

FIELD-BASED HYDRAULIC PERFORMANCE AUDIT OF A CENTRIFUGAL PUMP IN A FUEL DISTRIBUTION TERMINAL: CAVITATION ASSESSMENT, CAPACITY EVALUATION, AND VARIABLE SPEED DRIVE OPTIMIZATION

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Abstract

Centrifugal pumps are the operational backbone of fuel distribution terminals, where unplanned downtime or hydraulic failure can cascade into critical supply chain disruptions affecting national energy logistics. Despite their strategic importance, site-specific hydraulic performance evaluations of fuel terminal pumping systems remain underrepresented in the literature, leaving operators reliant on design-stage assumptions rather than field-verified evidence for maintenance decisions and efficiency improvements. This study addresses that gap by conducting a comprehensive hydraulic performance audit of the MCPK 125-100-400 CC centrifugal pump operating within the fuel distribution system at PT Pertamina Patra Niaga Fuel Terminal Tegal, Indonesia. Field observations and operational data collection were performed to evaluate critical hydraulic parameters, including flow velocity, actual and design flow rates, Reynolds number, head loss distribution, Net Positive Suction Head Available (NPSHa), Net Positive Suction Head Required (NPSHr), and Bernoulli equation compliance across the suction and discharge lines. Results confirm that the pump delivers a capacity of 165 m³/h, exceeding the actual operational demand of 1,600 L/min and establishing an adequate reserve for uninterrupted truck loading operations. A calculated NPSHa of 5.23 m exceeds NPSHr of 2.64 m, demonstrating a positive cavitation safety margin of 2.59 m under current field conditions. While the discharge flow velocity of 5.6 m/s satisfies standard hydraulic design criteria, fixed-speed operation against fluctuating demand introduces measurable energy inefficiency. Integration of a Variable Speed Drive (VSD) is proposed as a targeted retrofit to align pump output with real-time demand, reduce energy consumption, and extend equipment service life. The hydraulic audit methodology presented here offers a replicable framework for performance-based pump evaluation in fuel distribution infrastructure.

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1. Introduction

The demand for petroleum fuels in Indonesia has grown steadily in tandem with rapid expansion of the transportation sector, industrial activity, and broader economic development, intensifying pressure on national fuel distribution infrastructure to operate with greater reliability and efficiency [1]. As a critical node within this infrastructure, PT Pertamina Patra Niaga Fuel Terminal Tegal is responsible for the receipt, storage, and distribution of petroleum fuels across its designated service area. The operational continuity of this terminal depends primarily on the performance of its centrifugal pumping systems, which transfer fuel from storage tanks to tanker truck loading bays, a function that directly governs throughput capacity and supply chain reliability [1].

Centrifugal pumps are the most widely deployed pump type in the oil and gas industry, favored for their simple construction, high volumetric flow capacity, and operational robustness. Nevertheless, their performance is highly sensitive to deviations between design conditions and actual operating conditions [2]. When pump capacity is mismatched with real system demand, the consequences include elevated energy consumption, increased head losses, reduced hydraulic efficiency, and the onset of cavitation — a phenomenon in which localized pressure drops cause vapor bubble formation and subsequent collapse, resulting in mechanical damage to pump impellers and casings [2]. These failure modes are particularly consequential in fuel terminal settings, where operational continuity is non-negotiable and unplanned equipment downtime has direct logistical and economic repercussions.

Prior research has examined centrifugal pump performance across various industrial contexts. Efficiency analyses of pumps in fuel transportation systems have emphasized the importance of regular performance evaluation to sustain operational efficiency [3]. Other studies have also reported that non-optimal operating conditions can lead to increased operating temperatures and a decline in pump performance in fuel distribution systems [4]. Studies in related sectors have further demonstrated that non-optimal operating conditions elevate fluid temperatures and degrade pump performance, while the overall effectiveness of fuel loading operations has been shown to depend critically on the condition of the pumping system and its supporting infrastructure [5].

However, a specific and persistent gap remains in the literature. Existing studies rarely combine field-measured capacity assessment, flow velocity compliance against the API RP 14E standard, and cavitation risk quantification through direct comparison of Net Positive Suction Head Available (NPSHa) and Net Positive Suction Head Required (NPSHr) within an integrated hydraulic audit framework applicable to active fuel distribution terminals. Most published evaluations are conducted under controlled or simulated conditions, limiting their direct applicability to real terminal operations where demand fluctuates and pipeline configurations differ from design assumptions.

