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THEORETICAL ASSESSMENT OF OPERATING PARAMETERS IN DIESEL RAILWAY ENGINE CONVERTED TO HYDROGEN FUEL

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Abstract

This paper presents a theoretical evaluation of operating-parameter changes in a Cummins NTA855-R4 diesel engine converted to spark-ignited hydrogen operation. The objective is to quantify the attainable specific engine power after conversion and to characterise the accompanying shifts in combustion dynamics and mechanical loading. Thermodynamic modelling with parametric sweeps indicates that hydrogen increases peak cylinder pressure and pressure-rise rate, while a like-for-like conversion tends to reduce effective power. When internal mixture formation is applied, the indicated and brake efficiencies rise by approximately 7–10%. Complete substitution of diesel with hydrogen is achievable without power loss when using external mixture formation (port fuel injection) at lean excess-air ratios ($\lambda \approx 2$–3). However, the correspondingly rapid heat release can elevate loads on the crank-train, necessitating careful control of combustion phasing and pressure-rise limits to protect reliability and durability.

Keywords: railway, hydrogen, engine

1. INTRODUCTION

The decarbonisation of transport propulsion is driven by two converging imperatives: mitigating environmental impact and reducing strategic dependence on hydrocarbon fuels that underpin national economies [1]. Among near-term pathways, repurposing existing piston internal combustion engines (ICEs) to operate on hydrogen offers a pragmatic route that leverages mature platforms, established maintenance ecosystems, and global manufacturing bases [2, 3]. This option is particularly salient for railway applications – such as diesel multiple units (DMUs) and locomotive fleets – where legacy diesel engines combine high efficiency, durability, and wide deployment [4].

A well-recognised limitation of converting liquid-fuel engines to gaseous fuels is the power derate associated with the displacement of intake air by the gaseous fuel, which reduces the trapped fresh charge and, consequently, the charge energy density [1, 5]. The effect is accentuated for low-molecular-weight fuels such as hydrogen, where high specific volume exacerbates the volumetric-efficiency penalty. In externally premixed concepts (external mixture formation, e.g., port fuel injection), this manifests as a trade-off between lean operation – desirable for efficiency and NO_x control – and the available engine power.

Shifting to internal mixture formation (direct injection) can alleviate the air-displacement penalty by introducing hydrogen late in the compression stroke, thereby preserving volumetric efficiency [1]. However, such systems entail added hardware complexity, higher injection pressures, and stringent requirements on injector response and durability, which may challenge retrofit feasibility and cost. Consequently, external mixture formation remains an attractive baseline for conversions, provided that the acceptable power level can be demonstrated a priori through modelling and test planning. The present study addresses this design question for a representative DMU engine (Cummins NTA855-R4), quantifying the attainable power and efficiency under spark-ignited hydrogen operation with port fuel injection and delineating the control constraints that arise from altered combustion phasing and pressure-rise dynamics.

The purpose of this study is to provide an a priori, theory-based estimate of the ultimate specific engine power attainable by the Cummins NTA855-R4 engine – installed in the DM'90 (NS Class 3400) diesel multiple unit – following conversion to spark-ignited hydrogen operation with port fuel injection.

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"Ultimate specific engine power" is defined here as the maximum sustainable rating deliverable under steady-state conditions while respecting durability-relevant constraints on peak cylinder pressure, pressure-rise rate, exhaust temperature, and air-path limits, as well as the volumetric-efficiency penalty inherent to gaseous-fuel operation. The estimation is derived from thermodynamic cycle modelling and parametric sweeps over excess-air ratio and boost conditions representative of the base engine.

2. LITERATURE REVIEW

The first attempts to use hydrogen in transportation date back to the 19th century and saw active development throughout the 20th century, especially during periods of fuel shortages [11]. Today, hydrogen technologies are internationally supported as a strategic pathway toward sustainable development. Hydrogen is considered a key element of future energy systems, capable of enabling the decarbonization of the transport sector. According to projections by the Hydrogen Council, by 2050 approximately 25% of passenger cars and 20% of non-electrified rail transport will operate on hydrogen, potentially reducing oil consumption in transportation by 20% [12, 13].

