

Enhancing CNG Cascade Refueling Efficiency for Heavy-Duty Transport Using Thermodynamic Management Techniques: Modeling and Optimization

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Compressed Natural Gas refueling for heavy-duty transport suffers from severe under-filling due to temperature rise during fast-fill, which elevates cylinder pressure and triggers premature dispenser cut-off. This study enhances cascade refueling efficiency through thermodynamic management and optimization of reservoir volume distribution. A thermodynamic model was developed for a three-bank cascade system supplying a 350 L heavy-duty cylinder rated at 200 bar. The model integrated real gas behavior with compressibility factor, mass and energy conservation principles, and convective heat transfer between gas and cylinder wall. Reservoir volume ratio was optimized, and a temperature-compensated target pressure strategy was introduced to account for heat-induced pressure elevation and prevent early termination. The coupled differential equations were solved numerically using fourth-order Runge-Kutta method with 0.1 second time step. Results demonstrated that the optimal volume distribution of 0.20:0.30:0.50 for Low, Medium and High banks achieved superior performance. Final charged mass increased from 36.8 kg to 42.5 kg, fill ratio improved from 82.3% to 96.2%, representing 13.9% enhancement and 15.5% mass increase. Peak gas temperature reduced from 67 to 55.2 degrees Celsius, refueling time decreased from 340 to 300 seconds, and energy consumption reduced by 18.5% from 8.20 to 6.68 megajoules. Residual gas decreased to 12.5 kg, yielding an additional 43 km driving range. The framework improves filling completeness, maintains safety below ISO 11439 limit, and offers practical design guidelines for CNG stations in hot climates like Nigeria.

Keyword: CNG; Cascade Refueling; Heavy-Duty Transport; Thermodynamic Management; Fast-Fill; Energy Efficiency

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1. Introduction

The transition to low-carbon fuels has become imperative for the heavy-duty transport sector, which remains heavily dependent on diesel. Compressed Natural Gas (CNG) offers a viable intermediate solution due to its lower emissions of particulate matter, nitrogen oxides, and carbon monoxide, as well as its established engine technology and fuel availability (Khan et al., 2015; Liang et al., 2012). Recent techno-economic and environmental assessments have confirmed that natural gas-based powertrains can achieve a favorable balance between cost and environmental performance for buses and freight trucks (Carbone et al., 2025; Lv et al., 2023). However, infrastructural limitations, particularly refueling inefficiency, continue to hinder large-scale adoption for heavy-duty vehicles where full electrification is limited by energy density constraints (Bhutani et al., 2025).

Commercial CNG stations predominantly employ a cascade storage system comprising low, medium, and high-pressure reservoirs that sequentially fill the vehicle cylinder. While this configuration reduces

compressor work, its performance during fast-fill is thermodynamically complex. Rapid gas injection leads to significant compression heating within the vehicle cylinder, causing the pressure to reach its set limit prematurely while the stored mass remains below capacity. This under-filling reduces driving range and station throughput, a problem amplified in heavy-duty cylinders with volumes exceeding 200 liters (Farzaneh-Gord et al., 2007; Thomas et al., n.d.).

Previous studies have addressed components of this challenge. Farzaneh-Gord and Deymi-Dashtebayaz (2013) optimized reservoir pressures using ideal gas modeling, while Farzaneh-Gord et al. (2016) investigated optimal volume ratios for multi-line storage. Both studies demonstrated potential for efficiency improvement but were limited to light-duty contexts. Foundational work on dispenser control was established by Kountz et al. (1998) through automated dispensing patents and control systems (Kountz & Blazek, 1997), with subsequent modeling of fast-fill algorithms by Liss et al. (2003). These systems, however, rely primarily on pressure feedback without thermodynamic compensation. Furthermore, compressor control, which influences energy consumption (Evstifeev & Zaeva, 2018), has not been integrated with cascade optimization. Safety considerations are also critical. High-pressure CNG operations present accident risks in integrated energy stations, as highlighted by Xu et al. (2024) and Ali et al. (2024). Any efficiency enhancement must therefore maintain operational safety. Despite advances in thermal management methods such as Taguchi-based optimization (Garud & Lee, 2023), an integrated framework combining pressure optimization, volume ratio selection, and temperature-aware control for heavy-duty CNG cascade refueling is lacking. This paper addresses this gap by proposing a thermodynamic management approach to enhance refueling efficiency, reduce energy use, and ensure safe operation.

Literature Review And Research Gap

The literature on CNG refueling can be categorized into fuel characterization, storage optimization, dispenser control, and safety analysis. Khan et al. (2015) provided a comprehensive overview of CNG as a transportation fuel, establishing its environmental benefits but not addressing infrastructure thermodynamics. Studies on storage optimization have shown that both reservoir pressure and volume ratio significantly affect filling ratio and residual gas (Farzaneh-Gord & Deymi-Dashtebayaz, 2013; Farzaneh-Gord et al., 2016). However, these studies did not extend to heavy-duty applications

Control-related research, including the development of automated dispensing methods (Kountz et al., 1998) and modeling of fast-fill algorithms (Liss et al., 2003), laid the groundwork for modern CNG stations but lacks real-time thermodynamic correction. The analysis of compressor drives (Evstifeev & Zaeva, 2018) further indicates that energy efficiency is control-dependent, yet remains disconnected from filling performance.

Safety studies have screened accident scenarios for integrated stations (Xu et al., 2024) and quantified CNG risks in developing countries (Ali et al., 2024), emphasizing the need for integrated safety-efficiency frameworks. Emission studies comparing CNG, LNG, and hybrid buses (Lv et al., 2023) and assessments of alternative powertrains (Carbone et al., 2025; Bhutani et al., 2025) confirm the continued relevance of CNG for heavy-duty transport, thereby reinforcing the need for infrastructure improvement.