This study therefore aims to provide a field-based hydraulic performance evaluation of the MCPK 125-100-400 CC centrifugal pump at PT Pertamina Patra Niaga Fuel Terminal Tegal. Specifically, the objectives are to: (1) determine whether the actual pump capacity satisfies operational demand; (2) evaluate pipeline flow velocity against API RP 14E design criteria; (3) quantify head losses across the suction and discharge lines; and (4) assess cavitation risk by comparing the calculated NPSHa and NPSHr values under current field conditions [6]. The findings are intended to provide a verified characterization of the pump's operational status and to establish a replicable audit framework for performance-based assessment of centrifugal pumps in critical fuel distribution facilities.

2. Method

This study was conducted at PT Pertamina Patra Niaga Fuel Terminal Tegal, with the research object being a centrifugal pump used in the distribution system of Pertamina fuel to the filling shed. A quantitative research method was employed through field data collection and hydraulic analysis of the pumping system.

A. Data Collection

Data were obtained through direct field observations, interviews with operators and personnel from the Maintenance Plant and Service (MPS) Division, and company documentation reviews. The collected data included pump specifications, pump capacity, pump rotational speed, pump head, pipeline diameter, pipeline length, loading volume, and tanker truck loading time. These data were subsequently used to evaluate the hydraulic performance of the pumping system.



Figure 1. MCPK 125-100-400 CC centrifugal pump and its technical specifications used in the Pertamina fuel distribution system at PT Pertamina Patra Niaga Fuel Terminal Tegal. *B. Pump Capacity Analysis*

The specification of the pump as described in **Table 1**.

Table 1 Pump Specification

Parameter	Value
Model	MCPK 125-100-400 CC
Serial Number	049/01801011-009
Year	2018
Pump Capacity	165 m ³ /h
Pump Speed	1,451 rpm
Head	52.83 m
Pipe Diameter	4 inch
Pipe Length	101 m
Filling Volume	8,000 L
Filling Time	5 minutes

The actual system capacity was determined by measuring the volume of fuel transferred over a specified time interval, then comparing the resulting flow rate against the pump design capacity to assess the extent to which the pump met the operational requirements of the fuel distribution system [7]. The actual flow rate of the pumping system was determined using the following equation.

$$Q = \frac{V}{t} \quad (1)$$

where Q is the actual volumetric flow rate (L/min), V is the total volume of fuel transferred during one filling cycle (L), and t is the time required to complete the filling cycle (min). To facilitate comparison with the pump design capacity, which is expressed in cubic meters per hour (m³/h), the actual flow rate was converted using the following unit conversion equation:

$$Q_{\text{design}} = \frac{Q \times 1.000}{60} \quad (2)$$

where Q_{design} is the converted flow rate in m³/h, Q is the actual flow rate in L/min, the factor 1,000 converts liters to cubic meters (1 m³ = 1,000 L), and the factor 60 converts minutes to hours. This conversion ensures dimensional consistency when evaluating the pump's capacity relative to its rated design value of 165 m³/h.

C. Flow Characteristics Analysis

The flow characteristics within the piping system were analyzed by calculating the flow velocity and Reynolds number. The flow velocity was determined based on the relationship between the flow rate and the pipe cross-sectional area, while the Reynolds number was used to identify the flow regime occurring within the system.

$$V = \frac{Q}{A} \quad (3)$$

where V is the mean flow velocity (m/s), Q is the volumetric flow rate (m³/s), and A is the internal cross-sectional area of the pipe (m²), calculated as $A = \frac{\pi D^2}{4}$, where D is the internal pipe diameter (m). The calculated flow velocity was subsequently compared against the recommended velocity range specified in API RP 14E to evaluate the operational safety and suitability of the piping system [8].

$$Re = \frac{V \cdot D}{\nu} \quad (4)$$

where Re is the dimensionless Reynolds number, V is the mean flow velocity (m/s), D is the internal pipe diameter (m), and ν is the kinematic viscosity of the fluid (m²/s). In fluid mechanics, the Reynolds number is defined as the ratio of inertial forces to viscous forces and is commonly used to characterize the flow behavior of a fluid [9].

