MIXED-FLOW IRRIGATION PUMP DESIGN OPTIMIZATION FOR BANGLADESH

Yu, Su; Colton, Jonathan S.
Georgia Institute of Technology, United States of America

Abstract

Irrigation pumps are indispensable to the production of major crops in Southeast Asia. This study proposes a design method to optimize the design of a mixed-flow irrigation pump model in a cost-effective and energy-efficient manner through a case study in Bangladesh. This method deviates from standard ones in industrialized nations to account for resource limitations in developing countries. Based on this approach, the paper optimizes four design parameters using numerical simulation, computational fluid dynamics, and analysis of variance. The results of numerical simulations reveal that the pumping strength may be increased from 0.286 to 0.348 (22%), while the efficiency may be increased from 40.2% to 54.2% (14.2%) when operated at 1500 RPM. This study also ranks the effectiveness of each design parameter and their interactions. An optimized design is presented for the current model, and an optimization procedure is developed for all axial-flow and mixed-flow pumps.

Keywords: Numerical methods, Optimisation, Irrigation pumps, Developing countries, Simulation

Contact:
Su Yu
Georgia Institute of Technology
Mechanical Engineering
United States of America
syu61@gatech.edu

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1 INTRODUCTION

Globally, there are hundreds of different designs for irrigation pumps, each with its own strengths and limitations. Due to different applications and manufacturing conditions, irrigation pumps are constantly optimized to achieve the best performance. This research focuses on improving the design process for Thai mixed-flow pumps (MFP) in a cost-effective and energy-efficient manner for fabrication and use in Bangladesh.

Irrigation pumps are indispensable to the production of major crops. By supplying sufficient water for cultivation in non-rain-fed seasons, irrigation pumps increased annual harvests of rice from two (Aus and Aman) to three (and Boro) in Southeast Asia. This requires pumped irrigation between 11,700 m^3 ha^{-1} and 12,800 m^3 ha^{-1} per season (Sarker and Ali, 2010), which are difficult to acquire through manpower or other approaches. Hence, irrigation pumps, especially axial-flow pumps (AFP) and MFPs, are extremely important to most countries in this area, such as Thailand, Indonesia and Vietnam, where agriculture plays a dominant economic role.

AFPs and MFPs have been widely used in these regions since their introduction in 1960s (Chinsuwan, 1985). Advantages of these pumps include convenience, mobility, ease of manufacture, and low initial and continuing maintenance costs. In addition, AFPs and MFPs exhibit substantially higher pumping efficiency than centrifugal pumps at lift heights of 3 m or less (IRRI, 1983). As a result, AFPs and MFPs played pivotal roles in starting the "Silent Revolution" in monsoon Asia. The popularity of these two pumps continues to increase today (Biggs, 2011). In southwestern Bangladesh, 50% of the arable land can be intensified by these pumps (Schulthess et al., 2015). The government is thus providing a 25% subsidy to farmers to encourage their use (*Minor Irrigation Survey Report* 2012-13, June 2013).

The study of AFPs and MFPs for irrigation can be traced back to 1941, when the initial design was demonstrated (Chinsuwan, 1985). Two studies were performed by the International Rice Research Institute (IRRI) in the Philippines (IRRI, 1983) (Aban and IRRI, 1985). In 1983, the IRRI designed an AFP with an inner diameter of 15 cm. Two years later, an optimized design was developed with a higher efficiency. In 2014, an investigation on the Thai mixed-flow pump was conducted using numerical simulation (Kasantikul and Laksitanonta, 2014). The study attempted to optimize the energy efficiency of the original design based on results generated with computational fluid dynamics (CFD). At an impeller speed of 1000 RPM, the original had an efficiency of 56%, whereas the new design had an efficiency of 70%. Overall, by sacrificing 12.4% of flow rate, the new design increased the efficiency by 13.6% at a head increased by 12.9%. Due to similarities in the impeller model and approaches, this research is used as a reference for our parameter selection.

