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Development of domestic water pump to use as a turbine for pico hydro (PAT)

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Abstract

Pico hydro is one of the most appropriate and an economical way of providing electricity for rural areas anywhere in the world, where there is sufficient hydro energy potential. The present technology trend in relation to Pico hydro uses a domestic centrifugal water pump as the turbine. In this process, standard pump units are used as a low-cost alternative to conventional turbines to provide stand-alone electricity generation for isolated houses and remote communities who do not have electricity. Considering the high cost and unavailability of suitable axial flow pumps, the centrifugal pumps are used as turbines.

Deferent types of impellers for the turbine were designed, fabricated and tested. These impellers were developed by changing the number of vanes, vane angles, diameter of the impeller, type of impeller (closed type, semi open) and vane thickness. Brake load testing was carried out for those impellers to check their performance to be used as turbines.

Efficiency of an existing impeller when used as a turbine was 13% and maximum mechanical output power was 69 W running at 700 rpm, 4.0 l/s flow rate and 18 psi pressure head. The newly developed impellers, which were made of fiber reinforced plastic (FRP), reached a maximum mechanical output power of 130 W, running at 1000 rpm, flow rate of 4 l/s, pressure head of 21 psi and the efficiency was 22%. It is envisaged that the efficiency could be further improved beyond 30% by increasing the flow rate. Further research and development is done to optimize the impeller design.

Keywords: pump as turbine, pico-hydro, hydro power generation, centrifugal pumps

Introduction

Pico hydro is a term used for hydroelectric power generation of under 5 kW. It is involved in designing and installing small water-power schemes. Examples of devices which can be powered by Pico hydro are light bulbs, televisions, refrigerators and battery chargers. Pico hydro systems make use of the potential energy in water.

Pico hydro is the use of pump as a turbine (PAT) concept. It concerns the use of standard pump units as a low-cost alternative to conventional turbines to provide stand-alone electricity generation for isolated houses and remote communities that require only a small amount of electricity. In the recent researches through the world, propeller type axial pump concept has shown high efficiency rather than radial flow pump. But considering on the local situation, it is not economical to use an axial flow pump due to its high cost and unavailability.

And also, there are small Pelton turbine units for Pico hydro. But costs of those units are high.

Therefore, if the PAT concept can be developed in Sri Lanka,

- Cost of power generation units for Pico hydro can be reduced.
- The units can be easily installed in hydro sites.
- The units can be used for electricity power generation in low head and high flow rate hydro sites.

Materials and Methods

Methodology

The project was conducted, in two stages.

1. Checking the performance with the change of the flow rate for 2" commercial water pump when used as a turbine.
2. Design impellers for the above pump to improve its performance to use as turbine.

   Brake load testing (Figure 2.1 and Figure 3.16) was carried out for obtaining mechanical power of the pump shaft. In this case, inlet water flow rate at 4 l/s and pressure head at 18 psi were maintained approximately. And also theoretical equations to calculate the mechanical power and efficiency of the pump as turbine unit were used. Those equations are as shown below.

$$P = (T_1 - T_2) \times r \times \omega \quad ......01$$

P = Power (W)
T_1 = Tight Side Tension (N) of Friction Belt in Brake Load Testing
T_2 = Slack Side Tension (N) of Friction Belt in Brake Load Testing
r = Radius of Brake Load Pulley (m); ω = Rotational Speed of Pulley (rads-1)

$$\eta_m = (P_O/P_W)\times 100\% \quad ......02$$

η_m = Mechanical Efficiency of the Pump as Turbine Unit(%)
P_O = Maximum Output Power of the Pump as Turbine Unit (W)
P_W = Input Water Power to the Pump as Turbine Unit (W)

$$P_W = P\times Q \quad ....03$$

P = Pressure of the Inlet Flow to the Pump as Turbine Unit (N/m^2)
Q = Inlet Flow Rate to the Pump as Turbine Unit (m^3/s)

Impeller fabrication process

In this project, the main target was to design and fabricate new impeller for improving the performance of the pump as turbine unit. After some study about fiber it was decided to use the fiber as the material to fabricate new impellers. In fiber fabrication there are three elements; fiber, adhesive fiber gum; acid.

In the process firstly the adhesive gum and acid should be mixed together. The acid controls the bonding time of fiber. Then that gum and acid mixture should be applied as a layer (Figure 2.2). Next fiber layer should be lay on (Figure 2.3). Again gum and acid mixture should be applied as layer and these steps must be continuing until the required thickness is achieved. To do this fabrication, cement mould for lower part of the impeller (Figure 2.4) is needed and mould for required vane shapes are made by using aluminum sheet. In this case, firstly the impeller vanes are drawn by using Solid works software and according to the drawing, strip of aluminum sheet are curved until the required vane shape achieved. Then the moulds are filled by using fiber as mentioned above. After drying the fiber products, those are removed from moulds and do the required machining. Finally vanes should be fixed by using nut and bolts to lower part of the impeller which is produced by fiber.