Currently, there are two main approaches to hydrogen use in transportation: fuel cells [14] and hydrogen as a fuel for internal combustion engines (ICEs) [16–26].

At present, only a few automakers (Toyota, Hyundai, Honda) [15] offer commercial fuel cell vehicle models. In Germany (H2goesRail project) and Japan (HYBAR project), hybrid locomotives powered by hydrogen fuel cells are undergoing testing.

However, fuel cells are not designed for continuous operation at peak power. Their optimal efficiency is achieved under partial loads (30–50% of nominal), which is why they are typically integrated into hybrid systems with buffer batteries or supercapacitors. Additionally, fuel cell production relies on expensive materials such as platinum.

Hydrogen-fueled internal combustion engines are still being researched as alternatives to modern powertrains due to their high engine thermal efficiency (up to 45% [20]), near-zero harmful emissions, and complete absence of greenhouse gases in the exhaust [3, 21–25].

Compared to fuel cell technology, hydrogen ICEs offer several advantages: greater tolerance to fuel impurities, flexibility in switching between fuel types, reduced reliance on rare materials, and easier transition from conventional vehicles. Moreover, hydrogen ICEs benefit from the ability to utilize existing manufacturing infrastructure and processes developed for traditional ICEs, thereby reducing costs [17].

In 2009, BMW and its partners developed a hydrogen engine with spark ignition and direct hydrogen injection into the cylinder at pressures up to 300 bar, achieving a peak efficiency of 42%, comparable to turbocharged diesel engines [18]. This method avoids the issue of backfire typical of intake manifold hydrogen delivery, as injection occurs after the intake valves are fully closed.

Notable studies [21] have examined a turbocharged 2.0 L hydrogen engine with direct injection, achieving a maximum engine thermal efficiency of 42.6% with a slightly lean mixture (λ = 1.91) at 2000 rpm. NO_x emissions were reduced by over 99.5% at speeds below 2000 rpm and by approximately 90% at 4400 rpm. In two-thirds of operating modes, NO_x levels were below 20 ppm thanks to an ammonia-based selective catalytic reduction system (NH_3-SCR). These findings highlight the potential of DI hydrogen systems to deliver high power, efficiency, and near-zero emissions.

Since 2006, Mazda has been developing a hydrogen rotary engine that combines port and direct injection technologies [19].

Hydrogen combustion in ICEs differs fundamentally from diesel fuel:

- The flame speed of hydrogen is 7–8 times higher than that of diesel, resulting in faster flame propagation and earlier heat release [25]. This shortens combustion duration, which is beneficial for high-speed engines but requires precise ignition control.
- Hydrogen's wide flammability range (4–75% in air) allows for leaner mixtures, potentially improving efficiency. However, it also increases the risk of pre-ignition, detonation, and uncontrolled early ignition, especially under high temperature and pressure conditions [16].
- Hydrogen has a higher energy content per unit mass (≈120 MJ/kg) but lower volumetric energy density, complicating storage and transport. Diesel fuel, by contrast, has high volumetric energy density, making it more suitable for conventional fuel systems.
- Hydrogen's ignition energy (≈0.02 mJ) is significantly lower than that of diesel (≈0.25 mJ), increasing the risk of backfire and premature ignition. This necessitates careful mixture control and specialized ignition systems [16].

Special attention is given to hydrogen as an additive to conventional fuels–gasoline, diesel, natural gas, and ammonia [22–25]. Adding hydrogen to fuels like ammonia and natural gas significantly improves engine performance and combustion efficiency while reducing fuel consumption. However, this improvement comes with trade-offs: the increased combustion temperature due to hydrogen enrichment typically raises NO_x emissions, despite reductions in CO_2, HC, and soot. In some cases, elevated NO_x emissions can be mitigated by precisely adjusting the air excess ratio (λ) [25].