- Laminar Flow $Re < 2.300$
- Transition Flow $2.300 - 4.000$
- Turbulent Flow $Re > 4.000$

D. Head Loss Analysis

Energy losses due to friction along the pipeline were quantified using the Darcy–Weisbach equation, which is the standard formulation for major head loss in pipe flow. The Darcy friction factor f was determined from the Moody diagram by considering both the Reynolds number and the relative roughness of the pipe wall, defined as ε/D , where ε is the absolute pipe roughness (m) and D is the internal pipe diameter (m) [7]. The major head loss due to pipe friction was calculated as:

$$H_f = f \times \frac{L}{D} \times \frac{V^2}{2g} \quad (5)$$

where H_f is the major head loss due to friction (m), f is the dimensionless Darcy friction factor obtained from the Moody diagram, L is the total pipe length (m), D is the internal pipe diameter (m), V is the mean flow velocity (m/s), and g is the gravitational acceleration (9.81 m/s²). The ratio L/D represents the relative pipe length, and the term $\frac{V^2}{2g}$ represents the velocity head of the fluid. Head loss calculations were performed across both the suction and discharge lines to determine the total energy dissipated during the fuel distribution process, providing the basis for subsequent NPSHa evaluation and overall system efficiency assessment. The head loss calculation was performed to determine the amount of energy lost during the fuel distribution process.

E. NPSH Analysis

Cavitation potential was evaluated by calculating the Net Positive Suction Head Available (NPSHa) and comparing it against the Net Positive Suction Head Required (NPSHr) specified by the pump manufacturer [10]. The system is considered safe from cavitation when NPSHa exceeds NPSHr; failure to satisfy this condition results in vapor bubble formation at the pump inlet, leading to progressive mechanical damage to the impeller and casing [8]. The NPSHa was calculated using the following equation:

$$NPSHa = h_p - h_{vpa} + h_{st} - h_f - h_{vh} - h_a \quad (6)$$

Where h_p is the absolute pressure head at the fluid surface in the suction tank (m), h_{vpa} is the vapor pressure head of the fluid at the operating temperature (m), h_{st} is the static suction head representing the vertical distance between the fluid surface and the pump centerline (m), h_f is the major head loss due to friction along the suction line (m), h_{vh} is the velocity head at the pump suction inlet (m), and h_a is the minor head loss due to fittings and valves along the suction line (m). The pump specific speed n_s was calculated to characterize the hydraulic type of the pump and to determine the Thoma cavitation coefficient:

$$n_s = n \frac{QN_2^{1/4}}{Hn^3} \quad (7)$$

where n_s is the dimensionless specific speed, n is the pump rotational speed (rpm), Q is the volumetric flow rate at the best efficiency point (m³/s), and H_n is the net head delivered by the pump at the best efficiency point (m). The critical cavitation head H_{cv} , representing the minimum head margin required to suppress cavitation, was then determined as:

$$H_{cv} = \sigma \times H_n \quad (8)$$

where H_{cv} is the critical cavitation head (m), σ is the Thoma cavitation coefficient (dimensionless) derived from the specific speed n_s , and H_n is the net pump head (m). A system is confirmed cavitation-free when $NPSHa > H_{cv}$, providing an additional verification beyond the direct $NPSHa$ – $NPSHr$ comparison.

F. Research Methodology Flowchart

The research was conducted through a structured sequential procedure designed to systematically evaluate the hydraulic performance of the centrifugal pump at PT Pertamina Patra Niaga Fuel Terminal Tegal. The process began with a literature review and field study, encompassing a review of centrifugal pump theory, relevant hydraulic analysis standards including API RP 14E, and direct observation of the existing pumping system configuration and operational conditions at the terminal. Following the field study, parameter data collection was carried out to obtain all inputs necessary for hydraulic calculations, including pump nameplate specifications, pipe dimensions, fluid properties, filling volume, filling time, and field measurements. The collected data were subsequently processed through sequential hydraulic calculations covering actual flow rate, design flow rate, flow velocity, Reynolds number, head loss, $NPSHa$, $NPSHr$, and specific speed. The processed data were then subjected to pump performance analysis, in which the pump capacity was evaluated against system demand, flow velocity was assessed for compliance with API RP 14E criteria, and the cavitation safety margin was determined through comparison of $NPSHa$ and $NPSHr$ values. A decision criterion was then applied: if the pump was found to perform within all acceptable hydraulic criteria — that is, capacity meeting system demand, flow velocity within the recommended range, and $NPSHa$ exceeding $NPSHr$ — the findings were documented and the system was confirmed as operationally suitable. If any criterion was not satisfied, the analysis loop returned to the parameter data collection stage for re-examination of input data and recalculation. Based on the final analysis outcomes, engineering recommendations were formulated to address identified deficiencies or to propose performance optimization measures, including the potential integration of a Variable Speed Drive (VSD).

