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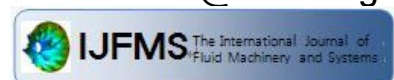
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Original Paper

Theoretical study on the efficiency of axial flow submersible pumps considering the influence of diffuser cone angle

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Abstract

This study presents a combined theoretical and experimental investigation into how the diffuser cone angle affects the hydraulic efficiency of submersible axial-flow pumps. Unlike conventional axial-flow designs, submersible configurations often require larger diffuser expansion angles due to structural constraints, potentially causing severe flow separation and energy losses. A theoretical model is proposed to quantify head loss due to cone angle variation, based on boundary layer theory and diffuser geometry. This model is validated through experimental testing on a full-scale HTCĐ 3000-3 pump under varying diffuser cone angles (0°, 42°, 60°). Results show that hydraulic efficiency (η_{hyd}) drops from 85.0% at 0° to 67% at 60°, with theoretical predictions deviating from experimental results by less than 2.3%. The findings highlight the critical role of diffuser geometry in optimizing pump design for high-flow, compact systems, offering a reliable tool for energy-efficient design in submersible pumping applications.

Keywords: Submersible pump, diffuser cone angle, hydraulic efficiency, axial-flow pump, head loss modeling, flow separation.

1. Introduction

Submersible axial-flow pumps are commonly used in irrigation, drainage, and water supply systems, where high flow rates and compact design are essential. Their performance is strongly influenced by diffuser geometry, especially the cone angle, which affects flow deceleration and pressure recovery. While conventional pumps use small cone angles (typically less than 8°) to minimize energy loss (Fig. 1b), submersible designs often require much larger angles (angles exceeding 30° (Fig. 1a)) due to structural constraints, increasing the likelihood of flow separation, vortex formation, and reduced efficiency.

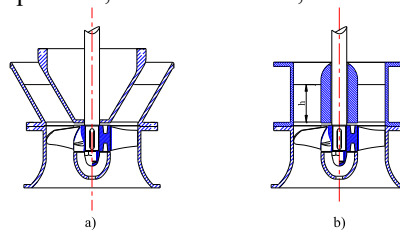


Fig. 1. Structure of the diffuser section

(a) Axial flow submersible pump (b) Conventional axial flow pump

Numerous studies have investigated how diffuser geometry affects pump efficiency. Hardial et al. [1] showed that even small expansion angles can cause flow separation at low Reynolds numbers, while Nguyen et al. [2] and Kim et al. [3] highlighted the importance of optimizing guide vanes and limiting cone angles to reduce turbulence and performance loss. Idelchik [4] also proposed optimal expansion angle limits to prevent separation.

Efficiency improvements have been reported through design optimization, such as blade and diffuser adjustments [5] and refined guide vane profiles [6]. Despite these advances, the impact of large-angle diffusers on hydraulic losses in submersible pumps remains

underexplored. To address this, the present study introduces a theoretical model to quantify head loss associated with large diffuser cone angles, validated through full-scale experiments. This approach improves predictive accuracy and provides practical insights for pump design and operation.

2. Methodology

2.1. Theoretical model development

The hydraulic performance of axial-flow submersible pumps is influenced by several factors, with the diffuser cone angle being one of the most significant. In conventional pumps, hydraulic losses are categorized based on flow conditions at the inlet, impeller, diffuser, and volute casing. However, in submersible pumps, the diffuser section experiences additional localized losses due to the steep expansion angle. The total head loss (h_{loss}) as the sum of the following components:

$$h_w = h_{in} + h_{imp} + h_{dif} + h_{vol} + h_{misc} \quad (1)$$

Where:

h_{in} : Losses associated with the fluid entering the pump, including losses due to the inlet geometry and flow development (m).

h_{imp} : Losses occurring within the impeller, such as shock losses, friction losses, and diffusion losses (m).

h_{dif} : Losses in the diffuser section, primarily due to flow separation and turbulence as the fluid decelerates (m).

h_{vol} : Losses in the volute casing, including friction losses and energy dissipation due to the spiral flow path (m).

h_{misc} : Miscellaneous losses from other components like bearings, seals, and other auxiliary parts (m).

