relations based on experimental approach [15,17,32] to predict the PAT behavior. However, the deviations between performance predicted by these methods and experimental results were found to be around ±20% or even more [9,33]. They concluded that the efficiency in turbine mode may vary from +2% to −8.5% compared to efficiency in pump mode [34]. In the present study, experimental investigations are carried out to improve the PAT performance by optimizing its geometric and operational parameters e.g. impeller diameter and rotational speed. An empirical correlation (in non-dimensional form) for prediction of PAT efficiency, based on impeller diameter and rotational speed, was developed by regressing the experimental data.

2. Experimental setup

For the present study, centrifugal pump with rated head, discharge and speed of 20 m, 0.0292 m^3/s and 1400 rpm respectively was selected for the experimental setup. At rated parameters, the maximum efficiency of the pump in pump mode was obtained as 77.5% [35]. Six numbers of backward curved vanes having inlet and outlet blade angles as 24° and 19° were provided in the pump impeller. The experimental test rig was developed at Institute of Technology, Nirma University, Ahmedabad as shown in Fig. 1. The experimental setup consisted of service pump, flow regulating valve, bypass line arrangement, electromagnetic flow meter, PAT with draft tube, pressure transmitters, DC shunt generator with load bank, tailrace channel, underground reservoir and piping system.

In the present study, draft tube was designed and fabricated as per IS 5496:1993 [36]. The service pump was taking water from the underground reservoir and supplying to the PAT which was recirculated through tailrace channel. To measure the electric power generated, digital voltmeter and ammeter were connected with resistive load in parallel and series respectively. The major specifications of different components of the experimental setup are given in Table 1.

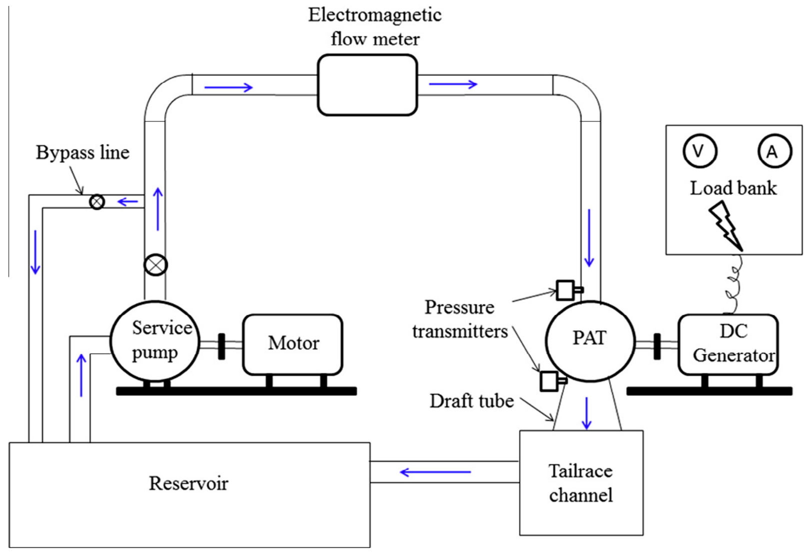

Fig. 1. Schematic diagram of experimental test rig.

Table 1
Specifications of various components and instruments.

Name of component	Major specifications	
Pump used as turbine	Type: End suction centrifugal Head: 20 m Discharge: 0.0292 m^3/s	
Service pump	Type: End suction centrifugal Head: 25 m Discharge: 0.05 m^3/s	
Draft tube	Inlet cone angle: 6° Height: 1 m Design: as per IS 5496:1993 [36]	
Electromagnetic flow meter	Discharge range: 0–0.05 m^3/s Sensitivity: ±0.0005 m^3/s	
Pressure transmitters	At PAT inlet Range: 0–40 psi Sensitivity: ±0.1 psi	At PAT outlet Range: −1–4 psi Sensitivity: ±0.1 psi
Digital speed sensor	Range: 0–3000 rpm Sensitivity: ±1 rpm	
DC shunt generator with load bank	Power: 4 kW Speed: 1450 rpm Resistive load range: 0–20 Amp Performance: as per IS:4722 [37] and IS:9320 [38]	
Voltmeter	Range: 0–1000 V (DC) Sensitivity: ±1 V	
Ammeter	Range: 0–20 Amp (DC) Sensitivity: ±0.01 Amp	

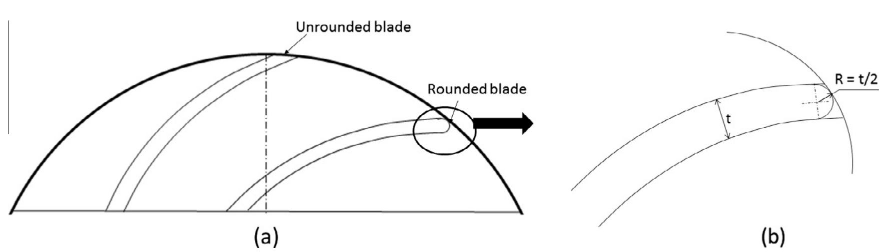

Fig. 2. (a) Unrounded and rounded blades (b) details of blade rounding. (t = blade thickness, R = radius of blade rounding).

All the measuring instruments were calibrated before put in use. The uncertainty in different performance parameters was estimated in accordance with the procedure given by Kline and McClintock [39]. The relative uncertainty in head, discharge, power output and overall PAT efficiency were found as 0.59%, 1%, 0.72% and 2.22% respectively.

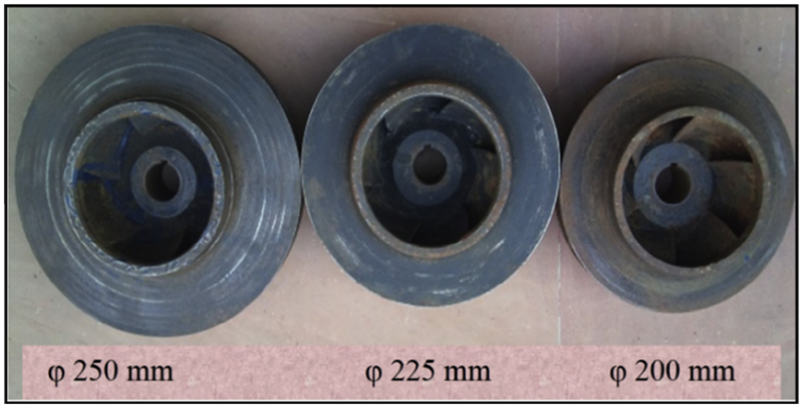

Fig. 3. Three impellers used for analysis.

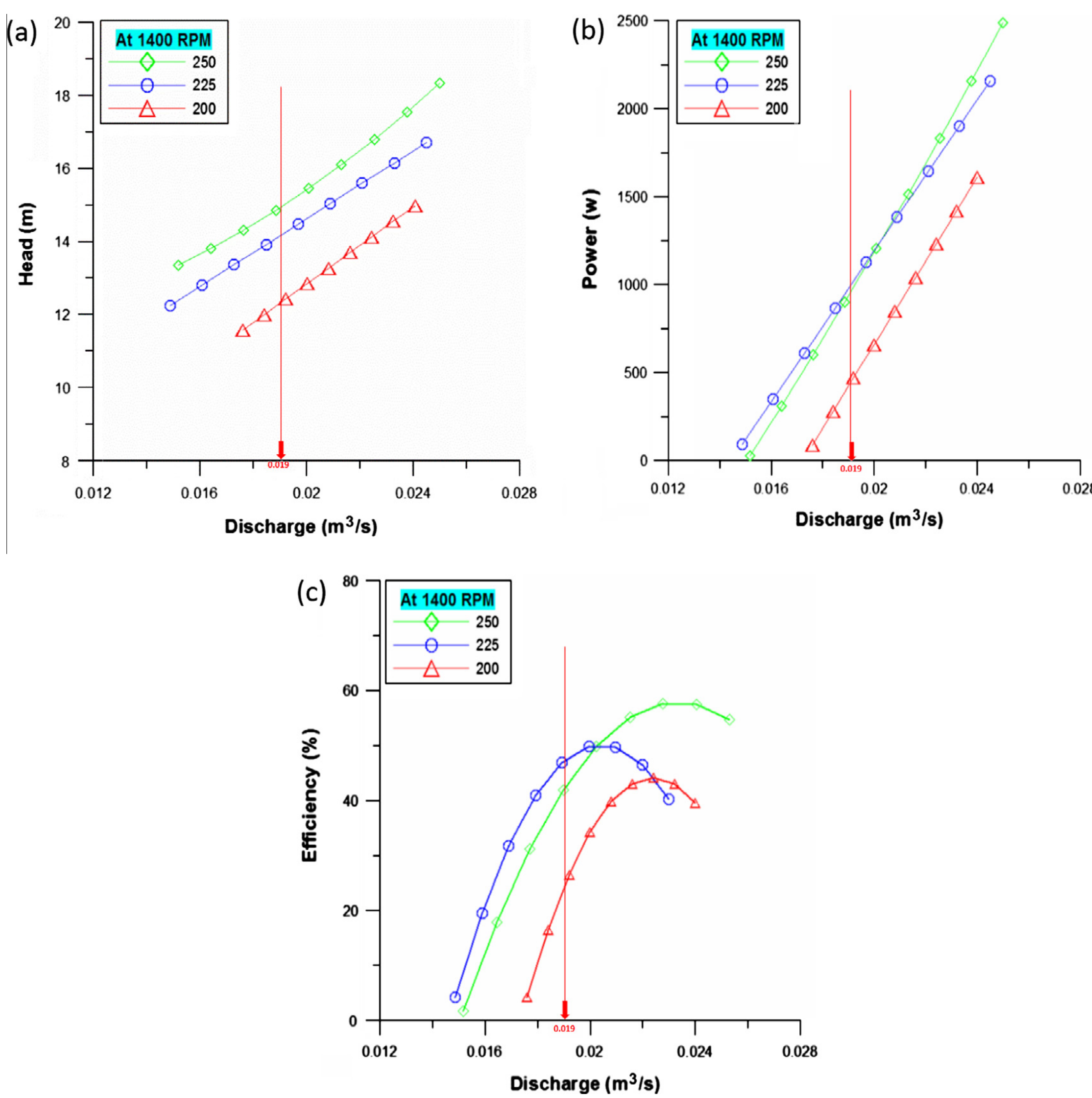

Fig. 4. Performance curves of PAT with different impellers (a) head versus discharge (b) power versus discharge (c) efficiency versus discharge.

3. Parametric studies on PAT

In the present study, effects of following parameters were studied on the performance of PAT:

- Effects of impeller trimming.
- Effects of rotational speed.
- Effects of impeller blade rounding.

Table 2
Parameters at part load ($Q = 0.019\ m^3/s$) for different impellers.

Impeller	H (m)	P (W)	η (%)
∅ 250 mm	14.91	932.29	42.14
∅ 225 mm	14.20	992.87	46.37
∅ 200 mm	12.29	422.44	24.19

The justifications for carrying out these investigations are discussed below.