Despite this wealth of research, the design of irrigation pumps remains incomplete due to a lack of field studies, especially among resource-limited regions. In this paper, a design approach incorporating the concept of “local design for local needs” for Bangladesh is developed through a case study conducted by Georgia Tech, Bangladesh Agricultural Research Institute (BARI), USAID, and CIMMYT. Two optimized MFP impeller designs with the greatest pumping strength and efficiency are presented. This paper is organized as follows: Section 2 introduces the theories associated with pump classification and specific speed. Section 3 presents the procedure used to conduct this study. Section 4 analyses the results. Finally, this study is summarized in Section 5.

2 PUMP THEORIES

This section introduces the theories used in this research. A pump is a piece of turbomachinery that adds energy to a fluid. Pumps may be classified into two categories: dynamic and displacement. Based on the direction of the flow, dynamic pumps are further categorized into AFPs that generate flow in the axial direction, $\hat{z}$; centrifugal pumps (CP) that generate flow in the radial direction, $\hat{r}$; and MFPs that generate flow in both directions (Wilcox, 2000). A commonly used term to describe the application range of different types of pumps is specific speed, N_{sd}, (Equation 1)

$$N_{sd} = \frac{\Omega\sqrt{Q}}{(H)^{3/4}} \tag{1}$$

where Ω is the shaft speed (rpm), Q is the volumetric flow rate (gpm or m^3/h), and H is the pressure head (ft or m). Physically, the specific speed is the operating speed at which a pump produces unit head at unit volume flow rate. Low specific speeds ($N_{sd(US\ gpm,ft)}$ < 4000) typically correspond to the most efficient operation of centrifugal-flow pumps, moderate specific speeds typically correspond to design points for mixed-flow pumps, and high specific speeds ($N_{sd(US\ gpm,ft)}$ > 9000) correspond to the most efficient operation of axial-flow pumps. Based on this concept, AFPs are suited for applications with high flow rate and low head, CPs are suited for that with large head and small flow rate, and MFPs are the intermediate solution of the previous two circumstances.

3 PROCEDURE

This section discusses the procedure to optimize the impeller design of a Thai MFP, also known as Thai-made irrigation pump (TmIP), for the most pumping strength and efficiency. When optimized, the TmIP is tested virtually using CFD simulation. This CFD approach is supported by past research, and validated by the physical testing results from previous work (Yu, 2017). Compared to physical testing as shown in Figure 1, the CFD approach saves time, cost, and labor.