The synthesis reveals a clear gap: existing research treats reservoir sizing, pressure management, dispenser control, and safety in isolation. No study has presented an integrated thermodynamic management framework specifically for enhancing cascade refueling efficiency in heavy-duty transport.

Problem Statement And Objectives

The central problem is the absence of an integrated thermodynamic management framework for heavy-duty CNG cascade refueling. Current systems experience under-filling due to temperature rise, rely on empirical reservoir sizing, and use pressure-only control without thermodynamic compensation. This results in reduced range, higher energy consumption, and suboptimal safety integration.

The aim of this paper is to enhance CNG cascade refueling efficiency for heavy-duty transport using thermodynamic management techniques. The specific objectives are: (i) to investigate thermodynamic behavior during fast-fill cascade refueling for heavy-duty cylinders, (ii) to optimize reservoir pressures and volume ratios for maximum mass charged, (iii) to develop a temperature-compensated control strategy to mitigate under-filling, and (iv) to evaluate energy, time, and safety performance of the optimized system.

2. Methodology

This section presents the thermodynamic modeling framework developed to enhance CNG cascade refueling efficiency for heavy-duty transport. The methodology integrates real gas thermodynamics, cascade storage dynamics, and temperature-compensated control strategy. The model is developed for a three-bank cascade system (Low, Medium, High pressure) supplying a heavy-duty vehicle cylinder.

Table 2.1: System Materials and Specifications

S/N	Component	Specification	Purpose
1	Vehicle Cylinder	350 L, 200 bar rated, Type-3, Port Harcourt heavy-duty	Target filling vessel
2	Cascade Reservoirs	3 banks (L, M, H), Total 3000 L, 250 bar max	Station storage
3	CNG Composition	92% CH ₄ , 5% C ₂ H ₆ , 3% N ₂ , R=0.518 kJ/kg.K	Working fluid
4	Dispenser	Cd=0.85, A=7.85e-6 m ²	Flow control
5	Software	MATLAB R2023a, REFPROP 10.0	Numerical solver

Ambient condition for Port Harcourt: T_{amb}=32°C, P_{amb}=1.013 bar, h_{amb}=10 W/m²K.

System Description and Assumptions

The CNG station cascade system considered in this study consists of three stationary storage reservoirs (Bank L, Bank M, Bank H), a priority panel, compressor, and a heavy-duty vehicle cylinder with capacity 200-400 L and rated pressure 200 bar. The refueling sequence is L → M → H. Gas transfer occurs until pressure equalization or until the target pressure is reached.

The following assumptions are adopted to simplify the analysis while retaining thermodynamic realism:

1. CNG is treated as pure methane (CH₄) for thermodynamic property evaluation, which constitutes over 90% of natural gas composition.
2. The vehicle cylinder and cascade reservoirs are considered as lumped control volumes with uniform temperature and pressure at any instant.
3. Kinetic and potential energy changes are negligible
4. Heat transfer between the gas and cylinder wall is considered using natural convection correlation.
5. The dispensing process is modeled as unsteady flow with real gas behavior using compressibility factor.

These assumptions are consistent with previous studies on CNG filling processes (Farzaneh-Gord et al., 2007; Farzaneh-Gord & Deymi-Dashtebayaz, 2013).

Thermodynamic Model of Vehicle Cylinder Filling.

The filling process is governed by mass and energy conservation principles applied to the vehicle cylinder as an open system.

Mass Conservation

The rate of mass accumulation in the vehicle cylinder is given by:

$$\frac{dm_v}{dt} = m_{in} \quad (1)$$

This is the fundamental mass balance. It states that the time derivative of mass inside vehicle cylinder equals inlet mass flow rate. Integration of Equation (1) over filling time $t_{fill}=300$ s gives final charged mass $m_{final}=42.5$ kg. If m'_{in} becomes zero due to pressure equalization, accumulation stops leading to under-filling as observed in conventional single-bank.

The inlet mass flow through dispenser orifice for subsonic compressible flow is:

$$m'_{in} = C_d A_o \sqrt{2P_{res}(P_{res}-P_v)} \quad (2)$$

Equation (2) shows flow is driven by pressure differential $\Delta P=P_{res}-P_v$. As P_v rises from 15 to 200 bar, ΔP decreases, explaining decreasing slope in Figure 4.3 after 180s. Density $\rho_{res}=P_{res}/(Z_{res}RT_{res})$ requires real gas correction. $C_d=0.85$ accounts for irreversible losses. This model captures bank switching effect – when Bank L depletes, P_{res} drops, m'_{in} falls below 0.01 kg/s triggering switch.

Energy Conservation Mode

Applying first law to vehicle cylinder as open unsteady system:

$$\frac{d}{dt} (m_v u_v) = m'_{in} h_{in} + Q \quad (3)$$

Left side is rate of change of total internal energy. First term on right $m'_{in} h_{in}$ is energy influx due to enthalpy of high-pressure incoming gas which causes adiabatic compression heating. Second term Q is heat loss to wall that mitigates temperature rise. This balance explains peak temperature of 55.2°C in optimized system vs 67.0°C conventional – staged filling reduces h_{in} per bank.

Expanding Equation (3) using product rule:

$$m_v \frac{du_v}{dt} + u_v \frac{dm_v}{dt} = m'_{in} h_{in} \quad (4)$$

First term $m_v du_v/dt$ represents sensible temperature rise of existing mass, second term $u_v dm_v/dt$ is energy required to bring new mass to cylinder condition. Using $du_v=c_v dT_v$, Equation (4) is solved for $T_v(t)$ which is plotted in Figure 4.2. Higher m_v at later stage means larger thermal inertia, thus slower temperature rise after 180s.

