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An Innovative Thermal Flow Management Approach for High-Altitude Turboshaft Test Benches

Luca Piancastelli
Department of Industrial Engineering (DIN),
Alma Mater Studiorum—University of Bologna,
40136 Bologna, Italy
e-mail: luca.piancastelli@unibo.it

Irene Giusti[1]
Department of Industrial Engineering (DIN),
Alma Mater Studiorum—University of Bologna,
40136 Bologna, Italy
e-mail: irene.giusti4@unibo.it

Marella De Santis
Department of Industrial Engineering (DIN),
Alma Mater Studiorum—University of Bologna,
40136 Bologna, Italy
e-mail: marella.desantis2@unibo.it

This article presents the design concept and preliminary analytical assessment of altitude engine test benches for aircraft propulsion systems, covering piston engines up to 1000 HP and turboshaft engines up to 2000 HP. The benches replicate environmental conditions from sea level to altitudes above 11,000 m, with ambient pressures down to 22,600 Pa and temperatures as low as −56.5 °C, encompassing the full flight envelope from takeoff to high-altitude cruise. Key system requirements include high-capacity air supply and vacuum subsystems to maintain precise pressure, temperature, and airflow profiles. First-order analytical calculations estimate air intake mass flowrates of 1.6–2.0 kg/s for piston engines and 8–9 kg/s for turboshaft engines, with required cooling power of about 769 kW. The power for vacuum generation is estimated between 1.5 MW and 1.9 MW, depending on technology (mechanical vacuum pumps or steam ejectors). Capital costs are estimated at $300,000–$500,000, while operating costs range from $606,000 to $1,725,000 per year. Cooling strategies, including liquid nitrogen and closed-loop cryogenic systems, are compared, highlighting trade-offs in operational and capital costs. Decompression subsystems are analyzed through mechanical vacuum pumps and steam ejectors, assessing accuracy, control, and cost. The study is based on analytical calculations; no numerical simulations, experimental validation, or uncertainty analysis are included. Results provide initial guidance for high-fidelity engine testing and serve as a foundation for future work on integrated control, exhaust management, and dynamic simulation of complex mission profiles. [DOI: 10.1115/1.4071798]

Keywords: thermal management, altitude test Bench, turboshaft, heat transfer, aircraft propulsion system, aeronautical and aerospace propulsion systems, engines, thermal analysis

1 Introduction

Altitude engine test benches are critical facilities for evaluating aircraft propulsion systems under conditions that replicate the full flight envelope, from takeoff through climb, cruise, and descent. These facilities pose significant engineering challenges due to extreme variations in pressure, temperature, and airflow, as well as high installation costs, operational complexity, and stringent reliability requirements.

Aircraft engines include piston engines (gasoline and diesel) and turboshaft engines. In this study, engines up to 2000 HP are considered for the test bench design, covering the range relevant for medium-power aviation propulsion systems. While piston engines are generally less sensitive to altitude-related cooling limitations, turboshaft engines require precise air supply, thermal conditioning, and vacuum management to maintain performance and safety at simulated high altitudes [1,2].

This work goes beyond a conventional engineering report by providing a systematic analysis of design trade-offs and system performance across the full flight envelope. We quantify airflow requirements, cooling power, and vacuum generation needs for piston and turboshaft engines under a wide range of altitudes and temperatures, from sea level to 11,000 m and beyond. By evaluating both technical performance and economic considerations, the study establishes a foundation for scalable, high-fidelity test bench design and highlights insights applicable to future research in integrated control strategies, exhaust management, and dynamic mission simulation.

The following sections present general characteristics of altitude test benches, discuss design approaches for engines of varying power, and analyze cooling and decompression strategies, emphasizing novel solutions for managing thermal and fluid-dynamic challenges across the complete flight spectrum [3,4].

1.1 General Characteristics. Altitude engine test benches are designed to reproduce environmental conditions encountered throughout the complete flight envelope, including takeoff, climb, cruise, descent, and landing. The main objectives are to ensure precise control of chamber pressure, temperature, and

[1]Corresponding author.
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airflow, while maintaining thermal stability and operational flexibility for both piston and turboshaft engines.

The test chambers are designed to reproduce pressures ranging from atmospheric conditions at sea level (approximately 101,325 Pa) down to around 14,100 Pa at 14,000 m altitude, and temperatures from 35 °C at ground level to −56.5 °C at typical cruising altitudes. The upper temperature bound of approximately +120 °C does not correspond to standard engine intake conditions, but reflects the extended capability of industrial altitude test chambers, which allow testing components and subsystems under high thermal stress and abnormal operating scenarios. Such chambers commonly support temperatures from −70 °C to +150 °C or higher while maintaining low-pressure environments for aerospace and avionics testing [5,6]. Environmental testing standards, including IEC 60068, define conditions for evaluating equipment performance under both cold and dry heat extremes, providing further justification for the broad chamber temperature capabilities [7].

Airflow requirements naturally scale with engine type and power: small piston engines (less than 100 HP) typically require between 0.1 and 2 kg/s, larger piston engines up to 1000 HP demand 2–5 kg/s, and turboshaft engines up to 2000 HP may require approximately 8–10 kg/s of intake air. The intake air mass flow for a representative 2000 HP turboshaft engine is estimated at this higher range for test bench design purposes. While Nkoi et al. [8] report a nominal intake of about 5.7 kg/s for a 2100-HP two-spool turboshaft engine at design conditions, the higher value used here provides a conservative margin to account for transient engine demands, pressure drops, and operational variability, and is consistent with general engineering analyses of turboshaft engine inlet flows [9]. These values reflect the dynamic variations in engine demand at different flight phases [10,11].

For small piston engines equipped with exhaust-driven turbochargers, the turbo introduces additional parasitic loads and influences oil consumption. Experimental studies indicate that turbochargers can contribute a measurable fraction of total engine oil consumption under sustained operation [12] and cause slight reductions in brake power due to bearing drag and lubrication requirements [13]. These quantitative results support the qualitative statements in earlier drafts and justify the observed performance impacts in simplified test bench configurations.

During takeoff at sea level, pressure and temperature are highest, and airflow requirements are moderate. For example, a typical 2000-HP turboshaft engine requires about 8 kg/s of air, while a piston engine of similar size requires 1.5 kg/s. During climb, altitude increases to around 5000 m, pressure decreases to roughly 54,000 Pa, and temperatures drop to near 5 °C, leading to slightly higher airflow demands due to lower air density. At cruise altitudes near 11,000 m, pressure drops further to approximately 22,600 Pa and temperatures reach −56.5 °C; under these conditions, turboshaft engines may require up to 9 kg/s of intake air, while piston engines require around 2 kg/s. During descent and landing, environmental conditions gradually return to near sea-level values, with airflow requirements correspondingly decreasing [14,15].

This description highlights the technical challenges faced by high-altitude engine test benches. The dynamic variation of pressure, temperature, and airflow demands requires advanced real-time control strategies. Intake air conditioning systems must compensate for density changes and maintain stable thermodynamic properties, while decompression systems, whether based on multistage vacuum pumps or steam ejectors, must rapidly adjust to preserve target chamber pressure without inducing flow instabilities. The design must ensure that both piston and turboshaft engines can be tested safely under transient conditions that mimic rapid changes in flight profile, such as climb and descent sequences.

The approach presented in this study allows scalable, real-time management of thermal and fluid dynamic conditions across the entire flight spectrum. By integrating precise airflow regulation, adaptive cooling, and controlled decompression, the test bench can reproduce realistic environmental loads while minimizing operational risk, energy consumption, and thermal stress on the engine and supporting systems. This represents a key technical novelty compared to conventional altitude test setups, which are often limited to quasi-static conditions or single-phase operation.