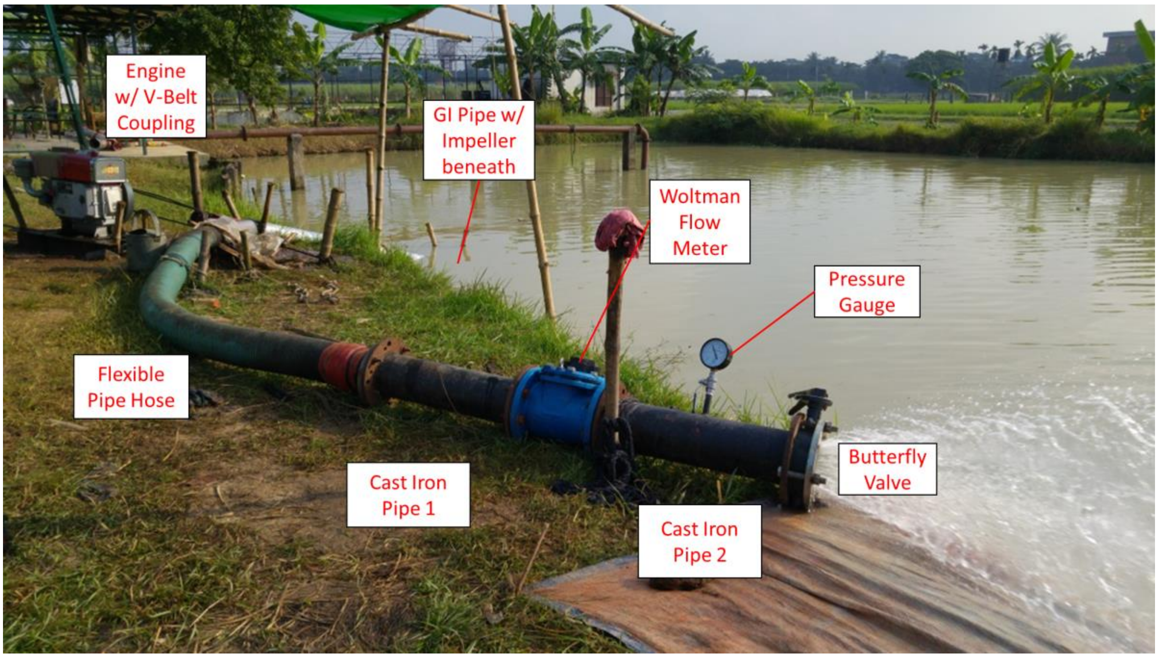

Figure 1. Physical testing of TmIP performed in Bangladesh

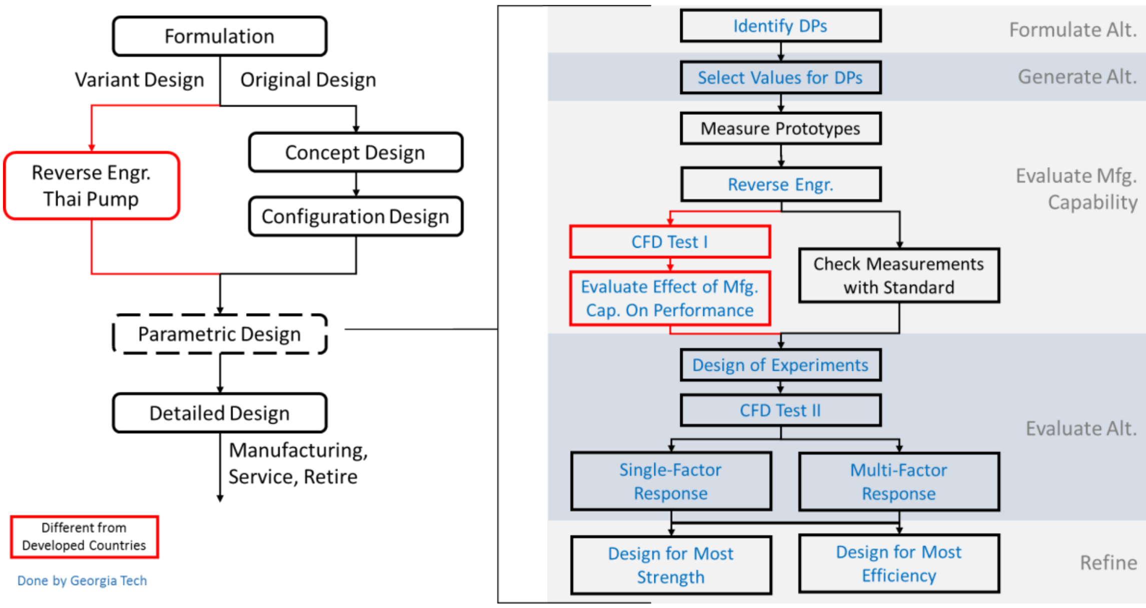

Figure 2. Flow chart describing the testing procedure

The design process is described in Figure 2. A complete engineering original design involves five phases – formulation, concept design, configuration design, parametric design, and detailed design (Eggert, 2010). Unlike this design method, our Bangladesh pump design process is simplified to a variant design process by reverse engineering the TmIP to reduce design expenses.
This paper focuses on the parametric design phase shown on the right, which determines values for controllable parameters to obtain an optimal performance. This phase can be classified into five stages: formulate alternatives, generate alternatives, evaluate manufacturing capability, evaluate alternatives, and refine. Compared to the parametric design process for developed industrialized countries, our design process is unique in the third stages by evaluating the effect of manufacturing capability on performance with CFD tests as opposed to the standard, industrialized country approach that checks geometric measurements against engineering drawings and industrial standards (Eggert, 2010). We adopt this approach because the manufacturing capability in many developing countries is that in industrialized countries, as evidenced during a visit to Bangladesh to observe the state of the art of local foundries and machine shops. In this section, the procedure of reverse engineering, CFD setup, and design optimization are presented.