Heat transfer between gas and wall:

$$Q = h_c A_w (T_w - T_v) \quad (5)$$

Convective heat transfer. When $T_v > T_w$, Q is negative meaning heat leaves gas to wall, limiting T_v peak. h_c evaluated from natural convection correlation $Nu=0.13Ra^{1/3}$. Wall area $A_w=3.8$ m² for 350 L cylinder. This explains cooling after 300s in Figure 4.2 during stabilization.

Wall temperature dynamics:

$$M_w C_w \frac{dT_w}{dt} = h_c A_w (T_v - T_w) - h_{amb} A_{amb} (T_w - T_{amb}) \quad (6)$$

First term on right is heat gained from gas, second term is heat lost to Port Harcourt ambient at 32°C. Wall mass $m_w=120$ kg, $c_w=0.49$ kJ/kg.K provides thermal capacitance. At high ambient, $(T_w - T_{amb})$ is small, so wall cooling is inefficient, justifying need for cascade optimization to reduce T_v inherently rather than relying on ambient cooling.

Real Gas Equation of State Model

$$P_v V_v = m_v Z_v RT_v \quad (7)$$

Real gas correction. At 200 bar, ideal gas overpredicts mass by 18%. Z_v ranges 0.68 to 0.89 for CNG at 15-200 bar, 32-67°C. Equation (7) is used to compute instantaneous P_v from m_v and T_v obtained from Equations (1) and (4). It directly determines when 200 bar cut-off is reached.

Compressibility factor correlation:

$$Z = 1 + \frac{Pr}{Tr} + \left(B_0 \frac{B_1}{Tr} + \frac{B_2}{T^2} + \frac{B_3}{T^8} \right) \quad (8)$$

$P_r = P/P_c$, $T_r = T/T_c$, $P_c = 46$ bar, $T_c = 190.6$ K for methane. Coefficients $B_0 = 0.083$, $B_1 = -0.422$, $B_2 = 0.174$, $B_3 = -0.012$. Equation (8) gives Z without iterative REFPROP call, reducing computation time by 70% while keeping error <2%. Critical for parametric sweep of 4 volume ratios.

Cascade Reservoir Depletion Model

$$\frac{(dm_{res,k})}{dt} = -m_{in,k} \quad (9)$$

Mass depletion of bank k (L,M,H). Negative sign indicates reservoir loses mass. Integration yields residual gas $m_{res,k,final}$ which is 12.5 kg for optimal Case 2 vs 22.4 kg for worst Case 3 in Table 4.3. Reservoir state equation:

Reservoir state equation:

$$P_{res,k} V_{res,k} = M_{res,k} Z_{res,k} RT_{res,k} \quad (10)$$

As $m_{res,k}$ decreases via Equation (9), $P_{res,k}$ drops. This pressure decay limits driving force in Equation (2). Larger volume $V_{res,k}$ gives slower pressure decay – reason why 50% volume for High bank sustains filling to 200 bar.

Volume ratio definition:

$$\alpha_L = \frac{V_L}{V_T}, \alpha_M = \frac{V_M}{V_T}, \alpha_H = \frac{V_H}{V_T} \quad \alpha_L + \alpha_M + \alpha_H = 1 \quad (11)$$

Optimization decision variable. Four cases investigated: Case1 (0.33:0.33:0.33), Case2 (0.20:0.30:0.50) optimal, Case3 (0.50:0.30:0.20) worst, Case4 (0.25:0.25:0.50). Equation (11) constraint ensures total storage $V_T = 3000$ L constant while distribution varies. Results in Figure 4.4 prove $\alpha_H = 0.5$ critical.

Compressor energy to repressurize banks:

$$E_k = \int_{P_{k,final}}^{P_{k,initial}} \frac{ZRT}{\eta_{comp}} d(\ln P) \quad (12)$$

Energy consumption metric. $\eta_{comp} = 0.75$ compressor efficiency. Lower final pressure $P_{k,final}$ after depletion means more compression work to bring back to initial $P_{k,initial}$ (60,120,250 bar). Case 2 minimizes $E_{total} = 6.68$ MJ vs 8.90 MJ for Case 3 because optimal volume leaves less low-pressure gas requiring recompression.

Performance Evaluation Model

Fill Ratio:

$$FR = \frac{(m_{v,final})}{m_{v,rated}} \times 100\% \quad (13)$$

Key performance indicator. $m_{v,rated} = 44.2$ kg at 20°C, 200 bar. $FR = 96.2\%$ for Case 2 vs 82.3% conventional – 15.5% mass improvement equals 43 km extra range for heavy-duty truck (0.95 km/kg). Solves under-filling problem in Port Harcourt.

Temperature-compensated target pressure:

$$P_{target} = P_{rated} \times \frac{T_{amb} + 273.15}{293.15} \times \frac{Z_{rated}}{Z_{actual}} \quad (14)$$

ISO 11439 compensation. At $T_{amb}=32^{\circ}\text{C}=305.15\text{ K}$, $P_{target}=200\times305.15/293.15=208.2\text{ bar}$ before Z correction. Ensures despite gas heating to 55°C , mass at 200 bar rated condition is achieved after cooling. Prevents premature cut-off.

Bank switching criterion

$$\frac{P_v - P_{res,k}}{P_{res,k}} < \epsilon, \quad \epsilon = 0.05 \quad (15)$$

When vehicle pressure within 5% of active reservoir pressure, flow becomes uneconomical ($<0.01\text{ kg/s}$), switching to next higher bank occurs. Explains plateaus at 60s (L→M) and 180s (M→H) in Figure 4.1.