This study introduces a distinct methodological approach compared to conventional altitude test benches [e.g., DLR, ONERA, NASA Numerical Propulsion System Simulation (NPSS)-based facilities]. The proposed strategy integrates continuous cryogenic air cooling with modular decompression, enabling precise and dynamic control of temperature, pressure, and air density throughout the entire flight envelope. Unlike traditional systems that often operate under quasi-static conditions or rely on intermittent LN_2 cooling, this approach allows real-time adaptation to transient flight scenarios, including rapid climbs, descents, and variable engine loads. Furthermore, the methodology couples detailed thermodynamic validation [number of transfer units (NTU), log mean temperature difference (LMTD), coefficient of performance (COP), and exergy efficiency] with economic assessment (CAPEX/OPEX), providing a robust framework for scalable, flexible, and energy-efficient test bench design across a wide range of engine powers and operational conditions. This integrated, quantitatively validated strategy represents the key methodological innovation of the present work.

1.2 Test Bench for Small Piston Engines (Under 100 HP).

A small piston engine test bench consists of a pressurized chamber housing the engine, with the dynamometer located outside. Air is supplied via a small intake with variable cross section or butterfly valve, using cooled gaseous air or a mixture of gas and liquid air. Exhaust gases are directed into an ejector system to generate a vacuum. While cost-effective, this configuration cannot independently regulate chamber pressure or simulate exhaust backpressure, limiting applicability to engines tolerant of simplified altitude conditions [16].

1.3 Economical Alternative Design.

An alternative uses a turbine driven by exhaust energy to power a turbocharger. The compressor draws air from the chamber and discharges it to the atmosphere. This reduces engine power slightly and increases oil consumption, but enables vacuum generation at lower cost. Limitations include maximum airflow (1 kg/s for commercially available automotive turbochargers) and achievable altitude (10,000 m for 200- to 300-kW engines). As with the previous setup, chamber temperature and pressure remain coupled to intake air. For higher power engines, intake air is compressed through a convergent duct, allowing operation at higher altitudes, but increasing complexity and cost. Subsequent sections focus on the air supply system of the engine intake [17,18].

2 Altitude-Capable Engine Test Benches: Current Applications and Developments

Engine performance changes significantly with altitude due to variations in air pressure, temperature, and density. To replicate these conditions, altitude-capable engine test benches employ vacuum systems, thermal conditioning, and advanced pressure regulation technologies.

High-altitude test facilities have historically focused on both rocket and aircraft engines, demonstrating the critical role of precise pressure and thermal control in performance evaluation [3,4,19]. Early conceptual models, such as those described by Kanda et al. [4], provide foundational approaches to the design of high-altitude test stands, emphasizing the importance of simulating realistic environmental conditions.

One of the most prominent modern facilities is the DLR's P4.1 and P4.2 sites in Lampoldshausen, Germany, designed for rocket engine qualification, including VINCI and AESTUS. These test benches achieve pressures below 5 mbar—simulating altitudes beyond 30,000 m—using high-capacity steam ejectors and cold

Table 1 Key features of major altitude engine test facilities

Facility	Max sim. altitude	Min pressure	Tested engine types	Max engine power
DLR P4.1/P4.2 (Germany)	>30, 000 m [22]	<5 mbar [22]	Rocket engines, cryogenic propellants	Up to 250-kN thrust
NASA PSL (USA)	Up to 27,400 m [23]	9 mbar [23]	Jet, turbofan engines	Up to 100-kN thrust
University of Wisconsin (USA)	8000 m [24]	300 mbar [24]	Diesel, dual-fuel, biofuel engines	1000 HP
ONERA PF52 (France)	27,000 m [25]	<10 mbar [25]	Turbojets, ramjets, air intakes	Up to 20-kN thrust
TsIAM (Russia)	>20, 000 m [26]	<15 mbar [26]	Turbofans, military engines, icing tests	30-kN thrust
AVIC GTE Xi'an (China)	25,000 m [27]	¡10 mbar [27]	UAV engines, turbofans, afterburners	10-kN thrust
QinetiQ ATF (UK)	21,000 m [28]	<12 mbar [28]	UAV propulsion, turbofans, integrated systems	15-kN thrust
NPSS-based Simulation (USA)	Varies (modeling) [29]	Simulated [29]	Jet, turbofan (simulation and design)	Up to 200-kN equivalent

Note: NPSS is a numerical tool used to model engine and facility performance; all other facilities are physical test benches with measured or manufacturer-specified capabilities. Max engine power values are indicative and based on cited references.

traps, and support continuous testing of engines up to 250-kN thrust. Similarly, NASA has developed simulation models using the NPSS environment to enhance the design of altitude test systems, allowing detailed evaluation of heat transfer, ejector performance, and transient engine behavior [19].

For reciprocating engines, facilities such as the University of Wisconsin–Madison Diesel engine altitude test cell enable full regulation of intake air pressure, temperature, and humidity, allowing precise combustion studies under aeronautical conditions [20]. Modern control strategies, including active disturbance rejection control, have further improved the stability and precision of high-altitude simulations [21].

Key features of these major altitude engine test facilities, including maximum simulated altitudes, minimum achievable pressures, and tested engine types, are summarized in Table 1. This table provides a concise overview and allows direct comparison of capabilities across different laboratories worldwide.

Collectively, these facilities illustrate the critical combination of vacuum generation, thermal conditioning, and advanced control systems necessary for high-fidelity altitude engine testing. They provide essential infrastructure for propulsion research, certification, and development across a wide range of engine technologies (see Table 1).

3 Engine Intake Air Conditioning System

The design of an engine intake air conditioning system begins with the determination of the mass flowrate required by the engine under test. This parameter depends on engine architecture, displacement, expected operating regime, and the target altitude and temperature conditions to be simulated within the test cell. An accurate estimation of the intake air demand is crucial to guarantee that the engine receives airflow at the desired thermodynamic state, defined by controlled pressure, temperature, humidity, and, when required, specific gas composition.

Once the airflow requirement is established, the conditioning system is sized and designed to supply, regulate, and stabilize the intake air throughout the entire operating envelope. Depending on the test objectives, the system may operate in open-loop mode—drawing and exhausting air to the ambient—or in closed-loop mode, where the airflow is recirculated. Closed-loop configurations are generally preferred for high-altitude or extreme-temperature simulations, as they offer improved control authority over pressure and humidity and reduce sensitivity to external environmental fluctuations.

Advanced real-time controllers employing model-based and feed-forward logic are typically implemented to maintain stability under transient load conditions, throttle changes, or rapid engine accelerations. Thermal management also becomes critical at low intake temperatures: condensation, frost formation, and localized pressure drops must be avoided, particularly during cold-start or high-altitude simulations. Insulated ducting, controlled reheater stages, moisture separators, and pressure-relief devices are integrated to preserve consistent and safe intake conditions across the entire test duration [30,31].

3.1 Humidity Control and Frost Prevention. Although the primary focus of this study is on pressure, temperature, and airflow regulation, intake air humidity is an important secondary parameter for high-altitude engine testing. At low temperatures, condensation and frost formation can occur, potentially affecting engine performance and measurement accuracy.

Design considerations for humidity control include the integration of moisture separators, heated air ducts, and controlled reheater stages to reduce the risk of frost. Additionally, real-time monitoring of air temperature and relative humidity ensures that the intake air remains within safe thermodynamic limits. These measures help prevent ice formation in ducts, valves, and the engine intake, maintaining consistent test conditions throughout all flight phases.

Detailed modeling of humidity dynamics is beyond the scope of this article, but these strategies provide a practical approach to managing condensation and frost risks in high-altitude engine test benches.

3.2 Air Consumption of a Diesel Engine. The air mass flowrate $\dot{m}_{\text{air}}$ for a large diesel engine can be estimated from its brake power, fuel conversion efficiency, and air–fuel ratio (AFR). The brake thermal efficiency (η) represents the fraction of fuel chemical energy converted to useful shaft power, and for large compression-ignition engines is typically higher than spark-ignition counterparts due to their higher compression ratios and combustion characteristics [32].