3.1 Reverse Engineering of Impellers

Reverse-engineering the manufactured impeller samples into CAD models is the first step. Six impeller prototypes manufactured in Bangladesh specifically for this study are adopted as the physical models. Eight of their design parameters are measured to describe the geometry, with values listed in Table 1. Using these measurements, a SolidWorks CAD model of the impeller is recreated. The actual impeller and the CAD model are shown together in Figure 3. As shown by Table 1, the measured standard deviation obviously exceeds the industrial standard of 0.4-50 µm, which emphasizes the necessity to evaluate manufacturing capability on impellers' performance (Braddock).

Figure 3. Impeller sample (left) and CAD model (right) of TmIP

Table 1. Design parameter measurements for sample TmIP in mm

	Vane Width	Cone Height	Impeller Height	Distance between Top of Vanes	Distance between Edges of Vane	Hole Diameter	Cone Diameter	Outer Diameter
Ave.	57.15	58.17	87.50	52.78	132.47	32.45	126.60	178.90
Std.	5.905	1.700	1.863	1.893	3.386	2.928	0.901	1.181

3.2 CFD Setup

Next, CFD models are generated in SolidWorks Flow Simulation, a CFD tool which performs fluid flow simulations on SolidWorks CAD models. The CFD model is constructed in several stages, including geometry modelling, mesh generation, and problem conditions.

3.2.1 Geometry Modelling

In the initial stage, the model is created by placing the impeller in a pipe segment, which is represented by a thin cylindrical shell with an inner diameter of 187 mm and a length of 270 mm. This is shown in Figure 4. An optimum clearance of 4 mm between the outer diameter of the impeller and the inner wall of the pipe is used, as suggested by IRRI (Aban and IRRI, 1985). In this case, the pipe and the impeller are defined to be solid, whereas all other regions within the pipe are defined as water. As required by the software, two additional lids of small thickness are generated at the inlet and outlet in order to seal the fluid region. Finally, a rotating region, as shown by the lower half of the fluid region, encloses the impeller is defined to rotate about the negative y-axis at a defined speed. When the system operates, the impeller is expected to move the water upwards in the figure.

3.2.2 Mesh Generation

In the second stage, the system generates mesh for the model by dividing the region of interest into small cells. The mesh is generated only in the fluid region inside the pipe, but outside of the impeller using structured cubic cells of different sizes. The mesh becomes more refined in the rotating region, where the geometry is more complex due to the impeller it contained. A three-dimensional view of the mesh profile is also shown in Figure 4. The mesh profile is also discussed in Section 4.1.

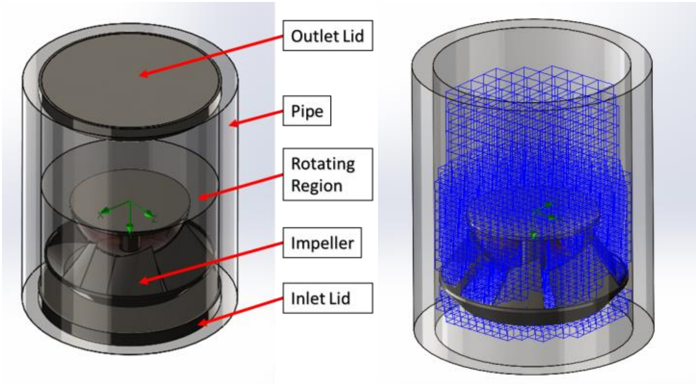

Figure 4. Geometry modelling (left) and mesh profile (right) of CFD model

3.2.3 Problem Conditions

In the third stage, the conditions are defined as a pumped internal pipe flow problem. It assumes no cavitation and pure liquid water as the only fluid. Based on the hydraulic diameter (pipe diameter), D_H, of 0.19 m and a reference flow velocity, *V*, of 2 m/s, the Reynolds number is calculated to be 3.9E+5. As the Reynolds Number is well beyond 4000, the software uses a modified k-ε two-equation turbulence model. The model defines a low turbulent intensity of 2%, and a turbulence length of 0.072 m, which is 3.8% of hydraulic diameter. The model also neglects any heat transfer in the system. Several boundary conditions are defined to describe the operation. As commonly practiced, the fluid in the rotating region and the pipe segment are defined to rotate about the negative y-axis at a constant speed, while keeping the impeller stationary. Because the rotation is relative between water and impeller, this approach is adopted to reduce the number of moving objects. The inlet lid is defined as an opening with an environmental pressure of 1 atm. When measuring the shut-off head, the outlet lid is defined as a wall with a smooth surface that rotates at the same angular velocity as the pipe wall; when measuring the maximum discharge rate, the outlet lid is defined as an opening like the inlet lid. All boundaries are assumed to be adiabatic.