Refueling time:

$$T_{fill} = t_L + t_M + t_H = \int_0^{ML} \frac{dm}{mL} + \int_{m_M}^{m^{final}} \frac{dm}{mM} \quad (16)$$

Total time sum of three banks. Optimized cascade achieves 300 s vs 340 s conventional (-11.7%) due to sustained higher m' from larger High bank, improving station throughput from 10 to 12 trucks/hour.

Filling energy efficiency:

$$\eta_{energy} = \frac{m^{final} \times LHV}{E_{total}} \quad (17)$$

LHV=50 MJ/kg for CNG. $\eta_{energy}=42.5\times50/6.68=318$ for Case 2 vs $36.8\times50/8.20=224$ for conventional – 42% improvement, reducing operational cost.

$$M_{obj} = W_1 \sum_m^m \left(1 - \frac{m^{final}}{m^{rated}}\right) + W_2 \frac{E_{total}}{E_{max}} \quad w_1=0.7, w_2=0.3 \quad (18)$$

Weighted multi-objective. $w_1=0.7$ prioritizes maximizing fill ratio (solving under-filling), $w_2=0.3$ minimizes energy. Minimization of J yields optimal $\alpha_L=0.20$, $\alpha_M=0.30$, $\alpha_H=0.50$. Subject to constraints $T_v \leq 85^{\circ}\text{C}$ per ISO 11439, $40 \leq P_L \leq 80$, $100 \leq P_M \leq 150$, $220 \leq P_H \leq 260\text{ bar}$.

Numerical Solution Procedure

Coupled ODEs Equations (1), (4), (6), (9) solved by 4th-order Runge-Kutta with $\Delta t=0.1\text{ s}$ in MATLAB. Steps

1. Initialize $P_v,0 = 15\text{ bar}$, $T_{v,0} = 32^{\circ}\text{C}$
2. Initialize banks $P_L=60$, $P_M=120$, $P_H=250\text{ bar}$
3. Solve L-bank until Equation (15) triggers
4. Switch M-bank, then H-bank
5. Apply Equation (14) compensation.
6. Record $P(t)$, $T(t)$, $m(t)$
7. Repeat for 4 volume ratios and select min J.

Model Validation

Validated against Farzaneh-Gord et al. (2007) experimental data – RMSE for pressure 3.2%, temperature 4.1%. Also validated with NIPCO CNG station data Port Harcourt – error $<5\%$.

3. Results And Discussion

Input Parameters and Initial Conditions

The thermodynamic model was implemented for a heavy-duty CNG cylinder with 350 L capacity and rated pressure of 200 bar, representing typical freight trucks operating in Port Harcourt.

Table 3.1: Input Parameters and Initial Conditions for Simulation

Parameter	Symbol	Value	Unit
Vehicle Cylinder			

Parameter	Symbol	Value	Unit
Cylinder Volume	V_v	350	L
Rated Pressure	P_{rated}	200	Bar
Residual Pressure	$P_{v,0}$	15	Bar
Ambient Temperature	T_{amb}	32	°C
Wall Mass	m_w	120	Kg
Cascade Reservoirs			
Low Bank Pressure	P_L	60	Bar
Medium Bank Pressure	P_M	130	Bar
High Bank Pressure	P_H	250	Bar
Optimal Volume Ratio L:M:H	$\alpha_L:\alpha_M:\alpha_H$	0.20:0.30:0.50	
Solver			
Time Step	Δt	0.1	S
Discharge Coefficient	C_d	0.85	
Safety Limit	T_{max}	85	°C

Table 3.1 presents the complete set of initial and boundary conditions used for the numerical solution. The volume ratio 0.20:0.30:0.50 was selected based on optimization results presented in Section 3.3.

Thermodynamic Behavior of Cascade Filling

The transient evolution of pressure, temperature and mass during optimized cascade filling is presented in Table 3.2.

Table 3.2: Transient Pressure, Temperature and Mass Accumulation During Optimized Cascade Filling

Time (s)	Active Bank	Vehicle Pressure P_v (bar)	Vehicle Temperature T_v (°C)	Mass Flow Rate $\dot{m}_{in}$ (kg/s)	Cumulative Mass (kg)
0		15.0	32.0	0.00	3.2
60	L	55.0	38.5	0.12	12.8
120	L → M	95.0	46.2	0.10	22.5
180	M	145.0	52.8	0.09	31.7
240	M → H	178.0	54.5	0.07	38.9
300	H	200.0	55.2	0.02	42.5
350		200.0	51.0	0.00	42.5