In the United States, the retrofitting of switcher locomotives into hydrogen–diesel hybrids is being explored as a means to reduce emissions in port areas. One such example is the Sierra Northern Railway project in California.

Experiments with diesel engines [22] operating with hydrogen added to the intake manifold have demonstrated increased combustion pressure, higher heat release rates, and faster pressure rise under full load conditions. As a result, thermal efficiency improved by up to 17% with 25% hydrogen addition. However, excessive hydrogen content leads to reduced engine stability and significantly increased NO_x and HC emissions. Moreover, this operating mode does not substantially reduce CO and particulate emissions, although CO_2 emissions decrease proportionally to the level of energy substitution. Nevertheless, when 46% of the diesel fuel was replaced with hydrogen, more than a 50% reduction in CO and smoke emissions was recorded compared to conventional diesel operation [27].

These findings highlight the importance of optimizing parameters such as compression ratio, injection timing, and hydrogen flow rate to ensure stable and efficient combustion.

3. MATERIALS AND METHODS

The Cummins NTA855-R4 is a 14-liter, turbocharged, inline six-cylinder compression-ignition engine equipped with a mechanical fuel injection system. The “R4” designation identifies the rail-traction variant of the engine, featuring a horizontal cylinder arrangement specifically configured for railway applications. The power engine is 360 kW at 1800 rpm. Engines of the 855 family have seen extensive deployment in on-road, industrial, marine, and power-generation applications, and the rail-specific R4 variant is widely used in locomotive and DMU service. Key specifications of the research subject are summarized in Table 1. Owing to its mature design, robust duty-cycle capability, and established maintenance ecosystem, the NTA855-R4 presents a suitable platform for hydrogen conversion studies focused on port-injected, spark-ignited operation.

Within the framework of the Latvian innovation project MASOC KC, the Cummins NTA855-R4 engine was upgraded into a hydrogen-fueled power unit, serving as a pre-prototype developed under the project’s applied research activity. The objective of this phase was to obtain additional empirical data not available from other sources and to refine the parameters required for the theoretical evaluation of engine operating characteristics.

This effort will ultimately enable the creation of one of the first passenger trains in Europe powered by a hydrogen internal combustion engine (H_2-ICE), by completing the retrofit design and full-scale installation during the project’s experimental development phase.

The locomotive retrofit system integrates five principal subsystems, including the hydrogen storage and supply system, the fuel regulation and injection system, the hydrogen-adapted internal combustion engine, the electronic control and telemetry units, and the integrated safety management system. All components are accompanied by conformity certificates and test reports confirming their suitability for safe H_2-ICE pre-prototype operation. In particular, the high-pressure hydrogen storage and piping assemblies were sourced from suppliers already providing certified equipment for authorized railway hydrogen vehicles, ensuring proven safety and reliability under real railway operating conditions.

Table 1. Brief technical characteristics of the engine Cummins NTA855-R4

Type		4-stroke; Horizontal In-line; 6-Cylinder
Aspiration		Turbocharged; Aftercooled
Bore	mm	140
Stroke	mm	152
Displacement	liters	14
Compression Ratio		14:1
Firing Order		1-5-3-6-2-4
Dry Weight	kg	1300

The installation of the hydrogen engine on the DM’90 multiple unit, viewed from the cylinder-head side, is shown in Figure 1. The photograph illustrates several design modifications, including the installation of ignition coils and spark plugs.

As a result, the DM’90 Cummins-based retrofit system has achieved the goals of the applied research stage of the project. The hydrogen engine control system pre-prototype has been installed on the engine and prepared for testing. Even at pre-prototype stage, the system design incorporates the relevant railway regulatory and safety requirements, including TSI LOC & PAS, EN 45545, and EC 79/2009, and is therefore ready to progress to the next phase of the project—experimental development, which includes final design, validation under real operating conditions, and demonstration.
To ensure predictable performance outcomes during operational testing, a dedicated research task has been established to model and predict hydrogen fuel consumption in service [8].

