

Article

Numerical Investigation and Design Optimization of Centrifugal Water Pump with Splitter Blades Using Response Surface Method

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Abstract: Centrifugal pumps are known to efficiently transport water from a certain point. However, they developed great concerns in water supply and distribution applications regarding their operating efficiency, which were caused by the accumulated losses and sudden power consumption growth. Thus, mitigating these concerns is important to improve the performance of the centrifugal pump. This study used ANSYS 2022 R2 for the optimization design process, combining the strengths of Computational Fluid Dynamics (CFD) and Response Surface Method (RSM), to come up with an optimal design for a centrifugal water pump. Splitter blades, with a length of 80% of the main blade, were included in the design to assess their effects on the performance of the pump. Design parameters such as the placement of the splitter blades, their ellipse ratios, and the volute tongue, were also investigated for further improvement. Results indicate that finding a perfect balance between the placement of the splitter blades, the design of the volute tongue clearance and thickness, and configuring the ellipse ratio of the splitter blades improves the pump's performance. The optimal design results in 27.35%, 15.70%, 28.18%, 16.67%, and 8.36% improvement in total efficiency, total head, static efficiency, static head, and power consumption, respectively.

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Keywords: centrifugal water pump; splitter blades; CFD; optimization; RSM

1. Introduction

A centrifugal pump has proven itself to become an efficient piece of equipment often used in the everyday activities of an industry that involves the production of industrial goods and other necessities [1]. It is characterized by its simple design but provides high performance and pressure, which can be effectively used for the distribution and transportation of fluids. However, centrifugal pumps often consume vast amounts of energy in industrial applications to ensure the system operates smoothly [2]. This significantly affects global energy consumption; hence, many researchers have focused on providing solutions to improve pump efficiency to reduce energy consumption.

Given that the industry relies on using centrifugal pumps through various applications, the energy loss that comes with its actual production and operations is being ignored by people in the industry. Ignoring these details cause damage to the centrifugal pump and cause severe concerns regarding its production capacity, leading to more serious economic problems. It is crucial to study the adjustment necessary to maintain the energy consumption of the centrifugal pump to a minimum [3]. In finding the best adjustment

method, the areas of concern, such as unstable operation, lower operating efficiency, and energy waste, can fundamentally solve the massive problems of parametric losses in centrifugal pumps.

The trend for global energy consumption is expected to increase from 2016 to 2030 by 30%. The continuous increase in energy demand for industrial applications has been shown to contribute to the global energy consumption percentage increase. Based on the record, the pumps account for nearly 22% of the energy power supplied by electric motors in the world, whereas centrifugal pumps alone consume 16% of the total energy consumption for pumps, which is shown to be higher compared to other types of rotodynamic pumps [4]. Given this percentage of energy consumption in centrifugal pumps, the concern must be addressed to overcome the energy deficit and develop a feasible solution to identify energy-saving potential. It is known that centrifugal pumps consume excessive power due to accumulated losses that minimize the fluid flow and gradually affect the pump's performance. The cause of inefficiencies in the performance of the pump must be investigated to find optimal solutions, contributing to the development of a comprehensive design that will address the concern [5].

Existing research studies highlight the significance of modifying the impeller geometry to reduce the accumulated losses in the pump's performance. Redesigning the impeller and changing the parameters of the blade profile improves the pump's hydraulic performance, leading to a more suitable design and adequate fluid flow. Simultaneously changing and altering the design of the impeller promotes a more uniform flow of fluid along the impeller passage [6,7]. Also, incorporating splitter blades in the design effectively improves the impeller outflow's stability and uniformity, reducing the inlet pressure pulsation. In general, a proper design of splitter blades helps decrease the clogging at the inlet of the impeller [8]. The result from a previous study concludes that an impeller with splitter blades improved the pump head by 8.2% while improving the overall efficiency by 3% [9]. This finding shows the relevance of considering the addition of splitter blades in the design of the impeller. However, blade regions, such as the trailing edge (TE) and leading edge (LE), should also be considered in the design as they affect the fluid flow within the pump. The TE profile of the blade is crucial for pressure pulsation, while the LE blade profile is vital for local flow separation and pressure distribution [10,11]. The impeller and the volute are the major components of a centrifugal pump. Hence, conducting a minor configuration on the volute should also be considered to have an efficient impeller–volute interaction, improving the hydraulic efficiency. A study observed that the spacing between the impeller outlet diameter and the location of the volute tongue causes a significant effect on the flow dynamics of the centrifugal pump [12]. Thus, minor changes in the volute tongue should also be investigated.

The CFD approach is one of the best and most efficient ways of designing a centrifugal pump. Present studies involving centrifugal pump design heavily rely upon the use of CFD for their respective design proposals as it is more efficient to use in a highly complex design having three-dimensional (3D) geometry models to use for simulating the behavioral flow and performance of the pump. Analyzing the flow of the centrifugal pump design is often a challenging task as it requires a critical understanding of a highly complex flow that is 3D in nature [13]. The Reynolds-Averaged Navier–Stokes (RANS) equation is commonly applied in CFD for modeling turbulent flows as this equation can predict the flow characteristics of fluid. The RANS equation is the deduced form of the Navier–Stokes (NS) equation. It aims to simplify the complexity of solving the NS equation by focusing on the mean flow and using empirical models to handle turbulence effects. The difference between the two equations is that the steady-state solution is removed from the time-varying fluctuations in the RANS method. Thus, the mean flow of velocity and turbulence are separated. In this

way, the numerical computation can easily be applied while efficiently determining the behavioral flow of the fluid. Equations (1) and (2) show the 3D, steady, and incompressible RANS equation [14].

$$\frac{\partial u_i}{\partial x_i} = 0, \quad (1)$$

$$u_j \frac{\partial u_i}{\partial x_j} = -\frac{1}{\rho} \left(\frac{\partial p}{\partial x_i} \right) + \nu_t \frac{\partial^2 u_i}{\partial x_j \partial x_j} + \frac{1}{\rho} \left(\frac{\partial (-\rho \overline{u'_i u'_j})}{\partial x_j} \right) \quad (2)$$

The RANS equation applies the continuity principle. It is widely used in fluid dynamics to determine the motion of fluid substances. The purpose of using the RANS method is to determine the number of turbulent fluctuations of the fluid by averaging the total Reynolds stress tensors. These Reynolds stress tensors are additional stresses or fluxes of momentum due to the unsteady turbulent motions. Through this equation the flow of the fluid can be numerically observed [15].

The RSM uses the data input defined in the Design of Experiments (DOE) to determine the cause-and-effect relationship between the known input and output parameters to attain an optimized response. The use of the RSM has an advantage in gaining the finest results through an array of exclusive experiments [16]. The visual outputs of the response surface are typically represented through a curved response surface diagram and contour plots. This representation contributes to a better interpretation of the response behavior to determine the conditions around the optimum response [17]. The Goodness of Fit (GOF) chart is also determined through the response surface method to measure the quality of the response model from the observed data. This criterion shows the proportionality degree or commonality between the variables obtained from the design points solutions defined in the DOE test environment and the verification points generated through the response surface. This aspect is crucial in the optimization process to determine the behavior of the input variables to their corresponding output response.

Integrating CFD with the RSM efficiently optimizes design. This approach is convenient for designing parameters with different variables to visualize the simulation trend in one diagram. Researchers have recently relied on using the RSM, which has progressively been applied to find an optimized design for centrifugal pumps. The previous study employed the RSM in their Pareto-based multi-objective optimization of centrifugal pump design to investigate the trend of the pump performance results of their research [18]. The RSM efficiently represented the system's response as a function of one or more factors. The relationship of the parameters was displayed using RSM graphical techniques, allowing the researchers to identify the optimization area conveniently. Pareto optimal is a set of solutions that are classified as non-dominated solutions, which are regarded as the practical solutions to be used in multi-objective optimization scenarios [19]. The application of the Pareto frontier in the multi-objective genetic algorithm in cultivating an efficient result investigation has shown to be a practical approach to deriving the best solution and the preferred point in an optimization model [20].

Achieving an optimal pump operation has been challenging in terms of minimizing energy costs and ensuring substantial head availability because the water supply and distribution system are heavy consumers of electrical energy [21]. Thus, the primary concern of this research study is to address the issues about the application of centrifugal pumps in water supply and distribution systems. In particular, this study aims to investigate the effects of modifying the impeller and the volute. Modifications in the impeller will incorporate splitter blades, varying their placement within the impeller and its TE and LE blade profiles, specifically at their corresponding ellipse ratios at the hub and shroud. On the other hand, minor modifications will be conducted in the volute, specifically the volute

tongue thickness and clearance, to promote efficient impeller–volute interaction with the design. Since investigations from previous studies had consistently focused on redesigning the impeller and changing the number of blades, considering these additional factors offers a more thorough approach to developing an optimal centrifugal pump.

The centrifugal pump's eminent suitability in the water supply and distribution industry accounts for an increase in excessive energy consumption. Thus, the significance of this study is to find a remedy for reducing the power consumption required to operate a centrifugal pump, especially in the continuous operation of the unit. In this way, people in the industry can save vast amounts of money on energy costs. The RSM will be utilized in the methodology of the study to determine how the changes in input parameters affect the corresponding output results, which can be visualized in a 3D graph that can help interpret and understand the generated results of the study, leading to an optimized design. This also helps point out what design parameters must be considered or avoided to maintain the pump's performance at its best. Through this initiative, an optimized centrifugal pump design will also help decrease the power demand of the industrial sector that heavily relies on centrifugal pumps. This will also help address a small portion of the country's electricity crisis. Additionally, an optimized design of centrifugal pumps will lead to efficient equipment performance, reducing the cost of maintaining or changing the whole pump unit. Hence, the consumers and operators will be able to maximize the usable life of the pump and achieve desirable operational effectiveness.

The design configurations of this study highlight its use for water supply and distribution applications. The fluid nature considered in making the design was only intended for centrifugal pumps that carry water. With this, the proposed design may only be limited to centrifugal pumps for water supply and distribution applications and may not apply to other pump types and usage. The study's results will depend on the simulations made in ANSYS 2022 R2, which will focus on improving the flow of the fluid within the centrifugal pump by determining the pressure of the fluid within the pump and the pressure induced in the meridional surface of the impeller blade along with the determination of the amount of turbulence present in the flow of the fluid. The CFD analysis that will be conducted will be limited to the normal condition of the water at one atmospheric pressure while neglecting the temperature changes in the water. Thus, the study will focus merely on the analysis of the flow of the fluid from the inlet to the outlet of the pump. Although the study intends to enhance the performance capabilities of centrifugal pumps, the findings of this research may still change if implemented and tested on an actual operation and scenarios. Hence, further validations are recommended for future research.

In this study, the optimization process plays an important role in pointing out the necessary adjustments needed to be applied in the design to achieve an optimal pump performance. This study gathered information from various previous studies, as mentioned, to carefully select what parameters are to be considered in the optimization process. Hence, revisiting existing fluid models and analysis has led this study to consider configuring numerous design parameters and assessing their individual effects on the performance of the pump rather than focusing on one design parameter. Having numerous design parameters to be investigated creates a new approach to centrifugal pump modeling as it provides a clear overview of how certain changes made to the design affect the overall pump performance. Thus, using the RSM in this study is beneficial as it can select the most suitable parameter to be applied in the design. The study offers significant advancements over previous research as the design process involves both experimental and statistical approaches in developing an optimal solution for the study. Other than configuring the design of the impeller, this study also considered redesigning the volute for better impeller–volute interaction, which improves the hydraulic efficiency of the pump. Considering the

impeller–volute interaction in the design makes this study more significant since previous research focuses only on redesigning the impeller blades of the pump, neglecting the friction losses between the impeller and volute.

2. Methodology

The research methodology highlights two processes that will be conducted in the study. The initial process will be based on the implementation of CFD, followed by the optimization process of the design. These processes will allow us to examine the effect of varying parameters in a centrifugal pump and compare the results and performance of the baseline and optimized design models. Both the CFD and optimization processes will be conducted using ANSYS 2022 R2. Figure 1 provides a summary of the whole research methodology.

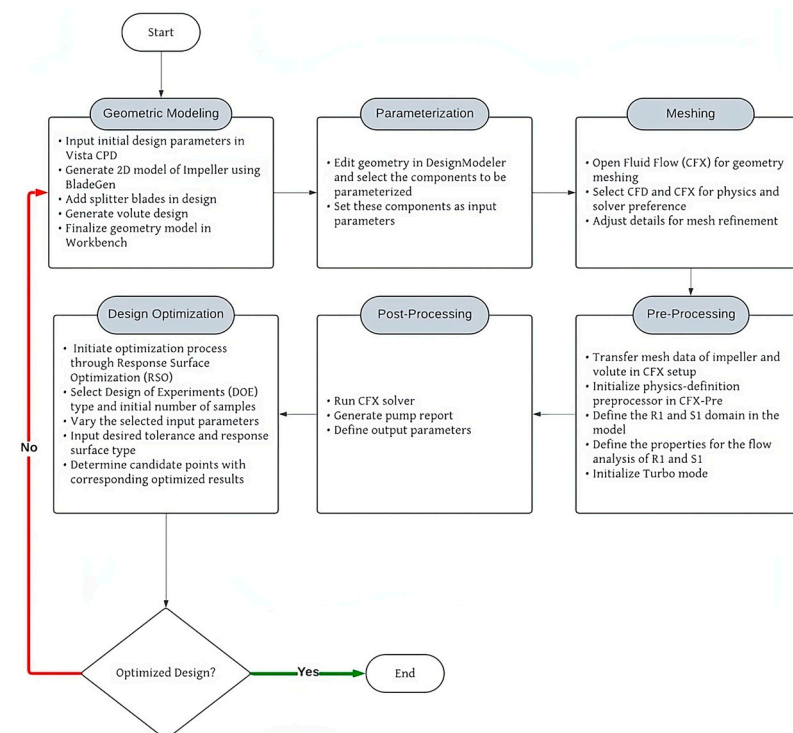


Figure 1. The methodological framework of the study.

2.1. CFD Simulation Process in Centrifugal Pump Design

This section will discuss the CFD simulation process for designing a centrifugal pump. It will also give insight into how CFD was implemented in the design process, highlighting every step involved to have a complete overview of the procedures to be conducted in the study. The creation of geometry, which refers to creating 3D models of the pump’s impeller and volute, will be the first to be discussed. This will be followed by parameterizing the pump components before proceeding with the geometry meshing. On the other hand, the pre-processing stage follows, which refers to finalizing the setup of the centrifugal pump before initializing the CFX-solver. Lastly, post-processing will also be presented to discuss the initial results of the baseline design of the centrifugal pump.

2.1.1. Geometric Modeling

The geometry of the pump’s impeller and volute can be generated using ANSYS Vista Centrifugal Pump Design (CPD) by defining the initial parameters of the centrifugal pump. This study utilized the Vista CPD design parameters presented by Alawadhi et al. [22] in their study. These parameters will serve as reference throughout the process of pump

geometric modeling up to the solver setup before the simulation commences. However, a few alterations will be implemented for the mentioned reference, which includes changing the number of vanes, as this study includes splitter blades in its design. The reference which corresponds with the number of vanes was based on the study conducted by Gölcü et al. [23]. Though the first mentioned reference paper deals with a slurry classification of fluid, this study will focus on the effects of using these configurations in means of water as the selected type of fluid for the pump design. Moreover, the parameters used in this study are provided in Table 1:

Table 1. Vista CPD input parameters.