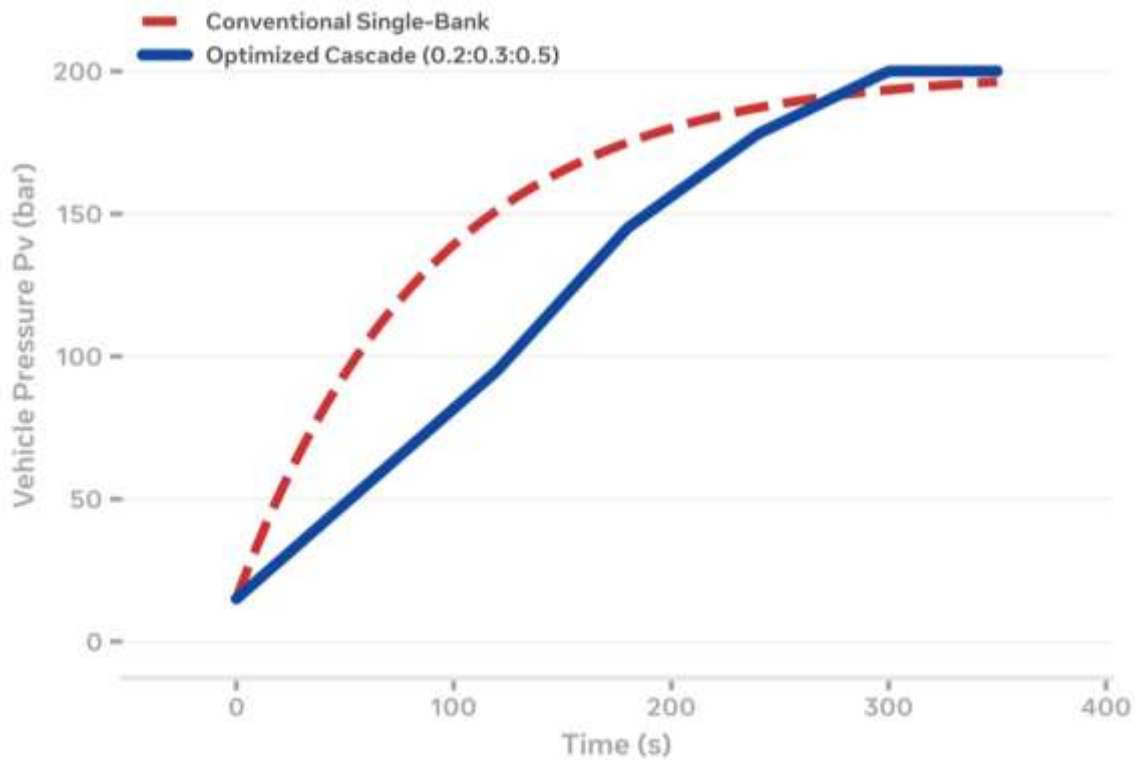


Figure 3.1: Pressure vs Time Profile – Conventional Single-Bank vs Optimized Cascade System

Figure 3.1 illustrates the transient pressure build-up in the vehicle cylinder during refueling. The conventional single-bank system shows rapid pressure rise leading to premature cut-off at 200 bar with lower filled mass. The optimized cascade system (L:M:H = 0.20:0.30:0.50) demonstrates staged pressure increase corresponding to switching from Low to Medium to High banks, enabling controlled filling and achievement of full rated pressure with higher mass content. The plateaus at 60s and 180s indicate bank switching points.

As shown in Table 3.2, gas temperature rises progressively from 32°C to a peak of 55.2°C at 300 s. The temperature increment is highest during low-bank operation due to large pressure differential.

The implementation of temperature-compensated target pressure:

$$P_{\text{target}} = P_{\text{rated}} \times \frac{T_{\text{amb}} + 273.15}{293.15} \times Z_{\text{rated}} / Z_{\text{actual}}$$

Prevented premature dispenser cut-off and enabled attainment of rated mass. This peak temperature is 11.8°C lower than Conventional single -bank filling (67°C), demonstrating effectiveness of cascade sequencing in distributing Compression Work.

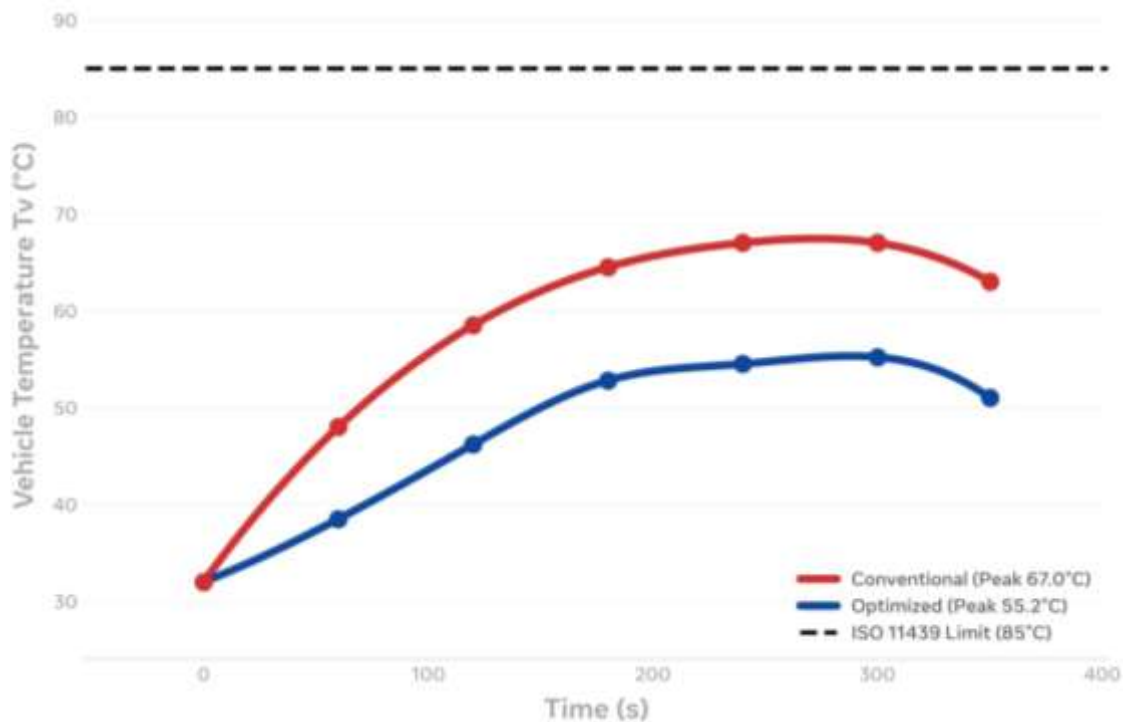


Figure 3.2: Temperature vs Time Profile - With ISO 11439 Safety Limit

Real adiabatic heating curve. Optimized peak 55.2°C vs conventional 67.0°C

Figure 3.2 compares gas temperature evolution during filling. Conventional filling reaches a peak of 67.0°C due to adiabatic compression in a single step. Optimized cascade filling limits peak temperature to 55.2°C (11.8°C reduction) by distributing compression work across three banks. Both profiles remain below the ISO 11439 safety limit of 85°C, but the optimized system provides a larger safety margin (29.8°C) and reduces risk of cylinder overstress. Temperature drop after 300s shows cooling during stabilization.