In axial flow submersible pumps experience a distinct localized head loss due to the steep diffuser cone angle. This additional loss is denoted as h_γ and modifies the general head loss equation as follows:

$$\Delta H = \sum h_w + h_\gamma \quad (2)$$

Where h_γ accounts for the additional losses caused by the large expansion angles in the diffuser, leading to higher energy dissipation.

Quantifying hydraulic losses in axial-flow submersible pumps is challenging due to the strong effect of the diffuser cone angle (γ). This study presents a hybrid analytical–empirical method that combines conventional pump performance correlations with a theoretical loss model for expanding pipes to estimate losses from steep diffuser geometries. Experimental validation on a physical test rig was used to calibrate the model, improving prediction accuracy and providing a framework to link theoretical estimates with real-world pump performance.

2.2. Theoretical background: Theoretical model for gradually expanding pipe

Consider the flow through a section with a gradually expanding cross-section, where the expansion angle is θ (as shown in Figure 2). When the expansion angle (θ) exceeds the optimal limit, flow separation can occur, leading to the formation of vortex regions within the flow channel. This phenomenon results in a reduction of the hydraulic efficiency of the system, as illustrated in Figure 2.

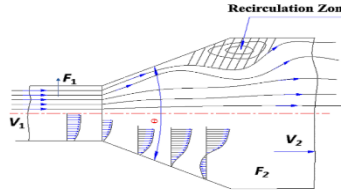


Fig. 2. Velocity profiles at different cross-sections of the diffuser section

Flow separation occurs when the kinetic energy within the boundary layer is insufficient to counteract the positive axial pressure gradient. The onset of separation is influenced by boundary layer thickness (δ) and the axial pressure gradient ($\partial p / \partial x$). To characterize this phenomenon, [7] introduced a dimensionless geometric parameter:

$$\Gamma' = \frac{dp}{dx} \frac{\delta}{\tau_0} \quad (3)$$

The expression represents the ratio of axial pressure drop dp/dx over a segment dx to the boundary layer thickness δ , divided by wall shear stress τ_0 reflecting the boundary layer's ability to maintain steady flow. Reducing the diffuser expansion angle (θ) while keeping a constant inlet-to-outlet area ratio (F_1/F_2) increases the diffuser length (l), raising frictional losses but lowering separation losses. An optimal (θ) exists to minimize total hydraulic resistance, typically 7° – 8° for circular or square cross-sections and 10° – 12° for rectangular ones.

By applying the energy equation to the flow through the diffuser section, the pressure increases (or head loss in the diffuser section, denoted as h_θ) is determined by the following expression:

$$\frac{p_2 - p_1}{\gamma} = \frac{v_1^2 - v_2^2}{2g} - h_\theta \quad (4)$$

The hydraulic resistance coefficient ξ of the diffuser section is determined by the following expression:

$$\xi = \frac{2gh_\theta}{v_1^2 - v_2^2} \quad (5)$$

In practical design, the total resistance coefficient is commonly expressed as:

$$\xi = \frac{2gh_\theta}{v^2} \quad (6)$$

Additionally, the loss coefficient k_θ is defined as the ratio between the head loss in the diffuser section and the theoretical loss due to impact according to the Borda–Carnot model [8], as shown by the following equation:

$$k_\theta = \frac{2gh_\theta}{(v_1 - v_2)^2} \quad (7)$$

The loss coefficient k_θ holds practical relevance as it remains independent of both the diffuser length and the inlet-to-outlet area ratio for a given expansion angle (θ). Values of (θ) above 8° in circular diffusers cause flow separation and performance loss. In axial-flow submersible pumps, a large motor downstream requires steep ($\theta \gg 8^\circ$). This constraint significantly increases hydraulic losses and must be considered in design and calibration.

From the above equation, we can determine the loss caused by the expansion angle θ as follows:

$$h_\theta = k_\theta \frac{(v_1 - v_2)^2}{2g} \quad (8)$$

Where:

v_1, v_2 : Flow velocities at the inlet and outlet cross-sections of the diffuser section (m/s); g : Gravitational acceleration (m/s^2); θ : Expansion angle (degrees); k_θ : Loss coefficient due to the expansion angle, determined by the empirical relationship [10]:

$$k_\theta = 0,011\theta^{1,22} \quad (9)$$

2.3. Calculation of head loss due to the diffuser angle γ in axial flow submersible pumps

Using the head loss formula for the diffuser and substituting $\theta = \gamma$ and $h_\theta = h_\gamma$, the hydraulic loss from the diffuser angle is expressed as follows:

$$h_\gamma = k_\gamma \frac{(v_1 - v_2)^2}{2g} \quad (10)$$

Where:

$$k_\gamma = 0,011\gamma^{1,22} \quad (11)$$

The formulation allows estimation of head loss h_γ from abrupt cross-sectional expansion in axial-flow submersible pumps. Accurate h_γ quantification is essential for performance optimization and design calibration.