Some initial conditions are defined as well. Fluid in all regions is defined initially at a pressure of 1 atm and room temperature. Fluid outside the rotating region is defined with an initial velocity of zero, whereas the fluid inside the rotating region is defined previously.

With the above conditions determined, the problem is then solved for a steady state solution until an analysis interval of 0.5 travels, which is the interval of past iterations over which the goals' convergence criteria are checked. Simulation parameters are presented in Table 2.

Table 2. Highlights of CFD simulation setup

Software	SolidWorks Flow Simulation
Problem type	Pumped internal pipe flow
Grid	Structured
Inlet	Opening at 1 atm
Outlet	Adiabatic wall / Opening at 1 atm
Other boundaries	Adiabatic wall
Turbulence model	Modified k-ε two-equation model
Convergence criteria	Analysis interval of 0.5 travels

3.3 Design Optimization

Next, the pumping power and efficiency of the current TmIP impeller are optimized using the CFD model. During optimization, four design parameters are studied to optimize the impeller's performance - cone height, cone diameter, overall height, and vane angle. The first three parameters are defined in Figure 5. The final parameter, the vane angle, is the angle of the taper helix used to define each blade for the impeller, assuming each blade has 0.1275 revolution, and a rise in height of 30 mm.

These four design parameters directly affect the flow profile, with their main contributions summarized as follows. Cone diameter determines the highest relative velocity of the liquid exiting the impeller. Cone height and cone diameter together determine the flow direction and potential cavitation. Vane angle and overall height affect the flow direction as well as the resistance encountered during rotation. With the effect of above four parameters analysed, the study proceeds to a grid independence study to determine an appropriate grid size prior to optimization.

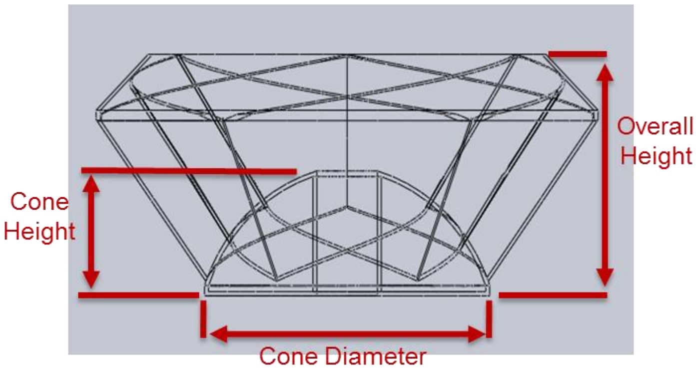

Figure 5. Definition of design parameters

During optimization, design of experiments is used to investigate the single-factor effect of each individual design parameter. The study captures the impeller performance of an individual parameter at various levels. It begins with creating impeller CAD models at 4-9 different levels above and below each design parameter in the original model, with a difference of 10% at each level. Then, the study runs these CAD models in SolidWorks Flow Simulation and collects their performance from CFD simulation. The operating shaft speed is set to 1500 RPM, typically used by Bangladeshi farmers. This approach allows the researcher to perform a detailed analysis of the behaviour with respect to this parameter, and thus to find its optimum value at the given operating condition. The design of experiments also establishes a set of full factorial experiments at two levels in order to study the interaction between the first three design parameters (Wu and Hamada, 2000). In this study, three parameters – cone height, cone diameter, and overall height – are set to “high” and “low” levels, with values determined from measured averages shown in Table 3. A total of eight CAD models are tested, and thus analysed for the behaviour of interactions. A model matrix is included as part of Table 5 by abbreviating the three design parameters Cone Height (A), Cone Diameter (B), Overall Height (C) and their High (+) and Low (-) values. As the number of experiments increases exponentially with factors, vane angle is not included in this discussion due to limited computing power.