3.1. Effects of impeller trimming

The performance of pump in turbine mode resembles the performance in pump mode i.e. high efficiency pump works better in turbine mode also. In centrifugal pumps, mainly three types of casings are used e.g. volute casing, vortex casing and diffuser casing with guide vanes. The fluid coming out from the impeller and entering the casing is highly turbulent and it suffers from higher amount of hydraulic losses while passing through the rapidly changing curvature of flow passage in the casing.

In vortex casing, annular space provided between the impeller and casing called 'vortex chamber' facilitate an additional space to the fluid for stabilization of the flow. This led to reduction in hydraulic losses to a considerable extent due to decrease in eddy formation in the casing. Hence, the efficiency of centrifugal pump with vortex casing is found to be better than that with volute casing [40–42]. In view of this, in the present study the performance of PAT was studied with original impeller (∅ 250 mm) and after 10% and 20% trimmed (∅ 225 and ∅ 200 mm) impellers respectively. The trimmed impeller in the existing volute casing facilitates an additional space in the casing, similar to vortex chamber, which may lead to stabilization of the turbulent flow and hence reduction in hydraulic losses due to eddy formation. Also, when head or discharge decreases, impeller trimming or replacing the existing impeller with smaller impeller could be the cost-effective solution [43].

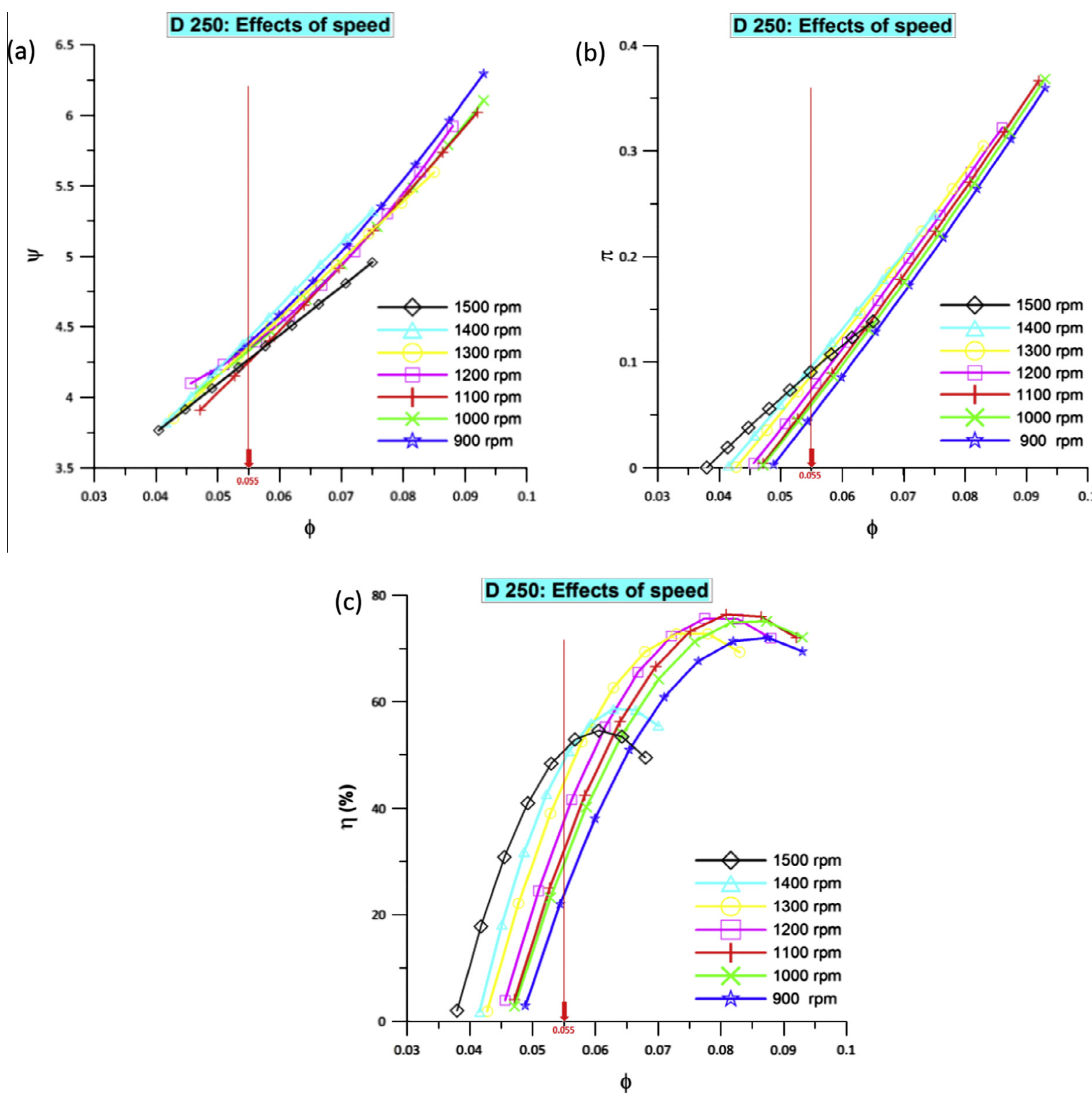

Fig. 5. Performance curves with original (∅ 250 mm) impeller at different speeds (a) ψ versus ϕ (b) π versus ϕ (c) η versus ϕ.

3.2. Effects of rotational speed

As mentioned earlier, the performance of pump in turbine mode directly depends on the performance of pump in pump mode. Most of the researchers have studied the performance in pump and turbine modes, either experimentally [15,32] or numerically [18,44], at same rotational speed. Usually, best efficiency point (BEP) of pump running in turbine mode is obtained at higher head and discharge compared to that in pump mode [4,14,18,45,46]. Consequently, the pattern of hydraulic loss distribution, flow instabilities and the radial unbalance generated in turbine mode are expected to be different compared to that in pump mode. Also, the studies on force analysis revealed that, there exists higher axial thrust in turbine mode compared to pump mode [47]. These parameters may affect the energy transformation process in turbine mode compared to that in pump mode at different rotational speeds. Hence, it is required to study the performance of PAT at different rotational speeds.

Very few researchers have studied the performance of PAT at different speeds e.g. Fernandez et al. [14], Rawal and Kshirsagar [46], Carravetta et al. [48]. In the present study, the performance of PAT was studied at 7 different speeds in the range of 900–1500 rpm with original, 10% trimmed and 20% trimmed impellers.

3.3. Effects of impeller blade rounding

The edges of the impeller blades at outer periphery are sharp. It is beneficial when pump runs in pump mode because fluid moves radially outward due to centrifugal force in diverging area. However, in turbine mode the sharp edges may lead to flow separation and hence wake formation as fluid moves radially inward in converging passage and may result in decrease in efficiency. By rounding the edges of impeller blades and making it bullet shaped as shown in Fig. 2, the losses due to flow separation and wake formation can be reduced which may result in improved performance of the PAT. Most of the researchers e.g. Singh [49], Derakhshan et al. [50], Suarda et al. [51] have applied this technique for improving the performance of PAT usually at rated speed. Singh and Nestmann [52] recommended to standardize the impeller blade rounding effects over the wide range of PATs. In the present study, blade rounding technique was applied on 3 different impellers and its effects were studied at 7 different rotational speeds.

4. Results and discussion

4.1. Performance parameters

In the present study, each experiment was repeated three times and results are plotted based on average of three readings.

The head is considered as the sum of pressure, kinetic and potential heads as given below:

$$H = \frac{p_1 - p_2}{\rho g} + \frac{V_2^2 - V_1^2}{2g} + \Delta z \tag{1}$$

where p_1 and p_2 are static pressure (N/m^2) at inlet and outlet, ρ is density of water (kg/m^3), g is acceleration due to gravity (m/s^2), V_1 and V_2 are fluid velocities (m/s) at inlet and outlet, Δz is the datum difference between two pressure transmitters (m).

Power input to PAT (P_i) is given by:

$$P_i = \rho g Q H \tag{2}$$

where Q = discharge passing through PAT (m^3/s), H = head acting on the PAT (m).

Power output from generator (P_o) is given by:

$$P_o = v \times I \tag{3}$$

where v = voltage (V), I = current (A).

The overall system efficiency (PAT-Generator combination) is calculated using following equation:

$$\eta_o = \frac{P_o}{P_i} \tag{4}$$

The efficiency of PAT is calculated as the ratio of the overall efficiency (η_o) and the generator efficiency (η_g) as mentioned below:

$$\eta = \frac{\eta_o}{\eta_g} \tag{5}$$

The experimental results are presented in the form of non-dimensional parameters e.g. head number (ψ), discharge number (ϕ), power number (π) which are defined below:

$$\psi = \frac{g \cdot H}{n^2 \cdot D^2} \tag{6}$$

$$\phi = \frac{Q}{n \cdot D^3} \tag{7}$$

$$\pi = \frac{P_s}{\rho \cdot n^3 \cdot D^5} \tag{8}$$

where n = rotational speed (rps), D = impeller diameter (m), P_s = shaft power (W).

Table 3
Parameters at ϕ = 0.055 at different speeds for original impeller.

Speed N (rpm)	Head number ψ	Power number π	Efficiency η (%)
900	4.39	0.049	24.02
1000	4.33	0.060	29.98
1100	4.25	0.064	32.46
1200	4.35	0.074	37.97
1300	4.36	0.088	45.24
1400	4.42	0.096	49.18
1500	4.27	0.091	51.10

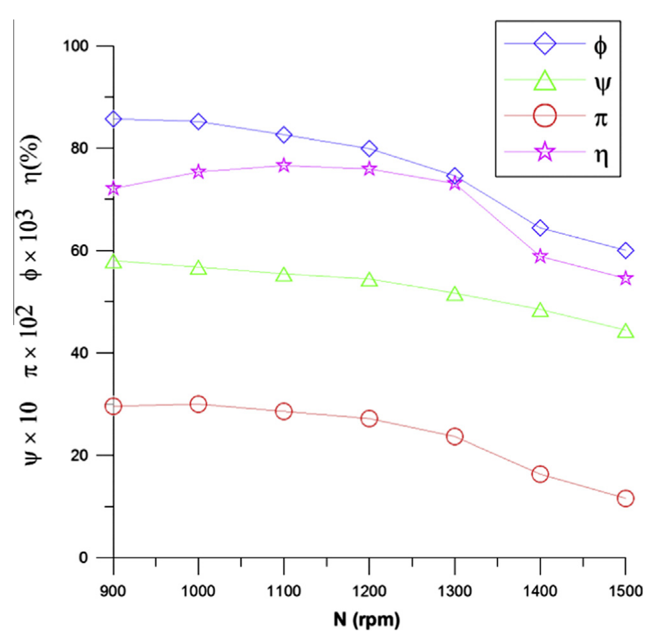

Fig. 6. BEP parameters at different speeds for original impeller.