Results and Discussion

Testing the existing impeller of the pump for pump as turbine unit

At the initial stage the pump was tested as a turbine with existing impeller (Figure 3.1) by changing the flow rate and load. According to Graph 3.1,

- Maximum mechanical power at 725 rpm = 69 W

By using equation 02 and 03,

- Efficiency of the pump as turbine at 725 rpm = 13.82 %

Brake load testing conditions: Flow rate = 4.125 l/s pressure= 17 psi

Table 1: Testing data of tension, speed and calculated power at flow rate = 4.125 l/s

Tight Side Tension (N)	Slack Side Tension (N)	Speed (rpm)	Power (W)
0	0	1137	0
5	0.7	930	53
7.4	1.75	890	67
8.8	2.25	773	67
11.2	3.25	633	67
14.4	4.5	399	53
16	5.75	160	22
17.8	7.25	69	10
22.6	12.75	0	0

By changing the flow rate it was able to obtain a relationship between the flow rate and the maximum power for existing impeller. It is shown that with the increase of the flow rate max power also increase. Further there should be specific flow rate for optimum value of maximum mechanical power of pump as turbine. So, further researches should be done for prove this.

Table 2: Relationship between the flow rate and maximum power

Flow Rate (l/s)	Maximum Power (W)
3.17	15.924
3.75	45.9144
3.94	62.2347
4.125	69
4.146	69.5529

Testing the reverse type impeller for pump as turbine unit

According to the above testing, existing impeller in the pump is not suitable for pump as turbine concept. So, next step was to fabricate a new impeller. As the initial step it was decided to fabricate the reverse type of the existing impeller (Figure 3.2) and the testing was carried out. But the obtainable speed was very low value to continue a brake load test. Hence the reverse type was not the area to develop.

Fabricating and testing the impeller with same specifications as the existing impeller

According to above testing results, it was decided to go for changing the impeller specifications. Then the impeller with same specifications as the existing impeller (Figure 3.3) was fabricated.

Inlet vane angle = 30^0; Outlet vane angle = 15^0; No of vanes = 4

Brake load testing conditions: Flow rate = 4.146 l/s pressure= 18 psi

Table 3: Testing data of tension, speed and calculated power for impeller-A2

Tight Side Tension (N)	Slack Side Tension (N)	Speed (rpm)	Power (W)
0	0	1440	0
3.4	0.25	1306	55
5.8	1	1204	77
7.2	1.75	1072	78
10.6	3	920	93
12	3.75	834	92
13.2	4	755	92
15.2	5	634	86
16.6	5.75	532	77
18	6.25	439	69
19.6	7	338	57
20.6	8	220	37
22	9	102	18
23.4	10.5	51	9

According to the Graph 3.2, the large improvement of the new impeller can be seen than original impeller. In this case, new impeller is less weight than original impeller. But it can't be said directly, weight is the main reason for this improvement. Because there are some other factors which can be affected for the improvement. Those are,

- New impeller has good finished (more smooth) than the original impeller.
- Vane thickness of the new impeller is larger than original impeller.

Testing the new impellers by changing the number of vanes

1. Impeller-A0 (Figure 3.4)
   Inlet vane angle = 30^0; Outlet vane angle = 15^0; No of vanes = 2
   Brake load testing conditions: Flow rate = 4.146 l/s pressure = 18 psi

Table 4: Testing data of tension, speed and calculated power for impeller-A0

Tight Side Tension (N)	Slack Side Tension (N)	Speed (rpm)	Power (W)
0	0	830	0
4.4	0.75	720	35
6	1.25	706	45
7.6	2	650	48
9	2.5	585	51
9.8	2.75	530	50
11.2	3.5	408	42
12	3.75	310	34
12.6	4	230	26
13.6	4.75	120	14
14.2	5.25	70	8
16.4	8	0	0

Brake load testing results (Graph 4.1):

- Maximum mechanical Power at 542 rpm =52.4135 W
- Efficiency of the pump as turbine at 542 rpm =10.05 %

2. Impeller-A1(Figure 3.5)
   Inlet vane angle = 30^0; Outlet vane angle = 15^0; No of vanes = 3
   Brake load testing conditions: Flow rate = 4.146 l/s pressure = 18 psi

Table 5: Testing data of tension, speed and calculated power for impeller-A1

Tight Side Tension (N)	Slack Side Tension (N)	Speed (rpm)	Power (W)
0	0	1350	0
5.4	1	1025	60
6.6	1.5	965	65
8.2	2.25	812	64
10.2	3	690	66
12.2	4	580	63
13.8	4.5	485	60
15.4	5.25	420	57
17	6	322	47
18.2	6.5	130	20
19.2	7.25	79	13
20.6	8.5	40	6
22.2	9.75	18	3