Table 3. Values of Design Parameters at Two-Level Full Factorial Experiments

Design Parameters (mm)	Low	High
Cone Height (A)	52.4	58.2
Cone Diameter (B)	126.6	139.3
Overall Height (C)	87. 5	96. 3

The experiments use two performance parameters, π_1 and η, to evaluate the pumping strength and the efficiency of the tested impeller model, respectively. Performance parameter π_1 is a function of H and Q. It quantifies the pumping strength of the impeller, and is defined in Equation (2) (Aban and IRRI, 1985).

$$\pi_1 = \frac{Q}{(gH)^{\frac{1}{2}} D_H^2} \qquad (2)$$

Performance parameter η quantifies the mechanical efficiency of the impeller, which may be defined as the ratio between water power (*w.p.*) and brake power (*b.p.*) as shown in Equations (3) - (5) (Lam, et al., 2015),

$$\eta = \frac{w.p.}{b.p.} \qquad (3)$$

where

$$w.p. = \rho * Q * \frac{v^2}{2} \qquad (4)$$

$$b.p. = T * N \qquad (5)$$

where ρ is water density, v is the average flow velocity at the outlet, T is the torque on the shaft in the axial direction, and N is the rotational speed. Both parameters are essential in evaluating the overall effectiveness of the impeller design.

4 RESULTS AND ANALYSIS

This section analyses results from numerical simulations. In order to validate the accuracy and reliability of the CFD model, this section first presents the grid independence study. Then, it analyses single-factor and multi-factor responses, and introduces the optimized designs.

4.1 Grid Independence Study

A grid independence study is performed to validate the accuracy and reliability of the computed results using the model setup in Section 3.2. This is done by comparing the pressure at closed valve conditions at multiple different grid sizes. When the change in pressure becomes insignificant, grid independence is achieved. In this grid independence study, pump system at four different grid sizes are solved. The results of this study are presented in Table 5, and graphs are generated in Figure 6.

Table 4. Grid Independence Study

Cell size /m	Shut-Off Pressure /Pa	Shut-Off Head /m	Error/Pressure
0.019	169592	6.97	
0.016	163106	6.30	4.0%
0.01	159414	5.93	2.3%
0.007	159767	5.96	0.2%

In Table 4, as the minimum cell size decreases from 0.019 m to 0.007 m, the change in shut-off head reduces, and the error drops from 4% to 0.2% of the pressure. Based on this trend, it is expected that the error from simulation calculated at the next level will be smaller than 0.2%, or approximately 0.03 m in shut-off head. This will not result in significant differences in future analysis. Thus, the results may now be concluded as grid independent at a minimum cell size of 0.007 m, which contains a total of 2285 cells with their volumes ranging between 8.8E-7 and 7.0E-6 m³.

4.2 Manufacturing Capability Evaluation

The evaluation of the effect of manufacturing capability on performance within the set of six impellers are shown in Table 5. Because the friction losses of other components within the system are not modelled, the volumetric flow rate is not calculated. Therefore, the simulation uses shut-off head to quantify the performance. From Table 5, the standard deviation is 4.4% of the average values. This demonstrates a high consistency of pump performance within manufacturing tolerances.

Table 5. Evaluation of Effect of Manufacturing Capability on Performance at 1500 RPM

Impeller						Average	Standard Deviation	STD/AVE
1	2	3	4	5	6			
6.06	6.76	6.89	6.45	6.65	6.52	6.55	0.29	4.4%

4.3 Single-Factor Response

The analysis starts with single-factor optimization. For each design parameter, the analysis attempts to fit the experimental data using linear regression to determine the behaviour of π_1 and η. Depending on the distribution of data points, a second order polynomial is fitted when a vertex is expected from the distribution, otherwise a first order polynomial is fitted. The results are plotted in Figure 6.