4.2. Performance of pump in turbine mode

4.2.1. Effects of impeller trimming

To analyze the effect of impeller diameter on PAT performance, the experiments were performed with following three impellers as shown in Fig. 3.

(1) Original impeller (∅ 250 mm).
(2) 10% trimmed impeller (∅ 225 mm).
(3) 20% trimmed impeller (∅ 200 mm).

The experiments were performed on PAT with these three impellers at rated speed of 1400 rpm. Due to impeller trimming, diameter decreased which led to increase in value of discharge number (∅) as it is inversely proportional to cubic power of diameter ($\varnothing \propto 1/D^3$). It resulted in shift in the range of ∅ for trimmed impellers. Hence, to study the effects of impeller trimming on head, power and efficiency at different discharge the performance curves are plotted in terms of dimensional parameters as shown in Figs. 4(a–c). As discharge increases, head and power output continuously increases; whereas, efficiency increases, reaches maximum and then decreases.

It can be seen that, impeller trimming resulted in decrease in head. The rate of decrease in head increased with increase in impeller trimming. BEP discharge shifted towards lower value and further trimming lead to backshift towards higher discharge. Similar variations in various parameters were also observed by Sheng et al. [53] and Sedlar et al. [54]. The various geometric

Table 4
Parameters at $\phi = 0.07$ at different speeds for 10% trimmed impeller.

Speed N (rpm)	Head number ψ	Power number π	Efficiency η (%)
900	4.83	0.059	20.39
1000	4.70	0.070	24.46
1100	4.70	0.080	28.62
1200	4.69	0.104	37.72
1300	4.95	0.118	41.76
1400	4.96	0.121	45.39
1500	5.01	0.125	44.67

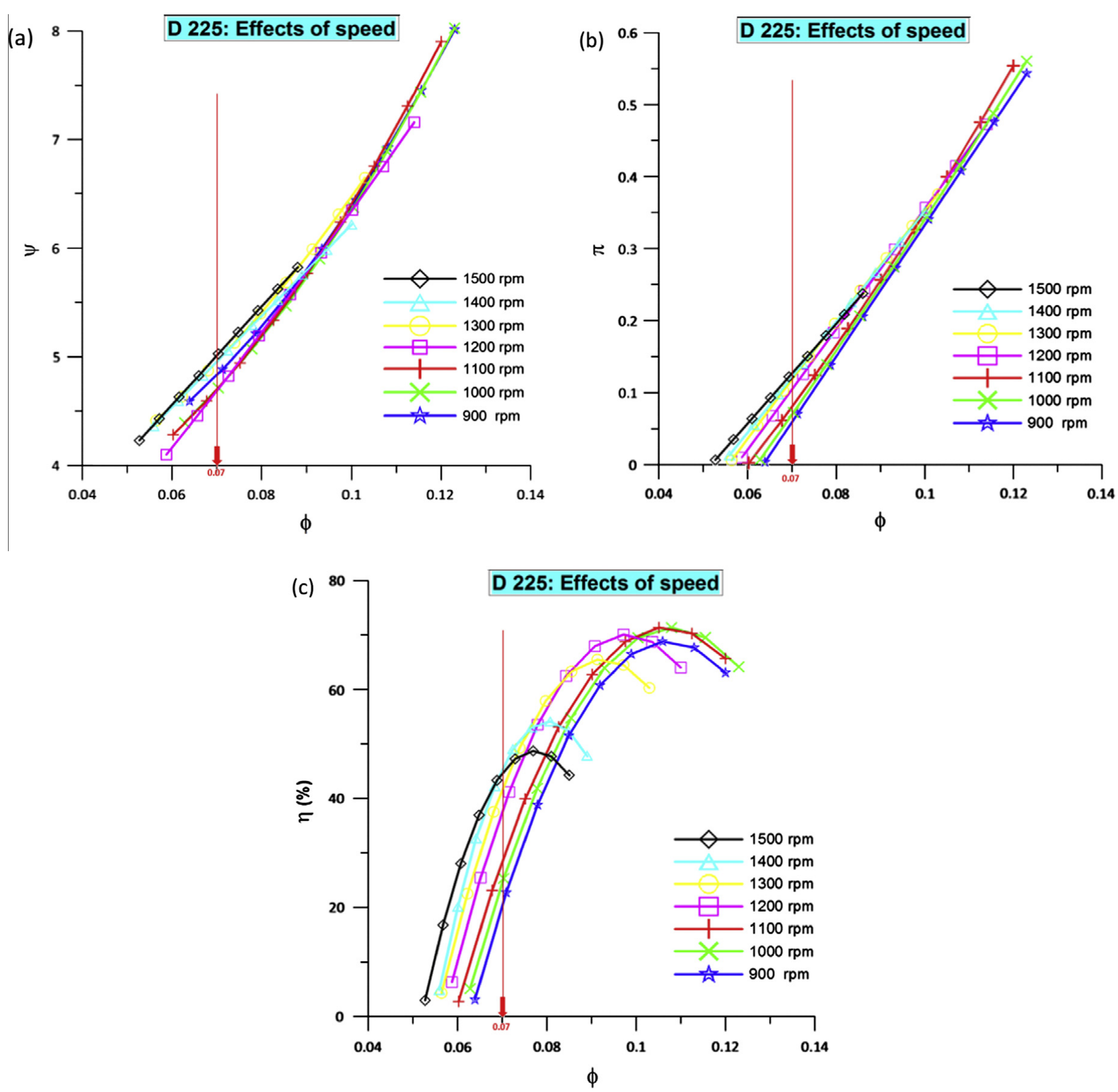

Fig. 7. Performance curves with 10% trimmed (∅ 225 mm) impeller at different speeds (a) ψ versus ϕ (b) π versus ϕ (c) η versus ϕ.

parameters such as inlet width, blade inlet angle and blade wrap angle changes due to change in impeller diameter. The performance of PAT with trimmed impeller depends on the combined effects of these geometric parameters.

Various parameters at part load operating conditions (e.g. $Q = 0.019\ m^3/s$) for different impellers are summarized in Table 2. It may be noted that, the power output and efficiency increased by 6.5% and 10% respectively with 225 mm diameter impeller, particularly at part load. This may be due to stabilization of the flow and suppression of eddies in the annular space created due to impeller trimming. Further trimming resulted in sudden drop in power output and efficiency over the entire range of discharge. The decrease in efficiency is attributed to the large global separations inside the impeller passages with trimmed impellers.

4.2.2. Effects of speed on different impellers

The rotational speed of PAT was varied by changing the discharge using the gate valve provided at the exit of the service pump. To study the effects of rotational speed on performance, the PAT was rotated at different speeds ranging from 900 to 1500 rpm in the step of 100 rpm i.e. 900, 1000, 1100, 1200, 1300, 1400 and 1500 rpm. The effects of speed were studied on original as well as on 10% and 20% trimmed impellers. The generated energy at different speeds was used in the resistive heaters provided in the resistive load bank.

The performance curves of PAT with original impeller (∅ 250 mm) at different speeds are shown in Fig. 5(a–c). It can be seen that, as ϕ increases ψ and π increases. The efficiency increases with ϕ, reaches maximum then decreases. As per the affinity laws, the non-dimensional parameters obtained at different speeds shall collapse [55]. It can be seen that, ψ and π curves collapsed to a larger extent; however, some deviation was observed in the η curve. This may be due to the fact that the affinity laws are basically an approximation which are derived based on the assumption of geometric similarity between the pump working under different conditions [56]. The deviations may also be due to different losses and uncertainty in the measurements. Day et al. [55] also reported some deviations in the performance curves at different speeds.

The non-dimensional parameters at different speeds are analyzed at part load conditions (e.g. at $\phi = 0.055$) as shown in Table 3. The study revealed that, as speed increased π and η also increased. However, the variation in ψ was negligible except at higher speeds. The BEP parameters at different speeds are plotted in Fig. 6. The power output and efficiency were found to be better in the speed range of 1000–1200 rpm. The rated speed for pump in pumping mode is 1400 rpm. However in turbine mode, while rotating in reverse direction, the conversion of hydraulic to mechanical energy may not be efficient at higher speeds. This could be the reason for better performance at lower speeds. It was also observed that the efficiency of PAT decreased at the speeds above rated speed; hence, experiments were not performed above 1500 rpm. The maximum efficiency was found to be 76.65% at 1100 rpm.

The performance curves for 10% trimmed (∅ 225 mm) impeller at different rotational speeds are shown in Fig. 7(a–c). The variations in ψ, π and η with ϕ are similar to original impeller. The efficiency range became wider at lower speeds. The η_{BEP} was found higher for all the speeds below the rated speed (below 1400 rpm) and it was obtained at higher value of ϕ. This may be due to cumulative effects of flow stabilization in the annular space created due to trimming and better energy conversion at lower speeds.

The non-dimensional parameters at different speeds are analyzed at part load operating conditions (e.g. at $\phi = 0.07$) as shown in Table 4. It can be seen that, as speed increases, π and η increases. The BEP parameters at different speeds are plotted in Fig. 8. The performance of PAT was observed better in the speed range of 1000–1200 rpm. The maximum efficiency was reported as 71.40% at 1100 rpm. The trend in results obtained with 10% trimmed impeller was almost similar to that obtained with original impeller.

The performance curves for 20% trimmed (∅ 200 mm) impeller at different rotational speeds are shown in Fig. 9(a–c). The non-dimensional parameters at different speeds are analyzed at partial discharge (e.g. at $\phi = 0.11$) as shown in Table 5. It can be seen that, as speed increases π and η increases. However, the variation in ψ is less except at higher speed. The BEP parameters at different speeds are plotted in Fig. 10. The performance of PAT was observed better in the speed range of 1100–1300 rpm. The possible reasons for the same are already reported in the earlier section. The maximum efficiency of the order of 59.84% was achieved at 1200 rpm.

4.2.3. Effects of impeller blade rounding

In the present study, the effects of blade rounding were studied on all the three impellers i.e. original impeller, 10% and 20% trimmed impellers at seven different speeds e.g. 900, 1000, 1100, 1200, 1300, 1400 and 1500 rpm. The leading edges of blades were given the bullet shape with rounding radius as 50% of the blade thickness (Fig. 2). The original impeller before and after blade rounding is shown in Fig. 11.

The comparison of results obtained with original (∅ 250 mm) impeller before and after blade rounding at different rotational speeds are shown in Fig. 12(a–c). It can be seen that, blade rounding led to decrease in head at all the speeds. At lower than rated speeds, the blade rounding found to be advantageous in the entire part load range though the maximum efficiency remained nearly same. Also, there is a significant increase of performance at all the speeds except for higher values of discharge.