Brake load testing results (Graph 4.1):

- Maximum mechanical Power at 748 rpm = 69.7562 W
- Efficiency of the pump as turbine at 748 rpm = 13.38 %

3. Impeller-A2 (Figure 3.3)
   Inlet vane angle = 30^0; Outlet vane angle = 15^0; No of vanes = 4
   Brake load testing conditions: Flow rate = 4.153 l/s pressure = 20 psi

Brake load testing results (Graph 4.1):

- Maximum mechanical Power at 858 rpm =95.5863 W
- Efficiency of the pump as turbine at 858 rpm =16.47 %

Table 6: Testing data of tension, speed and calculated power for impeller-A2

Tight Side Tension (N)	Slack Side Tension (N)	Speed (rpm)	Power (W)
0	0	1440	0
3.4	0.25	1306	55
5.8	1	1204	77
7.2	1.75	1072	78
10.6	3	920	93
12	3.75	834	92
13.2	4	755	92
15.2	5	634	86
16.6	5.75	532	77
18	6.25	439	69
19.6	7	338	57
20.6	8	220	37
22	9	102	18
23.4	10.5	51	9

4. Impeller-A3 (Figure 3.6)
   Inlet vane angle = 30^0; Outlet vane angle = 15^0; No of vanes = 5
   Brake load testing conditions:Flow rate = 4.072 l/s pressure = 20 psi

Table 7: Testing data of tension, speed and calculated power for impeller-A3

Tight Side Tension (N)	Slack Side Tension (N)	Speed (rpm)	Power (W)
0	0	1540	0
6.2	1.5	1285	80
8	2.25	1195	91
10.4	3.25	1045	99
13.8	4.85	815	97
16.4	6	630	87
17.8	6.75	515	76
19.8	7.5	415	68
22.4	9.75	195	33
24.6	11.5	90	16
25.2	12	60	11

Brake load testing results (Graph 4.1):
- Maximum mechanical Power at 934 rpm =105.387 W
- Efficiency of the pump as turbine at 934 rpm =18.52 %

5. Impeller-A4 (Figure 3.7)
   Inlet vane angle = 30^0; Outlet vane angle = 15^0; No of vanes = 6
   Brake load testing conditions:Flow rate = 4.167 l/s pressure = 20 psi

Table 8: Testing data of tension, speed and calculated power for impeller-A4

Tight Side Tension (N)	Slack Side Tension (N)	Speed (rpm)	Power (W)
0	0	1560	0
5.8	1.25	1312	79
8.2	2.25	1190	94
10.2	3.25	1089	101
12.4	3.75	970	112
13.8	4.5	855	106
15.4	5.25	725	98
16.8	6.25	545	76
18.8	7.25	468	72
20.4	8	354	58
21.6	8.75	240	41
23.2	10	130	23
24.8	11.5	70	12

Brake load testing results (Graph 4.1):
- Maximum mechanical Power at 947 rpm =108.839 W
- Efficiency of the pump as turbine at 947 rpm =18.69 %

Testing the new impellers by changing the vane angle

1. Impeller-B (Figure 3.8)
   Inlet vane angle $= 0^0$; Outlet vane angle $= 45^0$; No of vanes = 4
   Brake load testing conditions: Flow rate = 4.167 l/s pressure = 20 psi

Table 9: Testing data of tension, speed and calculated power for impeller-B

Tight Side Tension (N)	Slack Side Tension (N)	Speed (rpm)	Power (W)
0	0	1235	0
5.6	1.25	930	54
7.2	2	880	61
9.6	3	810	71
11.6	3.75	640	67
13.8	5	530	62
15.4	5.75	438	56
18	7.25	210	30
20	8.75	75	11
23.2	11.75	10	2

Brake load testing results (Graph 4.2):

- Maximum value of Power at 703 rpm =68.3358 W
- Efficiency of the pump as turbine at 703 rpm =11.74%

2 Impeller-C (Figure 3.9)
Inlet vane angle $= 60^0$; Outlet vane angle $= 0^0$; No of vanes = 4
Brake load testing conditions:Flow rate = 4.167 l/s pressure = 20 psi

Brake load testing results (Graph 4.2):

- Maximum value of Power at 876 rpm =105.334 W
- Efficiency of the pump as turbine at 876 rpm =18.09 %

Table 10: Testing data of tension, speed and calculated power for impeller-C

Tight Side Tension (N)	Slack Side Tension (N)	Speed (rpm)	Power (W)
0	0	1440	0
6.8	1.5	1192	84
8.4	2.25	1115	91
11.2	3.5	962	99
13.6	4.5	850	103
15.6	5.5	730	98
17	6	635	93
19.2	7.25	460	73
20.8	8.25	375	63
22.8	9.5	180	32
23.2	10	136	24
24.2	10.75	85	15