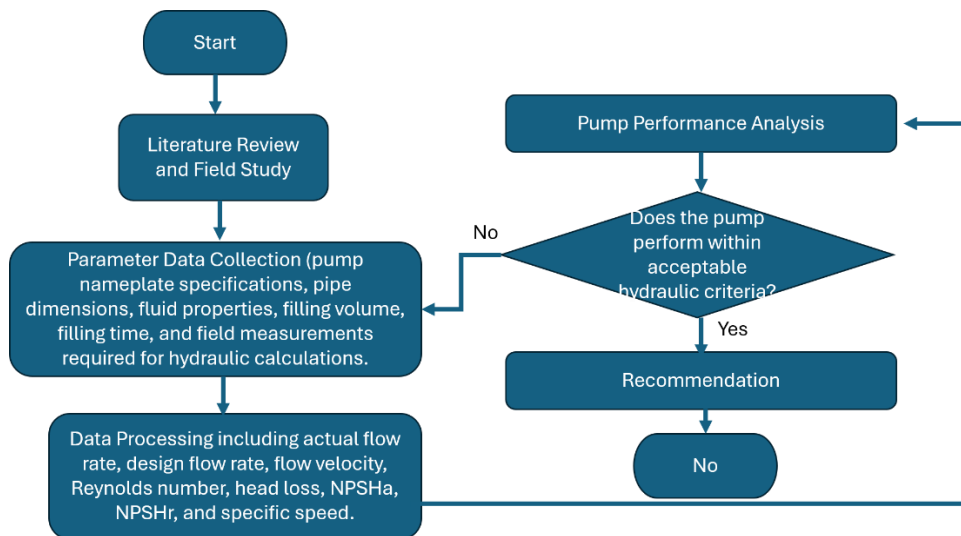


Figure 2. Research Framework

3. Result and Discussion

A. Pump Capacity Analysis

Field measurements recorded an actual flow rate of 1,600 L/min against a pump design flow rate of 2,750 L/min, yielding a reserve capacity of 1,150 L/min — equivalent to 41.8% of the rated design capacity (**Table 2**). This substantial margin confirms that the pump operates well below its maximum rated output under current loading conditions, indicating that the system is neither operating near its hydraulic limits nor at risk of capacity-driven failure.

Table 2. Actual and Design Flow Rate Comparison

PARAMETER	VALUE
Actual Flow Rate	1.600 L/min
Design Flow Rate	2.750 L/min
Capacity Difference	1.150 L/min

The 41.8% reserve capacity carries two practical implications. First, the pump retains sufficient operational flexibility to accommodate future increases in fuel distribution demand without requiring immediate equipment replacement or system redesign. Second, the persistent gap between actual and design flow rates suggests that the pump consistently operates at partial load, a condition known to reduce hydraulic efficiency and increase specific energy consumption relative to best-efficiency-point operation. This observation provides the primary technical justification for Variable Speed Drive (VSD) integration, which would allow pump output to be modulated in response to real-time demand, thereby reducing unnecessary energy expenditure while preserving the reserve capacity margin required for operational reliability.

B. Flow Velocity and Reynolds Number Analysis

Fluid flow velocity and Reynolds number were evaluated to characterize the hydraulic behavior of the fuel distribution piping system and to assess its operational safety and efficiency (**Table 3**). The calculated flow velocity of 5.6 m/s falls within the allowable erosional velocity limit specified by API RP 14E, confirming that the piping system operates under acceptable conditions with a low risk of flow-induced erosion or mechanical degradation of the pipe wall. The Reynolds number of 9.64×10^6 substantially exceeds the turbulent flow threshold of $Re > 4,000$, classifying the flow regime as fully turbulent. Fully turbulent flow is characteristic of high-capacity fuel distribution systems and is generally associated with enhanced fluid mixing and uniform velocity profiles across the pipe cross-section. However, fully turbulent conditions also produce higher frictional losses compared to transitional or laminar regimes, as the Darcy friction factor under fully turbulent flow becomes independent of Reynolds number and is governed primarily by pipe wall roughness. This has a direct bearing on head loss magnitude and, consequently, on the total energy demand of the pumping system — an interaction that is quantified in the subsequent head loss analysis.