The effects of blade rounding at different speeds are analyzed at part load condition (e.g. at $\phi = 0.055$). The percentage variations in ψ, π and η due to blade rounding in comparison with unrounded impeller at different speeds are tabulated in Table 6. It can be seen that, blade rounding led to decrease in ψ at all the speeds and the decreament was found to be higher at higher speeds. However, it resulted in increase in π and η, in the range of 3–25% and 6–31% respectively at different speeds. It was also observed that, blade

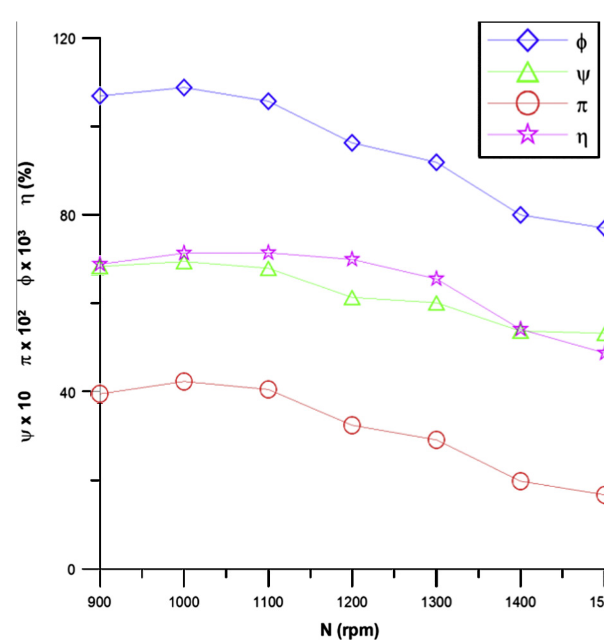

Fig. 8. BEP parameters at different speeds for 10% trimmed impeller.

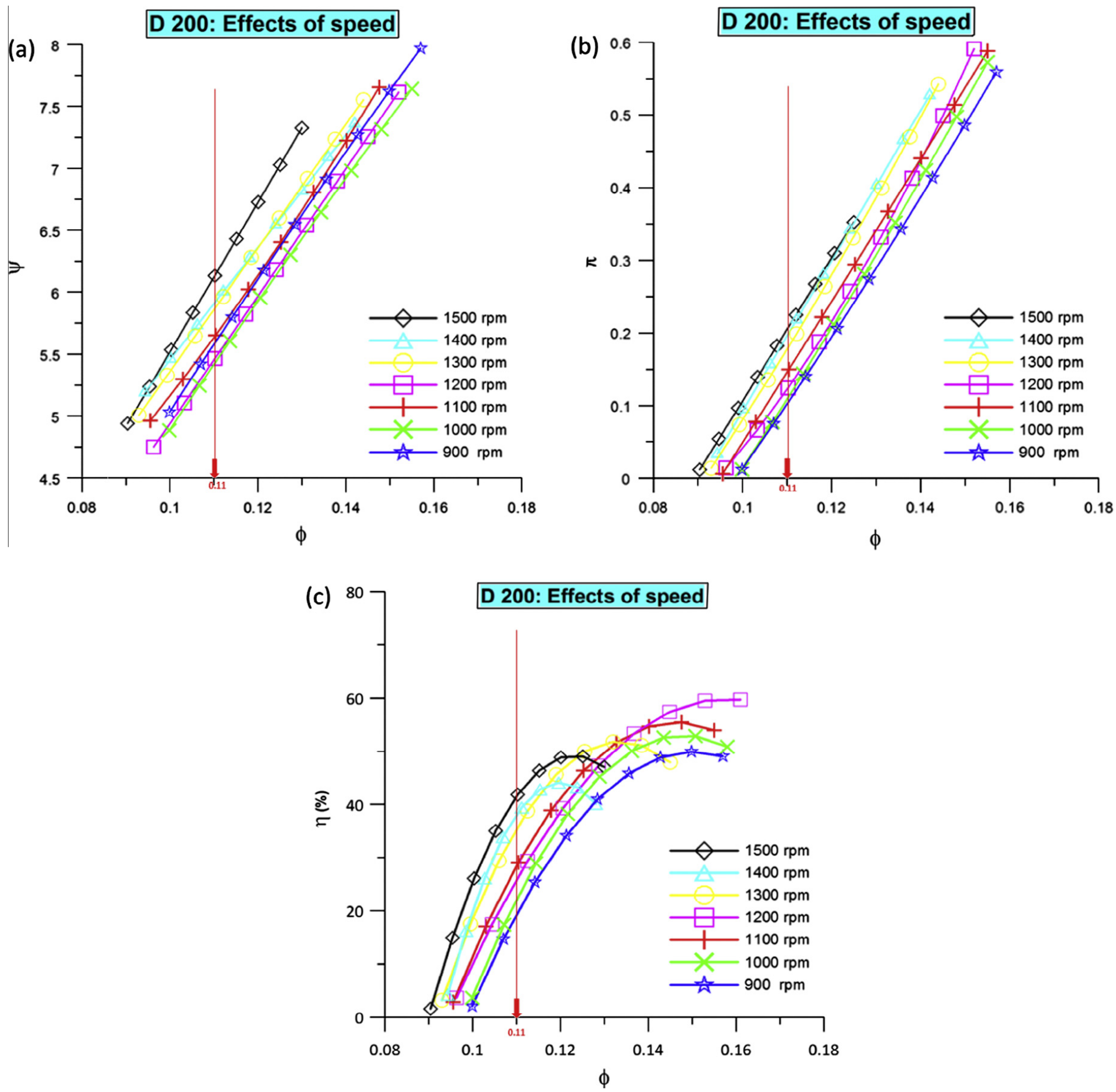

Fig. 9. Performance curves with 20% trimmed (∅ 200 mm) impeller at different speeds (a) ψ versus ϕ (b) π versus ϕ (c) η versus ϕ.

Table 5
Parameters at $\phi = 0.11$ at different speeds for 20% trimmed impeller.

Speed N (rpm)	Head number ψ	Power number π	Efficiency η (%)
900	5.578	0.102	19.39
1000	5.415	0.109	22.29
1100	5.633	0.146	28.53
1200	5.363	0.122	25.93
1300	5.741	0.177	35.49
1400	5.881	0.196	38.14
1500	6.120	0.205	41.57

rounding is advantageous usually at part load i.e. below BEP discharge. This may be due to decrease in losses due to wake formation and flow separation because of blade rounding. The blade rounding was found to be more advantageous in low speed range which may be attributed to the fact that in low speed range the flow separation gets suppressed.

Fig. 13 shows the effects of blade rounding on BEP parameters for original impeller at different speeds. The η_{BEP} remained nearly same below the rated speeds but slightly improved at rated and above rated speeds. Also, at rated speed η_{BEP} was obtained at lower discharge after blade rounding. Similar trend in efficiency was reported by Suarda et al. [51].

The comparison of results obtained with 10% trimmed (∅ 225 mm) impeller before and after blade rounding at different rotational speeds are shown in Fig. 14(a–c). It can be seen that, blade rounding caused decreases in ψ at all the speeds. Power and efficiency were improved over the entire discharge range above the speeds of 1300 rpm and the improvement was found to be higher at higher discharge. But, in low speed range blade rounding found advantageous only in high discharge range. Also, at rated speed the high efficiency range became wider.

The percentage variations in ψ, π and η due to blade rounding in comparison with unrounded impeller, at part load conditions (e.g. at $\phi = 0.07$) at different speeds, are summarized in Table 7. It can be seen that, blade rounding led to decrease in ψ at all

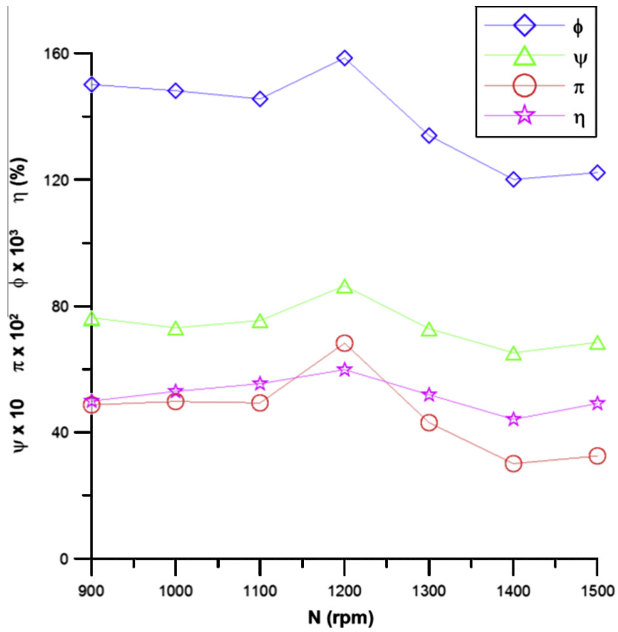

Fig. 10. BEP parameters at different speeds for 20% trimmed impeller.

the speeds and the decreament was found to be higher at higher speeds, similar to original impeller. It resulted in increase π and η in high speed range, above 1200 rpm, but then decreased below that speed.

Fig. 15 shows the effects of blade rounding on BEP parameters for 10% trimmed impeller at different speeds. It can be seen that, BEP was obtained at higher values of ϕ after blade rounding at all the speeds. At rated speed, the efficiency with 10% trimmed-blade rounded impeller was found to be better than the original impeller. Blade rounding resulted in increase in π and η at all the speeds and found to be more advantageous in high speed range, above 1200 rpm.

The comparison of results obtained with 20% trimmed (∅ 200 mm) impeller before and after blade rounding at different rotational speeds are shown in Fig. 16(a–c). The study of these curves revealed that, the head curve moved parallel down due to blade rounding at all the speeds and the drop was found to be higher than that with original and 10% trimmed impellers. Power was increased only at partial discharge in low speed range but decreased at higher speeds. Efficiency was increased at all the speeds and the increment was observed to be higher after BEP at most of the speeds except at 1500 rpm.

The percentage variations in ψ, π and η due to blade rounding in comparison with unrounded impeller, at part load conditions (e.g. at $\phi = 0.11$) at different speeds, are summarized in Table 8. It can be seen that, ψ is decreased at all the speeds and the decreament was found to be higher at higher speeds. But, η increased in most of the cases and the maximum improvement was obtained at 1200 rpm.

Fig. 17 shows the effects of blade rounding on BEP parameters for 20% trimmed impeller at different speeds. It can be seen that, BEP was obtained at higher values of ϕ after blade rounding in most of the cases, similar to 10% trimmed impeller. It resulted in improvement in π and η at most of the speeds. The maximum rise was obtained as 57% and 33% in power and efficiency respectively at rated speed. Similar results were obtained with 10% trimmed impeller.

4.2.4. Recommendations

After the detailed experimental investigations, three options are recommended for the performance enhancement of PAT based on geometric and operating parameters. The comparison of these modifications with original impeller (∅ 250 mm) at rated speed (1400 rpm) is shown in Fig. 18(a–c). As a first option, the blades of original impeller can be rounded from tip. That will result in improvement in power output and efficiency almost over the entire range of discharge. As a second option, the original impeller can be run at the speed of 1100 rpm after blade rounding. It may lead to decrease in required head and increase in power at part load operating conditions. That may result in improvement in efficiency over the entire range of discharge. The third option is to use 10% trimmed impeller (∅ 225 mm) at 1100 rpm after blade rounding. It gives the highest efficiency compared to other options in the whole operating range of PAT along with improvement in power at part load.