3 Impeller-D (Figure 3.10)
Inlet vane angle $= 15^0$ Outlet vane angle $= 30^0$; No of vanes = 4
Brake load testing conditions: Flow rate = 4.167 l/s pressure = 20 psi

Brake load testing results (Graph 4.2):

- Maximum value of Power at 753 rpm =78.6088 W
- Efficiency of the pump as turbine at 753 rpm =13.5 %

Table 11: Testing data of tension, speed and calculated power for impeller-D

Tight Side Tension (N)	Slack Side Tension (N)	Speed (rpm)	Power (W)
0	0	1280	0
5.8	1.25	1026	62
7.8	2.25	955	70
10.6	3.5	805	76
12.4	4.25	694	75
14.2	5	605	74
16	6	490	65
17.8	6.75	385	57
19.4	7.75	200	31
20	8.25	122	19
20.6	9	75	12

Testing the new impellers by changing diameter of the impeller

In this case the number of vanes as four and the inlet vane angle at 60^0 and outlet vane angle at 0^0 were kept as constant and changed the diameter of the impeller.

1 Impeller – E (Figure 3.11)
Diameter = 129 mm
Brake load testing conditions: Flow rate = 4.146l/s pressure= 18 psi

Table 12: Testing data of tension, speed and calculated power for impeller-E

Tight Side Tension (N)	Slack Side Tension (N)	Speed (rpm)	Power (W)
0	0	1210	0
4.6	0.5	1050	57
6.6	1.25	960	68
9.6	2.5	785	74
11.6	3.5	665	72
14	4.75	535	66
16	5.75	435	59
18.4	6.75	245	38
19.6	7.5	135	22
21.2	8.75	55	9

Brake load testing results (Graph 4.3):

- Maximum value of Power at 755 rpm =78.4805 W
- Efficiency of the pump as turbine at 755 rpm =15.24 %

2 Impeller – F (Figure 3.12)
Diameter = 134 mm
Brake load testing conditions: Flow rate = 4.146l/s pressure= 18 psi

Table 13: Testing data of tension, speed and calculated power for impeller-F

Tight Side Tension (N)	Slack Side Tension (N)	Speed (rpm)	Power (W)
0	0	1050	110
5.8	1	765	80
8.8	2.25	625	65
11	3.25	510	53
13.2	4	405	42
14.8	5	300	31
16.4	6.25	67	7

Brake load testing results (Graph 4.3):

- Maximum mechanical Power at 590 rpm =55.1855 W
- Efficiency of the pump as turbine at 590 rpm =10.72%

Testing the closed type impeller

All the above impellers were semi open type impellers. To find out the effect on maximum mechanical power of the pump shaft, rotational speed of the pump shaft related to maximum mechanical power of the pump shaft and efficiency of the pump as turbine, a closed type impeller was fabricated by keeping the number of vanes as 4 and inlet vane angle at 60^0 and outlet vane angle at 0^0.

1 Impeller – G (Figure 3.13)
Diameter = 139 mm
Brake load testing conditions: Flow rate = 4.146 l/s pressure= 18 psi

Table 14: Testing data of tension, speed and calculated power for impeller-G

Tight Side Tension (N)	Slack Side Tension (N)	Speed (rpm)	Power (W)
0	0	1115	0
5.2	1	900	50
6.6	1.5	865	59
7.4	2	830	60
9.6	3	770	68
12	4.25	670	69
13.2	4.75	625	70
15.2	5.75	560	70
18.4	7.5	450	65
20.2	8.25	385	61
20.4	9	330	50
22.6	10.5	150	24
24	11.25	95	16
25	12.75	45	7

Brake load testing results (Graph 4.4):

- Maximum mechanical Power at 630 rpm =71.5829 W
- Efficiency of the pump as turbine at 630 rpm =13.9 %

Testing the thin vane impeller

1 Impeller – I (Figure 3.14)
Diameter = 139 mm; Inlet vane angle = 60^0; Outlet vane angle = 0^0; No of vanes = 4
Brake load testing conditions:Flow rate = 4.146l/s pressure= 18 psi

Table 15: Testing data of tension, speed and calculated power for impeller-I

Tight Side Tension (N)	Slack Side Tension (N)	Speed (rpm)	Power (W)
0	0	1065	111
5.4	0.5	790	83
6.4	1	680	71
9.2	2.25	595	62
11	3	515	54
12.6	3.75	450	47
14.6	5	355	37
16	5.5	250	26
17.4	6.5	145	15
19.4	8	40	4

Brake load testing results (Graph 4.5):

- Maximum value of Power at 592 rpm =55.6566 W
- Efficiency of the pump as turbine at 592 rpm =10.81%

Testing the newly designed impeller

By considering all the testing results it could be concluded to a final design which having following specifications.