Input Parameter	Unit	Value
Rotational Speed	RPM	1600
Volume Flow Rate	m ³ /h	120
Density	kg/m ³	1000
Head Rise	m	20
Inlet Flow Angle	degrees	90
Merid Velocity Ratio	-	1.1
Number of Vanes	-	5

After finalizing the parameters in Table 1, these data will be used in ANSYS Vista CPD to generate the design specifications of the corresponding impeller and volute, which will be used in creating the geometry profile of the two after the program has successfully computed and analyzed the given input. The impeller and volute specifications are based on the computational assessment of the program, which can be seen in Tables 2 and 3. Also, the impeller export type setting was changed from isolated impeller to coupled to volute. This will ensure that the outlet radius of the impeller matches the inlet radius of the volute when exported from Vista CPD.

Table 2. Impeller specification.

Impeller Specification	Unit	Value
Shaft Diameter	mm	19.611
Hub Eye Diameter	mm	29.416
Eye Diameter	mm	125.75
Vane Thickness	mm	7.260
Tip Diameter	mm	242.01
Tip Width	mm	30.215
Number of Main Blade	-	5
Number of Splitter Blade	-	5

Table 3. Volute specification.

Volute Specification	Unit	Value
Inlet Width	mm	60.431
Base Circle Radius	mm	134.19
Tongue Clearance	mm	13.189
Tongue Thickness	mm	5.082
Diffuser Length	mm	151.79
Diffuser Cone Angle	degree	7
Diffuser Outlet Hydraulic Diameter	mm	91.868
Diffuser Outlet Height	mm	91.868

The geometry of the impeller can be extracted from ANSYS Vista CPD and transfer its data to BladeGen. The blade design extension of ANSYS BladeGen can be used to generate the 3D model of the pump impeller. The splitter blades were also added to the impeller in this process, as shown in Figure 2:

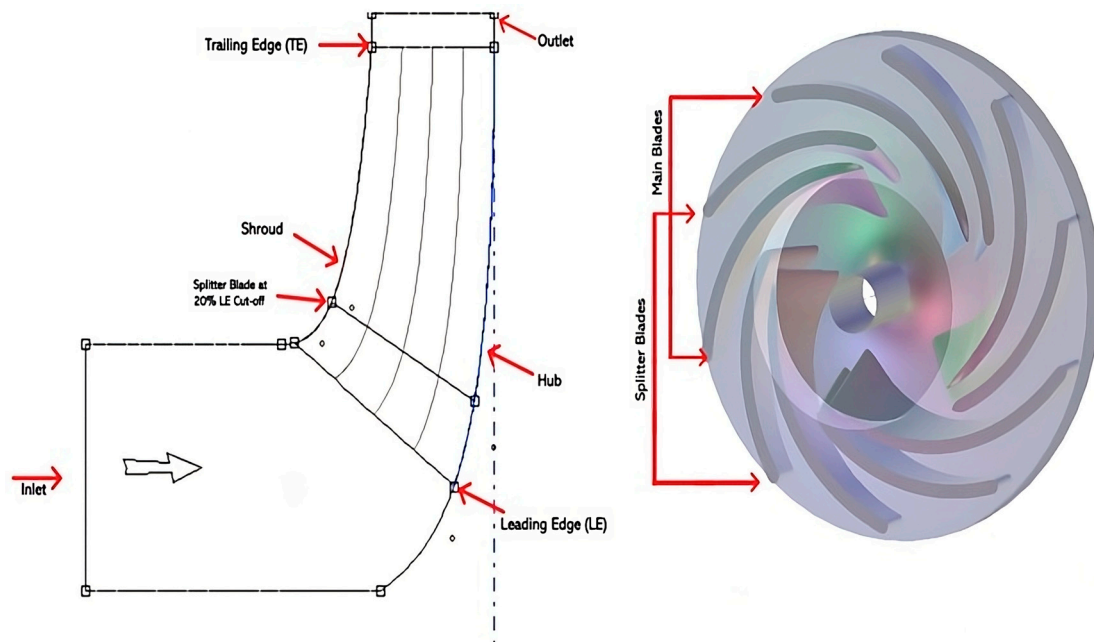


Figure 2. Blade profile and 3D model of the pump impeller.

The length of the splitter blades was reduced to 20% on the TE side, as shown in the above figure. This is in line with the suggested design of the study conducted by Gölcü et al., where the length of the splitter blades at 80% of the length of the main blades shows a preferable result [23]. Hence, the design suggestion was applied in this study. Additionally, the data from Vista CPD can be used to generate the geometry of the volute, which can be obtained through ANSYS DesignModeler. The generated geometry model of the volute is provided in Figure 3.



Figure 3. Three-dimensional model of the pump volute.

2.1.2. Parameterization

The TE ellipse ratio at hub (TE_{erh}), TE ellipse ratio at shroud (TE_{ers}), LE ellipse ratio at hub (LE_{erh}), LE ellipse ratio at shroud (LE_{ers}), along with the splitter blades' offset pitch fraction ($spltr_OPF$) or the placement of the splitter blades between the two main blades, are the components to be parameterized for the pump impeller. On the other hand, the components to be parameterized in the volute are the cutwater or tongue thickness (thk) and clearance ($clear$). The parameterization can be conducted through ANSYS DesignModeler, in which the software will define the selected components as the input parameters of the study. Figure 4 shows the location of the components to be parameterized in the design. In Figure 4a, the $spltr_OPF$ was set to an initial value of 0.5, which means the splitter

blades are located between the main blades by default. A value less than 0.5 refers to the placement of the splitter blade close to the pressure side of the main blade, while a value greater than 0.5 refers to the splitter blades' placement close to the main blade's suction side. Figure 4b,c shows the TE_{erh} , TE_{ers} , LE_{erh} , and LE_{ers} . These components will also be parameterized to determine their effect on the performance of the centrifugal pump, as an existing study shows that well-designed blade regions, such as an Ellipse on the Pressure Side (EPS) and Ellipse on Both Sides (EBS) profiles, contribute to improving the pump efficiency [24]. The modification of the EPS profile design was also observed to increase pump efficiency based on the study conducted by Zhang et al., which used an experimental approach with an impeller–volute combination [25]. Hence, configuring the ellipse ratio at the hub and shroud of TE and LE was also considered in this study. The $clear$ and thk were also parameterized in this study to improve the impeller–volute interaction further, leading to efficient hydraulic performance. This was based on Alemi et al.'s observation that the volute tongue significantly affects the absolute velocity in the impeller outflow [26]. Thus, redesigning the volute tongue was explicitly considered in the study of impeller–volute interaction. Figure 4d shows the location of the volute tongue parameters.

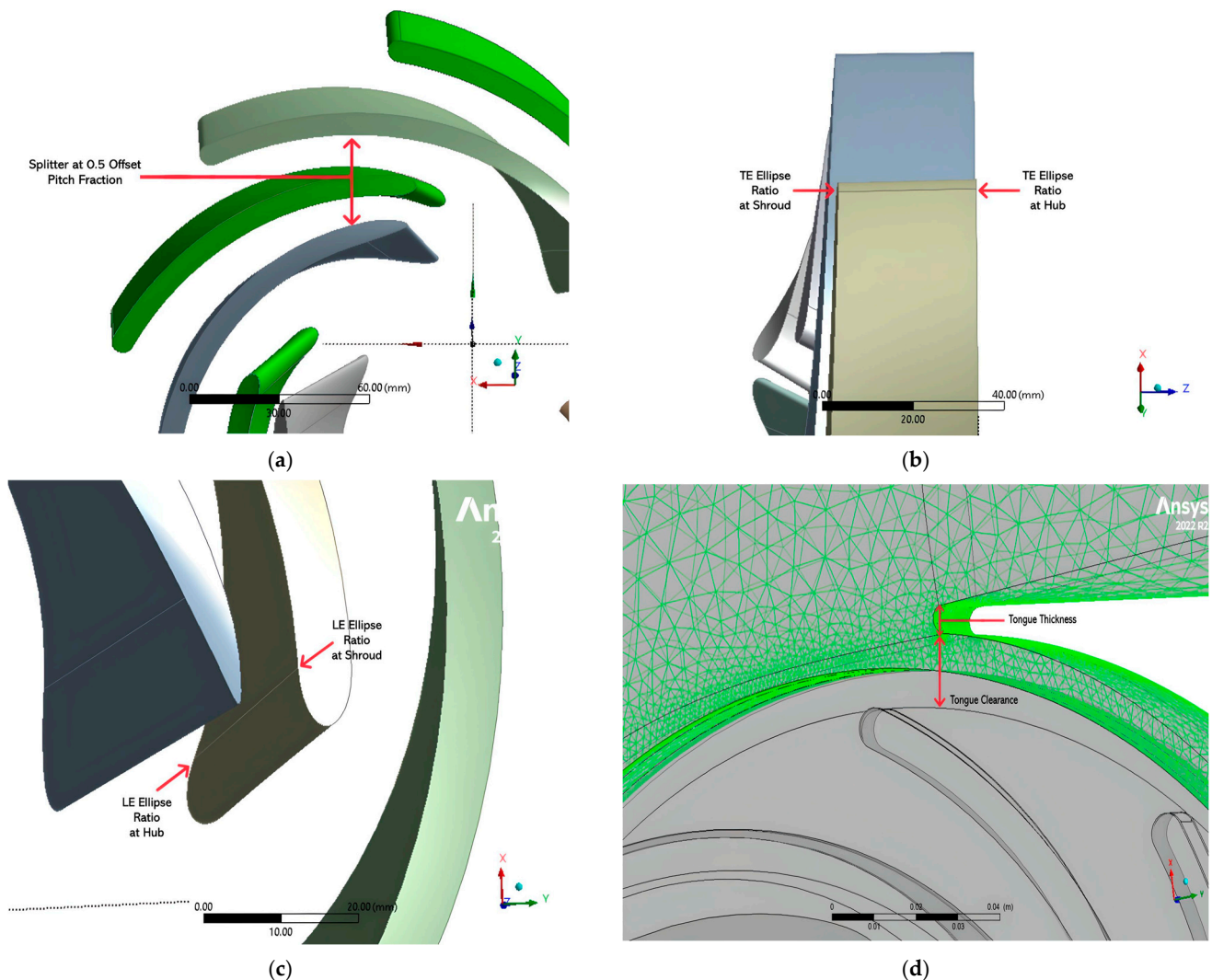


Figure 4. Components to be parameterized: (a) $spltr_OPF$; (b) TE_{erh} and TE_{ers} ; (c) LE_{erh} and LE_{ers} ; (d) $clear$ and thk .

The initial values of the parameterized components for the impeller and volute were generated through the geometric modeling process. The corresponding initial values of the parameters are provided in Table 4.

Table 4. Initial parameter values.

Parameter	Unit	Value
<i>spltr_OPF</i>	-	0.5
<i>LE_erh</i>	-	2
<i>LE_ers</i>	-	2
<i>TE_erh</i>	-	1
<i>TE_ers</i>	-	1
<i>clear</i>	mm	13.189
<i>thk</i>	mm	5.082

2.1.3. Meshing

Before the meshing process commenced, a mesh independence analysis was initiated to verify that the effect of enhancing the mesh would not affect the CFD results. In this analysis, the growth rates of the impeller and volute will be varied. The growth rate determines the density of the mesh. Thus, increasing it results in a fine mesh. The growth rate for the impeller varied from 1.1 to 1.4 and 1.2 to 1.5 for the volute. This was made intentionally to avoid having the same growth rate value for the impeller and volute and to limit computational cost. Additionally, only the total head was considered in this analysis due to resource constraints. Table 5 gives an overview of the data gathered from the analysis.

Table 5. Mesh independence analysis.

Growth Rate	Impeller Nodes	Elements	Growth Rate	Volute Nodes	Elements	H_t (m)	(%) Change
1.1	759,200	4,145,000	1.2	89,559	271,970	18.812	0
1.2	251,260	1,277,300	1.3	75,273	212,850	18.868	0.297
1.3	194,380	946,550	1.4	68,536	186,750	18.693	0.635
1.4	154,740	734,630	1.5	64,811	173,190	18.73	0.437

The defined growth rates for the impeller and volute on the initial mesh setting were 1.1 and 1.2, respectively. Subsequently, the growth rates were varied, and corresponding results were analyzed. Based on the results, the total heads of the succeeding mesh settings remained consistently closer to the result of the initial mesh setting, which also reveals a percentage change that is less than 1%. This shows that the solution converges, and the analysis indicates the independence of the CFD results on the mesh density. Also, utilizing a growth rate of 1.2 and 1.3 for the impeller and volute, respectively, yielded the lowest percent change out of the other mesh settings. Therefore, this mesh setting is deemed suitable for maintaining the quality of the mesh while ensuring that it will not affect the CFD results. With this, the mesh growth rates 1.2 and 1.3 will be applied in the study.

The impeller meshing process was conducted through ANSYS CFX Fluid Flow and ANSYS Mechanical for the volute meshing. Both components share the same physics and solver preferences, CFD and CFX, respectively. Table 6 summarizes the mesh metrics for both the impeller and volute. The total nodes and elements generated based on the meshing processes of both components are provided in Table 7. Tu et al. highlight the importance of using an unstructured triangular mesh for further refinement in a complex geometry. Although structured mesh tends to generate highly skewed cells, this type of mesh generally

leads to numerical unreliability that leads to the deterioration of computation results [27] (pp. 139–140). Hence, an unstructured (tetrahedral) type of mesh was used in the study.

Table 6. Impeller and volute mesh details.

Mesh Details	Impeller	Volute
Element Size (mm)	19.106	7.022
Growth Rate	1.2	1.3
Max Size (mm)	38.213	14.043
Defeature Size (mm)	0.096	0.035
Curvature Minimum Size	0.191	0.070
Curvature Normal Angle (°)	18	30
Bounding Box Diagonal (mm)	382.13	535.75
Average Surface Area (mm)	2.265	7.989
Minimum Edge Length (mm)	0.030	0.236
Target Skewness	0.9	0.9
Maximum Layers	5	5
Pinch Tolerance (mm)	0.172	0.063
Nodes	157,966	75,273
Elements	795,202	212,845

Table 7. Total mesh elements and nodes of the design.

Details	Nodes	Elements
Impeller	157,966	795,202
Volute	75,273	212,845
Total	233,239	1,008,047

The mesh metric in the previous table successfully meshed the pump impeller and volute. These parametric values must be defined to generate a mesh profile used in the CFX-pre setup. The meshed pump impeller and volute are shown in Figure 5.