4. RESULTS

From the theory of internal combustion engines, it is known that the effective power of the engine [N_e, W] is determined by the following ratio of its design and mode parameters [6]:

$$N_e = \frac{V_h \cdot n}{120} \times \frac{\eta_{VS} \cdot p_S}{T_S} \times \frac{\eta_e \cdot Q_{low}}{8314 \cdot \lambda \cdot M_0}. \quad (1)$$

In this formula: V_h – working volume of the engine, n – crankshaft speed, η_{VS} – cylinder filling factor by conditions before the valves, p_S, T_S – respectively pressure and absolute temperature of air (fresh mixture) before the valves at the inlet, η_e – effective efficiency of the engine, Q_{low} – the lowest mass

Fig. 1. Hydrogen engine installation (cylinder-head side view) on the DM'90 railcar

calorific value of the fuel used during combustion, λ – excess air ratio, M_0 – stoichiometric number of fuel.

The analysis of formula (1) shows that its right part can be divided into three components. The first component is related to the engine design parameters (total cylinder volume and speed) and remains constant when converting to hydrogen. For the Cummins NTA855-R4 engine, the values are: V_h = 14 liters, n = 1800 min^{-1} (nominal).

The second component - design parameters of the air supply system (supercharging units) - can also be assumed for this engine at its conversion to hydrogen constant: $\eta_{VS} \approx 0{,}95$, p_S = 2,59 bar, T_S = 333 K.

The third component is variable and depends on fuel parameters and efficiency of the working process depending on them. So, for diesel fuelQ_{low} = 42300 kJ/kg, M_0 = 0,5 kmol air/kg fuel,λ = 1,8, and for hydrogen Q_{low} = 120000 kJ/kg, M_0 = 1,2 kmol air/kg fuel. Besides, it is known that in order to avoid detonation combustion of hydrogen in the cylinder it should be burned at high values λ = 2,5–3,0 [7]. It is rather difficult to estimate the change in the value of the effective efficiency of engineηe without a refined calculation of the engine operating process.

It should be noted that using the set of the above parameter values it is also possible to theoretically estimate the cyclic fuel supply (for designing the fuel apparatus)

$$B_C = \frac{120N_e}{n \cdot z} \times \frac{1}{\eta_e \cdot Q_{low}}, \qquad (2)$$

and hourly fuel consumption (for estimation of economic feasibility of engine conversion):

$$G_F = \frac{3600 \cdot N_e}{\eta_e \cdot Q_{low}} \qquad (3)$$

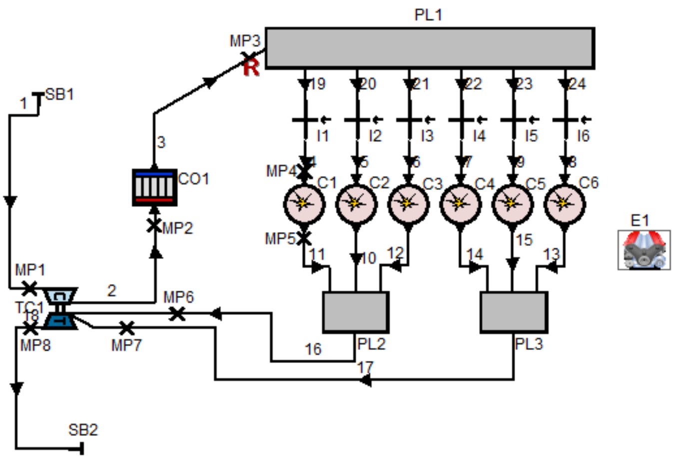

Fig. 2. Calculation diagram of the NTA855-R4 engine mathematical model

In order to evaluate the magnitude and variation of the effective efficiency and other important engine performance parameters, mathematical modelling of the operating process of the NTA855-R4 engine operating at nominal mode was performed for three cases - use of diesel conventional fuel ("DF"), use of hydrogen with internal blending ("H2int") and use of hydrogen with external blending ("H2ext").