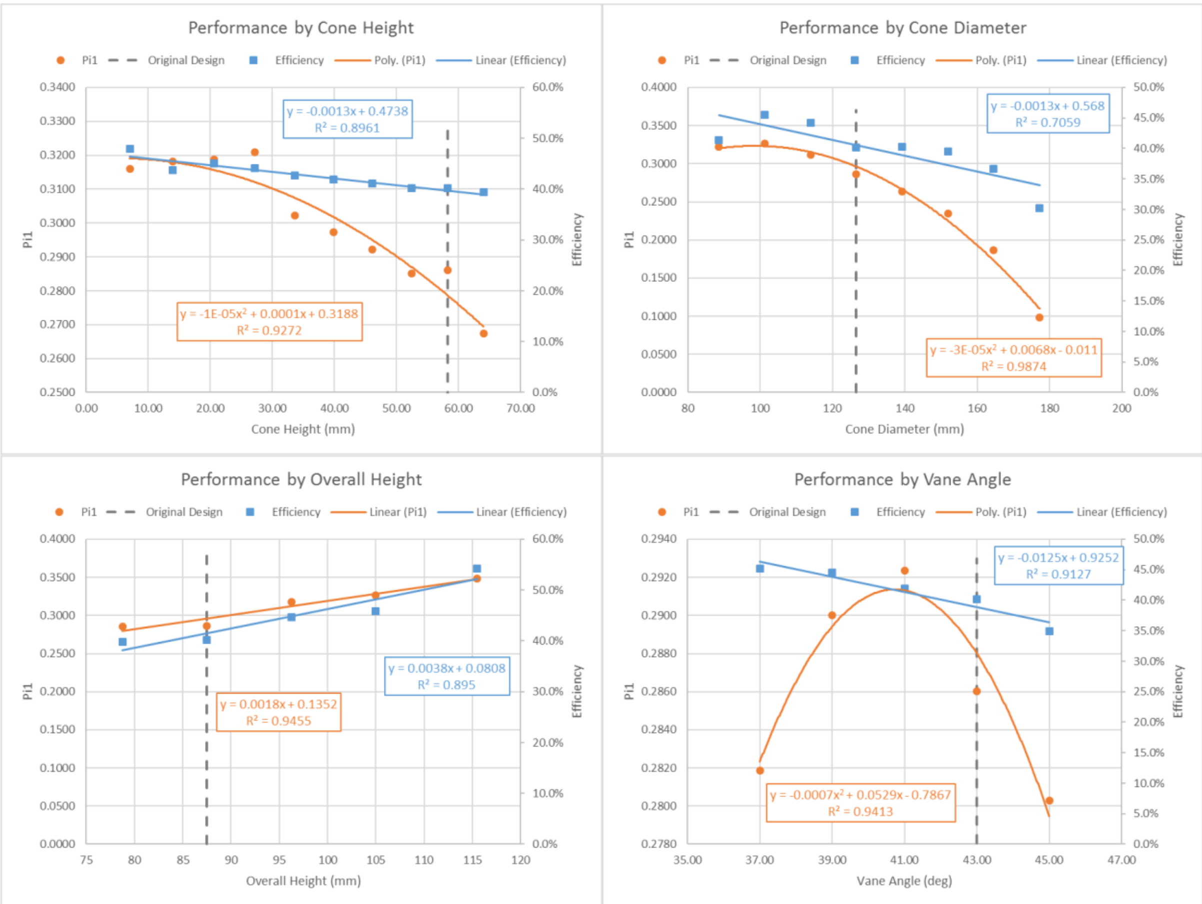

Figure 6. Single-factor response at 1500 RPM

Figure 6 demonstrates the behaviours of π_1 and η with respect to each design parameter. The behaviour of π_1 is coloured in orange, while that of η is coloured in blue. From Figure 6, the optimal TmIP should have a cone height of 27 mm, a cone diameter of 101 mm, a vane angle of 41 degrees, and maximized overall height (116 mm) for the greatest pumping strength at 1500 RPM. For the highest efficiency, the impeller design should minimize cone height (7 mm), cone diameter (89 mm) and vane angle (37°), and maximize overall height (116 mm) at 1500 RPM. After optimization, the π_1 of the original design may be increased from 0.286 to a maximum of 0.348, while the efficiency η may be increased from 40.2% to a maximum of 54.2%. The single-factor response investigates the behaviour of each design parameter by itself. Comparisons across different design parameters, nevertheless, are not yet included. These comparisons along with the effect of two-way interaction are covered in the multi-factor response analysis.