The advantage of first and second modifications would be the use of original impeller after blade rounding. But, the third option necessitates trimming of original impeller or replacement of original impeller by smaller impeller.

5. Development of correlation

It was observed that BEP efficiency (η_{BEP}) is a strong function of impeller diameter D and rotational speed N. The functional relationship between η_{BEP} and diameter ratio (D/D_r) and rotational speed ratio (N/N_r) can be expressed as:

$$\eta_{BEP} = fn(D/D_r, N/N_r)$$

An empirical correlation was developed for η_{BEP} by regression analysis of the experimental data based on blade rounded impeller. The monotonic variation in η_{BEP} with increasing D/D_r at different values of N/N_r is shown in Fig. 19(a). It was observed that, a second order polynomial equation best described the relationship between η_{BEP} and D/D_r. The values of Ln(η_{BEP}) versus Ln(D/D_r) were plotted as shown in Fig. 20(a) and a regression analysis to fit a second order polynomial equation yielded:

$$\mathrm{Ln}(\eta_{BEP}) = A_1 + n_1 \mathrm{Ln}(D/D_r) + n_2[\mathrm{Ln}(D/D_r)]^2 \quad (9)$$

Fig. 11. Impeller (a) before and (b) after blade rounding.

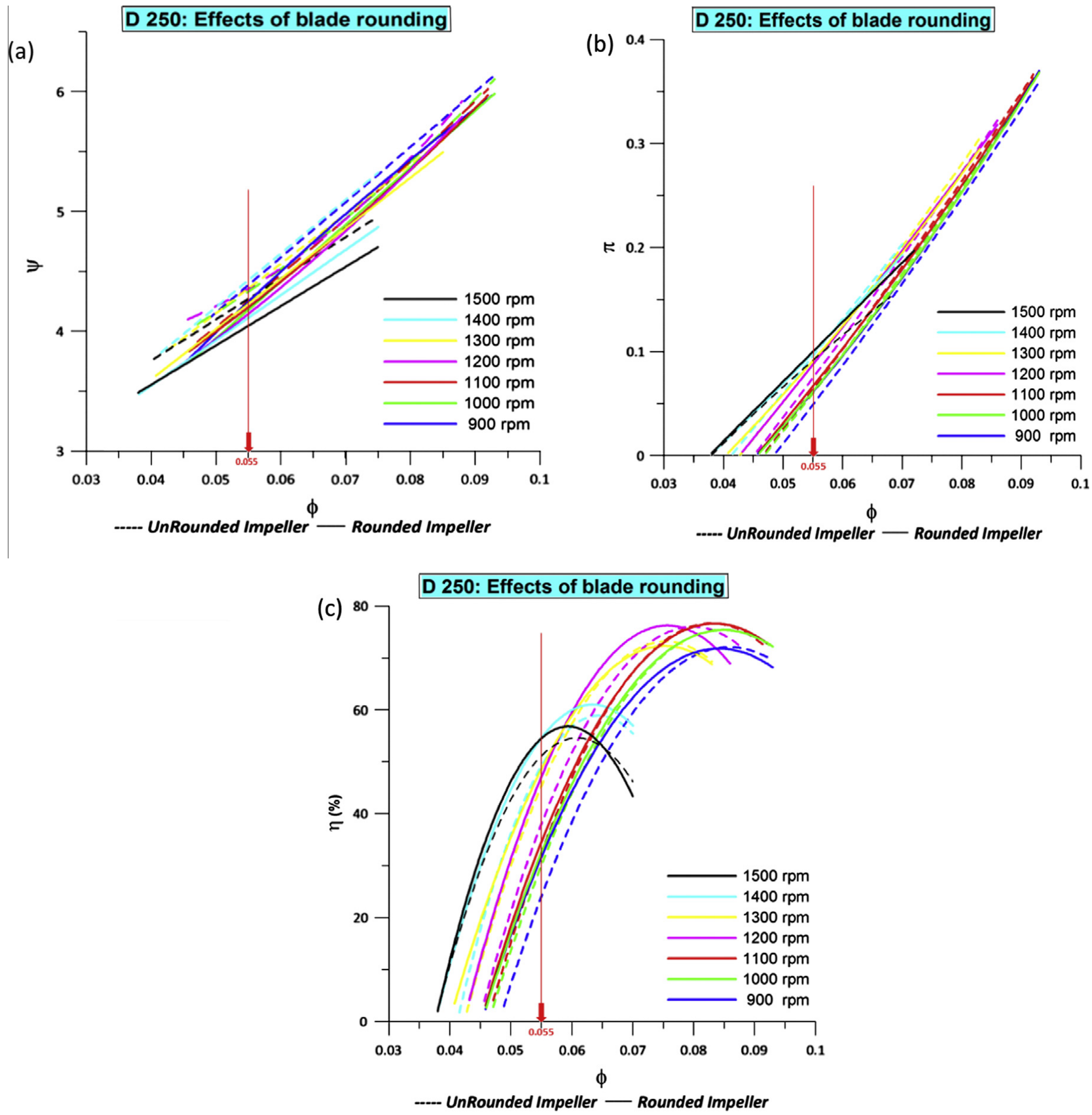

Fig. 12. Performance curves with original (∅ 250 mm) impeller at different speeds before and after blade rounding (a) ψ versus ϕ (b) π versus ϕ (c) η versus ϕ.

Table 6
Effects of blade rounding at ϕ = 0.055 at different speeds for original impeller.

Speed N (rpm)	% Change due to blade rounding in		
	Head number $\Delta\psi$	Power number $\Delta\pi$	Efficiency $\Delta\eta$
900	−3.17	25.76	31.57
1000	−3.57	3.26	8.39
1100	−1.20	4.49	6.20
1200	−4.83	17.23	23.88
1300	−2.94	3.82	7.32
1400	−7.07	3.16	10.91
1500	−5.25	9.05	6.79

Eq. (9) can be written as:

$$\eta_{BEP} = A_0(D/D_r)^{n_1} \cdot \exp(n_2[\mathrm{Ln}(D/D_r)]^2) \quad (10)$$

where A_0 is anti-Ln(A_1).

As shown in Fig. 20(a), a regression analysis to fit a second order polynomial equation through data points yielded the values of n_1 and n_2 as mentioned in Table 9.

In Eq. (10), the value of coefficient A_0 is a function of rotational speed ratio (N/N_r). Fig. 19(b) shows a monotonic increase of η_{BEP} with increasing N/N_r. The values of Ln(A_0) were plotted against Ln(N/N_r) as shown in Fig. 20(b) and a regression analysis to fit a second order polynomial equation yielded:

$$\mathrm{Ln}(A_0) = B_1 + m_1 \cdot \mathrm{Ln}(N/N_r) + m_2[\mathrm{Ln}(N/N_r)]^2 \quad (11)$$

Eq. (11) can be written as:

$$A_0 = B_0(N/N_r)^{m_1} \cdot \exp(m_2[\mathrm{Ln}(N/N_r)]^2) \quad (12)$$

where B_0 is anti-Ln(B_1).

As shown in Fig. 20(b), a regression analysis to fit a second order polynomial equation through data points yielded the values of B_0, m_1 and m_2 as mentioned in Table 9.

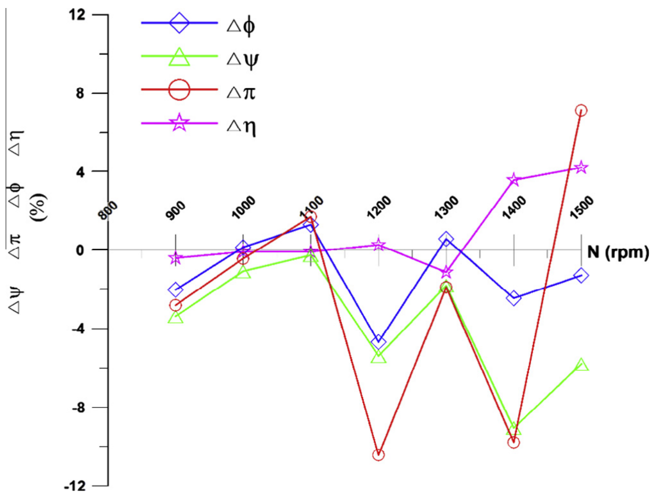

Fig. 13. Effects of blade rounding on BEP parameters at different speeds for original impeller.

Table 7
Effects of blade rounding at $\phi = 0.07$ at different speeds for 10% trimmed impeller.

Speed N (rpm)	% Change due to blade rounding in		
	Head number $\Delta\psi$	Power number $\Delta\pi$	Efficiency $\Delta\eta$
900	−9.27	−31.66	−23.85
1000	−5.20	−19.11	−16.64
1100	−7.38	−7.20	−0.20
1200	−6.71	−14.80	−9.68
1300	−10.71	2.31	15.51
1400	−9.44	10.15	12.86
1500	−12.30	8.51	24.33

From Eqs. (10) and (12),

$$\eta_{BEP} = B_o(D/D_r)^{n_1}(N/N_r)^{m_1}\exp(n_2[\mathrm{Ln}(D/D_r)]^2) \times \exp(m_2[\mathrm{Ln}(N/N_r)]^2) \quad (13)$$