Table 3. Fluid Flow Analysis Results

PARAMETER	VALUE
Flow Velocity	5,6 m/s
Reynolds Number	$9,64 \times 10^6$

C. Head Loss Analysis

Head loss was evaluated to quantify the frictional energy dissipated as fuel travels through the piping system and to assess its impact on overall pumping system performance (Table 4).

Table 4. Head Loss Analysis Results

PARAMETER	VALUE
Head Pump	5,6 m/s
Head Loss	1,61 m

The calculated head loss of 1.61 m represents approximately 3.4% of the total available pump head of 52.83 m, indicating that frictional resistance within the existing piping configuration is minimal and does not impose a significant energy penalty on the fuel distribution process. This low head loss-to-pump head ratio confirms that

the pipeline geometry, diameter, and length are appropriately sized relative to the operating flow rate, and that the current piping configuration remains hydraulically efficient under field conditions. The negligible proportion of energy lost to friction further implies that the pump head is largely available to overcome static elevation differences and deliver fuel to the loading bay, rather than being consumed by internal pipeline resistance.

D. NPSH Analysis and Cavitation Assessment

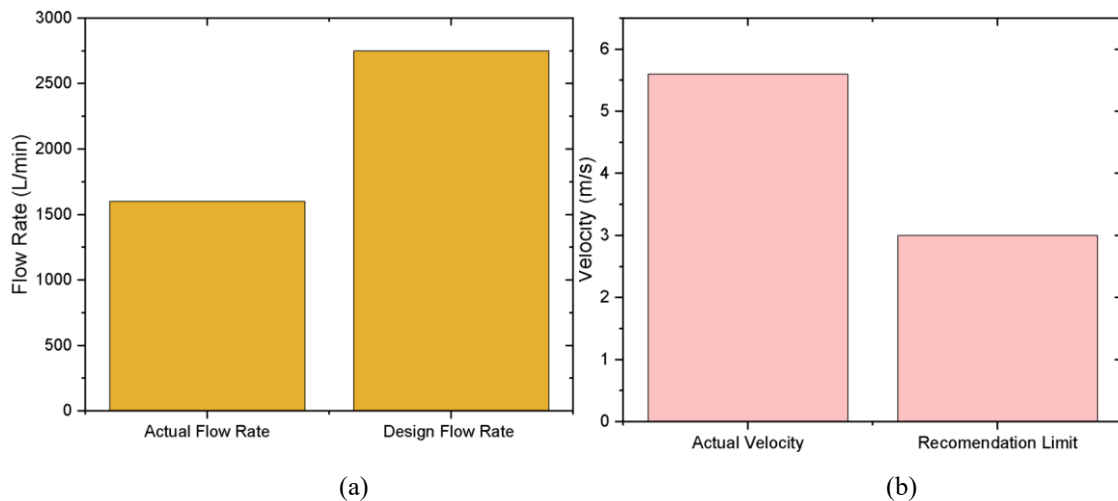
The NPSH analysis was conducted to evaluate the risk of cavitation at the pump suction inlet under current field operating conditions (**Table 5**). The calculated NPSHa of 5.23 m exceeds the manufacturer-specified NPSHr of 2.64 m by a margin of 2.59 m, confirming that the suction pressure at the pump inlet remains sufficiently above the vapor pressure of the fluid to prevent vapor bubble formation under current operating conditions. According to Karassik et al., the condition $NPSHa > NPSHr$ is the primary criterion for cavitation-free pump operation and is indicative of a high level of reliability against cavitation-induced mechanical damage, including impeller erosion, vibration, and noise. The positive NPSH margin of 2.59 m provides an additional safety buffer against transient suction pressure fluctuations that may occur during peak demand periods or thermal variations in the stored fuel. Therefore, the centrifugal pump at Fuel Terminal Tegal can be confirmed to operate safely with a low cavitation risk under the evaluated field conditions.