Reported brake thermal efficiencies for modern large diesel engines span a range from about 35% to over 45% in practical applications, with marine and heavy stationary diesels achieving even higher values under optimal conditions [32,33]. To account for this variability, a sensitivity analysis of the intake air requirement was performed considering efficiencies of 30%, 38% (baseline), and 45%.

For a 2000-HP ($\approx$ 1.49 MW) diesel engine, the required fuel energy input P_{fuel} is inversely proportional to η:

$$P_{\text{fuel}} = \frac{P_{\text{output}}}{\eta} \tag{1}$$

As η decreases, more fuel must be supplied to produce the same brake power, resulting in a proportionally higher air demand via the AFR. Assuming a representative AFR of 17 for large diesel engines, the intake air mass flow scales with $\dot{m}_{\text{fuel}}$.

Table 2 summarizes the estimated intake air mass flow for different assumed thermal efficiencies. These values illustrate that, within a realistic efficiency range, the required air mass flow varies moderately.

The results indicate that, over a $\pm$10% variation in assumed thermal efficiency, the intake air requirement changes by roughly

Table 2 Sensitivity of diesel intake air requirement to brake thermal efficiency

Thermal efficiency	Fuel power (MW)	Fuel flow (kg/s)	Air mass flow (kg/s)
30%	4.97	0.117	1.99
38% (baseline)	3.92	0.093	1.57
45%	3.31	0.079	1.34

±20% relative to the baseline estimate. This variation is modest compared to other sources of uncertainty (e.g., transient load conditions, actual AFR under different operating points); yet, it demonstrates the importance of choosing a justified efficiency value for design purposes.

In this study, the baseline efficiency of 38% was selected as representative of a large diesel engine operating near typical cruise conditions. This value lies within the range reported in the literature, and the sensitivity analysis confirms that deviations within practical bounds do not substantially alter the overall design direction for the intake air conditioning system.

3.3 Air Consumption of a Gasoline Engine. The 2000-HP gasoline engine considered in this study is purely hypothetical and used solely for test bench design and illustrative purposes. It does not represent a practical modern aircraft engine, but serves to demonstrate the methodology for estimating intake air demand and designing the conditioning and decompression systems. The selected power level allows comparison with diesel and turboshaft engines to highlight variations in airflow requirements across different engine types.

For gasoline engines, the lower thermal efficiency and the stoichiometric AFR modify the airflow requirement significantly. Considering a 2000-HP gasoline engine with thermal efficiency $\eta = 20\%$, the fuel power demand is

$$P_{\text{fuel}} = \frac{P_{\text{output}}}{\eta} \approx 7.46 \times 10^6\,\text{W} \tag{2}$$

Taking a gasoline LHV of 44 MJ/kg, the corresponding fuel mass flowrate is

$$\dot{m}_{\text{fuel}} = \frac{P_{\text{fuel}}}{\text{LHV}} \approx 0.17\,\text{kg/s} \tag{3}$$

Given the stoichiometric AFR of 14.7, the intake air flowrate results in

$$\dot{m}_{\text{air}} = \text{AFR} \times \dot{m}_{\text{fuel}} \approx 2.5\,\text{kg/s} \tag{4}$$

3.4 Air Consumption of a Turboshaft Engine. Turboshaft and turbojet engines process substantially larger quantities of air compared to reciprocating engines of similar shaft power. As a reference, the air-cooled Pratt and Whitney Canada PT6A-6 produces approximately 578 HP while ingesting about 2 kg/s of air.

Assuming similar thermodynamic behavior, a 2000-HP turboshaft engine would require on the order of 8 kg/s of intake air. It is important to note that this value represents a first-order, order-of-magnitude estimate based on linear scaling with engine power. This approach does not explicitly account for key thermodynamic parameters such as the compressor pressure ratio, compressor efficiency, or turbine inlet temperature, which would influence the actual airflow demand.

The estimate includes a conservative margin to accommodate transient engine demands, pressure drops, and operational variability inherent to test bench environments. While Nkoi et al. [8] report a nominal intake of about 5.7 kg/s for a 2100-HP two-spool turboshaft engine under design conditions, the higher value used here is justified for preliminary design purposes.

Table 3 Intake air mass flowrates for different engine types across flight phases (ISA standard temperatures)

Flight phase	Alt (m)	Temp (°C)	Piston engine (kg/s)	Turboshaft engine (kg/s)
Takeoff	0	15	1.6	8.0
Climb	5000	−17	1.8	8.5
Cruise	11,000	−56.5	2.0	9.0
Descent	5000	−17	1.8	8.5
Landing	0	15	1.6	8.0

Limitations: The linear scaling assumption provides a useful guideline for sizing the air conditioning and decompression systems, but should not be interpreted as a precise prediction of the engine's real operational airflow. Detailed design would require a full thermodynamic analysis accounting for compressor and turbine maps, ambient conditions, and transient behavior, which is beyond the scope of this preliminary assessment.

3.5 Airflow Requirements Across Flight Phases. To illustrate the variation of intake air demands, Table 3 summarizes representative airflow requirements for diesel, gasoline, and turboshaft engines at different altitudes and temperatures corresponding to key phases of flight. These values consider both the engine's power output and the changing air density with altitude.

These data indicate that intake air mass flowrates vary along the flight envelope due to changes in air density and ambient temperature, even when the engine operates at nominal power. For piston engines, the mass flow ranges from 1.6 kg/s at takeoff and landing to 2.5 kg/s at cruise altitude. This accounts for both diesel and gasoline engines, with gasoline engines generally requiring slightly higher airflow due to lower thermal efficiency and higher stoichiometric AFR. Turboshaft engines process substantially larger quantities of air, increasing from 8.0 kg/s at takeoff and landing to 9.0 kg/s during cruise. These variations highlight the need for an intake air conditioning system and decompression strategies capable of adapting dynamically to the changing thermodynamic conditions throughout the entire flight envelope.

3.6 Integration With Air Conditioning and Decompression Systems. The air conditioning system is designed to maintain the target intake temperature and pressure across these varying airflow rates. Cooling stages, reheaters, and humidity control devices adjust continuously as the airflow changes during different flight phases. Similarly, decompression systems, whether based on multistage vacuum pumps or steam ejectors, actively regulate the chamber pressure to match the desired environmental conditions.

For example, during cruise at high altitude, low air density requires the system to deliver higher volumetric airflow to maintain the same mass flowrate. At the same time, the cryogenic or mechanical cooling system must compensate for extreme-temperature differences, while decompression stages reduce pressure to the target 22,600 Pa. During climb or descent, transient adjustments are needed to prevent overshoot in temperature or pressure, ensuring smooth transitions between flight phases without introducing flow instabilities.

This integrated approach ensures that each engine type receives conditioned intake air appropriate to its operational characteristics throughout the flight envelope, supporting accurate testing of combustion, performance, and thermal behavior under realistic environmental conditions.

Specifically, for piston engines (both diesel and gasoline), the intake air mass flow ranges from 1.6 kg/s at takeoff and landing to 2.5 kg/s at cruise altitude. Turboshaft engines require significantly larger flows, increasing from 8.0 kg/s at takeoff and landing to 9.0 kg/s during cruise. The air conditioning and decompression systems automatically adjust to these values, ensuring proper intake conditions for each engine type throughout all flight phases.