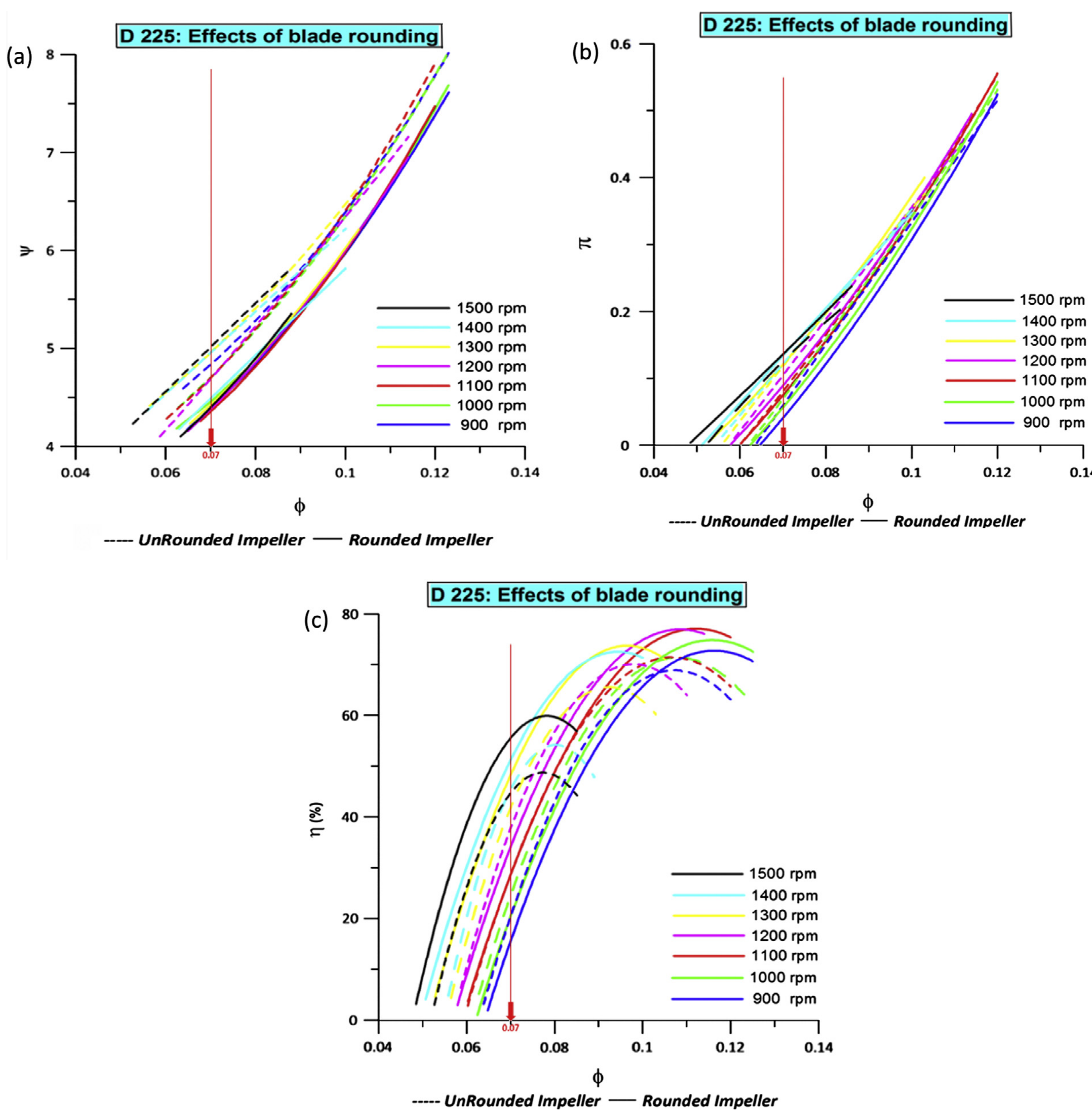

Fig. 14. Performance curves with 10% trimmed (∅ 225 mm) impeller at different speeds before and after blade rounding (a) ψ versus ϕ (b) π versus ϕ (c) η versus ϕ.

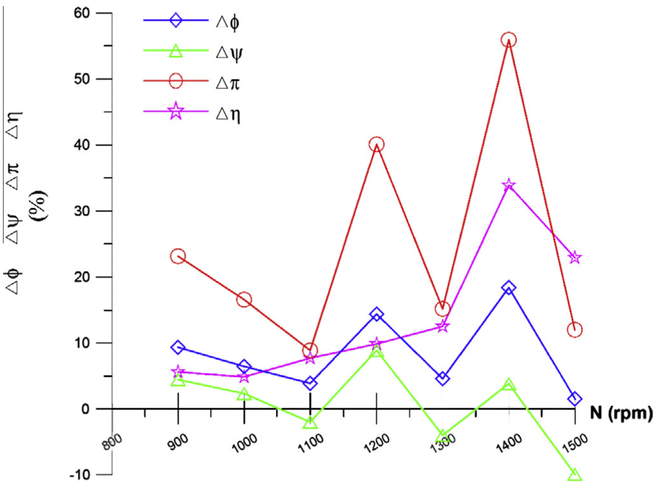

Fig. 15. Effects of blade rounding on BEP parameters at different speeds for 10% trimmed impeller.

Table 8
Effects of blade rounding at $\phi = 0.11$ at different speeds for 20% trimmed impeller.

Speed N (rpm)	% Change due to blade rounding in		
	Head number $\Delta\psi$	Power number $\Delta\pi$	Efficiency $\Delta\eta$
900	−1.91	10.18	14.12
1000	−3.78	6.77	9.78
1100	−7.70	−10.02	−3.38
1200	−3.22	31.46	32.72
1300	−10.16	3.89	14.53
1400	−10.66	−1.37	9.79
1500	−14.73	−8.20	8.53

Similar analysis was done for unrounded impeller and similar correlation was developed between η_{BEP}, D/D_r and N/N_r. The coefficients for unrounded and blade rounded impellers are tabulated in Table 9 [57].

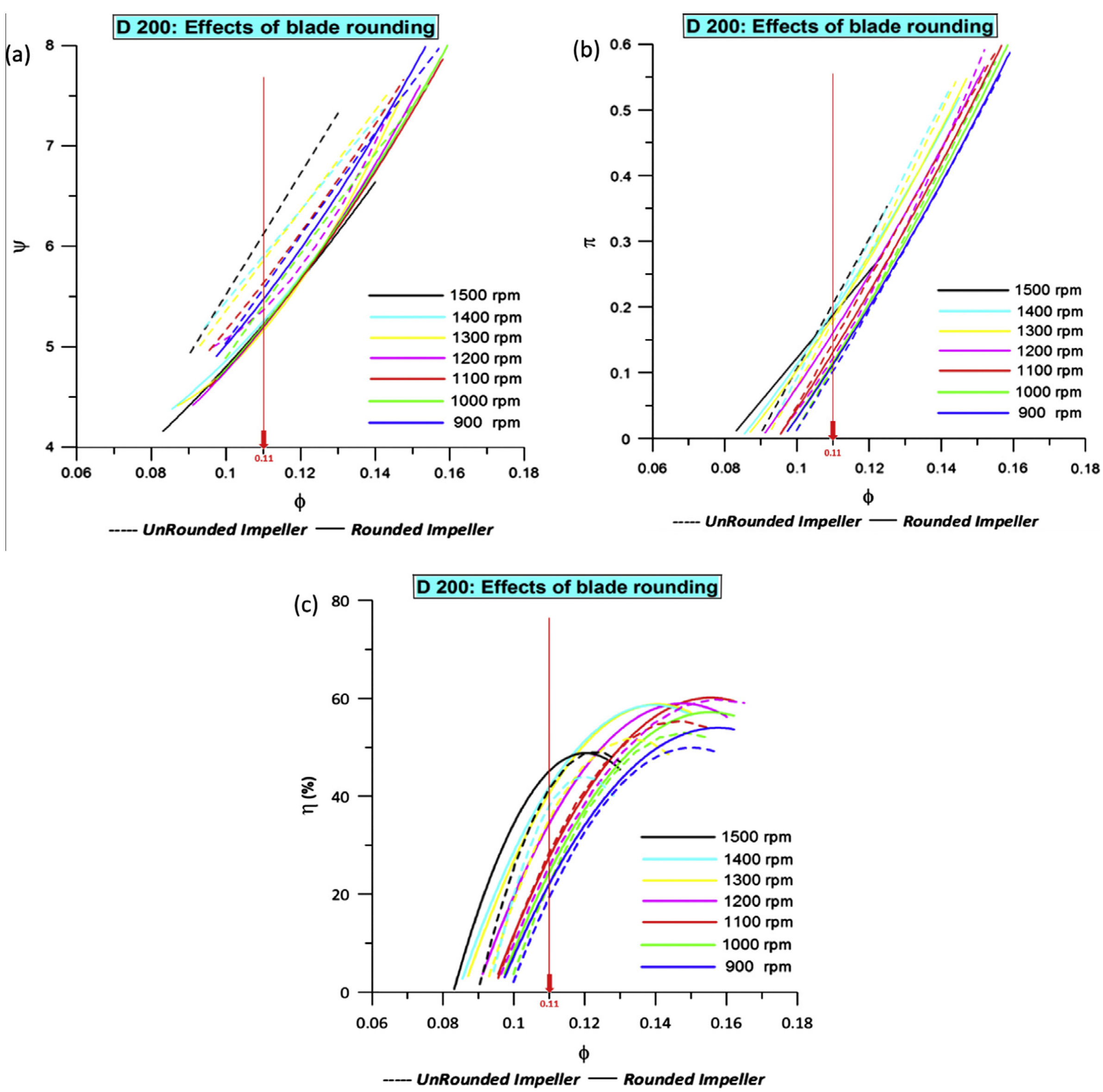

Fig. 16. Performance curves with 20% trimmed (∅ 200 mm) impeller at different speeds before and after blade rounding (a) ψ versus ϕ (b) π versus ϕ (c) η versus ϕ.

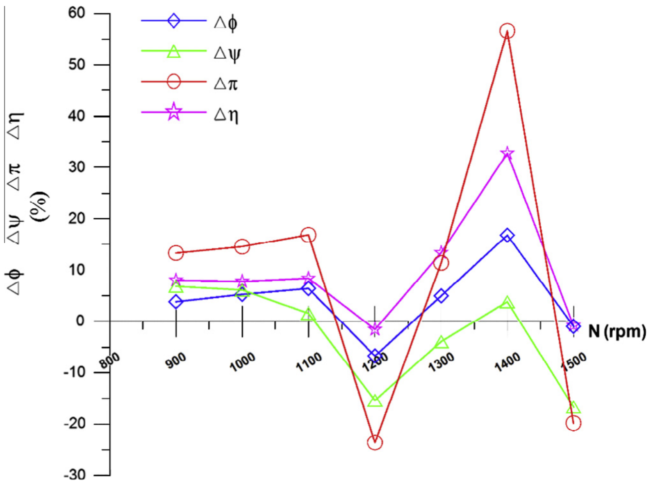

Fig. 17. Effects of blade rounding on BEP parameters at different speeds for 20% trimmed impeller.

A comparison between the η_{BEP} obtained from experimental investigation and those predicted by the empirical correlation, based on blade rounded impeller, is shown in Fig. 21(a). It was found that all the predicted data points lie within ±10% deviation lines of the experimental results. The correlation is applicable in the N/N_r range of 0.64–1.07 and D/D_r range of 0.8–1. The correlation developed, based on unrounded impeller, was applied to predict the η_{BEP} for different investigators e.g. Derakhshan and Nourbakhsh [32], Yang et al. [43], Fernandez et al. [44], Morros et al. [47], Sheng et al. [53], Sedlar et al. [54], Yang et al. [58] as shown in Fig. 21(b), which was found to vary in the range of ±13%. Carravetta et al. [48] predicted the PAT efficiency based on affinity laws and Suter parameters and the error between predicted and experimental efficiency was found less than 15%. More detailed investigations are recommended to minimize the error for the application of correlation in the wide range of N/N_r and D/D_r.