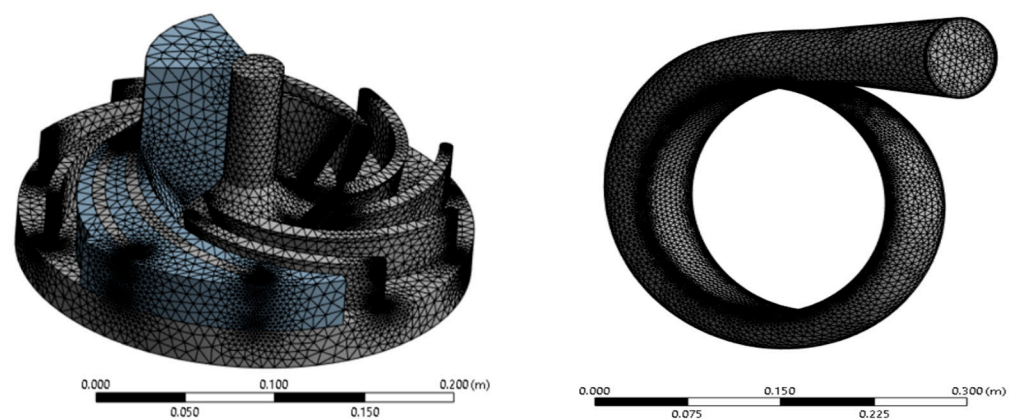


Figure 5. Meshed model of the pump impeller and volute.

2.1.4. Pre-Processing

The mesh profiles of the impeller and volute will be imported to CFX Pre Setup to define the respective domains and their corresponding boundaries. The whole centrifugal pump setup was divided into two domains, namely S1 and R1. R1 refers to the rotating pump component, which generally contains boundaries such as the pump inlet, blade, hub, and shroud. The angular velocity was also defined in R1 at 1600 RPM, the same speed used in Vista CPD. S1 refers to the stationary component of the pump, and this domain was used to define the pump volute. After determining the domain of the pump components,

the interfaces were generated through Turbo Mode options. The turbo mode enables the system to completely define the connection between the two domains involved in the model. Also, Turbo Mode gives the option to choose the type of solver, kind of fluid, and flow rate. In this setup, the Shear-Stress Transport (SST) solver was used as default, water was selected as the fluid type, and the flow rate was 33.333 m³/s. The steady-state k- ω SST RANS model was selected throughout this study by default, as specified by the solver, as it gives highly accurate numerical predictions that determine the amount of flow separation under adverse pressure gradients. The k- ω SST turbulence model is an approach that combines both the strengths of the k- ω and k- ϵ models. Thus, an accurate prediction for flows and separations can be employed in complex structures [28]. The k- ω SST turbulence model was developed by Menter (1994), where it employs equations of the flow field such as the continuity and momentum equations [29]. The k- ω SST equation is defined through Equations (3)–(6):

$$\frac{\partial \rho}{\partial t} + \frac{\partial}{\partial x_i}(\rho u_i) = 0, \quad (3)$$

$$\frac{\partial}{\partial t}(\rho u_i) + \frac{\partial}{\partial x_j}(\rho u_i u_j) = -\frac{\partial p}{\partial x_i} + \frac{\partial}{\partial x_j} \left(\Gamma_U \frac{\partial u_i}{\partial x_j} \right) + S_U, \quad (4)$$

$$\frac{\partial}{\partial t}(\rho k) + \frac{\partial}{\partial x_j}(\rho k u_j) = \frac{\partial}{\partial x_j} \left(\Gamma_k \frac{\partial k}{\partial x_j} \right) + \tilde{G}_k - Y_k + S_k, \quad (5)$$

$$\frac{\partial}{\partial t}(\rho \omega) + \frac{\partial}{\partial x_j}(\rho \omega u_j) = \frac{\partial}{\partial x_j} \left(\Gamma_\omega \frac{\partial \omega}{\partial x_j} \right) + G_\omega - Y_\omega + D_\omega + S_\omega, \quad (6)$$

In Equations (4)–(6), the effective diffusivities of the mean fluid speed, the turbulence kinetic energy, and the specific dissipation rate are given as Γ_U , Γ_k , and Γ_ω . The defined source terms for the transport equations are also given as S_U , S_k , and S_ω . On the other hand, equations representing the modeling of the turbulence production and dissipation are observed. The generation of the turbulence kinetic energy and the specific dissipation rate due to mean velocity gradients are represented in the given equations as $\tilde{G}_k$ and G_ω , while Y_k and Y_ω for the representation of the dissipation of kinetic energy and specific dissipation rate due to turbulence. In completing the equation, the SST model includes a damped cross-diffusion, which is represented by D_ω . This cross-diffusion term was introduced in the equation to blend the k- ω and k- ϵ models together.

Since the study will focus on the behavioral flow analysis of water within the pump, the material properties of water were set in its normal condition. Thus, physical changes such as the temperature and other properties of water within the pump will not be included. However, the pressure induced within the pump, the pressure distribution on its flow, and the amount of turbulence present in the fluid will be observed in the results of the CFD simulation. The material properties of water, as defined in the pre-processing setup, are listed in Table 8.

Table 8. Defined properties of water.

Data	Unit	Value
Density	kg/m ³	997
Specific Heat Capacity	J/kg-K	4181.7
Reference Temperature	°C	25
Reference Pressure	atm	1
Dynamic Viscosity	kg/m-s	8.899×10^{-4}

2.1.5. Post-Processing

In this stage the results of the CFD simulation were obtained. These results will be used as the data for the baseline design model of the study. Also, the pressure and velocity contour plots can be acquired in this stage. In this way, an initial evaluation of the simulated results can be conducted to understand the behavior and the performance of the centrifugal pump. The flow streamlines were also accessed to evaluate the fluid flow pattern in the pump's inlet and outlet. The pump and impeller report can also be generated, and the software will provide the corresponding data, such as the pump's total efficiency (η_t), static efficiency (η_s), total head (H_t), static head (H_s), and shaft/input power (SP). These acquired data will be assigned as the output parameters in preparation for the optimization process. The results of the baseline design model of the centrifugal pump from the conducted CFD simulation are provided in Table 9.

Table 9. CFD simulation results of the centrifugal pump baseline model.

Data	Unit	Value
η_t	%	58.458
H_t	m	13.041
η_s	%	40.592
H_s	m	11.621
SP	W	16,246

The performance results of the pump can also be obtained through the conducted CFD simulation. The volume flow rate (Q) and the total-static head (H_{t-s}) were determined in this process. By using these two components, the computation for the η_t and η_s can be conducted. In this way, it will provide an overview of how the two efficiencies were obtained. The equations for the η_t and η_s are shown in Equations (7) and (8), respectively.

$$\eta_t = \frac{\rho_w g Q H_s}{SP} \cdot 100\% \quad (7)$$

$$\eta_s = \frac{\rho_w g Q H_{t-s}}{SP} \cdot 100\% \quad (8)$$

2.2. Optimization Process in Centrifugal Pump Design

The defined input and output parameters from the conducted CFD simulation will be used to develop an optimized centrifugal pump design. The optimization process involves different procedures that will be followed according to the general workflow of the Response Surface Optimization (RSO) tool in ANSYS Workbench. Figure 6 provides a workflow summary of the optimization process using RSO.

2.2.1. DOE Setup

The DOE is the initial procedure for conducting an optimization process through ANSYS 2022 R2. It is known to explore data through computational algorithms efficiently. In this process, design points are created based on the defined data parameters of the study. The DOE used was the 'Custom + Sampling' type. The central composite design (CCD) was initially considered for use in the study. However, this DOE type was found unsuitable for this study as it contains numerous parameters to be considered, resulting in a higher computational cost and much time to process the input and output parameters. This finding also aligned with the suggestion of a related study, where Custom + Sampling ensures much smoother data processing for the DOE matrix [30]. Since the Custom + Sampling method processed the complexity of the given parameters, this DOE type was used in the

study. The total number of samples in this process was set to 15. With this, the DOE has generated 15 samples with different design parameters. The generated design points are provided in Table 10.

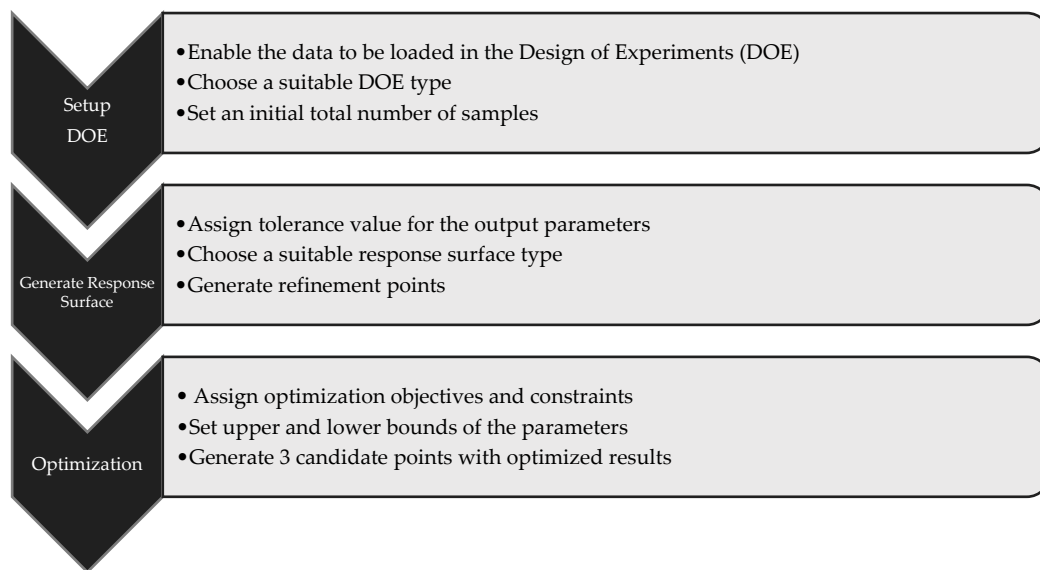


Figure 6. Workflow summary of the optimization process.

Table 10. DOE-generated design points.

DP	spltr_ OPF	LE_ erh	LE_ ers	TE_ erh	TE_ ers	Clear (mm)	thk (mm)	η_t (%)	H_t (m)	η_s (%)	H_s (m)	SP (W)
1	0.5	2	2	1	1	13.189	5.082	58.458	13.041	40.592	11.621	16,246
2	0.467	2.195	2.175	1.077	1.099	14.494	5.586	70.981	15.859	48.823	14.561	16,765
3	0.453	1.814	1.833	1.097	0.921	12.138	4.659	71.93	15.325	49.818	14.184	16,115
4	0.454	1.825	2.176	0.909	1.098	13.994	5.524	72.258	15.215	49.974	14.05	15,891
5	0.455	2.187	2.191	1.093	0.923	12.441	5.575	71.472	15.138	49.448	14.025	16,036
6	0.463	1.821	2.181	0.907	0.913	14.387	4.800	70.921	15.564	48.979	14.382	16,572
7	0.463	2.088	2.149	0.902	1.100	11.956	4.641	69.693	15.144	47.945	13.876	16,271
8	0.452	2.133	1.897	1.088	0.911	14.471	4.646	72.357	14.243	50.098	13.089	14,783
9	0.456	2.186	1.810	0.930	0.939	12.19	5.557	69.367	13.818	48.152	12.565	14,804
10	0.451	2.197	1.952	1.100	1.097	13.016	4.623	70.804	14.638	48.46	13.528	15,614
11	0.458	1.833	1.858	0.968	1.099	14.432	4.633	73.68	15.29	50.106	14.024	15,555
12	0.463	1.812	1.807	0.900	1.055	11.951	5.237	69.92	14.362	47.593	13.077	15,284
13	0.464	2.126	2.084	0.904	0.918	14.42	5.588	69.84	15.726	48.57	14.56	17,037
14	0.459	1.810	1.841	1.094	0.921	14.189	5.390	69.62	15.269	48.187	14.104	16,556
15	0.461	2.042	1.814	0.905	0.910	12.35	4.584	69.73	14.848	47.904	13.637	15,982

2.2.2. Response Surface Method

In this process, the software generates a surface by interpolating the design points created in the DOE process. The relationship between the input and output parameters of the created design points is statistically and numerically processed in the RSM, and their response is shown through either a 2D, 2D slice, or 3D graphical model. The type of response surface used in this study is the default response surface, 'Genetic Aggregation'. This type of response surface is ideal for optimization as it aims to find the suitable response level for each design point through an iterative genetic algorithm based on the preferred tolerance for the output parameters. The given tolerance for the response surface variables is provided in Table 11. It is essential to define the tolerance of the output parameters for the software to generate a reasonable response accurately. The tolerances were kept at a minimum value to retain the accuracy of finding a suitable response for the study.

Table 11. Indicated response surface tolerances.

Data	Unit	Value
η_t	%	1
H_t	m	0.5
η_s	%	1
H_s	m	0.5
SP	W	0.5

2.2.3. Optimization Process

The optimization process is the last procedure used in the study. It deals with further validating the response of each sample and producing 3 candidates with corresponding optimized design results. In this process, the upper and lower bounds of the input parameters along with the objectives and constraints of the output parameters must be set to allow the software to perform an optimization study and find optimal design parameters for the centrifugal pump. In Table 12, the boundary conditions of the input parameters are provided. The values listed in this table were based on the observed and predicted data through the DOE and RSM, respectively. The boundary conditions must be defined to limit the design parameters to a certain range. On the other hand, Table 13 shows the given objectives and constraints for the output parameters. The target objectives were based on the maximum response generated through the RSM, while the indicated constraints are set to values that are between the maximum and minimum responses generated. In this way, an optimal solution that is in line with the generated responses in the RSM can be achieved.

Table 12. Boundary conditions of the input parameters.

Data	Lower Bound	Upper Bound
$spltr_OPF$	0.45	0.55
LE_erh	1.8	2.2
LE_ers	1.8	2.2
TE_erh	0.9	1.1
TE_ers	0.9	1.1
$clear$ (mm)	11.87	14.508
thk (mm)	4.574	5.590

Table 13. Objectives and constraints of the optimization study.

Data	Objectives		Constraints
	Type	Target	
η_t	Maximize	85	$\geq 70\%$
H_t	Maximize	16	≥ 12 m
η_s	Maximize	60	$\geq 40\%$
H_s	Maximize	13	≥ 10 m
SP	Minimize	15,000	$\geq 10,000$ W

The data provided in Tables 12 and 13 will then be processed by the optimization method known as The Multi-Objective Genetic Algorithm (MOGA), which is a variant of the Non-dominated Sorted Genetic Algorithm-II (NSGA-II), which aims to determine the optimum design based on the given objectives and constraints. The RSO in ANSYS makes use of the MOGA as its optimization technique in finding the optimal solution of the design under multiple objectives and constraints [31]. The advantage of using the MOGA, that is a variant of the NSGA-II, provides a better sorting algorithm and better convergence rate [32]. Hence, this technique was selected for the optimization process. Initially, the configuration

allows generating 7000 samples consisting of 1400 samples per iteration at a maximum of 20 iterations and 33,600 estimated evaluations. The optimization process configuration allows a maximum Pareto and stability percentages of 70% and 2%, respectively. The Pareto percentage measures the amount of change per iteration of the given objectives and constraints, while the stability percentage determines the stability of the convergent points. The process measured a Pareto percentage at 0.07% and a convergence stability of 1.62% with no reported failure per iteration. The solution of the defined objective and constraints converged after 13 iterations. The process was completed after 23,329 evaluations. Given that the solution resulted in a convergence stability that is below the allowable percentage of 2% and it converged after 13 iterations, which is also below the given maximum of 20 iterations with no reported failure, it can, therefore, be validated that the solution generated in the optimization process is acceptable.