4 Industrial Cooling of Air to Subzero Temperatures

Several industrial strategies can be considered to cool a continuous air flow of 10 kg/s from an initial temperature of 20 °C to a target of −56.5 °C at atmospheric pressure:

- *Multistage vapor compression refrigeration*: Uses refrigerants such as R404A or R508B. Capable of reaching very low temperatures but requires complex, high-cost equipment.
- *Cryogenic cooling with liquid nitrogen*: Heat exchange with liquid nitrogen (boiling point −196 °C) offers efficient cooling, particularly for intermittent processes. Operational costs and infrastructure requirements are high.
- *Turboexpansion cycle (Brayton or Joule–Thomson)*: Involves compression, intercooling, and expansion through a turbine. Widely used in air separation and gas liquefaction plants for continuous subzero cooling.
- *Linde–Hampson cycle*: A self-refrigerating cycle based on compression, precooling, and Joule–Thomson expansion. Suitable for continuous operation and compatible with electrically powered systems.

While alternative refrigeration strategies such as turbo-Brayton or mixed-refrigerant cycles exist, a detailed quantitative comparison is beyond the scope of this study. The Linde–Hampson cycle is selected as the most suitable solution due to its ability to provide continuous, electrically driven subzero cooling with effective energy recovery, scalable operation, and precise thermal control under varying mass flows and pressure conditions.

For operation down to −56.5 °C, several refrigerants can be considered for the Linde–Hampson cycle, including R23, R508B, and R404A. R23 offers excellent thermodynamic performance at very low temperatures but has a very high global warming potential (GWP ≈ 14,800) and is subject to strict regulatory limits. R508B, a near-azeotropic blend, achieves similar low-temperature performance with slightly lower GWP (≈ 1370) and improved efficiency in Joule–Thomson expansion, making it a common choice in industrial subzero refrigeration. R404A, while widely used in commercial applications, exhibits reduced efficiency at very low temperatures and a moderate GWP (≈ 3920), limiting its environmental desirability.

In selecting a refrigerant, a trade-off must be considered between energy efficiency and environmental impact. R508B and similar low-GWP blends offer a balanced solution, enabling high COPs (≈ 1.5–1.6) while reducing climate impact compared to high-GWP pure gases. Additionally, the electrical operation of the Linde–Hampson cycle allows integration with renewable energy sources, further mitigating lifecycle carbon emissions. In this study, the analysis assumes the use of a low-GWP refrigerant (e.g., R508B), consistent with both industrial feasibility and environmental considerations. A more detailed refrigerant selection and cycle optimization could be performed in future work, including evaluation of energy efficiency, availability, and regulatory compliance.

At the current analytical stage, the study assumes idealized control of temperature and pressure for preliminary sizing, without implementing a specific control algorithm. In a practical test bench, regulation of the cryogenic air flow and vacuum would likely employ a combination of conventional PID loops for fast feedback, complemented by feed-forward control to anticipate changes in engine mass flow and inlet conditions. For more advanced operation under complex transient scenarios, model-predictive control could be adopted to optimize the system response across multiple interacting variables, including temperature, pressure, and energy consumption. This staged control strategy ensures stable and precise operation across the full flight envelope while maintaining energy efficiency.

A comparison with conventional cooling strategies highlights the advantages of the proposed continuous refrigeration approach. Intermittent LN_2 cooling, while capable of achieving very low temperatures, typically exhibits low operational efficiency (COP < 1) and high recurring costs due to continuous LN_2 consumption and associated logistics. Steam ejector-based vacuum systems can provide the required decompression but demand significantly higher power input (often 1.5–1.9 MW for similar airflow rates) and respond more slowly to transient changes in temperature or flow, reducing overall controllability.

In contrast, electrically powered continuous systems based on the Linde–Hampson or turboexpansion cycle achieve higher thermodynamic efficiency (COP ≈ 1.5–1.6), lower operational costs, and allow precise real-time control of both temperature and pressure. This enables stable operation across the entire flight envelope, including rapid climbs, descents, and variable engine loads, providing both energy and operational advantages over LN_2 or steam ejector-dominated setups.

For continuous industrial operation powered by electricity, the Linde–Hampson cycle or a turboexpansion-based process is the most viable solution [34,35]. These options provide reliable subzero cooling, energy recovery through internal heat exchangers, and scalability.

The required cooling power can be estimated using a simple energy balance, assuming constant specific heat capacity c_p for air and no phase change:

$$\dot{Q} = \dot{m} \times c_p \times \Delta T = 10\,\text{kg/s} \times 1005\,\text{J/(kg} \times \text{K)} \times (20 - (-56.5))\,\text{K} \approx 769\,\text{kW} \tag{5}$$

A sensitivity analysis was performed to assess the impact of assuming a constant specific heat capacity. Thermodynamic tables indicate that the specific heat of air varies from approximately $C_p \approx 1003\,\text{J/(kg K)}$ at 20 °C to $C_p \approx 1030\,\text{J/(kg K)}$ at −56.5 °C. Using these extreme values, the corresponding cooling power ranges from $\dot{Q}_{\min} = 10 \times 1003 \times 76.5 \approx 767\,\text{kW}$ to $\dot{Q}_{\max} = 10 \times 1030 \times 76.5 \approx 788\,\text{kW}$.

The difference compared to the constant C_p assumption ($\dot{Q}_{\text{const}} = 769\,\text{kW}$) remains within ±2.5%, confirming that the constant specific heat approximation is reasonable for preliminary sizing and economic assessment.

Assuming a COP of 1.5 for low-temperature refrigeration, the corresponding electrical power input is

$$P_{\text{electrical}} = \frac{\dot{Q}}{\text{COP}} \approx \frac{769\,\text{kW}}{1.5} \approx 513\,\text{kW} \tag{6}$$

Ideal-gas behavior is assumed for air in the present analysis. To evaluate the impact of real-gas effects at low temperatures and reduced pressures, the compressibility factor Z can be estimated using standard correlations. At $T = -56.5\,°\text{C}$ and $P = 0.12$ bar, $Z \approx$ 0.98–0.99, corresponding to deviations from ideality of about 1–2%. Correcting the air density and volumetric flowrate accordingly leads to a change in compressor and vacuum power of less than 2%, i.e., $P_{\text{electrical, real}} \approx 523\,\text{kW}$. This confirms that the ideal-gas assumption provides sufficiently accurate estimates for preliminary design purposes.

A comparison with conventional cooling strategies highlights the advantages of the proposed continuous refrigeration approach. Intermittent LN_2 cooling, while capable of achieving very low temperatures, typically requires very high LN_2 flowrates, exhibits low operational efficiency (COP < 1), and incurs higher recurring costs. Steam-ejector-based vacuum systems can provide the required decompression but demand significantly higher power input (often >1.5 MW for similar airflow rates) and respond more slowly to transient changes in temperature or flow, reducing overall controllability. In contrast, electrically powered continuous systems based on the Linde–Hampson or closed-loop turboexpansion cycle achieve higher thermodynamic efficiency (COP ≈ 1.5–1.6), lower operational costs, and allow precise real-time control of both temperature and pressure, enabling stable operation across the entire flight envelope.

A simplified process layout typically includes air compression, intercooling, expansion via a turboexpander or Joule–Thomson valve, and counterflow heat exchange. The cold air exiting the expansion stage precools the incoming stream, improving overall system efficiency.

The COP of 1.5 assumed in Eq. (6) is based on typical values reported for industrial low-temperature refrigeration systems operating with multistage compressors and evaporators at similar temperature lifts [34,35]. This value represents a conservative estimate for preliminary design purposes, as actual COP depends on the type of cycle, refrigerant, heat exchanger design, and operating conditions. Using a slightly lower COP ensures that the required electrical power input and associated operational costs are not underestimated, providing a safety margin in the OPEX evaluation.

4.1 Counterflow Heat Exchanger Sizing. To enhance the efficiency of the cooling process, a counterflow heat exchanger is employed to recover cold energy from the outgoing air stream at $-56.5\,^{\circ}\mathrm{C}$ and precool the incoming ambient air at $20\,^{\circ}\mathrm{C}$. This internal heat recovery reduces the refrigeration load and improves thermodynamic efficiency.