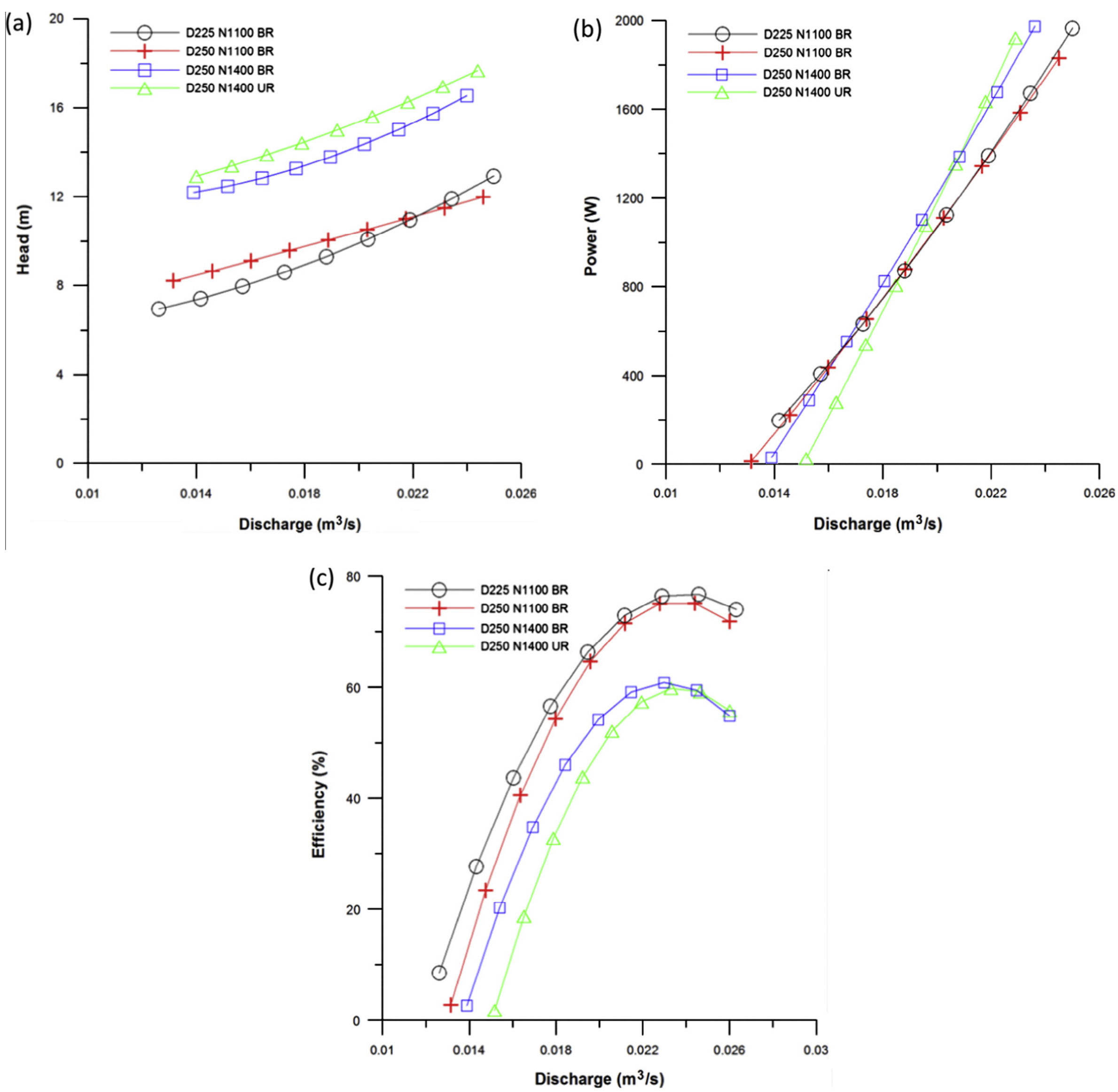

Fig. 18. Performance curves with different impellers (a) head versus discharge (b) power versus discharge (c) η versus discharge. (D = diameter (m), N = speed (rpm), UR = unrounded impeller, BR = blade rounded impeller).

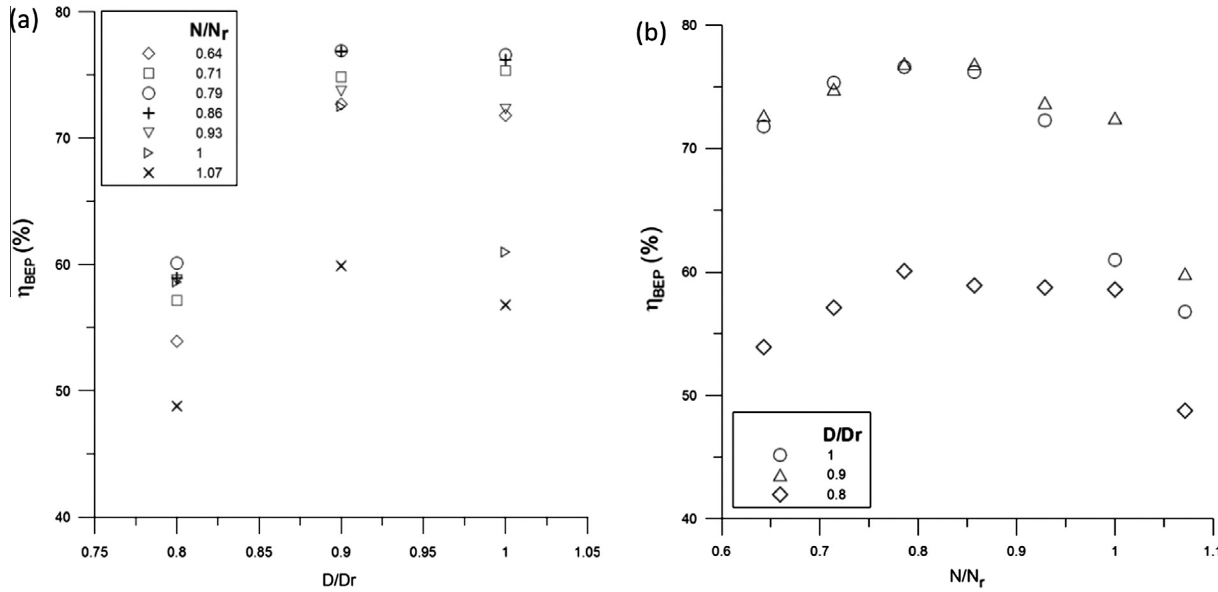

Fig. 19. (a) Effect of D/D_r on η_{BEP} (b) Effect of N/N_r on η_{BEP}.

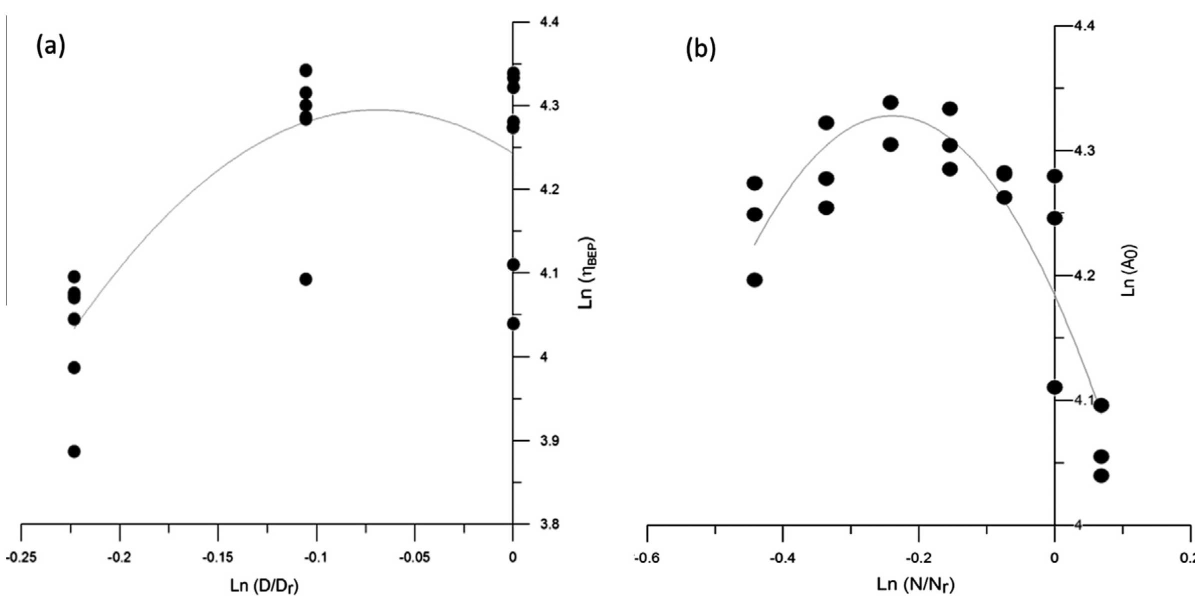

Fig. 20. (a) Plot of Ln (η_{BEP}) as a function of Ln (D/D_r) (b) Plot of Ln (A_0) as a function of Ln (N/N_r).

Table 9
Correlation coefficients.

Coefficients	For unrounded impeller	For blade rounded impeller
n_1	0.983	−1.516
n_2	−2.398	−10.99
m_1	−1.055	−1.202
m_2	−1.971	−2.513
B_0	67.98	65.63

6. Conclusions

In the present study, experimental investigations are carried out to improve the PAT performance by optimizing its geometric and operational parameters e.g. impeller diameter, and rotational speed. The experiments were performed in the wide range of rotational speeds varying from 900 to 1500 rpm with original (∅ 250 mm), 10% trimmed (∅ 225 mm) and 20% trimmed (∅ 200 mm) impellers before and after blade rounding. As a conclusion, three options are recommended for the performance enhancement of PAT e.g. use of original impeller after blade rounding, running the PAT at 1100 rpm with blade rounded impeller and use of 10% trimmed-blade rounded impeller at 1100 rpm. The maximum efficiency was obtained as 76.93% with 10% trimmed impeller at 1100 rpm. The empirical correlation was developed between η_{BEP}, D/D_r (range: 0.8–1) and N/N_r (range: 0.64–1.07) and applied for the present analysis. The predicted η_{BEP} was found to be within ±10% range of the experimental results.