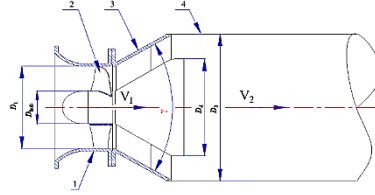


Fig.3. Schematic for calculating hydraulic loss through the guide vane system of a submersible axial-flow pump with expanding angle γ

The flow velocity v_1 at the inlet of the guide vane (the axial flow velocity from the impeller) is determined by the following equation:

$$v_1 = \frac{4Q_P}{\pi D_1^2 (1 - D_{hub}^2)} = \frac{4Q_P}{\pi (D_1^2 - D_{hub}^2)} \quad (12)$$

D_1 is the outer diameter of the impeller (m); D_{hub} is the diameter of the impeller hub (m); Q_P is the design flow rate of the pump (m^3/s); The flow velocity v_2 at the outlet of the guide vane is calculated by:

$$v_2 = \frac{4Q_P}{\pi (D_3^2 - D_4^2)} \quad (13)$$

D_3 is the outer diameter of the guide vane outlet (m); D_4 is the outer diameter of the guide vane hub (m).

The diameter D_4 varies with motor flange design and is typically larger in Vietnam, increasing the guide vane expansion angle (γ). This leads to higher hydraulic losses. The total hydraulic loss in the flow-guiding system is calculated by:

$$H = h_w + h_\gamma = k_1 k_2 H_P + k_\gamma \frac{(v_1 - v_2)^2}{2g} \quad (14)$$

Where: k_1, k_2 : Empirical loss coefficients; H_P : Design total head of the pump (m).

2.4 Hydraulic efficiency in axial-flow submersible pumps

Similar to conventional axial-flow pumps, the hydraulic losses in an axial-flow submersible pump are determined by the difference between the theoretical head and the total hydraulic loss, as expressed by the formula:

$$\eta_{hyd} = \frac{H_{th} - \Delta H}{H_{th}} = 1 - \frac{\Delta H}{H_{th}} \quad (15)$$

η_{hyd} : Hydraulic efficiency (%); H_{th} : Theoretical head of the pump (m).

By substituting the loss expressions into the above equation, the hydraulic efficiency of the axial-flow submersible pump is given by:

$$\eta_{hyd} = 1 - k_1 - \frac{0,011\gamma^{1,22}(v_1 - v_2)^2}{2gk_2H_P} \quad (16)$$

2.5 Overall efficiency of a submersible axial-flow pump

The overall efficiency of a submersible axial-flow pump, similar to that of conventional axial-flow pumps, is the product of three components: mechanical efficiency η_m , volumetric efficiency η_{vol} , and hydraulic efficiency η_{hyd} :

$$\eta_P = \eta_m \eta_{vol} \eta_{hyd} = \eta_m \eta_{vol} \left(1 - k_1 - \frac{0.011 \gamma^{1.22} (v_1 - v_2)^2}{2gk_2 H_P} \right) \quad (17)$$

Equation (17) forms a theoretical basis for assessing pump efficiency, highlighting the impact of guide vane expansion angle (γ). The model assumes steady, single-phase flow, constant diffuser geometry, and ignores thermodynamic complexities.

3. Input parameters and scope of analysis

The study examined the effect of diffuser cone angle (γ) on pump performance while keeping other parameters constant. Three configurations were analyzed: $\gamma = 60^\circ$ (steep), $\gamma = 42^\circ$ (moderate), and $\gamma = 0^\circ$ (no expansion) (fig 3).

The theoretical calculations were based on the following fixed design specifications: Pump model: HTCD 3000-3 with technical special to design: $N = 55$ kW; $n = 590$ rpm; $Q_P = 3.300$ m³/h; $H_P = 3$ m

The geometric dimensions and parameters are provided in Figure 3 and Table 1.