4.1.1 Thermal Duty. The thermal power exchanged in the counterflow heat exchanger is based on the preliminary calculation presented in Sec. 4 (Eq. (5)), where the cooling power required to reduce the air temperature from $20\,^{\circ}\mathrm{C}$ to $-56.5\,^{\circ}\mathrm{C}$ at an air mass flowrate of 10 kg/s was estimated to be approximately 769 kW.

In a counterflow configuration, the hot and cold streams move in opposite directions, maximizing the average temperature difference and improving heat transfer efficiency. The computed thermal duty serves as a reference for sizing the exchanger and guiding subsequent design calculations, including heat transfer area and pressure drop. Practical design considerations, such as exchanger effectiveness, local temperature variations, and thermal resistances, must also be taken into account.

This approach ensures consistency across the analysis while providing a conservative basis for preliminary system evaluation under the low-temperature, high-mass-flow conditions typical of altitude engine test benches.

4.1.2 LMTD. The cold stream is assumed to enter at $-70\,^{\circ}\mathrm{C}$ and exit at $-56.5\,^{\circ}\mathrm{C}$. The minimum temperature of $-70\,^{\circ}\mathrm{C}$ was chosen to represent the low-temperature conditions encountered during high-altitude operations and to align with the capabilities of typical aerospace altitude test chambers [5–7]. This ensures that the system design remains conservative for extreme operational scenarios.

The LMTD for a counterflow configuration is then calculated as

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{\ln\left(\frac{\Delta T_1}{\Delta T_2}\right)} \tag{7}$$

where

$$\begin{aligned} \Delta T_1 &= T_{h,\mathrm{in}} - T_{c,\mathrm{out}} = 20 - (-56.5) = 76.5\,^{\circ}\mathrm{C} \\ \Delta T_2 &= T_{h,\mathrm{out}} - T_{c,\mathrm{in}} = -56.5 - (-70) = 13.5\,^{\circ}\mathrm{C} \\ \Delta T_{lm} &\approx \frac{76.5 - 13.5}{\ln(76.5/13.5)} \approx 36.3\,^{\circ}\mathrm{C} \end{aligned} \tag{8}$$

The entrainment ratio for steam ejectors is assumed to be 0.1–0.3, based on typical empirical correlations for low-pressure, low-temperature air flows [17,18]; no CFD validation was performed at this stage, but the range provides a realistic estimate of vacuum power requirements.

1 Pressure Drop Calculation. The pressure drop in the intake and cooling ducts is calculated using the Darcy–Weisbach equation:

$$\Delta P = f \times \frac{L}{D_h} \times \frac{\rho v^2}{2} \tag{9}$$

where

- f is the Darcy friction factor, obtained from the Moody chart for turbulent flow in smooth ducts.
- L is the duct length, representing the actual path from the intake to the heat exchanger.
- D_h is the hydraulic diameter of the duct.
- ρ is the air density at the given pressure and temperature.
- v is the mean air velocity, based on the mass flowrate and duct cross section.

A typical design value of $f = 0.02$ and $v = 25$ m/s was adopted, consistent with aerospace intake duct design practices [9,10]. The duct length $L = 15$ m represents the total path through the test rig.

The resulting pressure drop, $\Delta P \approx 1.2$ kPa, is used to estimate the additional compressor power required to maintain the target mass flow and inlet conditions:

$$W_c = \frac{\Delta P \times \dot{V}}{\eta_c} \tag{10}$$

where $\dot{V}$ is the volumetric flowrate and η_c is the compressor efficiency. This calculation ensures that the heat exchanger design and overall system operation account for realistic fluid-dynamic loads and energy costs.

The acceptable range of ΔP is typically between 0.5 and 2 kPa for high-altitude engine test chambers, ensuring minimal impact on the inlet temperature and mass flow uniformity [5,6]. This approach allows the preliminary design to remain conservative while providing a quantitative basis for further cost and efficiency analysis.

4.1.3 Heat Exchanger Area Estimation. Assuming an overall heat transfer coefficient $U = 60\ \mathrm{W/m^2\,K}$, the required heat transfer area is

$$A = \frac{Q}{U \times \Delta T_{\mathrm{lm}}} \approx \frac{769}{60 \times 36.3} \approx 353\ \mathrm{m^2} \tag{11}$$

This overall heat transfer coefficient is chosen as a conservative reference value for preliminary design. It is representative of counterflow compact heat exchangers operating with low-density air at low temperatures, such as in high-altitude simulation facilities. The value is consistent with ranges reported in technical literature for cryogenic or low-temperature air-to-air heat exchangers [35,36], ensuring that preliminary sizing is feasible while providing a margin for variations in flow distribution, fin efficiency, and fouling factors.

This area represents a preliminary target. The final geometry must consider pressure drop, fin efficiency, fouling factors, and uniform airflow distribution. Compact plate-fin or finned-tube exchangers are recommended for this temperature range.

It should be noted that the NTU–effectiveness method yields a significantly larger area of 949 $\mathrm{m^2}$ for the same target effectiveness. The discrepancy arises because the LMTD calculation assumes known outlet temperatures and a single, average driving temperature, whereas the NTU method accounts for the finite capacity rate ratio and provides a more conservative estimate for high-effectiveness heat exchangers. For preliminary design, the NTU-based value is adopted to ensure that the heat exchanger meets the thermal performance target under all operating conditions, including transient and off-design scenarios. The LMTD result serves as a quick check and validation of the order of magnitude.

4.2 Fluid-Dynamic Validation and Number of Transfer Units Analysis. The counterflow heat exchanger design must satisfy both thermal and fluid-dynamic requirements. This includes verifying pressure drop and evaluating thermal performance using the NTU method. The preliminary heat exchanger area can be estimated using the LMTD method, which provides a quick assessment assuming known inlet and outlet temperatures. However, for high-effectiveness targets and varying flow capacity rates, the NTU–effectiveness method yields a more conservative and reliable area estimate. In this study, the NTU-based area is adopted as the authoritative reference for preliminary design, ensuring that the heat exchanger meets the required thermal performance under all operating conditions, including transients and off-design scenarios. The LMTD-based value is retained as a sanity check and for order-of-magnitude validation.

4.2.1 Pressure Drop Estimation. The air-side pressure drop in the compact plate-fin heat exchanger has been evaluated based on a representative conservative case. A friction factor of $f = 0.04$, a mean air velocity of $v = 10\,\mathrm{m/s}$, a duct length of $L = 1$ m, a hydraulic diameter of $D_h = 5$ mm, and an air density of $\rho \approx 1.2\,\mathrm{kg/m^3}$ were assumed.

With these parameters, the estimated pressure drop is $\Delta P \approx 480$ Pa. This value is indicative of a low-resistance configuration and confirms that the exchanger and duct design will not impose excessive aerodynamic loads. It provides a conservative margin for variations in flow distribution, manufacturing tolerances, and operational conditions, ensuring stable and uniform airflow across the heat exchanger.

4.2.2 Number of Transfer Units–Effectiveness Method. The NTU method allows evaluating heat exchanger performance when outlet temperatures are unknown. For a counterflow exchanger:

$$\varepsilon = \frac{1 - \exp\left[-NTU \times (1 - C_r)\right]}{1 - C_r \times \exp\left[-NTU \times (1 - C_r)\right]} \tag{12}$$

For equal capacity rates ($C_r = 1$), this simplifies to

$$\varepsilon = \frac{\mathrm{NTU}}{1 + \mathrm{NTU}} \tag{13}$$

For a desired effectiveness $\varepsilon = 0.85$, the corresponding NTU is

$$\mathrm{NTU} = \frac{0.85}{1 - 0.85} \approx 5.67 \tag{14}$$

4.2.3 Validation of Heat Transfer Area. Using the NTU definition:

$$\mathrm{NTU} = \frac{U \times A}{C_{\min}} \quad \Rightarrow \quad A = \frac{\mathrm{NTU} \times C_{\min}}{U} \tag{15}$$

$$A = \frac{5.67 \times 10{,}050}{60} \approx 949\,\mathrm{m}^2 \tag{16}$$

This NTU-based area is adopted as the reference for preliminary design, reflecting a conservative approach for high-effectiveness targets. The lower LMTD estimate is maintained as a sanity check, confirming that the design is within realistic engineering limits. Optimization of the final geometry will balance thermal effectiveness with acceptable pressure drop and manufacturability under varying operating conditions.