3. Results and Discussion

This section, which contains the pump data, such as the efficiencies, heads, and power, will discuss interpreting the study's results. Contour plots will also determine the pump's pressure and fluid flow behavior. The evaluation will begin with the results of the baseline or initial design of the centrifugal pump through CFD simulation and compare them to the findings gathered in the optimized design through the RSM.

3.1. Baseline Design Results

The summary of the pump performance for the baseline design of the centrifugal pump was previously provided in Table 9. The η_t generated was 58.458%, with a H_t of 13.041 m. On the other hand, the η_s resulted in 40.592% with a H_s of 11.621 m. The SP resulted in 16,246 W. Here, the η_t includes the outlet velocity, and the static pressure generated in the pump, while the η_s measures only the static pressure in the inlet and outlet of the pump, hence the difference in the results.

3.1.1. Results Validation

The acquired results were compared to the experimental results presented in the study by Gölcü et al. [23] to validate the results of the baseline model obtained through CFD simulation. Both the mentioned related study and CFD simulation used five main blades and five splitter blades in the design, with the length of the splitter blades making 80% of the length of the main blades. Both designs selected water as the fluid type. However, slight differences in the properties of water were observed. The reference study used a density of 998 kg/m³ at 20 °C, while this study defined a density of 997 kg/m³ at 25 °C. While these differences can be observed, it is expected that slight deviances can occur. The η_t of the CFD was compared to the data observed in the reference experimental study at 80% nondimensional splitter blade length, while H_t of the CFD was compared to the design parameter of the head as defined in the reference experimental study. As shown in Table 14, the results were validated with a percent difference of 0.974% for the total efficiency and 0.315% for the total head.

Table 14. Comparison of η_t and H_t from related study and CFD simulation.

Parameter	Experimental	CFD	Percent Difference (%)
η_t (%)	59.03	58.458	0.974
H_t (m)	13.00	13.041	0.315

3.1.2. CFD Analysis of the Baseline Design Model

To understand what led to the simulated results of the baseline design model, a CFD analysis will be conducted to determine the fluid flow behavior within the pump. A series of contour plots and a streamline diagram will be provided in this section to provide an in-depth understanding of the root causes that affect the pump's performance. This analysis is crucial in the design process of a centrifugal pump as it determines not just the flow of the fluid but also the amount of turbulence and pressure present within the pump.

1. Pressure contour plot

The pressure contour plot in Figure 7a shows the pressure contour plot of the baseline design model. The figure shows the difference in the pressure distribution between the impeller and the volute. The pressure contour in the volute is way too high, considering that there is a low-pressure energy obtained from the impeller. A sudden pressure drop may result in further damage to the blades. This irregularity in pressure distribution clearly affects the pump's performance. It risks obtaining several issues, such as an unstable flow due to pressure fluctuations, excessive stress on the components, and unstable pumping operation. When it comes to improving the design of a centrifugal pump, this specific issue should be addressed to improve the quality of the pump's performance. Thus, the optimization design process should achieve an improved pressure distribution.

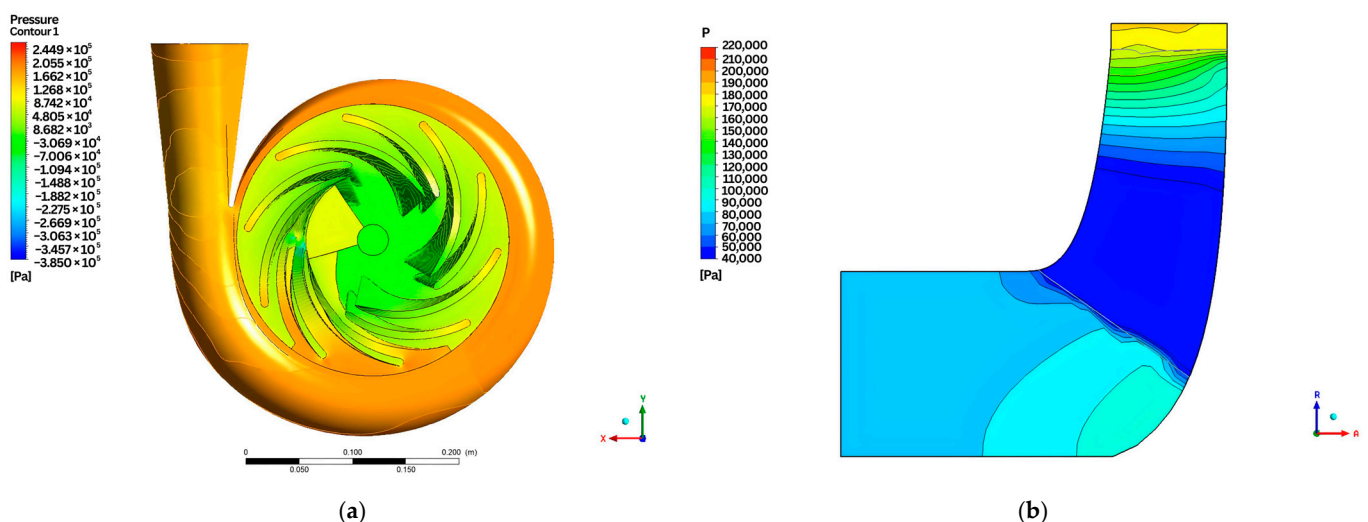


Figure 7. The (a) pressure contour plot of the baseline design model and the (b) area-averaged pressure contour on the meridional surface of the blade.

To understand the fluid's interaction with the pressure side of the blade, the area averaged pressure contour plot on the meridional surface is provided in Figure 7b. Based on the figure, a huge drop of pressure is evident in the middle section of the blade. This implies that the middle section of the blade may be subjected to strains, which may lead to further damage within the surface of the blade. Thus, it is vital to address this issue concerning the damage that may affect the impeller blades to maintain the efficiency and performance of the pump.

2. Volume rendering of the entire domain

The cloud contour plot of the entire domain can be accessed through volume rendering. The volume rendering helps visualize the 3D representation of the internal structure within the whole domain of the simulated baseline centrifugal pump design. In Figure 8a, the accumulated pressure of the entire domain is not equally distributed. Irregular pressure distribution is evident between the pump impeller and the volute, judging by the color

contour of the given domain. An equal pressure distribution is essential to maintaining the system's fluid pumping efficiency within its entire domain. In this case, pressure fluctuations will likely occur between the impeller and volute, affecting fluid flow within the pump. Additionally, the impeller–volute interaction gradually affects the pump performance. Hence, this uneven pressure distribution may lead to inefficient pump performance. On the other hand, the eddy viscosity cloud contour in Figure 8b shows the number of eddies present in the entire system. Determination of the number of eddies in the system is essential as these accumulated eddies generate turbulence in the pump. Thus, many eddies result in higher turbulence in the system. Based on the given figure, spots of eddies can be seen within the pump impeller, containing a higher value of eddy viscosity compared to the other sections within the pump. It is also evident that the eddy viscosity gradient is unevenly distributed within the system. Given this situation, the fluid flows turbulently due to the number of eddies in the system. A higher amount of turbulence within the fluid flow may increase energy loss. This finding may also be the cause of the low pressure present in the impeller blade surface, as previously shown in Figure 7b. Additionally, the turbulence caused by the number of eddies disturbs the path of the fluid and causes a decrease in pressure. With this, the system will demand more pumping power to attain the desired flow performance. Thus, this will increase the input power delivered by the shaft.

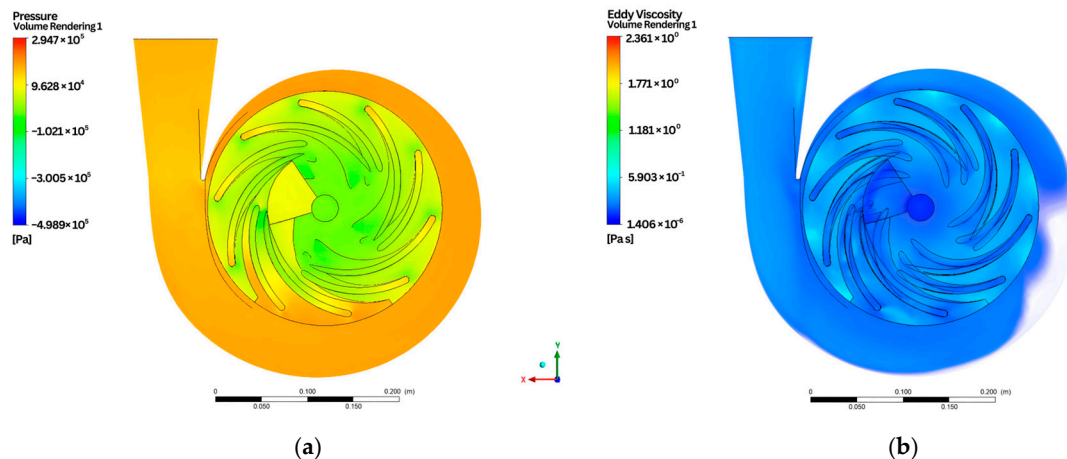


Figure 8. The (a) pressure and (b) eddy viscosity cloud contour of the entire domain.

3. Velocity streamline of the baseline design model

The velocity streamline diagram visualizes the flow path of the fluid. Figure 9 shows the pattern flow of the fluid from the inlet and the outlet of the impeller blades. It can be seen in the given figure that the flow of fluid deviates from the profile of the blade at some point. The flow of the fluid also forms an indefinite pattern, which depicts that the flow of the fluid in the impeller is inconsistent, and the velocity reduces due to the irregularity of the flow pattern of the fluid, which can also have a significant effect on the hydraulic efficiency and performance of the pump.

3.2. Optimization Result

The optimization result involves presenting the findings in the conducted optimization study through the RSM. This will provide valuable insight into how the optimal design of the centrifugal pump was achieved by analyzing the effect of modifying the design, determining the response of varying the input parameters, and evaluating the improved design's performance from the centrifugal pump's baseline design model. With this, the study's objectives can be fulfilled, and an insightful conclusion and recommendation can be constructed.

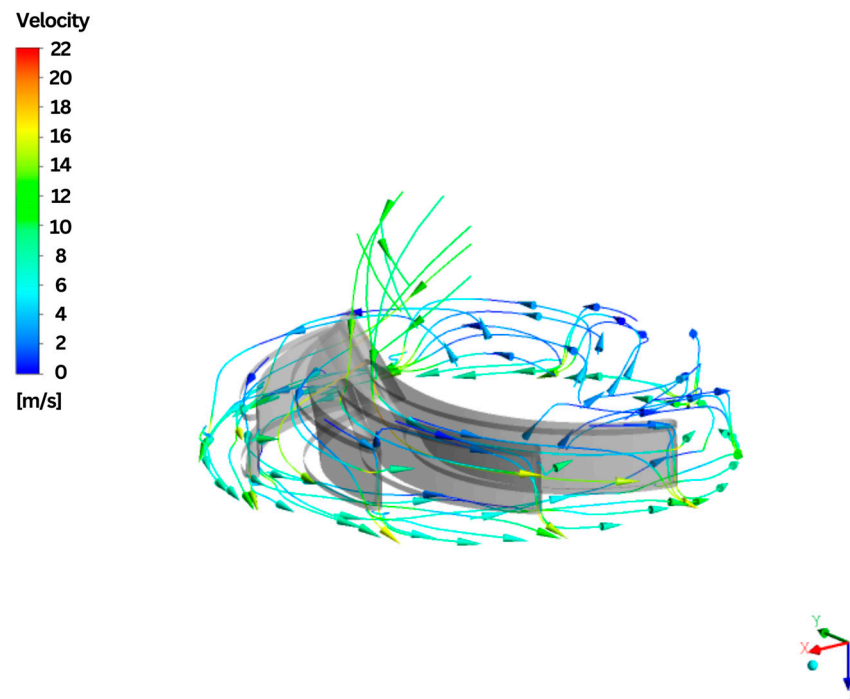


Figure 9. The velocity streamline diagram of the baseline design model.

3.2.1. Minimum and Maximum Generated Response

The minimum and maximum responses are generated through the RSM according to the given tolerances provided in Table 11. The response surface type Genetic Aggregation was used to create 90 refinement points having different input parameters with their corresponding output or response. The calculated minimum and maximum responses are shown in Table 15. The η_t gathered a minimum response of 53.527% and a maximum of 86.262%. The minimum response for the H_t gathered a response of 6.976 m and a maximum of 18.383 m. The calculated minimum and maximum response for the η_s are 33.893% and 58.479%, respectively. The H_s obtained a minimum response of 4.961 m and a maximum of 17.401 m. The SP obtained a minimum response of 4936.6 W and a maximum of 20,954 W. These responses generated are beneficial in the optimization study as they will help determine the effect of different variations in parameters. In this way, the objective of the study to develop an optimized centrifugal pump can be achieved.

Table 15. Generated minimum and maximum response.

Parameter	Minimum	Maximum
η_t (%)	53.527	86.262
H_t (m)	6.976	18.383
η_s (%)	33.893	58.479
H_s (m)	4.961	17.401
SP (W)	4936.6	20,954

3.2.2. Goodness of Fit (GOF) Chart

The GOF Chart in Figure 10 shows the relationship between the observed and the predicted values. This criterion measures the degree of proportionality of the two values and determines how close the two values are to each other. The horizontal axis represents the observed points from the design points generated in the DOE test environment, and the vertical represents the estimated or predicted points extracted from the RSM. A diagonal regression line assesses the relationship between the observed and the predicted points to

determine whether the two indicate a strong fit. The observed and predicted values for the SP , η_t , and η_s are linear, as the scattered points of these parameters closely align with the diagonal regression line. This means that the two values obtained possess a strong similarity. However, both the H_t and H_s move slightly away from the regression line. This indicates that the relationship between the observed and predicted values for the H_t and H_s contains a small number of discrepancies. Nonetheless, the deviation between the observed and predicted values of the two heads is still tolerable as they converged within the regression line at some point in the GOF chart. With these findings, it can be verified that through the GOF chart, the observed and predicted values are adequate results.