Table 1. Basic dimensions of the guide vane system for different diffuser angle cases

No.	γ (°)	D_1 (m)	D_{hub} (m)	D_4 (m)	D_3 (m)
1	60°	0.58	0.23	0.67	1.1
2	42°	0.58	0.23	0.67	1
3	0°	0.58	0.23	0.23	0.58

Using the loss and efficiency expressions derived in Section 2.3 (Equations 10, 11, and 16), the impact of γ on hydraulic losses and pump performance was calculated.

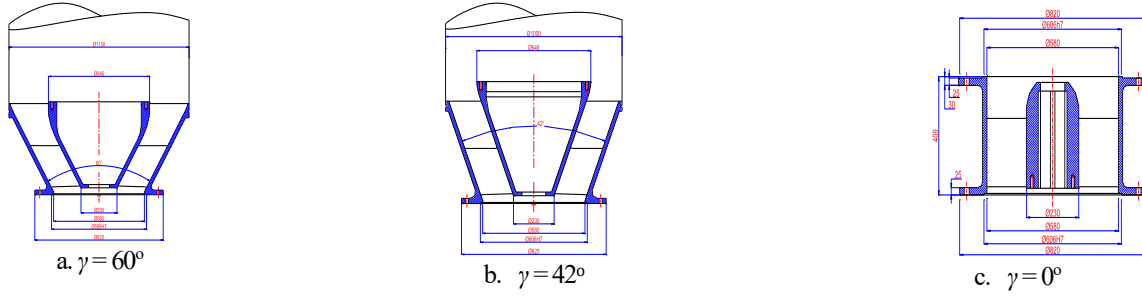


Fig. 4. Structure of the diffuser section in the submersible axial-flow pump

4. Results and discussion

4.1 Theoretical calculation results

Table 2. Efficiency and hydraulic parameters for HTCD 3000-3 pump

γ (°)	η_P (%)	η_m (%)	η_{vol} (%)	η_{hyd} (%)	k_1	k_2	H_P (m)	k_γ	V_1 (m/s)	V_2 (m/s)
60	63.79	97	98	67.0	0.15	1.08	3	1.63	4.12	1.47
42	73.77	97	98	77.0	0.15	1.08	3	1.05	4.12	2.00
0	80.8	97	98	85.0	0.15	1.08	3	0	4.12	4.12

Table 2 presents the calculated hydraulic and overall efficiency parameters for the HTCD 3000-3 pump model under the three diffuser configurations.

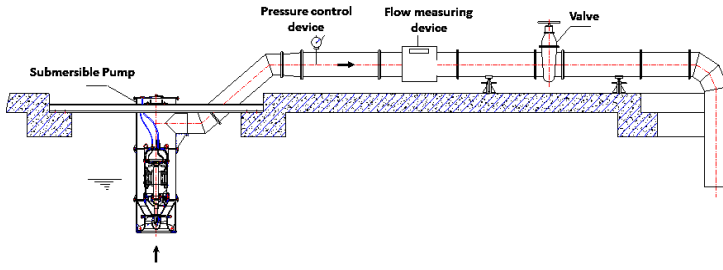
The results demonstrate a clear trend: increasing the diffuser angle (γ) leads to a reduction in hydraulic efficiency due to enhanced flow separation and associated energy losses. Specifically, η_{hyd} decreased from 85% in the non-expanded configuration (0°) to 67% in the steep-angle configuration (60°), as reflected in the higher loss coefficient ($k_\gamma = 1.63$) for $\gamma = 60^\circ$. The 42° diffuser configuration (Case 2) represents a practical compromise, achieving a hydraulic efficiency of 77% and an overall efficiency of 73.77%.

4.2 Experimental results

To validate the theoretical model, experimental tests were performed on three HTCD 3000-3 model pumps with identical operating parameters ($N = 55$ kW, $n = 590$ rpm, $Q = 3,000$ m³/h), γ values (60° , 42° , 0°) and exhaust pipe diameter DN600mm.