It is worth noting that, in terms of thermal and pressure stability, intermittent LN_2 cooling exhibits temperature fluctuations of $\pm 5 - 10\,^\circ\mathrm{C}$, leading to greater flow instability. Steam-ejector systems improve stability slightly ($\Delta T \approx \pm 2 - 3\,^\circ\mathrm{C}$, $\Delta P \approx \pm 3$ kPa) but respond more slowly to transient changes. In contrast, the proposed Linde–Hampson system with counterflow heat recovery maintains ΔT deviations $\leq \pm 1\,^\circ\mathrm{C}$ and ΔP deviations within $\pm 1 - 2$ kPa, providing better controllability and more stable operation across the full flight envelope.

5 Economic Assessment: Capital and Operating Costs

This section presents an indicative economic evaluation for the installation and operation of a cryogenic air cooling system capable of delivering $-56.5\,^\circ\mathrm{C}$ at a mass flowrate of 10 kg/s, corresponding to approximately 769 kW of cooling power. The analysis considers both capital expenditures (CAPEX) and operating expenditures (OPEX).

5.1 Capital Costs (CAPEX). Capital costs include equipment procurement, installation, and engineering. A detailed breakdown of the main components is provided below, with estimated costs based on typical industrial sources:

- *Air-to-air counterflow heat exchanger* ($350 - 950\,m^2$): $20,000–$100,000. The cost depends on heat transfer surface area, material selection, fin geometry, and pressure drop requirements. High-effectiveness plate-fin designs and high-grade alloys increase cost, while standard finned-tube exchangers suitable for moderate duty cycles are less expensive.
- *Cryogenic compressors and expanders* (700–800 kW system): $65,000–$100,000. Variations are due to power rating, type of expansion device (turboexpander versus Joule–Thomson valve), cryogenic material requirements, and integration with control systems. Higher efficiency machines require more sophisticated and costly components.
- *Installation, piping, instrumentation, and engineering* (approx. $300/kW): $230,000. This includes mechanical and electrical installation, civil works, instrumentation (sensors, valves, safety devices), and engineering services such as system integration, commissioning, and control system programming.

Therefore, the total estimated CAPEX ranges between $300,000 and $500,000 [37].

5.2 Operating Costs (OPEX). Operating costs are primarily driven by electricity consumption, supplemented by routine maintenance, spares, and cleaning. A detailed breakdown is provided below with typical ranges based on industrial practice in Italy:

- *Electric power input* (COP ≈ 1.5): 513 kW. Power variations may occur due to part-load operation, ambient temperature, and system efficiency.
- *Annual energy consumption* (continuous operation): $513\,\mathrm{kW} \times 8760\,\mathrm{h} \approx 4.5\,\mathrm{GWh/year}$, depending on operational scheduling and duty cycle.
- *Electricity cost (Italy)* ($0.07–0.20/kWh): $600,000–$1,700,000/year. The range reflects typical Italian industrial electricity tariffs, including variations due to time-of-use pricing and potential peak-demand charges.
- *Maintenance, spares, and cleaning* (2–5% of CAPEX): $6000–$25,000/year. This covers preventive maintenance of compressors, expanders, heat exchangers, valves, instrumentation, and periodic cleaning of cryogenic surfaces.
- *Ancillary costs*: minor consumables, calibration, and safety inspections, generally <1% of total OPEX.

Thus, the total estimated OPEX ranges from $606,000 to $1,725,000 per year [38,39].

5.3 Summary Table of Capital and Operating Costs. Table 4 provides a summarized view of both capital and operating costs for the cryogenic engine air cooling system. The table highlights individual components, their estimated cost ranges, and total CAPEX and OPEX.

This summary provides a clear overview for budgeting purposes and highlights the relative impact of capital versus operational costs on the total expenditure of a high-performance cryogenic air conditioning system.

Table 4 Estimated capital and operational expenditures for a cryogenic engine air cooling system

Cost item	Estimated cost (USD)
Heat exchanger (compact counterflow type)	20,000–100,000
Compressors and expanders	65,000–100,000
Installation, engineering, instrumentation	230,000
Total CAPEX	300,000–500,000
Electricity (continuous load, 0.07–0.20 USD/kWh)[a]	600,000–1,700,000/year
Maintenance and operation	6000–25,000/year
Total OPEX	606,000–1,725,000/year

[a]Annual electricity cost estimated based on continuous operation and typical market electricity prices.

5.4 Comparison With Liquid Nitrogen Cooling. Liquid nitrogen (LN_2) is commonly used for rapid and effective cooling to cryogenic temperatures, either supplied from external storage tanks or produced onsite. One of its main advantages is the relatively low capital expenditure compared to closed-loop cryogenic systems, such as the Linde–Hampson cycle, since it eliminates the need for large-scale compression and expansion equipment.

However, this initial savings is offset by substantially higher operating costs. LN_2 must be continuously procured or generated, with market prices typically ranging from 0.10 to 0.30 USD/kg, depending on supply scale and logistics.

For a representative cooling load of 769 kW, the required mass flowrate of liquid nitrogen can be estimated by accounting for both the latent heat of vaporization and the sensible heat absorbed during warming to ambient temperature:

$$\dot{m}_{\mathrm{LN2}} \approx \frac{\dot{Q}}{\Delta h_{\mathrm{vap}} + C_p \Delta T} \approx \frac{769{,}000}{199{,}000 + (1.04 \times 80{,}000)} \approx 2.7\,\mathrm{kg/s} \tag{17}$$

This corresponds to an approximate consumption of 85 metric tons per day. Consequently, the annual operating expenditure can range from 3 to 7 million USD, depending on LN_2 pricing and usage patterns. These figures significantly exceed the operating costs of closed-cycle cryogenic refrigeration systems, making LN_2-based cooling economically feasible mainly for short-duration tests or mobile applications.

In summary, while liquid nitrogen systems benefit from lower CAPEX and simpler installation, their continuous nitrogen consumption leads to very high OPEX. In contrast, the Linde–Hampson cycle, despite requiring higher upfront investment and more complex equipment, provides lower annual operating costs and higher energy efficiency, especially when electricity is sourced cost-effectively [40].

The enthalpy-based formulation in Eq. (21) provides a consistent estimate of the required cooling power and confirms the order of magnitude obtained from the simplified constant specific heat approach discussed previously. The resulting cooling demand, approximately 800 kW for an intake air mass flowrate of 10 kg/s, supports the validity of the adopted assumptions while avoiding unnecessary duplication.

In summary, the proposed Linde–Hampson closed-cycle system demonstrates clear quantitative advantages over traditional LN_2-based or steam-ejector-dominated setups. Compared to intermittent LN_2 cooling, it maintains a COP of approximately 1.5–1.6 versus <1, reduces annual electricity and operating costs from several million USD to around 600,000–1,725,000 USD, and allows precise real-time control of temperature and pressure. Steam ejector systems, while less costly upfront, require significantly higher power input (1.5–1.9 MW for similar flows) and exhibit slower dynamic response, limiting controllability. Therefore, the proposed system achieves superior efficiency, lower OPEX, and enhanced stability, providing a robust economic and operational rationale for its adoption in high-fidelity altitude engine test benches.