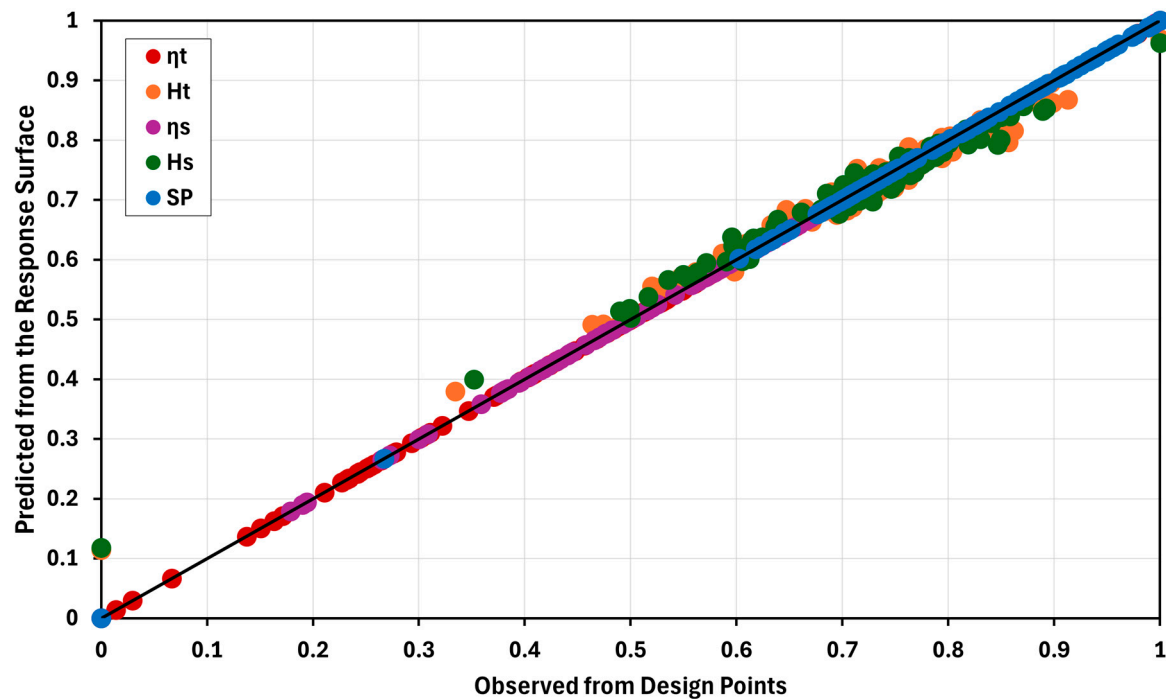


Figure 10. Normalized value chart of the predicted vs. observed values of the study.

3.2.3. Sensitivity Analysis

The sensitivity analysis can be formulated through the local sensitivity chart in the RSM. The local sensitivity helps understand how sensitive an output response is to changes in specific input parameters, which is beneficial in an optimization process. This sensitivity analysis can be used to determine the impact of varying the input parameters on the output's response. The chart typically shows a positive and a negative range. A positive value implies a positive relation between the input and output parameters, while a negative value suggests the opposite. This means that an increase in the input parameter reduces the output value. To further analyze the sensitivity of the parameters, the local sensitivity chart is provided in Figure 11, illustrating the sensitivity percentage between the input and output parameters. Based on the figure, increasing the $spltr_OPF$ further decreases the η_t . The same results can be seen for the H_t , η_s , and H_s . Increasing the TE_erh increases the η_t . On the other hand, increasing the LE_erh is more suitable for increasing the H_t , η_s , and H_s . In terms of the thk and $clear$, the same findings were extracted from the investigation of Morabito et al., where modifying the volute tongue affects the pump's performance. Thus, making it the most crucial variable in the design modification of the volute [33]. In keeping the SP at a minimum level, increasing almost every input parameter will be beneficial since one of the study's objectives is to reduce the power consumption of the centrifugal pump. In terms of practical applications, the use of sensitivity analysis will be beneficial in

assessing the positive and negative effect of modifying the design parameters. This gives an advantage in making design considerations or decision making.

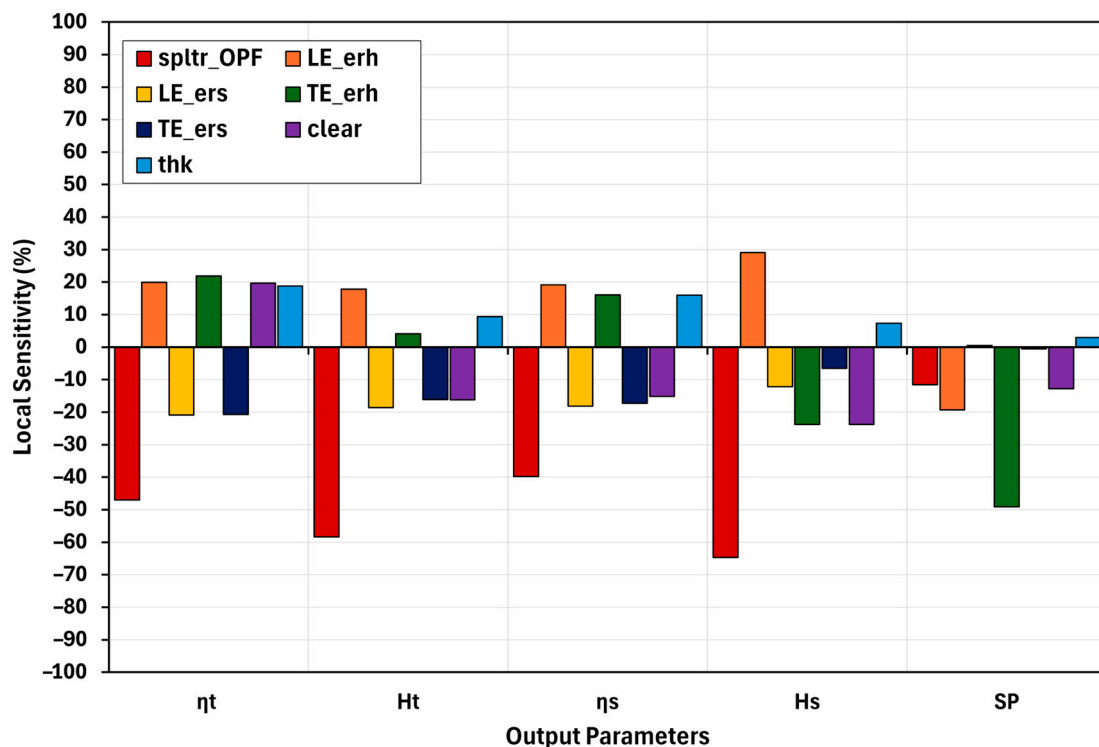


Figure 11. Local sensitivity chart of the input and output parameters.

3.2.4. Response Surface Charts

This section highlights the use of response surface charts to further understand how varying the input parameters affects the output parameters in a positive and negative way. The results provided by the response surface charts will be used to determine the appropriate design modifications that must be made to optimize and fully boost the performance of the centrifugal pump. Combining these positive modifications of the parameters will also help shed light on how the pump performance improved from its baseline design model.

1. *spltr_OPF*

Figure 12 provides the effect of varying the *spltr_OPF* in different output parameters. These response charts make it easier to understand how the changes in the position of the splitter blades affect the pump's performance. When the value of the *spltr_OPF* is greater than 0.5, the η_t , H_t , η_s , and H_s increase initially then decrease continuously. In contrast, these output values increase when the value of the *spltr_OPF* is lower than 0.5. This means that there is a better flow of the fluid that leads to an improved hydraulic performance in the pump when the *spltr_OPF* is below 0.5. The slight increase with the output parameters that were observed when the *spltr_OPF* is above 0.5 might be due to a better flow of fluid at some point, but because of the inconsistency of flow or the occurrence of pressure fluctuation within the impeller, the output parameters continue to decrease. Regarding the *SP*, the highest value was obtained when the *spltr_OPF* was near 0.49 and gradually decreased as the *spltr_OPF* decreased. The initial value of the *spltr_OPF* was set at 0.5. This means that the location of the splitter blades lies between the pressure and the suction side of the main blade. A value less than 0.5 refers to the location of the splitter blade close to the pressure side of the main blade, while a value greater than 0.5 refers to the splitter blades' location close to the main blades' suction side. Based on the results,

it is suggested that the position of the splitter blades should be maintained closer to the pressure side at a reasonable distance to maintain a high value for the efficiencies and head, which also leads to a decrease in the power consumption of the pump. This observation shared the same sentiment as the findings of Zakeralhoseini and Schiffmann [34].

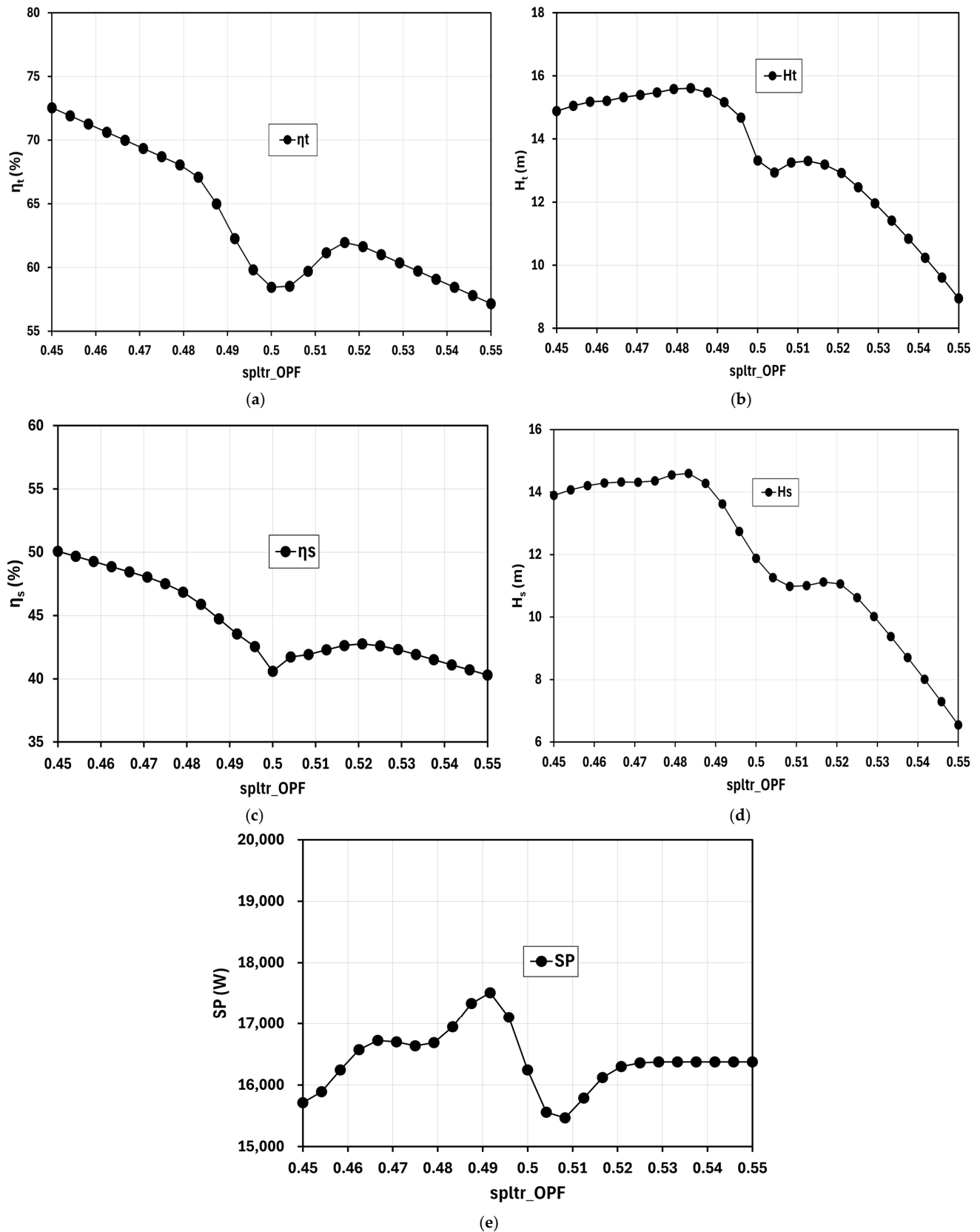


Figure 12. The effect of varying $spltr_OPF$ in (a) η_t ; (b) H_t ; (c) η_s ; (d) H_s ; (e) SP .

2. *thk* and *clear*

The effect of modifying the *thk* and *clear* was also observed in this section of the study. Figure 13 shows that the η_t , H_t , η_s , and H_s increase when the *thk* increases and the *clear* is maintained to a minimum value, which, in this case, is at a value of 11.87 mm and is represented by a violet trendline in the response chart. On the other hand, the *SP* can be reduced when the *thk* is reduced with an increase in the *clear*. Based on the findings provided in the response charts, it is evident that a minor change in the volute tongue profile has a significant effect on the pump's performance. The result shows that the *thk* should increase while maintaining a lesser value for the *clear* to attain a better pump performance. Thus, it is crucial to consider the perfect balance between the design of the *thk* and *clear* to achieve high efficiency and head while maintaining a minimum amount of power consumption, as modifying these two parameters of the volute contributes to a better impeller–volute interaction that improves the hydraulic efficiency of the pump. With this, friction losses and pressure fluctuations from fluid flow at the impeller and volute are reduced, leading to a desirable performance of the pump. This finding proves that instead of redesigning only the impeller blades, a minor volute modification should also be considered and not neglected when designing an efficient centrifugal pump. When it comes to the usual setup of centrifugal pumps, the impeller blades are accompanied by the volute or the pump's casing and should not be neglected in the design. This is because in practical applications, the impeller–volute interaction affects the hydraulic performance of the pump. Hence, designing the impeller and volute should be considered simultaneously to enhance the pump's performance.

3. Splitter blades' *LE_erh*, *LE_ers*, *TE_erh*, and *TE_ers*

In this section, the effect of varying the splitter blades' *LE_erh*, *LE_ers*, *TE_erh*, and *TE_ers* on each output parameter will be determined. This will allow a suitable combination of LE and TE modification to be selected for the centrifugal pump's design optimization. Configuring the blade regions of the splitter blades is a crucial aspect of blade design that should be investigated. The shape of the blades controls the direction of the fluid from inlet to the outlet of the pump. In some cases, the LE and TE side of the blade can cause excessive vibration and noise in industrial applications if not designed properly. Thus, improving their design significantly enhances the performance of the pump.

- Total Efficiency (η_t)

Figure 14 shows the effect of varying the LE and TE ellipse ratios in η_t . The initial values of the ellipse ratios of the LE were set to 2 and 1 for the ellipse ratios of the TE. Both response charts depict an increase in η_t from their initial values when the *LE_erh* and *TE_erh* increase, and the *LE_ers* and *TE_ers* decrease. However, a slight decrease in efficiency is evident in the TE response chart when the *TE_erh* decreases, and *TE_ers* maintains its initial value of 1. Additionally, minimum efficiencies are observed for the LE and TE response charts when their respective ellipse ratios at the hub and shroud are on their initial values.

- Total Head (H_t)

The effect of configuring the splitter blades LE and TE ellipse ratios to the H_t of the pump can be observed by visualizing the response of the output parameters to the configured input parameters. Figure 15 shows the effect of varying the LE and TE ellipse ratios in H_t . The maximum head can be attained if both *LE_erh* and *TE_erh* increase and both the *LE_ers* and *TE_ers* decrease. The results of the response charts for the H_t are similar to the analysis mentioned in the η_t . However, in this case, the minimum head can be seen when the *LE_erh* decreases and the *LE_ers* increases. Meanwhile, increasing the

TE_{erh} and TE_{ers} minimizes the head. In this analysis, it is evident that configuring the LE and TE of the splitter blades creates a significant impact on the H_t of the pump. To attain a desirable pump's H_t , the results suggest that the LE_{erh} and TE_{erh} must be kept greater than their initial values while keeping the LE_{ers} and TE_{ers} lesser than their initial values.

