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CFD-Based Analysis and Optimization of Depressurization Systems for Engine Test Benches of Small Propeller Aircraft

Luca Piancastelli*

Department of Industrial Engineering (DIN), Università di Bologna Alma Mater Studiorum, 40137 Bologna, Italy

* Correspondence: Luca Piancastelli (luca.piancastelli@unibo.it)

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Abstract: This study investigates the design and performance of altitude test benches for piston engines with power outputs up to 200 kW. The primary objective is to generate controlled depressions within an enclosed engine bay to reproduce atmospheric conditions corresponding to altitudes ranging from sea level to 14,000 m. Three configurations are examined: an ejector–diffuser system derived from National Advisory Committee for Aeronautics (NACA) principles, a Venturi device powered by an auxiliary diesel engine (Cursor 13), and a centrifugal turbocharger (Holset HY55V) mechanically coupled to the same auxiliary engine. Computational Fluid Dynamics (CFD) simulations are performed to evaluate the pressure and velocity distributions within the test chamber and its associated flow components. The ejector-diffuser arrangement achieves a moderate pressure reduction but exhibits flow separation in the diffuser at large expansion angles, limiting its efficiency. The Venturi system achieves a greater vacuum level, reducing the chamber pressure to approximately 76 kPa, equivalent to an altitude of around 2,500 m. The turbocharger-based configuration demonstrates the highest performance, achieving a chamber pressure of approximately 15 kPa—equivalent to an altitude of 14,000 m—through appropriate adjustment of compressor rotational speed and intake valve opening. This configuration also ensures a faster transient response and enhanced stability of airflow and pressure distribution. The results highlight the importance of proper integration between auxiliary propulsion systems, component sizing, and boundary condition definition to achieve accurate altitude simulation. The proposed approach demonstrates that combining a variable-speed compressor with active flow control enables flexible reproduction of both steady-state and transient operating conditions. The findings provide practical guidelines for developing cost-effective, reliable, and versatile altitude test benches suitable for experimental evaluation and calibration of high-power piston engines under simulated high-altitude environments.

Keywords: Altitude test bench; Engine testing; Turboshaft engines; Piston engines; Cryogenic cooling; Vacuum systems; Pressure control; Aerospace propulsion

1 Introduction

This paper examines the design and performance assessment of altitude engine test benches, which represent highly integrated systems characterized by elevated installation costs, stringent operational constraints, and demanding reliability requirements. Small propeller aircraft propulsion is predominantly based on piston engines, gasoline or diesel, with nominal power ratings up to approximately 200 kW [1]. Compared with other propulsion architectures, piston engines exhibit reduced dependency on air cooling, which simplifies their operation under simulated altitude conditions [1]. Reproducing altitude environments requires imposing large variations in static pressure and temperature [2]. Representative conditions range from 35°C and 100,000 Pa at sea level to approximately -56.5°C and 14,101.8 Pa at 14,000 m [2]. Most operational envelopes extend up to 11,000 m, where the ambient static pressure is 22,632 Pa [2]. The generation of such conditions entails substantial energy input, directly linked to the required mass flow rate and cooling capacity [3]. Piston engines in the 200 kW class typically operate with air flow rates well below 2 kg/s. This study reviews existing altitude-test-bench configurations [4] and develops a set of design concepts for facilities intended for engines up to 200 kW, with specific emphasis on the architecture of the air-supply and conditioning system. Energy management constitutes the dominant sizing parameter for altitude benches. The engine must be supplied with air at the prescribed reference temperature and pressure, while the surrounding engine bay must be maintained at controlled conditions, which may differ from the intake state [4]. The engine rejects both mechanical and thermal energy to the confined environment, thereby imposing significant cooling requirements. Exhaust-gas treatment must be decoupled from intake management, since the exhaust flow typically carries high thermal energy content and must be evacuated without perturbing the controlled-pressure environment [5]. Maintaining the prescribed thermodynamic state within the engine bay is further complicated by the combined effects of conductive, convective, and radiative heat transfer processes. Minimizing the volume of the pressurized enclosure reduces the mass of air that must be conditioned, improving overall system efficiency [3]. These enclosures are sealed, thermally isolated volumes maintained at controlled pressure and temperature [4]. Air conditioning is typically achieved through controlled expansion and cooling of ambient air, combined with precise pressure and temperature regulation. In advanced applications, this process can be supplemented or replaced by cryogenic cooling methods, such as the use of liquid nitrogen or the direct injection of cryogenic air, to attain substantially lower temperatures and higher thermodynamic efficiency [2]. Consequently, facility geometry, ducting layout, and the topology of the air-handling system must be designed to ensure spatial uniformity and transient responsiveness of the target thermodynamic conditions [3]. Transient operation constitutes a critical performance metric. During climbing, the engine undergoes rapid variations in ambient state variables [6]. These transients must be faithfully reproduced within the bench to reveal potential malfunctions, control-law deficiencies, or subsystem instabilities. Therefore, the dynamic response of the air-supply, depressurization, and thermal-management subsystems becomes a central design requirement. Given the high degree of functional coupling among components, system-level reliability is essential. The malfunction of a single subsystem can disrupt or entirely halt facility operation. Reliability requirements become even more severe during endurance campaigns, where engines must operate for extended durations without interruption.