1 Impeller – H (Figure 3.15)
Number of vanes = 6; Vane angles = Inlet 60^0, Outlet 0^0;
Semi open type
Diameter = 139 mm (as same as the original impeller)
Brake load testing conditions: Flow rate = 3.7 l/s pressure= 21 psi

Table 16: Testing data of tension, speed and calculated power for impeller-H

Tight Side Tension (N)	Slack Side Tension (N)	Speed (rpm)	Power (W)
0	0	1415	148
4.8	0.5	1230	129
6.6	1.25	1165	122
9.8	2.5	1035	108
12	3.75	920	96
15.2	5	720	75
17.2	6	595	62
20	7.5	425	44
22	8.75	265	28
24.2	10.25	115	12
25.4	11	60	6

Brake load testing results (Graph 4.6):

- Maximum value of Power at 870 rpm =106.779 W
- Efficiency of the pump as turbine at 870 rpm =19.93 %

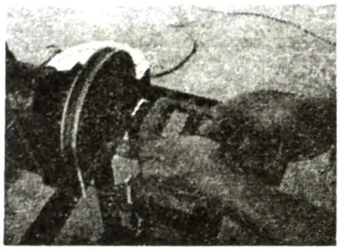


Fig. 2.1: Brake load testing

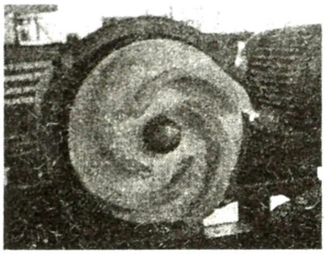


Fig. 3.2: Reverse Type Impeller

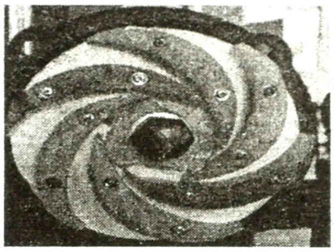


Fig. 3.7: Impeller-A4

Fig. 2.2. Applying Adhesive Gum

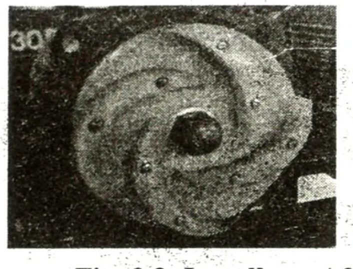


Fig. 3.3: Impeller – A2