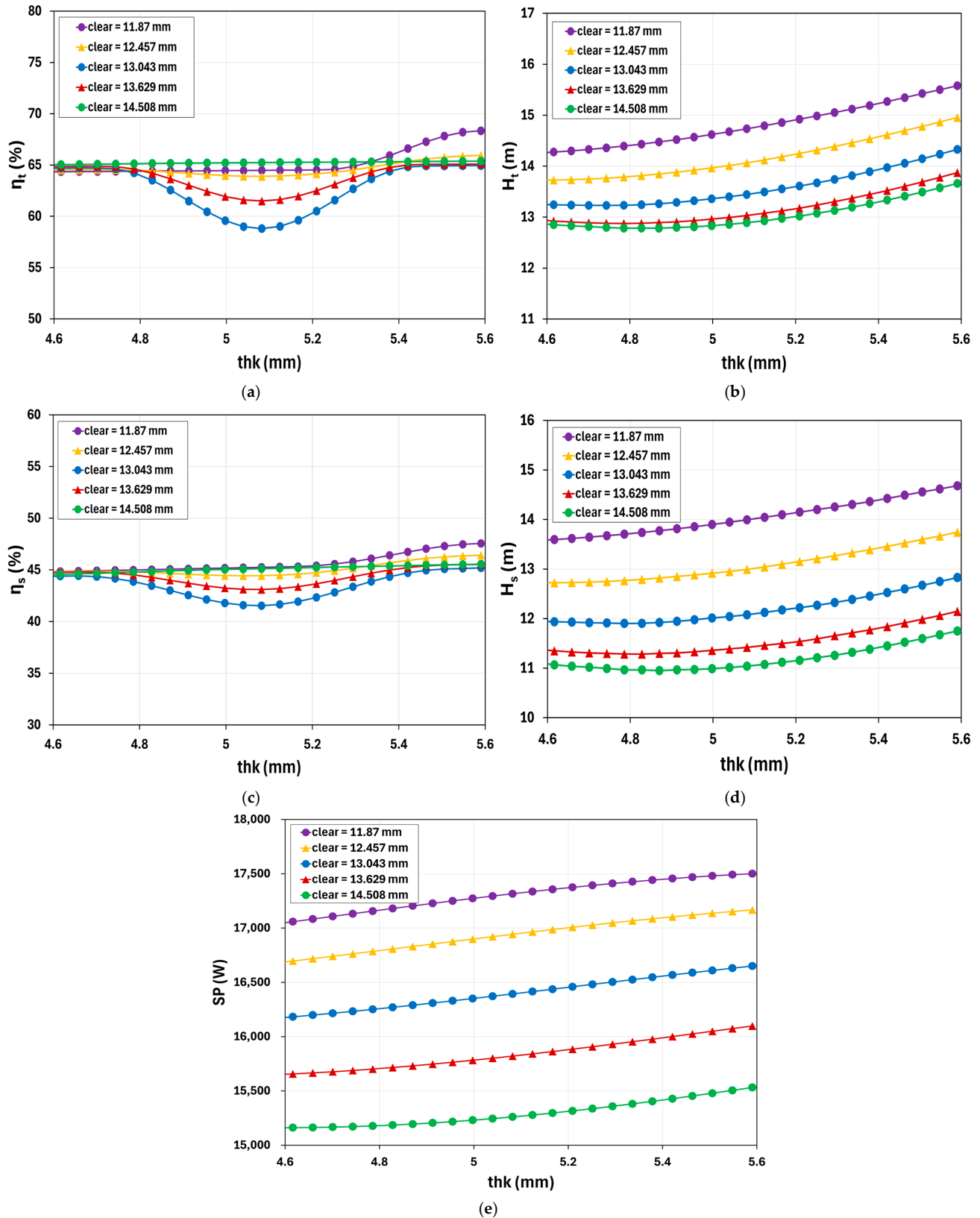


Figure 13. The effect of modifying the thk and $clear$ in (a) η_t ; (b) H_t ; (c) η_s ; (d) H_s ; (e) SP .

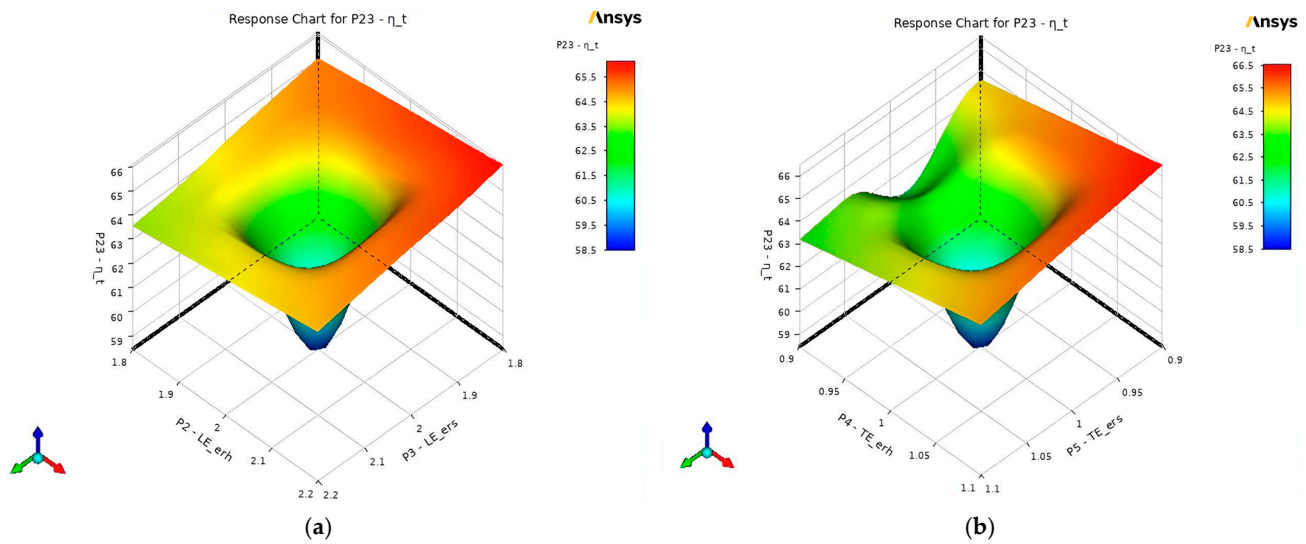


Figure 14. The effect of varying (a) LE and (b) TE ellipse ratios in η_t .

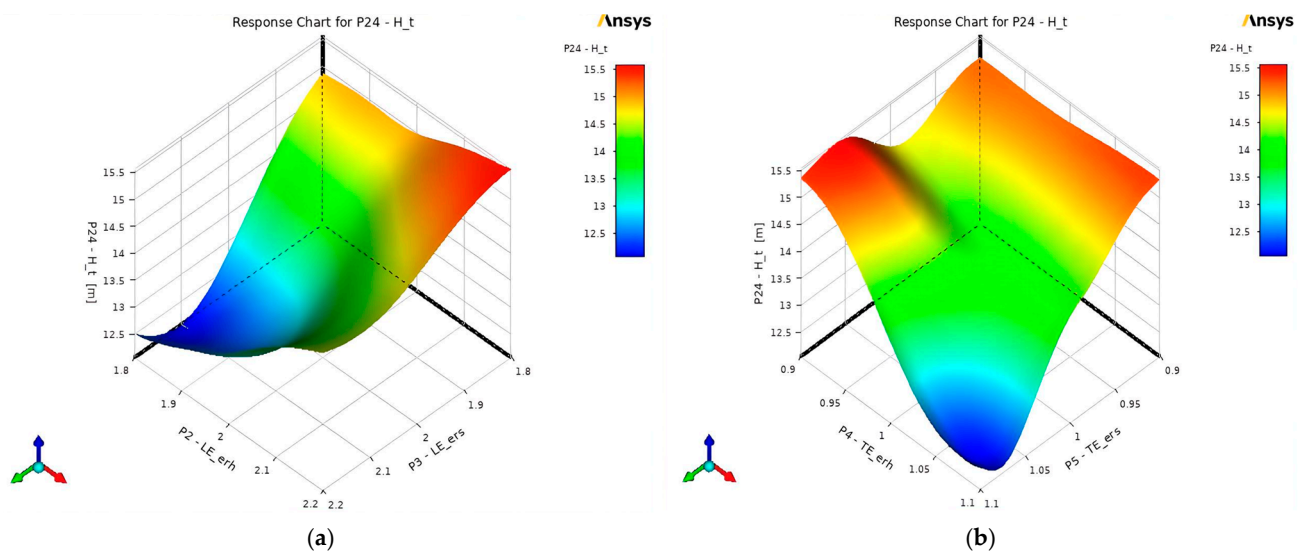


Figure 15. The effect of varying (a) LE and (b) TE ellipse ratios in H_t .

- Static Efficiency (η_s)

In Figure 16, the response surface charts show the response of the η_s to the changing input variables of LE and TE. The response of the η_s is shown to be similar to the response of the η_t observed in the previous discussion. Based on the given response charts, the maximum η_s can be attained if both LE_{erb} and TE_{erb} are greater than their initial values, and both the LE_{ers} and TE_{ers} maintain a value lesser than their initial values. The minimum η_s can also be observed when the LE and TE ellipse ratios are at their respective initial values. Therefore, both η_t and η_s results suggest that the LE and TE ellipse ratios should be drawn away from their respective initial values to ensure an increase in η_t and η_s .

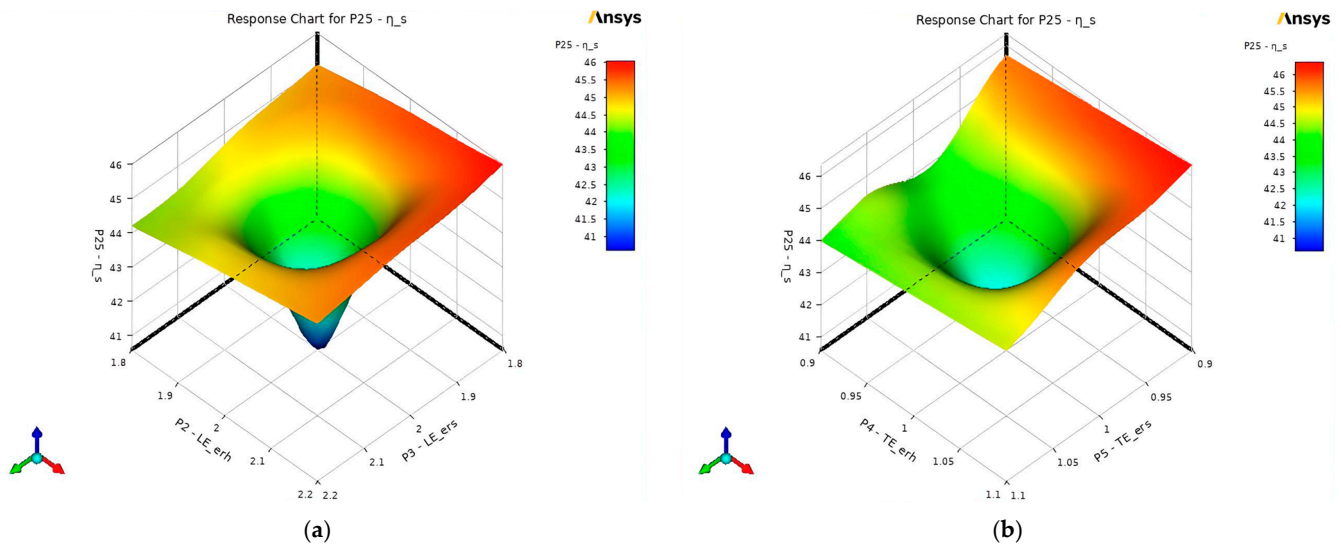


Figure 16. The effect of varying (a) LE and (b) TE ellipse ratios in η_s .

- Static Head (H_s)

Figure 17 provides the LE and TE configuration response charts in line with the H_s . Based on the response charts, the behavioral response of the H_s seems to be similar to the response observed in H_t . The H_s increases if both LE_{erb} and TE_{erb} increase from their initial values while decreasing the values for both the LE_{ers} and TE_{ers} . However, it was also observed in the response chart of the TE that the H_s can also increase if the TE_{erb} decreases from its initial value and the TE_{ers} increases from its initial value. This new finding shows how sensitive the H_s is to the configuration of the TE ellipse ratios. Hence, finding a suitable balance between the ellipse ratios of the splitter blades' LE and TE is suggested when it comes to design modification to maintain a desirable value of H_s .

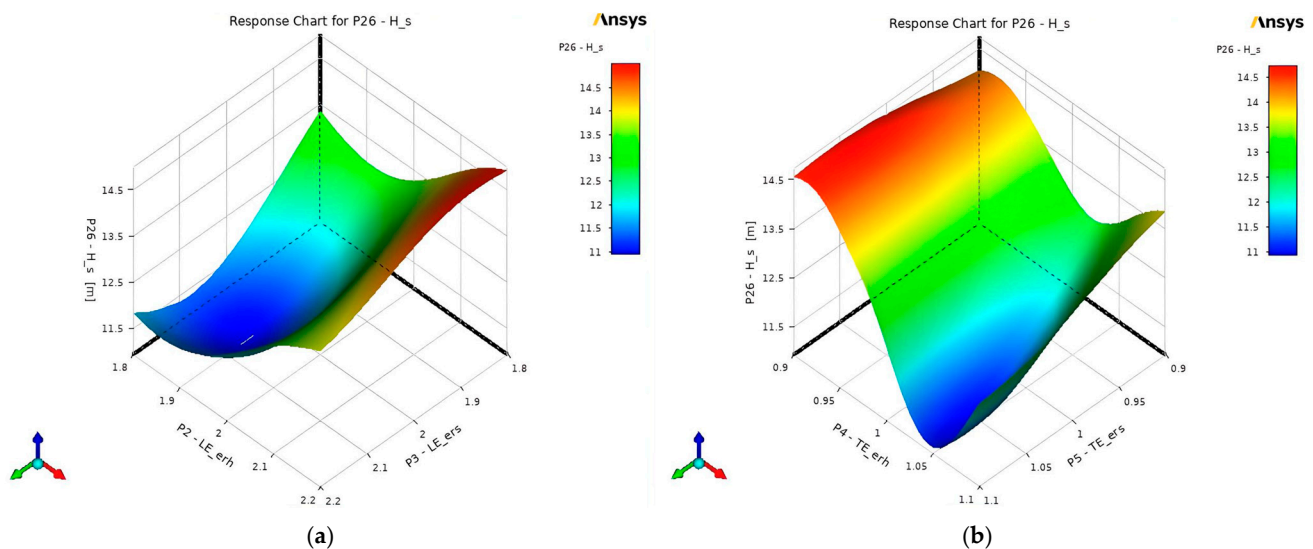


Figure 17. The effect of varying (a) LE and (b) TE ellipse ratios in H_s .