The external view of the model created in the AVLBoost®, is shown in Fig. 2. The elements I1–6 shown in the diagram are individual injectors of external hydrogen supply. They were ignored in the calculation of the diesel cycle and the hydrogen cycle with internal mixing.

The diagram also indicates the individual structural components of the engine: E1 – Engine assembly; TS1 – Turbocharger; CO1 – Charge air cooler; PL1 – Intake manifold; C1–C6 – Engine cylinders; PL2 – Left exhaust manifold; PL3 – Right exhaust manifold; 1–29 – Pipelines and gas channels; MP1–MP8 – Parameter measurement points; SB1 – Inlet boundary conditions; SB2 – Outlet boundary conditions.

The main results of the conducted mathematical modelling of the nominal operating mode of the Cummins NTA855-R4 engine for three variants of the fuel used in it are presented in Table 2.

Table 2. Main results of modelling

Parameter	DF	H2int	H2ext
Peak Fir.Pres. [bar]	128.5	154.7	142.6
Peak Pres.Rise [bar/deg]	3.10	6.19	5.24
Peak Fir. Temp. [K]	1865	1807	1827
BMEP [bar]	17.11	12.98	10.48
Indicated Eff. [-]	0.441	0.483	0.450
Brake Power [%]	41.2	44.1	40.2
Eff. Rel. Energy [kJ]	58.34	41.25	36.34
Fuel Massfl. [g/s]	20.7	5.175	4.58
Excess Air Ratio λ [-]	1.79	2.99	2.99
Effective Power [kW]	360.4	273.4	220.7

5. DISCUSSION

The results indicate that, under hydrogen operation, the peak firing pressure (“Peak Fir. Pres.”) rises by 20% and 11% for internal and external mixture formation, respectively, while the maximum rate of pressure rises (“Peak Pres. Rise”) increases by 100% and 69% in the same order. Concurrently, the peak in-cylinder temperature (“Peak Fir. Temp.”) decreases slightly owing to the higher excess-air ratio.

For the nominal hydrogen settings modelled, the brake means effective pressure (BMEP) and, consequently, the engine power (“Effective Power”) are reduced by 24% and 39%, respectively. The cycle efficiency increases with internal mixture formation: both the indicated efficiency (“Indicated Eff.”) and the brake engine efficiency rise by approximately 7–10%. With external mixture formation, the efficiencies remain essentially unchanged relative to diesel (±2%).

These quantities provided the missing inputs needed to evaluate equations (1) – (3). The corresponding calculations are summarized in Table 3.

The close agreement between Table 3 and the simulation outputs validates both the model synthesis and the theoretical relations introduced earlier.

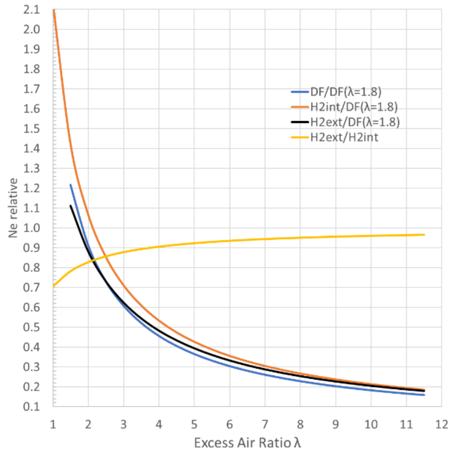

Fig. 3. Relative change of engine power with change of excess air ratio during combustion in the cylinder

In addition, using equation (1), an analytical exploration of the effect of excess-air ratio on engine power for the Cummins NTA855-R4 was performed; the relative power change versus diesel is shown in Fig. 3. To obtain the relative change in power, we used the equation:

$$N_{e\ relative} = \frac{N_e}{N_{e\ DF(\lambda=1.8)}}. \qquad (4)$$

Here $N_{e\ DF(\lambda=1.8)}$ – is the rated power of the diesel engine.