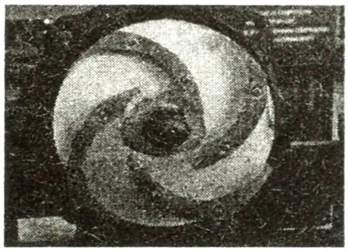


Fig. 3.8: Impeller – B

Fig. 2.3: Applying Fiber

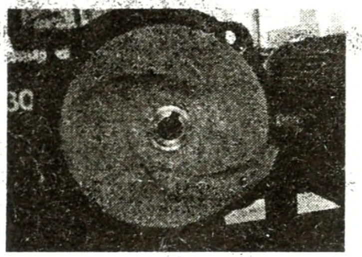


Fig. 3.4: Impeller – A0

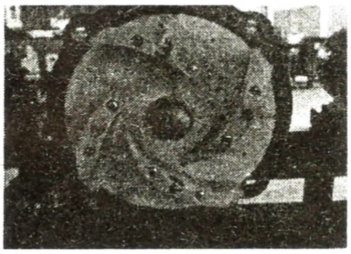


Fig. 3.9: Impeller – C

Fig. 2.4: Cement mould

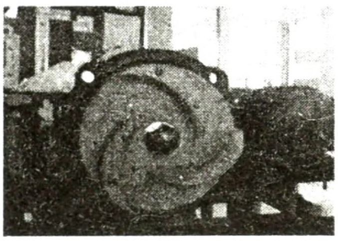


Fig. 3.5: Impeller – A1

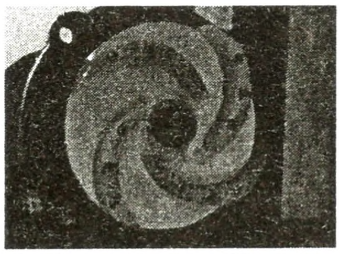


Fig. 3.10: Impeller – D

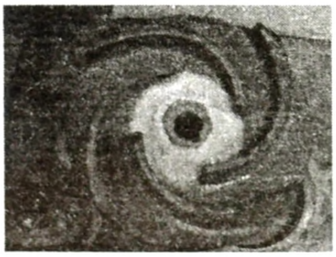


Fig. 3.1: Existing Impeller

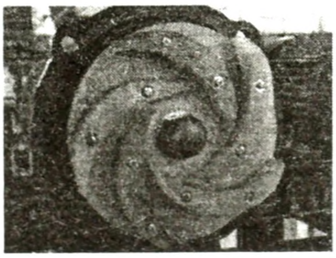


Fig. 3.6: Impeller – A3

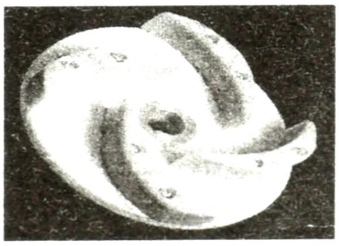


Fig. 3.11: Impeller – E

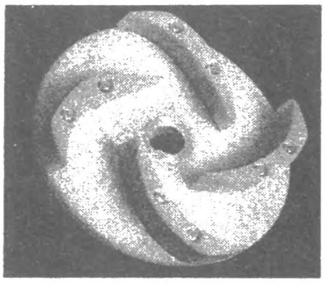


Fig. 3.12: Impeller – F

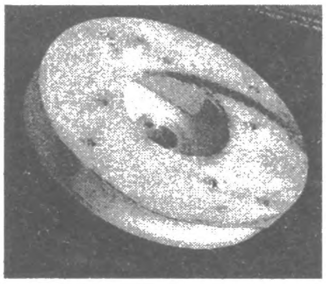


Fig. 3.13: Closed Type Impeller

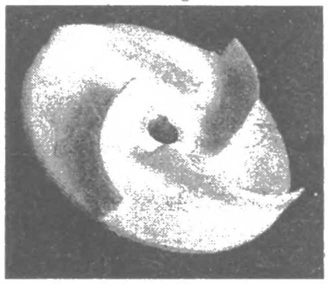


Fig. 3.14: Thin Vane Impeller

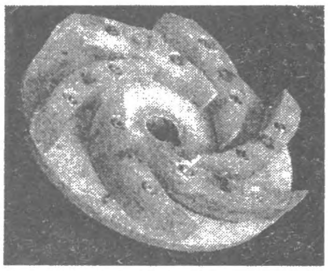


Fig. 3.15: Newly Designed Impeller

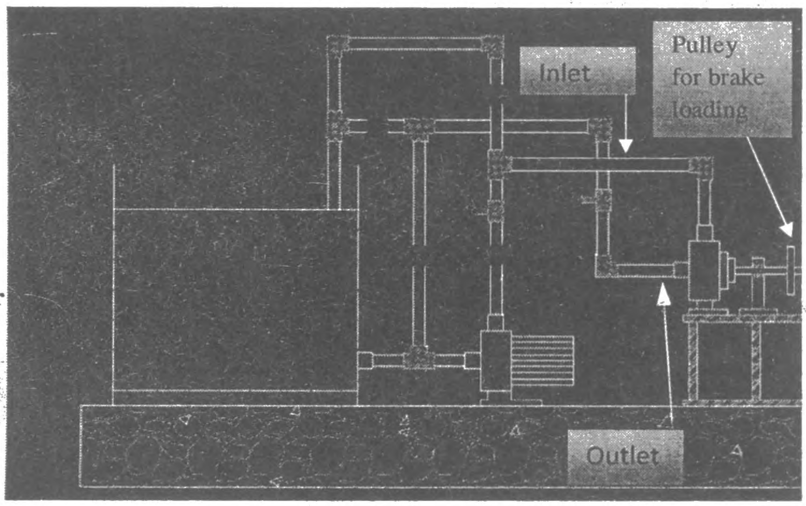
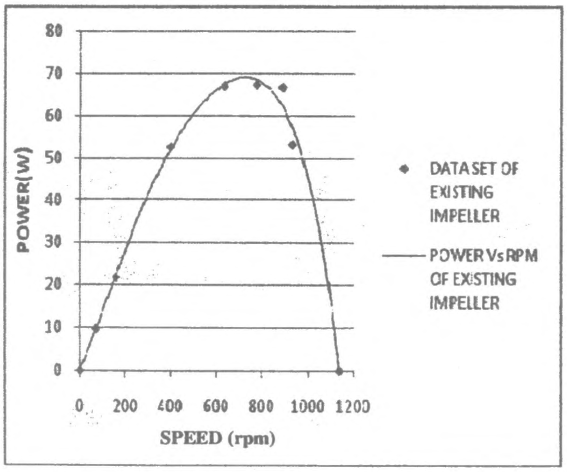
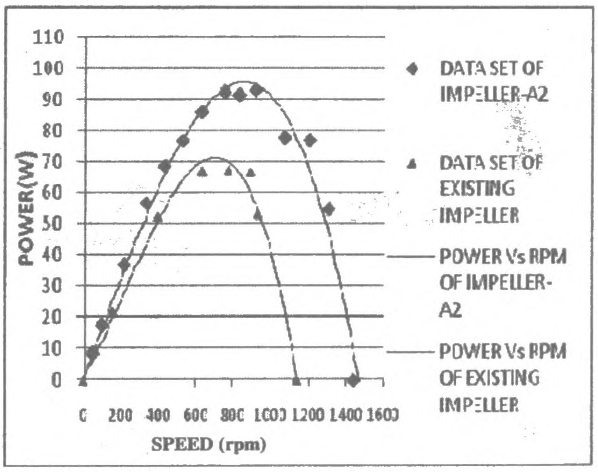