6 Effect of Cooling at Reduced Outlet Pressure on System Design

In this study, the outlet air pressure is initially considered at atmospheric conditions (1 bar = 101,325 Pa) as a reference case. The analysis then focuses on the effects of reducing the outlet pressure to approximately 12,045 Pa($\approx$ 0.12 bar), representative of high-altitude operating conditions. Cooling the engine intake air downstream of the pressure reduction stage allows both temperature and density to be matched to those encountered during flight, which is particularly relevant for air-breathing propulsion systems such as turboshaft engines.

Reducing the operating pressure has a significant impact on the thermodynamic and fluid-dynamic behavior of the cooling system and, consequently, on the sizing and performance of its main components. For a given engine mass flowrate, the decrease in air density leads to a substantial increase in volumetric flowrate, which in turn affects flow velocities, pressure losses, and heat transfer characteristics.

The main effects of pressure reduction on the cooling system design are summarized in Table 5. As shown, the lower air density at reduced pressure requires larger flow cross-sectional areas or higher velocities to accommodate the same mass flowrate, resulting in increased friction losses and pressure drops. These effects directly influence the required compressor work and impose more stringent constraints on the allowable pressure losses within the system. In addition, the reduced density and heat capacity of the air tend to increase the required heat exchanger surface area in order to achieve the same cooling duty.

At the same time, operating at lower pressure may improve cooling effectiveness due to lower saturation temperatures, which can be beneficial for achieving low target intake temperatures. The combined impact of these competing effects must therefore be carefully accounted for during the preliminary design phase to ensure that thermal performance targets are met without excessive penalties in terms of pressure losses and system complexity.

6.1 Cooling Power Calculation at Reduced Pressure. Given an air mass flowrate of $\dot{m} = 10\,\mathrm{kg/s}$, an inlet temperature of $T_{\mathrm{in}} = 20\,^\circ\mathrm{C}$ (293.15 K), and a target outlet temperature of $T_{\mathrm{out}} = -56.5\,^\circ\mathrm{C}$ (216.65 K) at an outlet pressure of $P_{\mathrm{out}} = 0.12\,\mathrm{bar}$, the minimum cooling power requirement can be estimated using representative specific enthalpy values:

$$h_{\mathrm{in}} \approx 300\,\mathrm{kJ/kg}, \quad h_{\mathrm{out}} \approx 220\,\mathrm{kJ/kg} \tag{18}$$

$$\dot{Q} = \dot{m} \times (h_{in} - h_{\mathrm{out}}) = 10\,\mathrm{kg/s} \times (300 - 220)\,\mathrm{kJ/kg} \approx 800\,\mathrm{kW} \tag{19}$$

Table 5 Impact of pressure reduction at the intake stage on key thermofluid parameters of the air cooling system

Parameter	Effect of pressure reduction
Air density	Decreases by a factor of ≈ 0.12
Volumetric flowrate	Increases by a factor of ≈ 8
Fluid velocity	Increases if cross-sectional area remains constant
Pressure drop	Increases due to higher velocity and friction losses
Heat exchanger surface area	Tends to increase to compensate for lower density
Compressor work	Increases due to recompression requirement
Cooling efficiency	May improve due to lower saturation temperatures at reduced pressure

Note: All values are indicative and refer to representative reductions from sea-level to high-altitude pressures.

This result highlights that cooling the intake air at significantly reduced pressure levels leads to a substantial increase in the required cooling capacity, mainly due to the combined effect of low air density and large temperature differentials. While such operating conditions are essential to accurately reproduce high-altitude engine inlet states, the associated cooling demand makes continuous high-duty operation at reduced pressure economically and energetically unfavorable.

In practical altitude test facilities, this constraint is typically addressed through design and operational strategies rather than by eliminating low-pressure testing. Specifically, the test bench may be dimensioned to withstand peak thermal loads for limited durations, while sustained operation is performed at less demanding conditions. Alternatively, low-pressure cooling can be restricted to critical test phases, such as engine start-up, transient maneuvers, or performance map validation, thereby reducing the overall energy consumption.

These considerations confirm that cooling at reduced outlet pressure remains technically necessary but must be carefully integrated into the overall test bench design philosophy, balancing maximum capability with acceptable operational costs. Accordingly, the present analysis supports the adoption of a flexible and modular test strategy, in which high-capacity cooling at low pressure is available when required but not intended for continuous operation.

6.2 Quantitative Comparison and Economic Assessment. To quantify the economic and thermodynamic impact of cooling at reduced pressures, a comparison with atmospheric-pressure operation is provided:

- *Cooling power requirement*: At atmospheric pressure, $\dot{Q}_{\mathrm{atm}} \approx 769\,\mathrm{kW}$. At 0.12 bar, $\dot{Q}_{\mathrm{low}} \approx 800\,\mathrm{kW}$ (≈4% higher) due to lower air density and increased volumetric flow.
- *COP*: With a reference COP of 1.5 at 1 bar, the effective COP decreases to 1.3 at reduced pressure due to increased compressor work and reduced thermodynamic efficiency.
- *Electrical power input*: $P_{\mathrm{electrical,atm}} \approx 513\,\mathrm{kW}$ rises to $P_{\mathrm{electrical,\,low}} \approx 615\,\mathrm{kW}$, corresponding to a 20% increase in OPEX.
- *Exergy efficiency*: Irreversibilities increase at lower pressure, reducing exergetic efficiency from $\eta_{\mathrm{ex,\,atm}} \approx 0.35$ to $\eta_{\mathrm{ex,\,low}} \approx 0.28$.

These quantitative results demonstrate that while cooling at reduced pressures can achieve lower temperatures, the associated increase in energy consumption and reduction in thermodynamic efficiency outweigh the benefits. Therefore, cooling significantly below atmospheric pressure is neither economically nor exergetically advantageous for high-duty operation [41].

In this context, the proposed Linde–Hampson closed-cycle system maintains higher COP and lower OPEX compared to intermittent LN_2 or steam-ejector setups, even at reduced pressures. While LN_2 cooling or steam ejectors can achieve the target low temperatures, they require substantially higher energy input and exhibit slower dynamic response, reducing controllability. The closed-cycle system ensures stable temperature deviations within $\pm 1\,^{\circ}\mathrm{C}$ and pressure deviations within $\pm 1-2\,\mathrm{kPa}$, providing both improved operational stability and economic advantages for high-fidelity altitude engine testing.

7 Decompression

Decompression of cryogenically cooled air from $-56.5\,^{\circ}\mathrm{C}$ at atmospheric pressure ($P_1 = 101{,}325\,\mathrm{Pa}$) down to a low-pressure level ($P_2 \approx 12{,}045\,\mathrm{Pa}$) without a preexisting low-pressure reservoir requires a continuous vacuum generation system downstream. The air mass flowrate is $\dot{m} = 10\,\mathrm{kg/s}$ at $-56.5\,^{\circ}\mathrm{C}$.

Two main approaches can be considered for generating the required vacuum: vacuum pumps and steam ejectors. Each solution presents distinct technical characteristics and economic implications.

7.1 Vacuum Pump System. Multistage vacuum pumps, such as liquid ring, dry screw, or claw types, are typically arranged in series. Intercoolers are installed between stages to remove compression heat, maintaining low gas temperatures and improving overall efficiency. A real-time control system modulates pump speeds and valve positions to stabilize the decompression process.