Test Setup

The experimental setup is illustrated in Fig. 5. Performance parameters, including flow rate, head, absorbed power, and efficiency, were recorded and analyzed for each configuration. Tables 3, 4, and 5 summarize the key technical specifications obtained from the experimental tests.



a) Test schematic diagram



(b) HTCĐ 3000-3 submersible axial-flow pump

Fig. 5. Testing setup for pumps at Hai Duong pump manufacturing JSC

Table 3. Technical specifications of HTCĐ 3000-3 pump with $\gamma = 60^\circ$

n	U	I	N_{motor}	N_{shaft}	Q_p	V	H_p	N_{hyd}	η_{total}
rpm	V	A	kW	kW	m ³ /h	m/s	m	kW	%
590	390	59	25.77	22.67	2.090	2.05	1.74	9.91	43.60
590	390	58	26.15	23.01	2.017	1.98	2.15	11.82	50.00
590	390	58	26.29	23.14	1.903	1.87	2.60	13.48	58.10
590	390	61	27.23	23.97	1.751	1.72	3.19	15.22	63.50
590	390	67	30.31	26.67	1.617	1.59	3.61	15.91	60.07
590	390	69	31.15	28.29	1.356	1.33	3.65	16.52	58.00

Table 4. Technical specifications of HTCĐ 3000-3 pump with $\gamma = 42^\circ$

n	U	I	N_{motor}	N_{shaft}	Q_p	V	H_p	N_{hyd}	η_{total}
rpm	V	A	kW	kW	m ³ /h	m/s	m	kW	%
590	390	90	42.56	37.88	3.730	3.67	2.13	21.60	57.03
590	390	94	44.45	39.56	3.580	3.52	2.37	23.13	58.47
590	390	96	45.39	40.40	3.490	3.43	2.54	24.15	59.79
590	390	99	46.81	41.66	3.380	3.32	2.90	26.73	64.17
590	390	104	48.94	43.56	3.310	3.17	3.25	28.64	65.75
590	390	109	51.54	45.87	2.960	2.91	3.67	29.61	64.56
590	390	115	54.38	48.40	2.760	2.71	4.02	30.20	62.40
590	390	118	55.80	49.66	2.308	2.27	4.51	28.18	56.75
590	390	120	56.97	50.75	2.070	2.03	5.06	28.54	55.95

Table 5. Technical specifications of HTĐ 3000-3 pump with $\gamma = 0^\circ$

n	U	I	N_{motor}	N_{shaft}	Q_p	V	H_p	N_{hyd}	η_{total}
rpm	V	A	kW	kW	m ³ /h	m/s	m	kW	%
590	390	90	42.56	37.88	3.986	3.92	2.22	24.14	63.73
590	390	92	43.50	38.72	3.750	3.69	2.46	25.16	64.99
590	390	93	43.98	39.14	3.575	3.51	2.75	26.78	68.44
590	390	95	44.68	39.77	3.353	3.30	3.14	28.72	72.33
590	390	98	46.34	41.24	3.262	3.21	3.32	29.55	71.64
590	390	99	46.81	41.66	3.105	3.05	3.46	29.32	70.36

The experimental results show that the overall pump efficiency varies with the diffuser vane angle γ . The pump with a 60° diffuser achieved a unit efficiency of $\eta_{\text{total}} = 58.10\%$, corresponding to a pump efficiency $\eta_p = 64.55\%$ (assuming motor efficiency $\eta_{\text{motor}} = 90\%$). The 42° diffuser configuration showed a higher unit efficiency of $\eta_{\text{total}} = 65.75\%$, yielding $\eta_p = 73.05\%$. The conventional axial-flow pump with no diffuser expansion (0°) attained the highest performance with $\eta_p = 80.25\%$.

Figures 7, 8, and 9 present the performance curves of the HTCĐ 3000-3 pump for the 60° , 42° , and 0° diffuser configurations, respectively. These findings validate the theoretical predictions and underscore the critical role of diffuser geometry in axial-flow pump optimization. The experimental agreement enhances confidence in the proposed model's applicability to design and performance forecasting.

Comparison with theoretical results

Experimental results closely align with theoretical predictions, confirming the model's reliability in capturing key hydraulic behaviors. As shown in Table 4, larger diffuser angles significantly reduce efficiency due to flow separation and associated head loss.