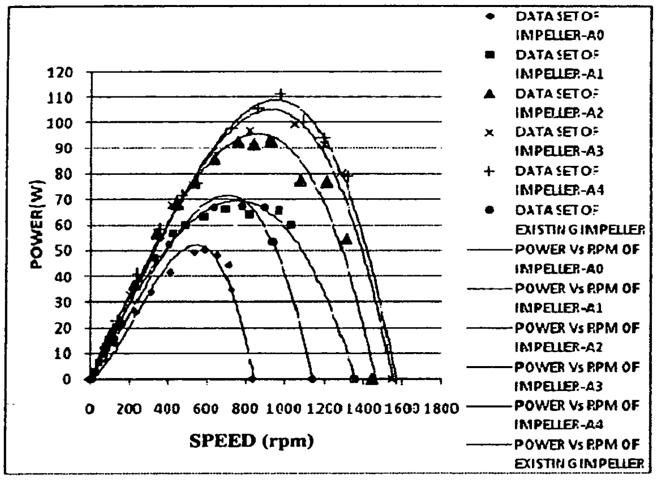
Fig. 3.16: Auto CAD Drawing of Test Bench for Brake Load Testing



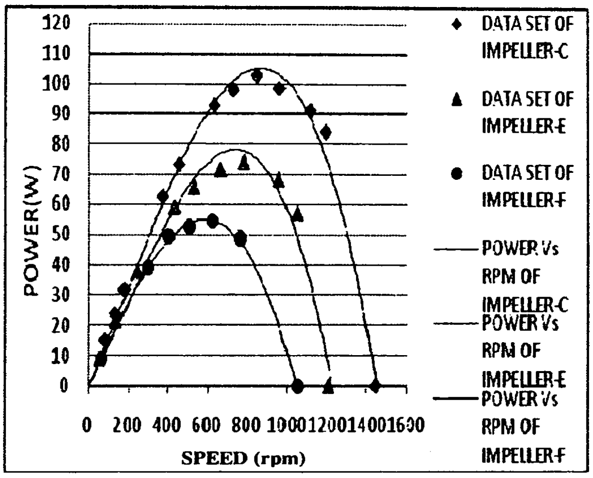
Graph 3.1: Power vs. rpm at Flow rate= 4.125 l/s



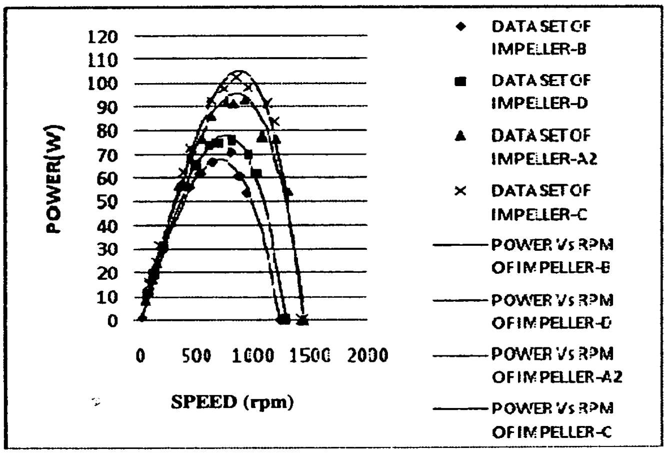
Graph 3.2: Power vs. rpm of Impeller-A2 and Existing Impeller



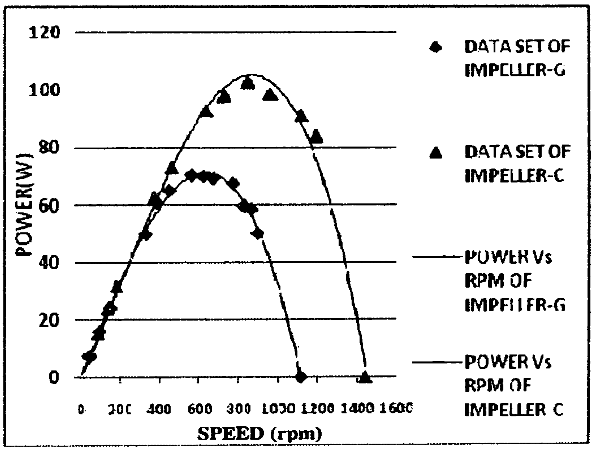
Graph 4.1: Power vs. rpm of Existing Impeller, Impeller-A Impeller-A1, Impeller-A2, Impeller-A3 and impeller-A4