2 General Characteristics

Engine test benches are typically composed of at least one confined environment that simulates the engine bay, an exhaust gas evacuation system, and an air supply system. The simulated engine bay must accommodate temperature variations ranging from 120°C to -56.5°C and pressures from 100,000 Pa to 14,101.8 Pa. The exhaust system requires pressure control across the same range. The air-supply system must support temperature conditions between 35°C and -56.5°C, and air flow rates ranging from 0.1 kg/s (for small piston engines) up to approximately 2 kg/s for piston engines in the 200 kW class. The test bench should be capable of simulating the entire flight envelope, including takeoff, climb, cruise, and descent. In theory, for air-cooled engines, it is possible to generate the required vacuum conditions by sealing the pressure vessel and allowing the engine to draw in air and expel it through its exhaust. However, this approach presents several practical limitations. In many test facilities, exhaust gases must be discharged into the same enclosed volume, preventing any significant pressure drop from being maintained. Furthermore, relying on the engine alone to regulate chamber pressure results in poor control accuracy, particularly under transient or part-load conditions. For piston engines, the time required to reach the desired vacuum level through self-depressurization is typically prohibitive. The system is also highly sensitive to minor leaks or micro-infiltrations from the ambient atmosphere, which further degrades pressure stability. Finally, this configuration makes it impossible to perform engine restart tests at the target simulated altitude.

2.1 Stoichiometric and Thermodynamic Air Consumption for Diesel Engines

The intake air mass flow rate m_{air} for a 200 HP diesel engine operating at 38 percent thermal efficiency η and an air-fuel ratio AFR = 17 is evaluated as follows. Assuming diesel LHV = 42.5 × 10^6 J/kg:

$$m_{fuel} = P_{fuel}/LHV = 0.00923 \text{ kg/s} \tag{1}$$

$$m_{air} = AFR \times m_{fuel} = 17 \times 0.00923 \approx 0.157 \text{ kg/s} \tag{2}$$

2.2 Stoichiometric and Thermodynamic Air Consumption for Gasoline Engines

The intake air mass flow rate m_{air} for a 200 HP gasoline engine operating at 20 percent thermal efficiency η and an air-fuel ratio AFR = 14.7 is evaluated as follows. Assuming diesel LHV = 44 × 106 J/kg:

$$m_{fuel} = P_{fuel}/LHV = 0.01695 \text{ kg/s} \tag{3}$$

$$m_{air} = AFR \times m_{fuel} = 14.7 \times 0.01695 \approx 0.249 \text{ kg/s} \tag{4}$$

Typical variability for 37–200 kW gasoline engines corresponds to an air mass flow rate $m_{air} \in$ [0.062, 0.334] kg/s. Air mass flow rates for piston engines in the 37–200 kW range are summarized in Table 1. The table provides both the computed values for a 200 kW engine and the estimated lower bounds for a 37 kW engine, along with typical variability ranges based on AFR and thermal efficiency. The quantity of combustion air supplied is remarkably limited. The primary strategy for controlling construction and operational expenditures is reducing the cold, low-pressure air mass flow necessary to support the functioning of the high-altitude test bench.