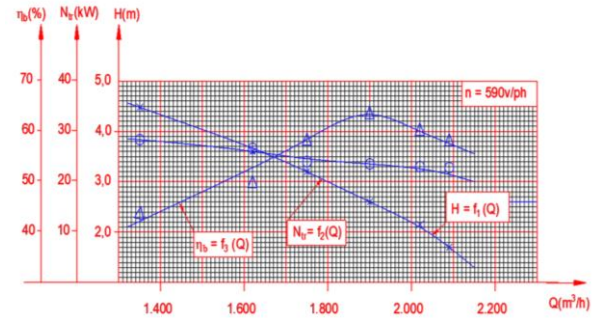


Fig. 6. Performance curve of HTCĐ 3000-3 submersible axial-flow pump with $\gamma = 60^\circ$

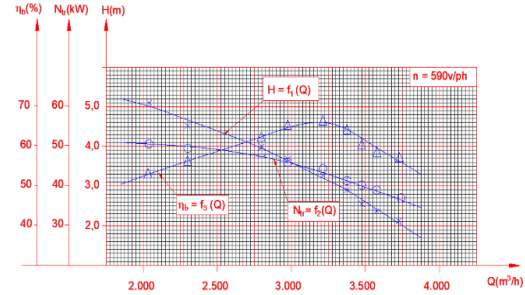


Fig. 7. Performance curve of HTCĐ 3000-3 submersible axial-flow pump with $\gamma = 42^\circ$

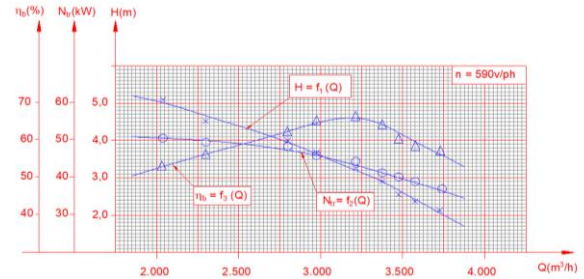


Fig. 8. Performance curve of HTĐ 3000-3 conventional axial-flow pump with $\gamma = 0^\circ$

Table 4. Comparison of theoretical and experimental results

γ	Theoretical results		Experimental results		Error	
	η_{hyd}	η_P	η_{hyd}	η_P	η_{hyd}	η_P
(°)	(%)	(%)	(%)	(%)	(%)	(%)
60	67.10	64.55	63.79	58.10	4.65	6.56
42	77.00	73.05	73.77	65.75	4.07	9.96
0	85.00	80.25	80.80	75.19	0.94	6.33

Hydraulic efficiency errors increase with larger diffuser angles, reaching 4.65% at 60° due to possible flow separation effects. The smallest error (0.94%) occurs at 0°, where flow conditions are simpler and more predictable.

Overall efficiency deviations peak at $\gamma = 42^\circ$ (9.96%), likely from combined mechanical, volumetric, and unsteady flow effects. Most deviations remain within 0.9% - 2.3%, influenced by measurement uncertainty, unsteady flow, and varying conditions [9].

Model limitations also stem from assumptions like steady-state and single-phase flow particularly for steep diffuser configurations [3]. These simplifications reduce accuracy, especially for steep diffuser configurations.

Future improvements should include unsteady flow modeling, better instrumentation, and multiphase considerations.

Comparisons with previous studies confirm the validity of these findings.

The combined theoretical–experimental approach provides a solid framework for pump optimization. The proposed model demonstrates reliable predictive accuracy for vertical submersible pumps with diffuser angles between 0° and 60°

However, broader applications will require CFD-based recalibration and further experimental validation.

5. Conclusion

In conclusion, the study confirms that diffuser cone angle significantly affects pump efficiency, with smaller angles yielding better performance. Specifically, the 60° diffuser configuration exhibited the lowest hydraulic efficiency (67.1%), while the 0° configuration achieved the highest performance (85.0%). A 42° diffuser was identified as a practical compromise, balancing spatial constraints with acceptable efficiency levels.

The proposed model shows high accuracy and serves as a useful design tool, though its applicability is limited by simplifying assumptions. Future work should expand the model using CFD and more complex flow conditions to improve its reliability and generalizability.

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Nomenclature

h_w	The total headloss [m]	N	Submersible electric motor power [kW]
H_p	Design total head of pump [m]	n	Rotational speed [rpm]
Q_p	The design flow rate of pump [m ³ /h]	θ, γ	Cone angle [degree]
η_{hyd}	Hydraulic efficiency [%]	v_1	The flow velocity at the inlet guide van [m/s]
v_2	The flow velocity at the outlet guide van [m/s]		

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