Table 3. Results of theoretical analysis

		DF	H2int	H2ext	Dimensionality
Initial data	Engine litre	14	14	14	L
	Speed	1800	1800	1800	rpm
	Number of cylinders	6	6	6	
	Boost pressure	2.59	2.59	2.59	bar
	Charge air temperature	60	60	60	C
	Filling factor	0.95	0.95	0.95	
	Effect. efficiency	0.412	0.441	0.402	
	Lower calorific value	42319	120000	120000	kJ/kg
	Stoichiometric ratio	0.5	1.2	1.2	kmol air/kg fuel
	Excess Air Ratio λ	*1.8*	*3*	*3*	
Calculation results	Effective power	362	274	220	kW
	Cycle fuel supply	0.230	0.058	0.051	g/cycle
	Hourly fuel consumption	74.65	18.66	16.39	kg/hour

As expected for quality-governed control, power decreases approximately inversely with increasing excess-air ratio. Over the range λ = 2–3, the power difference between external and internal mixture formation is modest (≈12–18%) and diminishes further as λ increases. This implies that, at part-load conditions governed by mixture quality (hydrogen enabling operation at λ up to ~10 and above), external mixture formation can preserve the average operating power while offering lower hardware complexity.

Moreover, within λ = 2–3, the externally premixed hydrogen configuration is predicted to deliver approximately the same effective power as the baseline diesel (see Fig. 4), supporting the feasibility of complete diesel-to-hydrogen substitution without compromising power capability. Nevertheless, the rapid heat-release characteristics of enriched hydrogen–air mixtures elevate crank-train forces and mechanical loads, necessitating stringent control of combustion phasing and pressure-rise limits to protect reliability and durability.

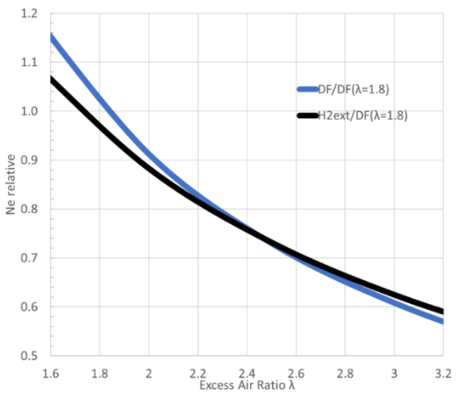

Fig. 4. Relative change of engine power in λ=2–3

6. CONCLUSIONS

1. Despite their reliability and widespread use, diesel locomotives significantly contribute to greenhouse gas and particulate emissions. Hydrogen, as a clean energy carrier, has the potential to deliver near-zero exhaust emissions and substantial reductions in CO_2, NO_x, and particulate matter, aligning with global decarbonization goals and increasingly stringent emission standards in the transport sector. However, large-scale implementation requires further optimization of design and control parameters, development of ignition management systems, and advancements in hydrogen storage technologies.
2. The proposed is the method of a priori analysis of the change of power parameters of the Cummins NTA855-R4 diesel engine operation at transition to hydrogen fuel allows to estimate its future technical and economic operational performance.
3. Upon conversion to hydrogen, the peak in-cylinder (firing) pressure increases by 20% and 11% for internal and external mixture formation, respectively, and the maximum pressure-rise rate increases by 100% and 69%, respectively. The peak in-cylinder temperature decreases slightly owing to the higher excess-air ratio, while the brake mean effective pressure (BMEP) and engine power decrease by 24% and 39%, respectively. The indicated and engine efficiencies increase by approximately 7–10% with internal mixture formation, whereas with external mixture formation they remain essentially unchanged relative to diesel (±2%).
4. The study confirms the feasibility of complete diesel substitution with hydrogen without deterioration of power capability for the Cummins NTA855-R4 operating with external mixture formation in the range of excess-air ratio λ = 2–3. However, the rapid heat-release characteristics of enriched hydrogen–air mixtures increase gas forces and mechanical loads on the crank-train and may affect reliability and durability.

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