Table 5. NPSH Analysis Results

PARAMETER	VALUE
NPSHa	5,23 m
NPSHr	2,64 m
Reserve capacity	1.150 L/min

E. Comparative Graphical Analysis

The calculated hydraulic parameters were synthesized into a comparative visual analysis to provide a consolidated assessment of pump performance against design specifications and applicable technical standards (Figure 3).



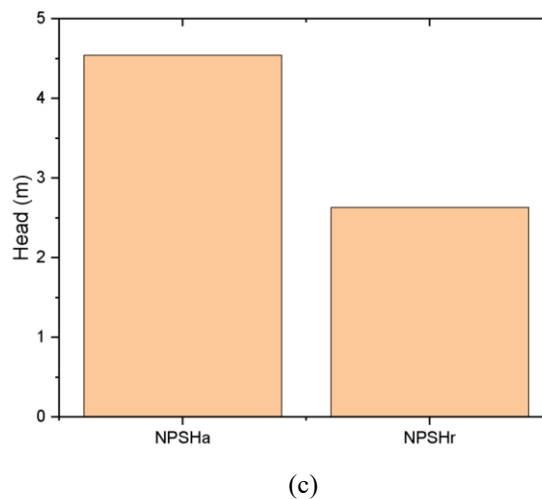


Figure 3 Hydraulic performance comparison: (a) flow rate, (b) flow velocity vs. API RP 14E limit, and (c) NPSHa vs. NPSHr.

Figure 3 presents three key comparisons: (1) NPSHa versus NPSHr, confirming the cavitation safety margin; (2) actual flow velocity against the API RP 14E recommended limit, assessing erosional velocity compliance; and (3) actual flow rate against the pump design flow rate, evaluating capacity adequacy. The comparative analysis confirms that the pumping system satisfies all three hydraulic performance criteria under current field conditions. The design flow rate of 2,750 L/min substantially exceeds the actual system demand of 1,600 L/min, establishing a reserve capacity margin of 41.8% that ensures uninterrupted fuel distribution operations. The NPSHa of 5.23 m exceeds the NPSHr of 2.64 m by a margin of 2.59 m, confirming cavitation-free operation with an adequate suction pressure buffer against transient demand fluctuations. Although the measured flow velocity of 5.6 m/s approaches the upper boundary of the API RP 14E recommended range, it remains within acceptable limits and does not present an immediate erosion risk under current operating conditions. Collectively, these results demonstrate that the centrifugal pump exhibits adequate capacity, hydraulic safety, and operational reliability for supporting fuel distribution activities at Fuel Terminal Tegal. Nevertheless, the persistent partial-load operation implied by the 41.8% capacity surplus reinforces the recommendation for Variable Speed Drive (VSD) integration to optimize energy efficiency without compromising system reliability.

4. Conclusion

A field-based hydraulic performance audit of the MCPK 125-100-400 CC centrifugal pump at PT Pertamina Patra Niaga Fuel Terminal Tegal yielded three main findings. The pump capacity of 165 m³/h exceeds the actual operational demand of 1,600 L/min, producing a reserve margin of 1,150 L/min (41.8% of rated capacity). This confirms that the pump is adequately sized for current fuel distribution requirements and retains sufficient flexibility to accommodate demand growth without equipment replacement. The NPSHa of 5.23 m surpasses the NPSHr of 2.64 m by a margin of 2.59 m, confirming cavitation-free operation under current field conditions. This positive margin demonstrates that suction pressure remains consistently above the fluid vapor pressure during normal loading cycles, ensuring mechanical integrity of the impeller and long-term pump reliability. The flow velocity of 5.6 m/s complies with API RP 14E erosional velocity criteria, and the head loss of 1.61 m accounts for only 3.4% of the total available pump head of 52.83 m, confirming that the piping configuration is hydraulically efficient with minimal frictional energy penalty. Despite satisfactory hydraulic performance across all evaluated criteria, the 41.8% capacity surplus indicates chronic partial-load operation under the current fixed-speed configuration, which reduces hydraulic efficiency and increases specific energy consumption. Variable Speed Drive (VSD) integration is therefore recommended to align pump output with real-time demand, reduce energy waste, and extend equipment service life without compromising the safety margins established by this audit.

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