- Shaft Power (SP)

One of the study's objectives is to develop an optimized centrifugal pump design that will decrease the power consumption for the pump's operation. In Figure 18, the response chart of the LE shows that the SP can be reduced if the LE_{erb} and LE_{ers} both increase from their initial values of 2. In contrast, a value of 2.1 for the LE_{erb} and a decreasing

value of the LE_{ers} increase the SP . On the other hand, the TE response chart shows a reduction in SP when the TE_{erh} was 1.06, and the TE_{ers} was greater than its initial value of 1. Decreasing the TE_{erh} from its initial value while increasing the TE_{ers} from its initial value will increase the SP . Through these results, the limitations of the LE and TE modifications can be determined to develop a reduced power consumption for the pump. A suitable combination of LE and TE parameters should be carefully selected to attain the given objectives of the study.

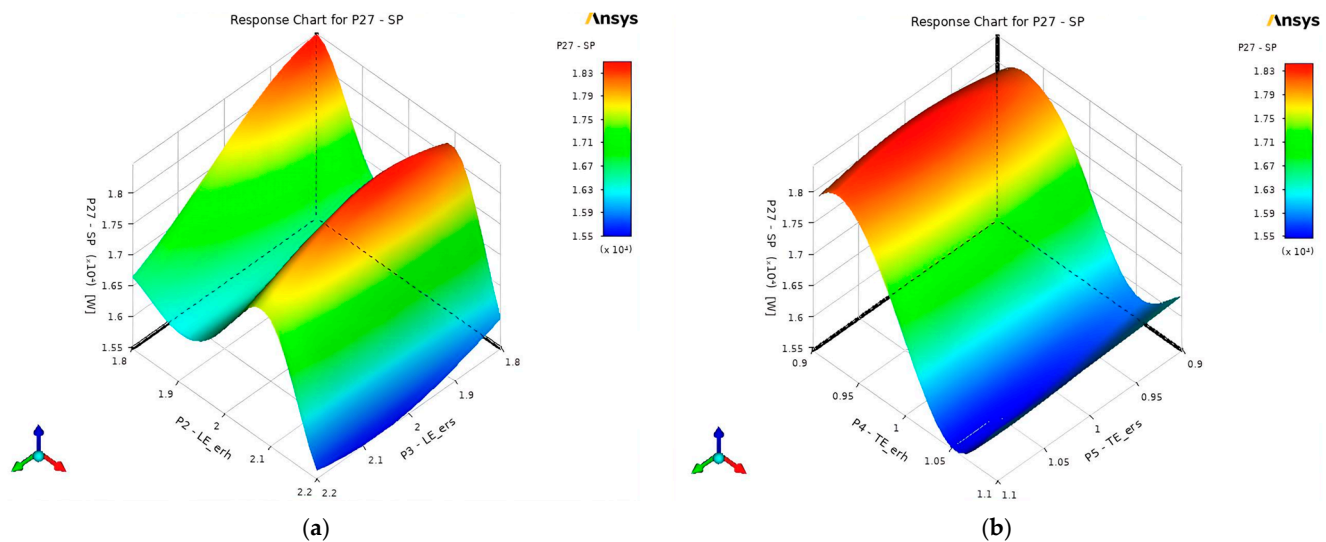


Figure 18. The effect of varying (a) LE and (b) TE ellipse ratios in SP .

3.2.5. Optimal Candidate Points

The optimization study generated three (3) candidate points with corresponding improvements for the centrifugal pump design. These candidate points could be determined through the defined objectives and constraints. The optimization process solution was able to run using the MOGA method, which aims to find optimal results out of all the generated design and response points from the DOE and RSM, respectively. The optimization process ended with 23,239 evaluations at 13 iterations with no reported failures. Hence, the suggested candidate points are exemplary. The three candidate points with optimal input design parameters are provided in Table 16, while their corresponding output parameters are provided in Table 17. Based on the data, the first candidate point is the best option for this study as it contains the highest η_t and static η_s while having the lowest power consumption compared with the two other candidate points. However, candidate points 2 and 3 are also acceptable as these three points only possess a small number of discrepancies. Nonetheless, the parameters obtained in candidate point 1 will be used in this study.

Table 16. Optimal input design parameters of the study.

Input Parameter	Candidate Point 1	Candidate Point 2	Candidate Point 3
$spltr_OPF$	0.45	0.45	0.45
LE_{erh}	2.110	2.095	2.092
LE_{ers}	1.810	1.816	1.819
TE_{erh}	1.015	1.012	1.014
TE_{ers}	0.901	0.901	0.901
$clear$ (mm)	12.994	12.888	12.772
thk (mm)	5.579	5.589	5.584

Table 17. Optimal design output results of the study.

Output Parameter	Candidate Point 1	Candidate Point 2	Candidate Point 3
η_t (%)	74.445	74.315	74.293
H_t (m)	15.088	15.107	15.095
η_s (%)	52.029	51.972	51.971
H_s (m)	13.558	13.59	13.573
SP (W)	14,888	14,914	14,891

3.3. Comparison of Results

In this section, the optimized design results will be compared to the baseline design model of the centrifugal pump. This will determine the improvement of the optimized design achieved by comparing their performance and highlighting the crucial aspects in the acquired results. The comparison will include the analysis of pressure distribution, amount of turbulence, and streamline diagram of the two designs to highlight the improvement of the optimized design from the baseline design model.

Figure 19 shows the difference between the pressure contour of the baseline and the optimized design. An improved pressure distribution can be seen in the contour plot of the optimized design, where the amount of pressure in the impeller and the volute is closely similar compared to the pressure distribution of the baseline design. A more reasonable pressure distribution results in improved stability regarding the relative motion of the impeller and the volute. Hence, this results in further improvement of the pump's performance and the flow dynamics of the fluid within the pump.

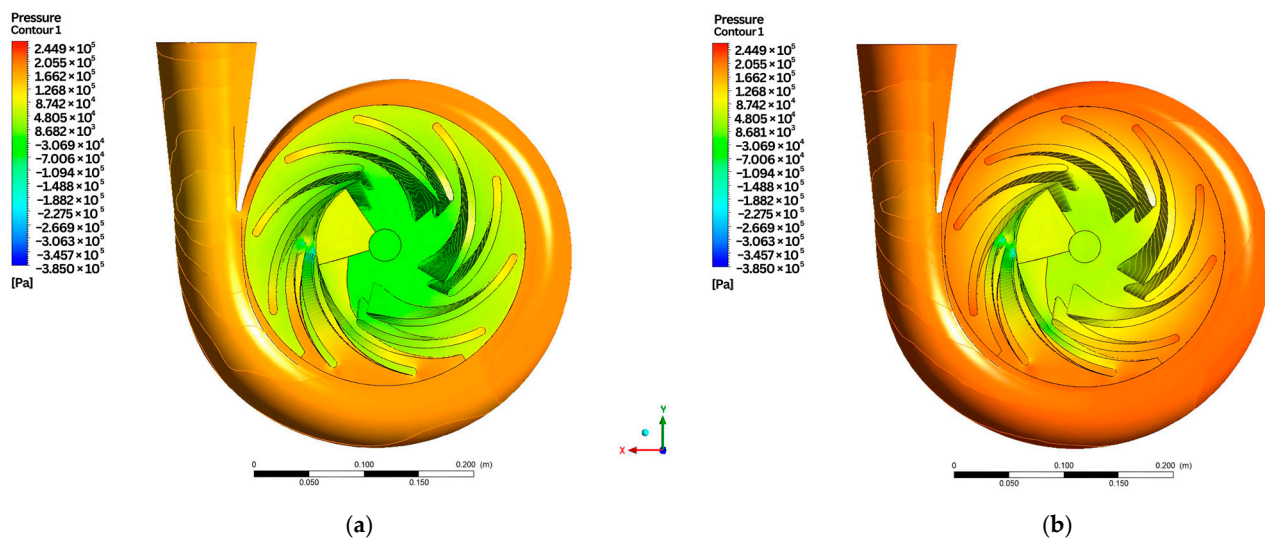


Figure 19. The pressure contour plot of the (a) baseline design vs. (b) the optimized design.

In Figure 20, a comparison between the meridional surface of the blade for the baseline design and the optimized design can be observed. There are huge changes involved that can be analyzed by just looking at the contour plots of the presented figures. In the optimized design, it shows that the low pressure in the middle section of the blade was significantly reduced compared to what can be observed in the baseline design. Also, the leveling of the pressure contour plots in the surface of the blade shows to be properly distributed. In terms of the amount of pressure subjected to the surface of the blade, the optimized model is shown to have an improvement since the levels of pressure are more significant. This implies that when the pressure from the flow of the fluid is applied on the surface of the blade, it will be properly distributed, which will help achieve a consistent flow of the fluid within the impeller.

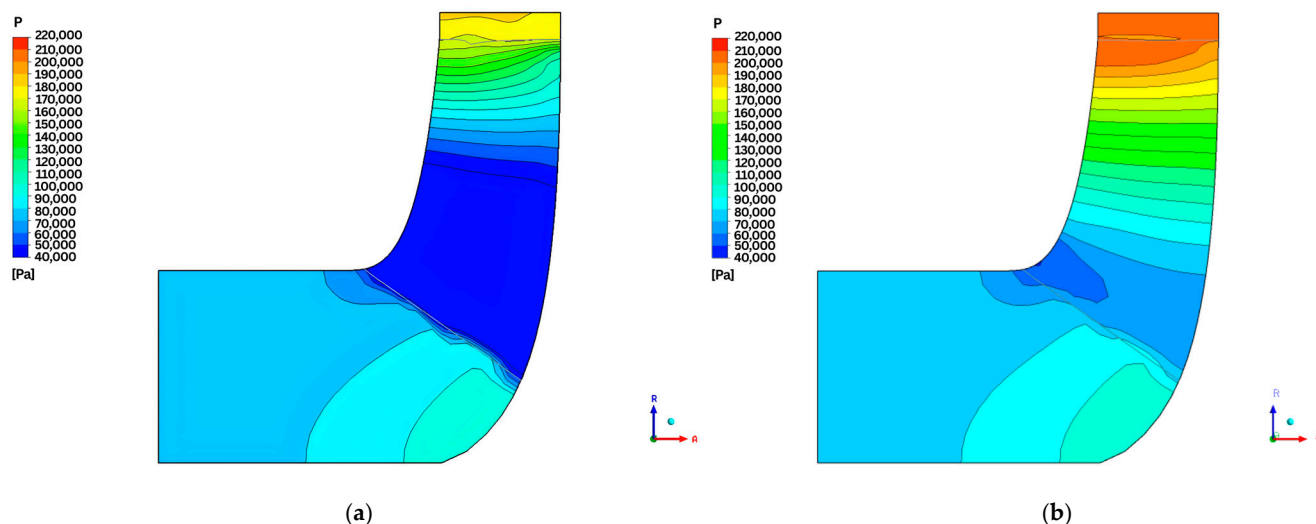


Figure 20. The area-averaged pressure contour on the meridional surface of the (a) baseline design vs. (b) the optimized design.

To visualize the amount of pressure inside the pump, the pressure cloud contour provides a clear representation of the pressure distribution for each pump component. Figure 21 shows the difference between the pressure distribution of the baseline design and the optimized design. A large gap of pressure between the impeller and the volute can be seen in the baseline design. This shows an imbalanced pressure distribution between the two main components of the pump. The imbalanced pressure attained in the baseline design contributes to the unsteady flow of the fluid due to an inconsistent transfer of energy. In the optimized design, the pressure distribution shows improvement as the impeller and the volute have a nearly similar amount of pressure attained, judging by the color contour of the optimized design. The impeller–volute interaction plays a crucial role in improving the pump’s performance as it enhances the behavioral flow of the fluid. Therefore, a uniform pressure distribution between the volute and the impeller will reduce pressure fluctuation and increase the hydraulic efficiency of the pump.

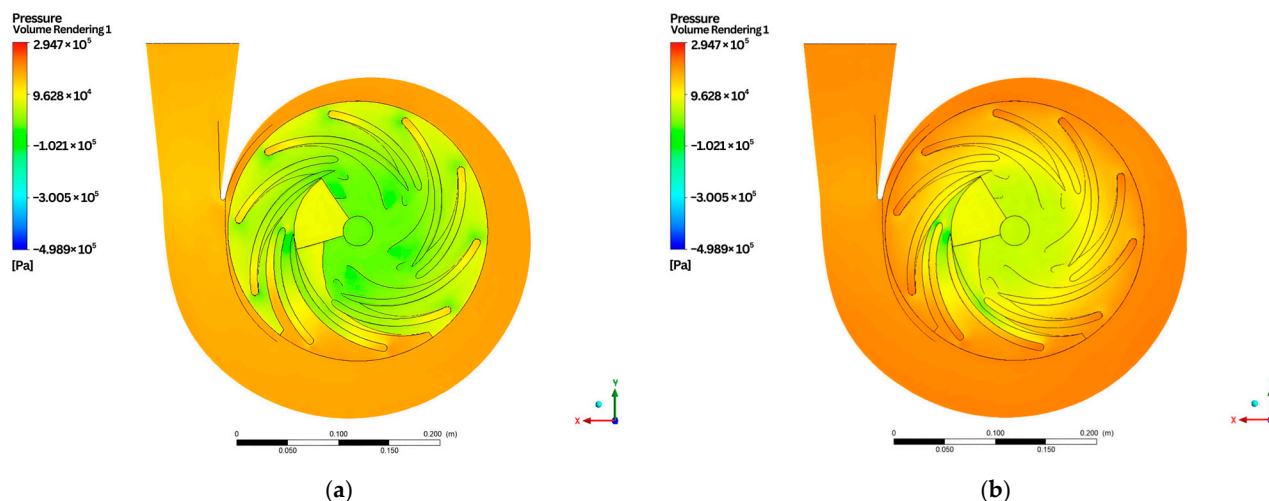


Figure 21. The pressure cloud contour of the (a) baseline design vs. (b) the optimized design.

The eddy viscosity contour plot of the baseline design and the optimized one are shown in Figure 22. The baseline design shows a higher eddy viscosity within its system. Colored spots, which correspond to higher eddies, are present between the impeller blades of the pump. This means there is an excessive amount of turbulence in the impeller, which

affects the flow of the fluid. In the optimized design, the number of eddies is reduced to a minimum level, which means that the turbulence affecting the fluid flow is reduced. The number of eddies is also well distributed compared to the baseline design. This corresponds to the amount of turbulence that results in an efficient change in pressure and velocity within the fluid, which leads to minimizing energy losses. Additionally, eliminating turbulence helps reduce the energy required to pump the fluid, reducing the work exerted by the system. Thus, more efficient power consumption can be achieved.