4.4 Multi-Factor Response

A set of full factorial experiments at two levels is conducted to determine the multi-factor response. The results are shown in Table 6.

Table 6. Multi-Factor Response at 1500 RPM

Case No.	A	B	C	AB	AC	BC	π_1	η
1	–	–	–	+	+	–	0. 2851	40. 2%
2	–	–	+	+	–	+	0. 3073	41. 1%
3	–	+	–	–	–	+	0. 2588	39. 3%
4	–	+	+	–	+	–	0. 2619	39. 3%
5	+	–	–	–	+	–	0. 2806	38. 3%
6	+	–	+	–	–	+	0. 3177	44. 6%
7	+	+	–	+	–	+	0. 2635	40. 2%
8	+	+	+	+	+	–	0. 2659	39. 9%

Analysis of Variance (ANOVA) is performed using Matlab for π_1 and η to analyse the results. In this case, the null hypothesis assumes that π_1 or η is independent of each design parameter, while the alternative hypothesis suggests π_1 or η is affected by these design parameters. The analysis is performed at a confidence level $\alpha = 0.10$. Depending on the effect of interaction terms, a Type II or Type III sum of square is used.

Table 7. ANOVA Results of π_1 and η

	Source	Sum of Sq.	d.f.	Mean Sq.	F	p
π_1	A	0.00003	1	0.00003	0.88	0.5211
	B	0.00247	1	0.00247	81.23	0.0703
	C	0.00052	1	0.00052	17.25	0.1504
	AB	0.00000	1	0.00000	0.03	0.8869
	AC	0.00003	1	0.00003	0.83	0.5290
	BC	0.00036	1	0.00036	11.89	0.1797
	Error	0.00003	1	0.00003		
	Total	0.00344	7			
η	A	1.2012	1	1.2012	0.30	0.6829
	B	3.7812	1	3.812	0.93	0.5114
	C	5.9513	1	5.9513	1.47	0.4396
	AB	0.0013	1	0.0013	0.00	0.9888
	AC	3.2513	1	3.2513	0.80	0.5353
	BC	7.0312	1	7.0312	1.73	0.4173
	Error	4.0613	1	4.0613		
	Total	25.2787	7			

Table 7 shows the ANOVA with respect to π_1 and η. The last column gives the p-value associated with each term. The smaller its p-value, the more significant the parameter is. In the case of π_1, the main effect of cone diameter has a p-value 0. 0703, which is less than $\alpha = 0.10$. Thus, the main effect of cone diameter is the only significant effect with respect to π_1 at a confidence interval of 90%. At some lower confidence intervals, such as 80%, the main effect of the overall height and interaction effect between cone diameter and overall height, with p-values of 0.1504 and 0.1797, may also become significant. π_1 is independent of the other design parameter and interactions. Because interactions are not significant, a Type II sum of squares is used for calculation.

Similarly, in the case of η, none of the effects has a p-value less than $\alpha = 0.10$. Thus, there is not enough evidence to reject the null hypothesis. In other words, all design parameters as well as their interactions at the 90% confidence interval do not have significant effect on mechanical efficiency of the impeller. Because interactions are once again insignificant, Type II sum of squares is used for efficiency calculations.

Based on this analysis, the effectiveness of each design parameter and their interactions may be ranked, and optimized designs can be determined. For example, when the design parameters discussed in Section 4.2 cannot be optimized simultaneously, one should prioritize cone height, overall height, and their interaction effect for the greatest power. Following the same principal, one should prioritize the interaction between cone diameter and overall height, overall height, and cone diameter for the highest efficiency.

5 CONCLUSION

This study proposes a numerical testing and optimization approach for the design of AFPs and MFPs, which deviates from standard approaches in industrialized nations by accounting for resource limitations in developing countries, such as Bangladesh. The proposed design process, which incorporates local manufacturing capability, improves the current design of a TmIP in terms of pumping strength and efficiency through CFD simulation. The results reveal that the pumping strength parameter π_1 may be optimized from 0.286 to 0.348 (22%), while the efficiency parameter η may be optimized from 40.2% to 54.2% (14%) when operated at 1500 RPM. The effectiveness of each design parameter and their interaction based on ANOVA of multi-factor responses are ranked.

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