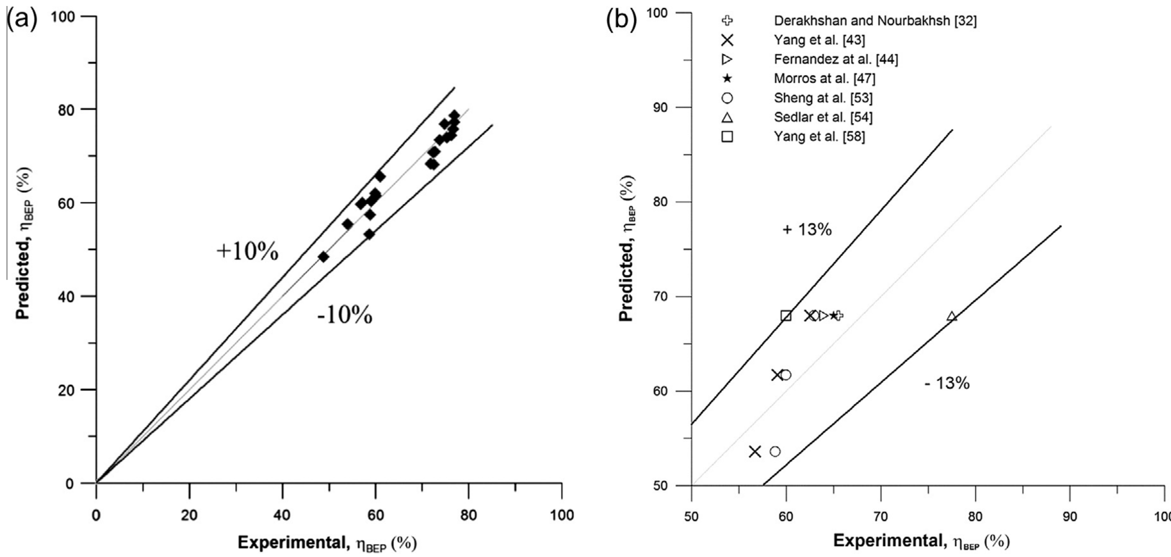

Fig. 21. Comparison experimental and predicted values of η_{BEP} (a) for present study (b) for other authors.

Acknowledgements

The authors are sincerely acknowledge the financial support received from Gujarat Council on Science and technology (GUJ-COST), Gandhinagar, Gujarat, India for the Minor Research Project entitled "Experimental Investigations on Pump as Turbine for Micro Hydropower Plants" wide letter No. GUJCOST/MRP/ 201574/10-11/3718 dated 1/12/2010.

References

[1] Yassi Y, Hashemloo S. Improvement of the efficiency of the Agnew micro hydro turbine at part loads due to installing guide vanes mechanism. Energy Convers Manag 2010;51:1970–5.
[2] Saxena P, Kumar A. Hydropower development in India. In: Proceedings of the international conference on hydraulic efficiency measurement. Roorkee, India; 2010, p. 1–6.
[3] Yassi Y. The effects of improvement of the main shaft on the operating conditions of the Agnew turbine. Energy Convers Manag 2009;50:2486–94.
[4] Prasad V, Shukla SN, Joshi SG. Performance characteristics of pump as turbine. Indian Pumps 2006:5–9.
[5] Small Hydropower Development. In: International course on small hydropower development. Roorkee, India; 2004.
[6] Obaseki TG. Hydropower opportunities in the water industry. Int J Environ Sci 2010;1(3). 392–02.
[7] Khurana S, Kumar A. Small hydro power – a review. Int J Therm Technol 2011;1(1):107–10.
[8] Sammartano V, Arico C, Carravetta A, Fecarotta O, Tucciarelli T. Banki–Michell optimal design by computational fluid dynamics testing and hydrodynamic analysis. Energies 2013;6:2362–85.
[9] Williams A. Pumps as turbines – a user's guide. 1st ed. London: Intermediate Technology; 1995.
[10] Williams AA. Pumps as turbines for low cost micro hydropower. Renew Energy 1996;9(1–4):1227–34.
[11] Nautiyal H, Varun Kumar A. Reverse running pumps analytical, experimental and computational study: a review. Renew Sustain Energy Rev 2010;14:2059–67.
[12] Patel N. Cavitation and draft tube analysis of pump running in turbine mode for micro hydropower plants [MTech thesis]. Nirma University; 2014.
[13] Carravetta A, Fecarotta O, Sinagra M, Tucciarelli T. Cost-benefit analysis for hydropower production in water distribution networks by a pump as turbine. J Water Resour Plann Manage 2013;140:1–8.
[14] Fernandez J, Blanco E, Parrondo J, Stickland MT, Scanlon TJ. Performance of a centrifugal pump running in inverse mode. Proc IMechE Part A J Power Energy 2004;218(4):265–71.
[15] Nautiyal H, Varun Kumar A, Yadav S. Experimental investigation of centrifugal pump working as turbine for small hydropower systems. Energy Sci Technol 2011;l(1):79–86.
[16] Singh P, Nestmann F. An optimization routine on a prediction and selection model for the turbine operation of centrifugal pumps. Exp Therm Fluid Sci 2010;34:152–64.
[17] Derakhshan S, Nourbakhsh A. Theoretical, numerical and experimental investigation of centrifugal pumps in reverse operation. Exp Therm Fluid Sci 2008;32:1620–7.
[18] Barrio R, Fernandez J, Parrondo J, Blanco E. Performance prediction of a centrifugal pump working in direct and reverse mode using computational fluid dynamics. In: Proceedings of the international conference on renewable energies and power quality. Granada, Spain; 2010, p. 1–5.
[19] Lueneburg R, Nelson RM. Hydraulic power recovery turbines. In: Lobanoff VS, Ross RR, editors. Centrifugal pumps: design and applications. Texas: Gulf Publishing Company; 1985. p. 246–82.
[20] Stepanoff AJ. Centrifugal and axial flow pumps. New York: John Wiley; 1957.
[21] Childs SM. Convert pumps to turbines and recover HP. Hydro Carbon Process Pet Refin 1962;41(10):173–4.
[22] Hancock JW. Centrifugal pump or water turbine. Pipe Line News 1963:25–7.
[23] McClaskey BM, Lundquist JA. Hydraulic power recovery turbines. In: Proceedings of the ASME conference 1976; 19: 76.
[24] Sharma K. Small hydroelectric project-use of centrifugal pumps as turbines. Technical report. Bangalore, India: Kirloskar Electric Co.; 1985.
[25] Schmiedl E. Serien-Kreiselpumpen im Turbinenbetrieb. Pumpentagung Karlsruhe October 1988 [Section A6].
[26] Alatorre-Frenk C. Cost minimization in micro hydro systems using pumps-as-turbines, PhDME Thesis. University of Warwick; 1994.
[27] Buse F. Using centrifugal pumps as hydraulic turbines. Chem Eng 1981:113–7.
[28] Kittredge CP. Centrifugal pumps used as hydraulic turbines. J Eng Power 1961:74–7.
[29] Diederich H. Verwendung von kreiselpumpen als turbinen. KSB Technische Berichte 1966;12:30–6.
[30] Grover KM. Conversion of pumps to turbines. Katonah, New York: GSA InterCorp; 1980.
[31] Lewinsky-Keslitz H-P. Pumpen als Turbinen fur Kleinkraftwerke. Wasserwirtschaft 1987;77(10):531–7.
[32] Derakhshan S, Nourbakhsh A. Experimental study of characteristic curves of centrifugal pumps working as turbines in different specific speeds. Exp Therm Fluid Sci 2008;32:800–7.
[33] Chapallaz JM, Eichenberger P, Fischer G. Manual on pumps used as turbines. Braunschweig: Vieweg; 1992.
[34] Jain S, Patel R. Investigations on pump running in turbine mode: a review of the state-of-the-art. Renew Sustain Energy Rev 2014;30:841–68.
[35] Motwani KH. Experimental and numerical investigations on pump as turbine for micro hydropower plants [M Tech thesis]. Nirma University; 2012.
[36] IS 5496:1993. Guide for preliminary dimensioning and layout of elbow type draft tubes for surface hydroelectric power stations. Bureau of Indian standards.
[37] IS 4722: 2001. Rotating electrical machines – specification. Bureau of Indian standards.
[38] IS 9320: 1996. Guide for testing direct-current machines. Bureau of Indian standards.
[39] Kline SJ, McClintock FA. Describing the uncertainties in single sample experiments. Mech Eng 1953:3–8.
[40] Bansal RK. A textbook of fluid mechanics and hydraulic machines. 9th ed. New Delhi: Laxmi Publications; 2005.
[41] Kumar DS. Fluid mechanics and fluid power engineering. 7th ed. New Delhi: S.K. Kataria & Sons Publications; 2008.
[42] Introduction to Pump. <http://engineers4world.blogspot.in/2009/10/pump-part-1.html> [accessed 11.10.14].

[43] Yang SS, Liu HL, Kong FY, Dai C, Dong L. Experimental, numerical, and theoretical research on impeller diameter influencing centrifugal pump-as-turbine. J Energy Eng, ASCE 2013;139:299–307.
[44] Fernandez J, Barrio R, Blanco E, Parrondo JL, Marcos A. Numerical investigation of a centrifugal pump running in reverse mode. Proc IMechE Part A J Power Energy 2010;224:373–81.
[45] Shukla A. Performance evaluation of pump as turbine [M Tech thesis]. Indian Institute of Technology Roorkee; 2008.
[46] Rawal S, Kshirsagar JT. Numerical simulation on a pump operating in a turbine mode. In: Proceedings of the twenty-third international pump users symposium. Houstan, Texas; 2007. p. 21–7.
[47] Morros CS, Oro JMF, Diaz KMA. Numerical modelling and flow analysis of a centrifugal pump running as a turbine: unsteady flow structures and its effects on the global performance. Int J Num Methods Fluids 2011;65(5):542–62.
[48] Carravetta A, Fecarotta O, Martino R, Antipodi L. PAT efficiency variation with design parameters. Proc Eng 2014;70:285–91.
[49] Singh P. Optimization of internal hydraulics and of system design for pumps as turbines with field implementation and evaluation [PhD thesis]. University of Karlsruhe; 2005.
[50] Derakhshan S, Nourbakhsh A, Mohammadi B. Efficiency improvement of centrifugal reverse pumps. ASME J Fluids Eng 2009;131:21103–9.
[51] Suarda M, Suarnadwipa N, Adnyana WB. Experimental work on the modification of impeller tips of a centrifugal pump as a turbine. In: Proceedings of the 2nd joint international conference on sustainable energy and environment. Thailand, Bangkok; 2006, p. 1–5.
[52] Singh P, Nestmann F. Internal hydraulic analysis of impeller rounding in centrifugal pumps as turbines. Exp Therm Fluid Sci 2011;35:121–34.
[53] Sheng YS, Yu KF, Ming JW, Yun QX. Effects of impeller trimming influencing pump as turbine. Comput Fluids 2012;67:72–8.
[54] Sedlar M, Soukal J, Komarek M. CFD analysis of middle stage of multistage pump operating in turbine regime. Eng Mech 2009;16(9):413–21.
[55] Day SW, Lemire PP, Flack RD, McDaniel JC. Effect of Reynolds number on performance of a small centrifugal pump. In: Proceedings of the 4th ASME JSME joint fluids engineering conference. Honolulu, USA; 2003, p. 1–7.
[56] Energy design resources, centrifugal pump application and optimization. <http://energydesignresources.com/media/1678/EDR_DesignBriefs_centrifugalpumps.pdf?tracked=true> [accessed 11.10.14].
[57] Swarnkar A. Performance enhancement of pump as turbine for micro hydropower plants [M Tech thesis]. Nirma University; 2013.
[58] Yang SS, Derakhshan S, Kong FY. Theoretical, numerical and experimental prediction of pump as turbine performance. Renew Energy 2012;48:507–13.