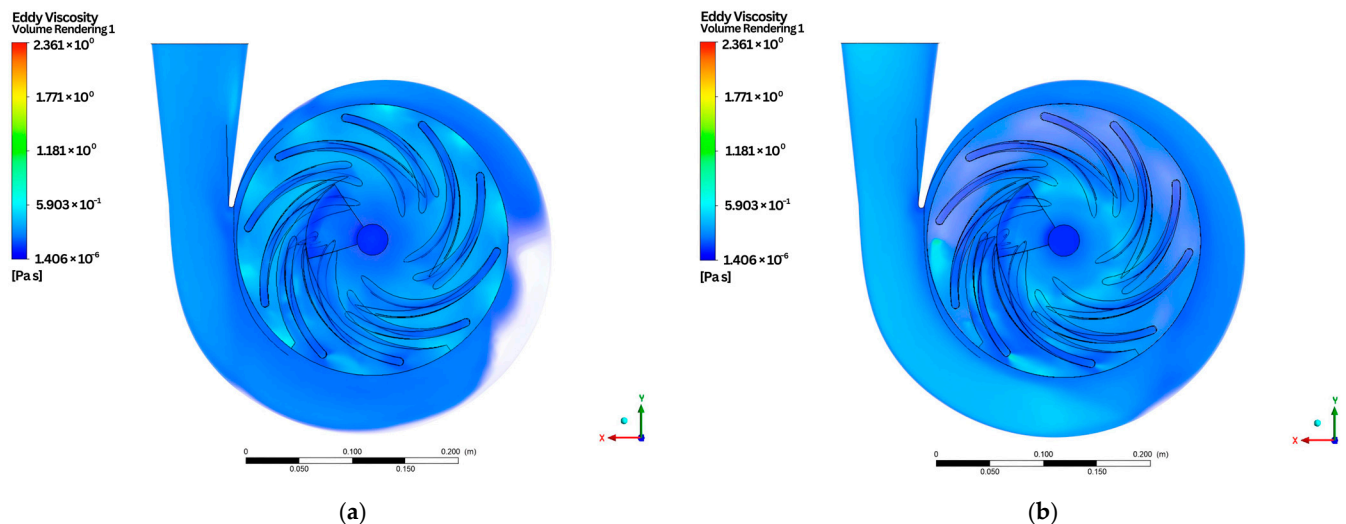


Figure 22. The eddy viscosity cloud contour of the (a) baseline design vs. (b) the optimized design.

The difference in fluid flow between the baseline and the optimized design is shown in Figure 23. It is evident in the given figure that the flow of fluid was improved in the optimized design. The fluid flow in the baseline design forms an indefinite pattern, which depicts that the fluid flow in the impeller is inconsistent, and the velocity reduces due to the irregularity of the flow pattern of the fluid. This irregularity has a great effect on the pump, resulting in inefficient performance. In terms of the optimized design, the scattered flow pattern of the fluid is shown to be reduced, and the velocity of the affected flow path also improves, judging by the consistency of the flow from the tip of the impeller blades. Therefore, the optimized design promotes a better fluid flow pattern than the baseline design, as it depicts flow stability in the inlet and outlet of the impeller. This enhances the hydraulic efficiency of the pump as the optimized design contains a better flow path of the fluid.

To visualize the improvement of the flow behavior of the fluid, an overall view of the streamline is provided in Figure 24. The streamline diagram shows the fluid's behavior at the impeller and the pump's volute. In the baseline design model, huge gaps can be observed in the streamlines, specifically in the impeller area. Other areas also seem to not be entirely covered by these streamlines, which may result in a decrease in the velocity of the fluid. These gaps that are not covered by the streamlines depict an inconsistent flow of the fluid throughout the subjected areas. By comparing the streamline flow within the impeller for both the baseline and optimized design, it shows that in optimized design, the flow from the inlet of the impeller up to the volute is properly directed compared to that of the baseline design. This finding may have caused the reduction in pressure distribution in the fluid flow, as observed in the previous results and discussions. On the other hand, the streamline diagram shown in the optimized design is highly compact compared to the streamline diagram of the baseline model. By looking closely at the difference between the two streamline diagrams, the fluid circulation is much more consistent on the optimized design compared to the fluid flow in the baseline design. The stableness of the flow in the

optimized design results in an increase in the velocity of the fluid. This finding justifies how the pump's performance was improved judging by the behavior of the fluid within the pump.

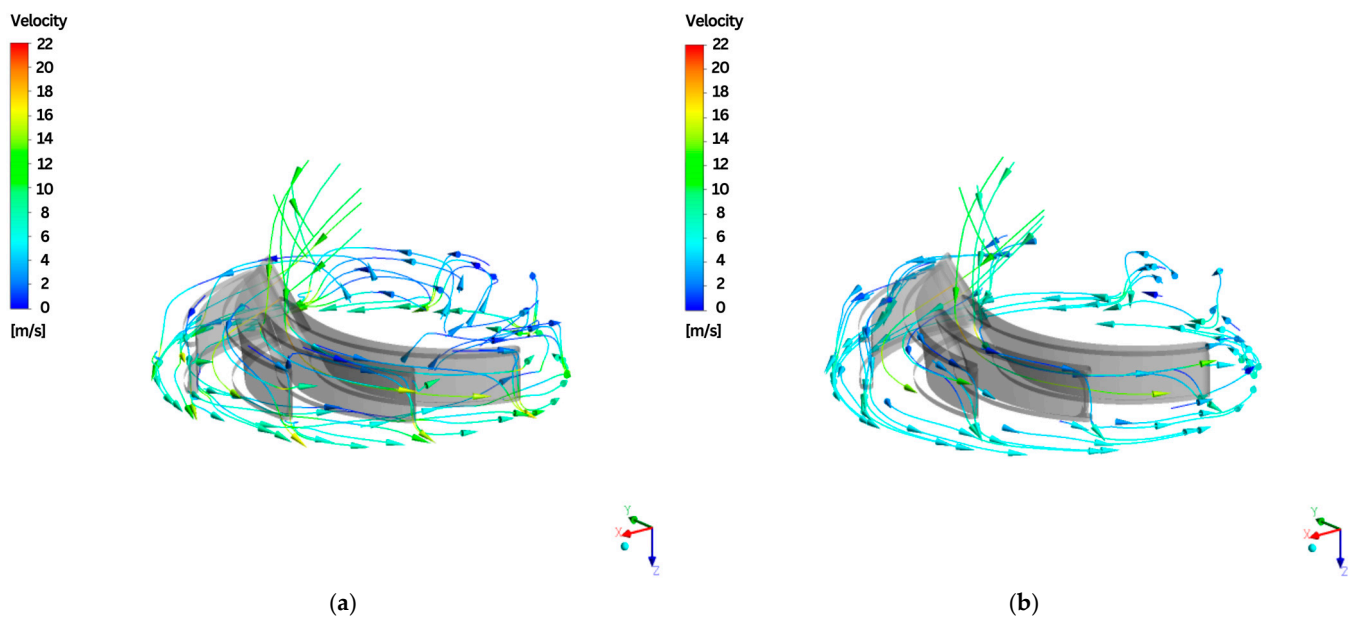


Figure 23. The velocity streamline diagram of the (a) baseline design vs. (b) the optimized design.

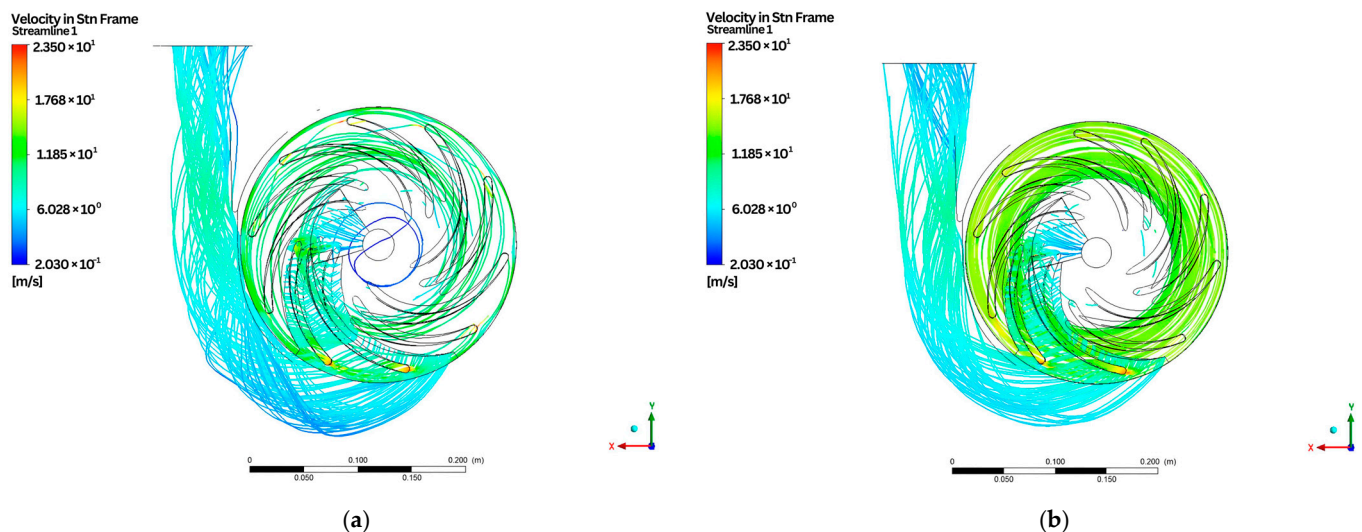


Figure 24. The overall velocity streamline diagram of the (a) baseline design vs. (b) the optimized design.

To determine the percentage improvement of the optimized design from the baseline design, the comparison of results and their corresponding input parameters are listed in Table 18. Through this table, it can be determined how much the initial values of the input parameters have changed by simply comparing the corresponding values for both the baseline and optimized design and see how much the design improved through the comparison of the output parameters for both designs. The η_t obtained a 27.35% improvement, with 15.70% on the H_t . The η_s improved by 28.18%, with 16.67% on the H_s . The optimized design achieved a reduced SP , with an 8.36% improvement. The ability to minimize the power consumption of the pump and increase its operating efficiency provides a clear alternative in enhancing the design of centrifugal pumps in water supply

and distribution applications. The findings of this study provide a new design approach that will help deviate from the traditional design of centrifugal pumps. Additionally, having a well-designed centrifugal pump offers practicality as it corresponds to having low maintenance cost while having an increased usable life of the pump.

Table 18. Comparison of results for baseline and optimized design.

Input Parameter	Baseline Design	Optimized Design	Percentage Improvement (%)
<i>spltr_OPF</i>	0.5	0.45	-
<i>LE_erh</i>	2	2.110	-
<i>LE_ers</i>	2	1.810	-
<i>TE_erh</i>	1	1.015	-
<i>TE_ers</i>	1	0.901	-
<i>clear</i> (mm)	13.189	12.994	-
<i>thk</i> (mm)	5.082	5.579	-
Output Parameter			
η_t (%)	58.458	74.445	27.35
H_t (m)	13.041	15.088	15.70
η_s (%)	40.592	52.029	28.18
H_s (m)	11.621	13.558	16.67
<i>SP</i> (W)	16,246	14,888	8.36

4. Conclusions

After subsequent performance analysis and result observation between the baseline and optimized centrifugal pump design, an insightful conclusion can be formulated:

- The result established that *spltr_OPF* should maintain a value of less than 0.5. As per the result of the optimized design, the *spltr_OPF* should be at a value of 0.45 as this value improves the η_t , H_t , η_s , and H_s while also minimizing the *SP*. This value also coincides with the graphs presented in Figure 12.
- Based on the analysis conducted on the graphs presented in Figure 13, the performance of the pump can be improved when the *thk* increases while maintaining the *clear* at a minimum level. This analysis shows to be coherent with the results of the *clear* and *thk* provided in Table 18. In the table, the value of the *thk* increases from 5.082 mm to 5.579 mm while having a reduced value for the *clear* from 13.189 mm to 12.994 mm. This explains the significance of the optimization results. Having these modifications for the volute tongue improves the dynamics between the impeller and the volute, enhancing the pump's hydraulic efficiency. This finding highlights the significance of the impeller–volute interaction. Hence, it can be concluded that the volute of the pump should also be considered when modifying the impeller.
- For the LE and TE ellipse ratios, improvements in the performance of the pump, such as the η_t , H_t , η_s , and H_s were achieved when both *LE_erh* and *TE_erh* increased and both the *LE_ers* and *TE_ers* decreased from their initial values of 2 and 1 for LE and TE, respectively.
- Reducing the *SP* requires different LE and TE configurations. In LE, setting the *LE_erh* and *LE_ers* to a value greater than their initial value of 2 will help minimize the power required by the shaft. On the other hand, the *TE_erh* should maintain a value closer to 1.06, and there should be an increasing value of the *TE_ers* that is greater than 1 to minimize the *SP*.
- Configuring the LE and TE of the splitter blades to reduce the *SP* to its full extent may compromise the efficiencies and heads due to the differences in their required

configurations. Therefore, finding the perfect balance between these parameters is essential to achieve a desirable pump performance with conflicting parametric objectives and constraints.

- The optimal design of the study resulted in 27.35%, 15.70%, 28.18%, and 16.67% improvement in η_t , H_t , η_s , and H_s , respectively, while achieving an 8.36% decrease in SP .
- Although this study was able to obtain improvement with the design of centrifugal water pump, the need for actual implementation of this study remains to be a possibility for further research. Researchers may consider applying the findings of this study to an actual setup where changes in temperature and sudden drops in pressure may become a challenging aspect for future work.

5. Recommendation

The study employed numerical analysis to investigate the results and develop an optimal centrifugal pump design. With this, the acquired results of the study were solely based on numerical approximations. Thus, the outcome of implementing the study in a real-world situation may deviate from the findings gathered in this study. Future studies may incorporate other existing optimization techniques to determine which technique generates a solution relatively closer to the results when implemented in a real-world setup. It is also recommended that researchers consider the type of material to be used in the design of the pump's impeller and volute in the scope of the study. In this way, the effects on the pump's performance can be observed when the stress induced in the material of the pump is reduced. Additionally, the study was performed under normal water conditions, and the change in temperature of the water was neglected. To conduct a complete evaluation of the behavior of the fluid within the pump, considering the changes in the properties of water, such as the temperature, should be included in the analysis. Moreover, future researchers may uncover other design alternatives that will reduce the computational time needed to generate the optimal solution for the study.

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Nomenclature

The following is the list of symbols used in this manuscript:

$clear$	volute tongue clearance
g	gravity, 9.81 m/s ²
H_s	static head
H_t	total head
H_{t-s}	total-static head, 8.069 m
k	turbulence kinetic energy
LE_{erh}	leading edge ellipse ratio at hub
LE_{ers}	leading edge ellipse ratio at shroud
p	static pressure
Q	volume flow rate, 0.084 m ³ /s

SP	shaft/input power
$spltr_OPF$	splitter blades' offset pitch fraction
TE_{erh}	trailing edge ellipse ratio at hub
TE_{ers}	trailing edge ellipse ratio at shroud
thk	volute tongue thickness
U	mean fluid speed
u	flow velocity
ν_t	kinematic viscosity
ρ	density of fluid
ρ_w	density of water, 997 kg/m ³
η_s	static efficiency
η_t	total efficiency
ω	specific dissipation rate

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