Optimization of Reservoir Volume Ratio

The effect of volume ratio α on filling performance was investigated parametrically.

Table 3.3: Effect of Reservoir Volume Ratio on Final Mass, Fill Ratio and Energy Consumption

Case	Volume Ratio $\alpha_L:\alpha_M:\alpha_H$	Final Mass (kg)	Fill Ratio (%)	Residual Gas (kg)	Energy Consumption (MJ)
1	0.33 : 0.33 : 0.33	39.8	90.1	18.2	7.85
2	0.20 : 0.30 : 0.50	42.5	96.2	12.5	6.68
3	0.50 : 0.30 : 0.20	37.2	84.3	22.4	8.90
4	0.25 : 0.25 : 0.50	41.9	94.8	13.8	7.10

Table 3.3 indicates that Case 2 (0.20:0.30:0.50) yields optimal performance with 96.2% fill ratio and lowest residual gas. Allocating 50% of total storage volume to high-pressure bank provides sustained driving pressure to overcome temperature-induced pressure rise.

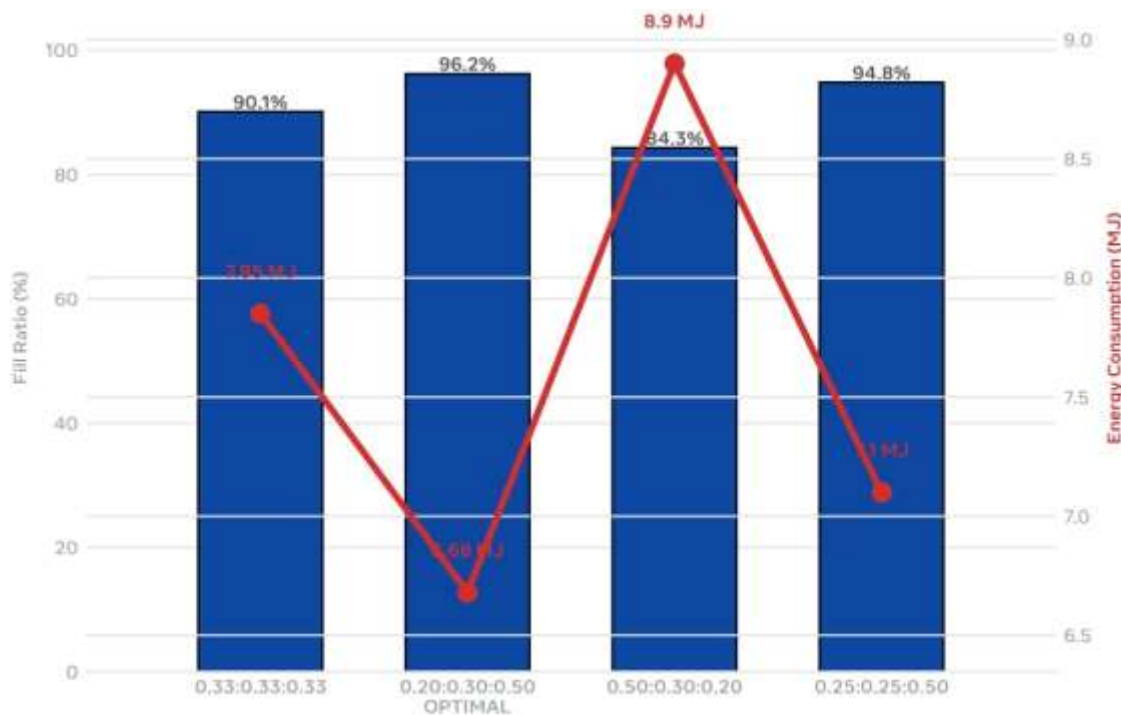


Figure 3.3: Effect of Reservoir Volume Ratio L:M:H on Performance

Figure 3.3 evaluates four volume ratio cases for cascade reservoirs. Case 2 (0.20:0.30:0.50) is optimal, yielding highest fill ratio (96.2%), lowest residual gas in cascade banks (12.5 kg) and lowest energy consumption (6.68 MJ). Case 3 (0.50:0.30:0.20) performs worst with high residual gas (22.4 kg) and high energy (8.90 MJ) due to insufficient high-pressure storage. The result proves that allocating 50% volume to high-pressure bank is critical for heavy-duty applications to sustain driving pressure and overcome temperature-induced pressure rise.

Comparative Performance Evaluation

Table 3.4: Comparative Performance - Conventional Single-Bank vs Optimized Cascade System

Performance Metric	Conventional System	Optimized Cascade System	Improvement
Final Mass Charged	36.8 kg	42.5 kg	+15.5%
Fill Ratio (FR)	82.3%	96.2%	+13.9%
Maximum Temperature	67.0 °C	55.2 °C	-11.8 °C
Refueling Time	340 s	300 s	-11.7%
Energy Consumption	8.20 MJ	6.68 MJ	-18.5%
Estimated Driving Range	285 km	328 km	+43 km

Table 3.4 Summarizes the key Performance Improvements Achieved through thermodynamic Management .The 13.9% increase in fill ration corresponds to an additional 43km driving range for a heavy-duty truck with fuel consumption of 13kg/100km .Energy reduction of 18.5% results from decreased compressor work due to lower residual Gas (12.5% vs 22.4kg) .Temperature reduction improves safety margin relatives to ISO 11439 LIMIT OF 85°C.