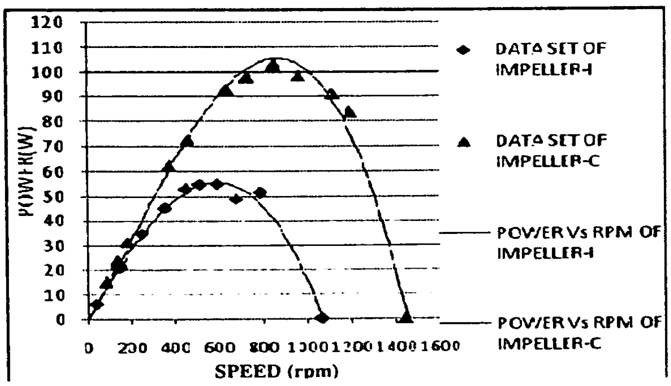
Graph 4.3: Power vs. rpm of Impeller–F, Impeller–E and Impeller-C



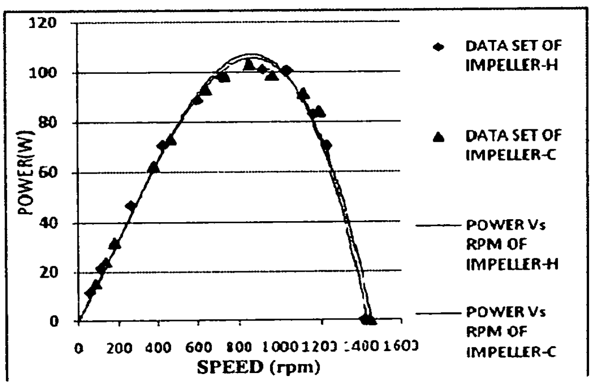
Graph 4.2: Power vs. rpm of Impeller-D, Impeller-B and Impeller-C



Graph 4.4: Power vs. rpm of Impeller-G and Impeller-C and Impeller-A2



Graph 4.5: Power vs. rpm of Impeller-I and Impeller-C



Graph 4.6: Power vs. rpm of Impeller-H and Impeller-C

Conclusion

Changing the no of vanes

In this section several notations are used for convenience. Those are,

P_{max}= Maximum mechanical power of the pump shaft

N_r = Rotational speed of the pump shaft related to maximum mechanical power of the pump shaft

η_{max} = Efficiency of the pump as turbine unit

Type1 impeller = inlet vane angle is smaller than the outlet vane angle

Type 2 impeller = inlet vane angle is larger than the outlet vane angle.

According to the Graph 4.1, some conclusions can be given which are depending on testing conditions. Those are,

- When the no of vanes are being increased, P_{max} and N_r increased. But from 4 vane impeller, increment of the P_{max} for vane 5 and vane 6 are small.
- When the no of vanes are being increased, η_{max} also increased. But the increment of the η_{max} is small.

Changing the vane angle

To check the effect of the vane angle, no of vane 4 for all different vane angle impellers were used. So according to the Graph 4.2, some conclusions can be given which are depending on testing conditions. Those are,

- P_{max}, N_r and η_{max} for the Type1 impeller, is smaller than Type2 impeller
- When difference of the value of inlet vane angle and outlet vane angle is being increased for Type1 impeller, P_{max}, N_r and η_{max} decreased.
- When difference of the value of inlet vane angle and outlet vane angle is being increased for the Type2 impeller, P_{max}, N_r and η_{max} increased.

Changing the diameter of impeller

The effect on the P_{max}, N_r and η_{max} by changing the diameter is fluctuating. Therefore further research should be carried out with in this area. According to the Graph 4.3 it can be said that by decreasing the diameter, P_{max}, N_r and η_{max} reduce.

Closed type impeller

Considering the Graph 4.4, it can be said that P_{max}, N_r and η_{max} reduce when the closed type impeller is used.

Changing the vane thickness

Considering the Graph 4.5, it can be said that by decreasing the vane thickness P_{max}, N_r and η_{max} reduce.

Newly designed impeller

According to the Graph 4.6, P_{max}, N_r and η_{max} for both impellers are approximately same. This is happened due to difference of the inlet flow rate for each impeller (flow rate for impeller-H =3.7 l/s and impeller-C =4.167 l/s). Because of using the centrifugal pump for supply inlet water flow to the pump which we use as turbine, we can't keep the flow rate and pressure at constant for each impeller (water flow area for each impeller is differ). The efficiency of the newly developed impeller was nearly 20% and maximum output power of 106 W, running at 870 rpm at the flow rate of 3.7 l/s. If the flow rate can be increased up to 4 l/s, 22% efficiency and maximum output power of 130 W, running at 1000 rpm can be obtained. And further increment depends on the available flow rate.

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