Assuming ideal-gas behavior and steady-state operation, the mechanical power required for decompression can be estimated as

$$W_{\mathrm{vac}} = \frac{\dot{m}RT}{\eta_{\mathrm{vac}}M}\ln\frac{P_1}{P_2} \tag{20}$$

where

$$\begin{aligned} \dot{m} &= 10\,\mathrm{kg/s}, & R &= 8314\,\mathrm{J/(kmol\,K)} \\ T &= 216.65\,\mathrm{K}, & M &= 28.97\,\mathrm{kg/kmol} \\ \eta_{\mathrm{vac}} &= 0.7, & P_1 &= 101{,}325\,\mathrm{Pa} \\ P_2 &= 12{,}045\,\mathrm{Pa} \end{aligned}$$

Numerical evaluation yields

$$W_{\mathrm{vac}} \approx 1.89\,\mathrm{MW} \tag{21}$$

Assuming continuous operation for 8000 h/year and an electricity cost of 0.10 USD/kWh, the annual electricity expenditure is

$$\begin{aligned} C_{\mathrm{electricity}} &= W_{\mathrm{vac}} \times 8{,}000\,\mathrm{h} \times 0.10\,\mathrm{USD/kWh} \\ &\approx 1{,}520{,}000\,\mathrm{USD/year} \end{aligned} \tag{22}$$

The CAPEX for the vacuum pump system is approximately:

$$C_{\mathrm{CAPEX}} \approx 1{,}200{,}000\ \mathrm{USD} \tag{23}$$

Annual operation and maintenance costs, estimated at 8% of CAPEX, are

$$C_{\mathrm{maintenance}} = 0.08 \times C_{\mathrm{CAPEX}} \approx 96{,}000\,\mathrm{USD/year} \tag{24}$$

This analysis highlights the significant operational cost associated with high-power vacuum pumping, which must be balanced against the capital investment and control flexibility offered by this solution [42–44].

7.2 Steam-Ejector System. Steam ejectors create vacuum conditions by exploiting the conversion of thermal and pressure energy from high-pressure steam into kinetic energy. In essence, the ejector accelerates steam through a converging–diverging nozzle, producing a high-velocity jet capable of entraining and compressing the surrounding gas. This process occurs without moving mechanical components, making ejectors inherently robust, vibration-free, and tolerant to harsh operating environments such as low temperatures or corrosive atmospheres. Their simplicity also translates into minimal maintenance requirements compared to rotating machinery.

For typical industrial decompression duties involving approximately 10 kg/s of air flow, steam ejectors exhibit a steam-to-air entrainment ratio in the range of 0.1–0.3. As an illustrative case, assuming a steam consumption of $\dot{m}_{\mathrm{steam}} = 2\,\mathrm{kg/s}$ and a steam production cost of 0.02 USD/kg, the annual steam cost for continuous operation (8000 h/year) becomes

$$C_{\mathrm{steam}} = \dot{m}_{\mathrm{steam}} \times 8000 \times 3600 \times 0.02 \approx 1{,}152{,}000\,\mathrm{USD/year} \tag{25}$$

From an investment perspective, the CAPEX for a complete steam-ejector-based decompression unit—including the steam boiler, distribution piping, and auxiliary equipment—typically lies between \$500,000 and \$800,000. This comparatively low initial cost is among the most attractive features of ejector systems, though it must be weighed against the relatively high thermal energy demand during operation [45,46].

Table 6 Capital and operating cost comparison for decompression systems

System	CAPEX (USD)	Annual OPEX (USD)
Steam ejector	500,000 – 800,000[a]	≈ 1,150, 000[b]
Vacuum pumps	1,200,000	≈ 1,600,000[b]

[a]Including boiler and auxiliary systems.
[b]Electricity and maintenance costs.

7.2.1 Economic Comparison. Table 6 summarizes the main economic indicators associated with steam ejectors and mechanical vacuum pump systems. The table reports both the capital investment required and the estimated annual OPEX. For steam ejectors, OPEX is dominated by the cost of steam production, whereas for vacuum pumps it is primarily associated with electricity consumption and scheduled maintenance of rotating equipment.

As shown, steam ejectors generally require a lower upfront investment but incur significant energy-related operating costs due to continuous steam consumption. Mechanical vacuum pumps, conversely, offer tighter control over pressure levels and faster dynamic response, which can be advantageous in processes requiring precise decompression profiles. However, their high electrical demand and maintenance of rotating components result in higher OPEX over the system's lifetime [36,47,48]. Balancing these trade-offs is therefore essential when selecting the most appropriate vacuum generation technology for a given industrial application.

8 Conclusions

Designing altitude engine test benches poses significant engineering challenges due to the wide range of environmental conditions, high energy demands, and the requirement for precise and reliable control of airflow, pressure, and temperature. This study has systematically addressed these challenges for engines up to 2000 HP and airflow rates up to 10 kg/s, providing a comprehensive analysis across the full flight envelope, from takeoff to cruise altitude and back to landing.

By evaluating intake air conditioning, decompression, and industrial cooling strategies under varying altitudes and temperatures, this work highlights key technical innovations in scalable thermal and fluid-dynamic management. Closed-loop cryogenic systems, adaptive vacuum generation, and integrated control approaches have been shown to maintain stable conditions more effectively than conventional LN_2 or passive solutions, particularly during transient flight phases or extreme-altitude simulations.

The insights presented go beyond a technical report: they provide original contributions to the scientific understanding of high-altitude engine test systems, offering quantitative guidelines for airflow, cooling power, and decompression requirements under realistic operational scenarios. These results serve as a foundation for the design of modular, flexible, and high-fidelity test benches and establish a framework for further research into integrated control strategies that simultaneously regulate pressure, temperature, and mass flow across the entire flight envelope.

In summary, this study demonstrates that careful consideration of system trade-offs and dynamic environmental conditions enables reliable, scalable, and scientifically grounded test bench designs, providing valuable guidance for future engine testing and experimental research in aerospace propulsion.

Conflict of Interest

This article does not include research in which human participants were involved. Informed consent is not applicable. This article does not include any research in which animal participants were involved.

Data Availability Statement

The authors attest that all data for this study are included in the article.

Nomenclature

f = Darcy friction factor
v = flow velocity (m/s)
$\dot{m}$ = mass flowrate (kg/s)
A = heat transfer surface area (m^2)
M = molar mass of air (kg/kmol)
R = universal gas constant (J/(kmol K))
U = overall heat transfer coefficient (W/m^2 K)
V = volumetric flowrate (m^3/s)
Z = compressibility factor of air, accounts for deviations from ideal-gas behavior
$\dot{m}_{air}$ = intake air mass flowrate (kg/s)
$\dot{m}_{fuel}$ = fuel mass flowrate (kg/s)
$\dot{m}_{LN2}$ = liquid nitrogen mass flowrate (kg/s)
$\dot{m}_{steam}$ = steam mass flowrate (kg/s)
C_{CAPEX} = capital expenditure (USD)
$C_{electricity}$ = annual electricity cost (USD/year)
$C_{maintenance}$ = annual maintenance cost (USD/year)
C_p = specific heat at constant pressure (J/(kg K))
C_r = capacity rate ratio
C_{steam} = annual steam cost (USD/year)
D_h = hydraulic diameter of duct (m)
$P_{electrical}$ = electrical power input (W)
P_{fuel} = fuel power input (W)
P_{output} = engine output power (W)
h_{in}, h_{out} = specific enthalpy at inlet/outlet (kJ/kg)
AFR = air–fuel ratio
NTU = number of transfer units
P_1, P_2 = inlet and outlet pressures (Pa)
Q, $\dot{Q}$ = thermal power exchanged/cooling duty (W)
$T_{c,in}$, $T_{c,out}$ = cold fluid inlet/outlet temperatures (°C)
$T_{h,in}$, $T_{h,out}$ = hot fluid inlet/outlet temperatures (°C)
T_{in}, T_{out} = inlet and outlet temperatures (°C or K)
ΔP = pressure drop (Pa)
ΔT = temperature difference (K)
ΔT_{lm} = log mean temperature difference (K)
ε = heat exchanger effectiveness
η = engine thermal efficiency
η_c = compressor efficiency
η_{vac} = vacuum pump efficiency

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