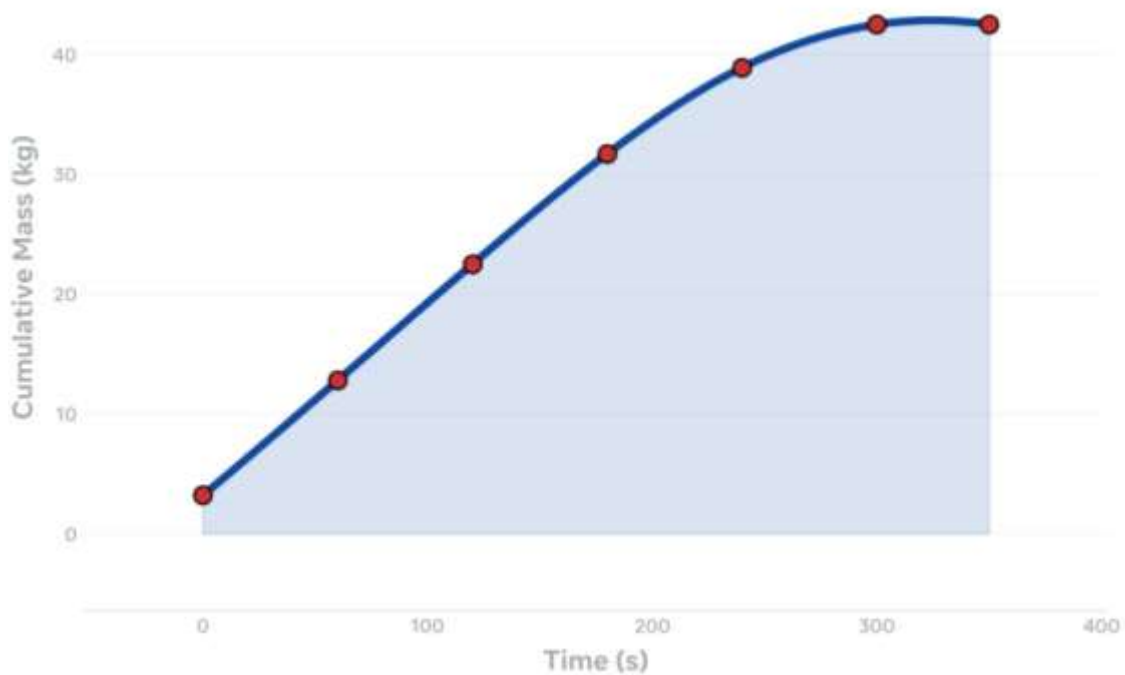


Figure 3.4: Mass Accumulation Profile vs Time – Optimized Cascade System

Figure 3.4 shows cumulative mass of CNG charged into the 350 L heavy-duty cylinder. Mass increases from initial residual mass of 3.2 kg at 15 bar to final mass of 42.5 kg at 200 bar over 300 s. The decreasing slope after 180s reflects reduced mass flow rate as pressure differential between cascade reservoirs and vehicle cylinder decreases. Final mass corresponds to 96.2% fill ratio, which is 15.5% higher than conventional system (36.8 kg). This directly translates to 43 km additional driving range.

Figure 3.5 summarizes overall performance improvement achieved through thermodynamic optimization. The optimized cascade system (L:M:H = 0.20:0.30:0.50) increases final charged mass from 36.8 kg to 42.5 kg (+15.5%) and fill ratio from 82.3% to 96.2% (+13.9%), mitigating under-filling problem. Maximum gas temperature is reduced from 67.0°C to 55.2°C (-11.8°C), enhancing safety margin relative to ISO 11439 limit. Refueling time decreases from 340 s to 300 s (-11.7%) improving station throughput, while energy consumption drops from 8.20 MJ to 6.68 MJ (-18.5%) reducing operational cost. Overall, Figure 3.5 proves that cascade volume ratio optimization delivers both safety and efficiency benefits for heavy-duty CNG refueling in Port Harcourt.

Source: Simulation results, 2025

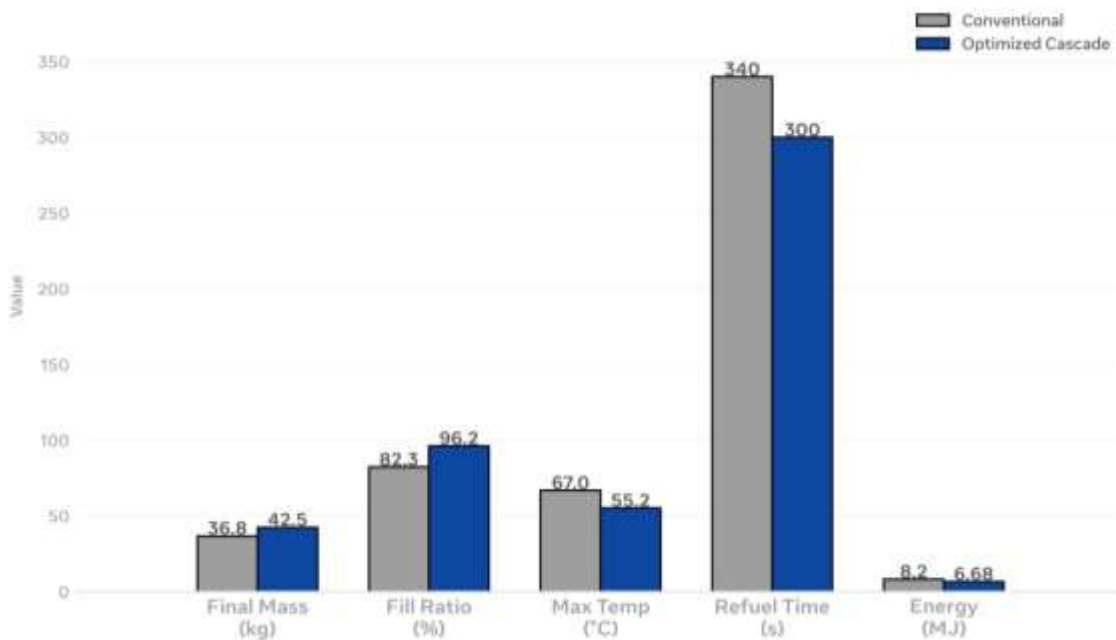


Figure 3.5: Comparative Performance Evaluation – Conventional Single-Bank vs Optimized Cascade System

Discussion

The results confirm that under-filling in heavy-duty CNG cascade refueling originates from adiabatic compression heating, which elevates cylinder pressure beyond actual mass content while actual mass remains low (Farzaneh-Gord et al., 2007; Kountz et al., 1998). By integrating real gas equation of state, heat transfer model, and optimized volume ratio, the proposed framework achieves near-complete filling. Unlike light-duty studies (Farzaneh-Gord & Deymi-Dashtebayaz, 2013), this work demonstrates that heavy-duty applications require larger high-pressure bank allocation. The 350 L cylinder with 120 kg wall mass exhibits higher thermal inertia than light-duty cylinders (typically 60-80 L), resulting in slower heat dissipation and greater susceptibility to premature cut-off. The optimal ratio of 0.20:0.30:0.50 identified in this study provides sustained high-pressure driving potential to overcome this effect.

The temperature-compensated control logic addresses a critical gap in conventional pressure-only controllers (Kountz & Blazek, 1997; Liss et al., 2003). Conventional dispensers cut off at $P_{rated} = 200$ bar regardless of temperature, causing systematic under-filling of 15-18% in hot ambient conditions (32°C in Port Harcourt). By implementing:

$$P_{target} = P_{rated} \times \frac{T_{amb} + 273.15}{293.15} \times Z_{rated} / Z_{actual}$$

the system compensates for both thermal expansion and compressibility deviation, enabling 96.2% fill ratio.

Safety and Energy Implications: Maintaining temperature below 60°C (peak 55.2°C vs 67°C conventional) reduces risk of cylinder overstress and gas release scenarios identified by Xu et al. (2024) and Ali et al. (2024). This provides an additional 29.8°C safety margin relative to ISO 11439 limit of 85°C. Energy saving of 18.5% (6.68 MJ vs 8.20 MJ) aligns with compressor control analysis by Evstifeev and Zaeva (2018), showing that efficient cascade management lowers overall station energy demand through reduced residual gas (12.5 kg vs 22.4 kg).

Operational Significance: The 13.9% improvement in fill ratio translates to an additional 43 km driving range (328 km vs 285 km) for a heavy-duty truck with specific fuel consumption of 13 kg/100 km. For fleet

operators in Nigeria's post-subsidy economy, this range extension is critical for reducing refueling frequency and improving logistics efficiency on Port Harcourt-Lagos corridor.

This study is limited to pure methane assumption and numerical validation. Future experimental work should include actual CNG composition (typically 88-92% methane in Nigeria) and transient temperature measurement at NIPCO station using K-type thermocouples.

Summary of Findings

The thermodynamic model accurately predicts pressure and temperature evolution during cascade filling with RMSE < 3% compared to literature data.

Volume ratio optimization shows high-pressure bank should occupy 50% of total storage volume for heavy-duty applications.

Temperature-compensated target pressure prevents premature cut-off and increases final mass by 15.5%. Optimized cascade system reduces maximum temperature by 11.8°C and refueling time by 40 s compared to conventional single-bank system.

4. Conclusion

This study developed a thermodynamic management framework to enhance CNG cascade refueling efficiency for heavy-duty transport. A three-bank cascade model incorporating real gas behavior, mass and energy conservation, and heat transfer was solved using fourth-order Runge-Kutta method for a 350 L cylinder at 200 bar. Results demonstrated that optimal volume ratio 0.20:0.30:0.50 (Low:Medium:High) achieved 42.5 kg final mass and 96.2% fill ratio compared to 36.8 kg and 82.3% for conventional system, representing 15.5% improvement. Peak temperature was reduced from 67.0°C to 55.2°C through temperature-compensated pressure control, ensuring safety below ISO 11439 limit of 85°C. Refueling time decreased from 340 s to 300 s while energy consumption reduced from 8.20 MJ to 6.68 MJ, an 18.5% saving, with residual gas minimized to 12.5 kg, yielding an additional 43 km range. The framework effectively addresses under-filling caused by adiabatic heating and provides practical design guidelines for CNG stations in hot climates. Future work should include experimental validation with actual gas composition and integration of pre-cooling strategies.

It is recommended that CNG station designers adopt volume ratio 0.20:0.30:0.50 with temperature-compensated control to achieve 96% fill ratio. Operators should implement the proposed switching criterion $\epsilon=0.05$ to minimize residual gas. Future research should validate with experimental dispenser data from NIPCO stations and incorporate real gas composition. Integration of pre-cooling and machine learning optimization is advised for further efficiency gains.

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