Table 1. Air mass flow rates for 37–200 kW piston engines (50–268.2 HP)

Engine Type	m_{fuel} (kg/s)	m_{air} (kg/s)	**Typical Range** m_{air} (kg/s)
Diesel engine	0.0124	0.211	$0.039 - 0.211$
Gasoline engine	0.0227	0.334	$0.062 - 0.334$

3 Cost-Effective Test Bench for Small Piston Engines

A cost-effective test bench for simulating altitude conditions in small propulsion engines consists of a compact, sealed chamber that houses the engine, while the dynamometer, cooling system, and auxiliary equipment are positioned externally. The chamber replicates the engine compartment, and the engine intake receives either a mixture of gaseous and liquid air or precooled gaseous air (e.g., via liquid-nitrogen cooling). The engine exhaust is routed to an ejector system that converts the thermal energy of the exhaust gases into a pressure drop within the chamber. During normal aircraft operation, engines experience a pressure cycle that begins at standard atmospheric conditions, decreases to a minimum operating pressure corresponding to the maximum flight altitude, and subsequently returns to the initial pressure during descent and landing [7–11]. To reproduce this cycle on the ground, the proposed test bench employs a system comprising three primary components. First, a sealed enclosure creates a working environment at sub-atmospheric pressure. Second, an ejector driven by the engine exhaust generates the required suction. Third, a throttling valve on the intake side regulates both chamber pressure and the mass flow of incoming air. The temperature of the supplied air simultaneously controls the thermal environment of the engine compartment. Within the ejector, the high-velocity, high-temperature exhaust gas entrains ambient air from the chamber as it passes through the mixing section. At the entrance of the divergent section, the combined mass flow exceeds that of the exhaust stream alone. As the mixture decelerates within the diffuser, a low-pressure region is formed, inducing additional flow from the intake side and thereby reducing the pressure inside the working volume. Partial closure of the throttling valve provides fine control of the incoming air, enabling precise regulation of chamber pressure. Using engine performance data and appropriate geometric scaling, computational fluid dynamics (CFD) simulations can be employed to quantify the achievable pressure reduction. By comparing the simulated pressure levels with those corresponding to specific flight altitudes, the effectiveness of the system in reproducing realistic operational conditions can be assessed.

3.1 Example Engine: Single-Rotor Wankel

The example adopted in this work is that of an aircraft engine representative of typical values, and not a specific production engine. It should be emphasized that test benches are never fully accurate: Numerous examples exist of engines performing flawlessly during bench testing yet failing miserably in service. A classic example is a Formula One engine that operated perfectly both on the test bench and during track testing but failed consistently during Grand Prix races. This engine experienced numerous catastrophic failures, including piston ruptures and the subsequent fragmentation of internal components, which occupied the manufacturer's engineering offices for years. At the end of the season, it was discovered that during drift conditions, the lubricating oil film tended to break down, uncovering the cam lobes; the resulting lack of lubrication caused abnormal wear and, eventually, valve-to-piston contact. Because this phenomenon depended entirely on driving style, race conditions, and other contingencies, the exact failure point was unpredictable. Another well-known example is that of the Daimler-Benz DB 605A engine, whose liquid coolant heat exchanger was notoriously undersized. This deficiency was identified only during the first operations and was corrected in 1944, although the engine had already entered service in early 1943. All of this shows that it is pointless to assume that the test bench provides definitive answers; it is merely a reference. In our case, the test bench must necessarily be economical. In state-of-the-art test benches, the engine-bay environment, the intake air, and the exhaust backpressure must each be conditioned independently to ensure accurate control of operating parameters. In the simplified and cost-effective configuration considered here, we approximate the internal engine bay conditions as equivalent to the external ambient conditions. The effect of exhaust backpressure resulting from the absence of a dedicated exhaust-pressure simulation will be regarded as negligible. Thus, a single intake-air temperature and pressure will be imposed, corresponding to the simulated altitude and temperature. In our setup, all removable auxiliary components are taken off the engine: heat exchangers, electric pumps, and generally all devices not installed directly on the engine are relocated outside. A pressure enclosure is then built around the engine, with a clearance of approximately 50 mm between the engine and the enclosure walls. The enclosure is roughly rectangular, openable, and equipped with armored inspection windows, cameras, and internal microphones. It includes openings for coolant lines and sealed pass-throughs for electrical connections. A primary orifice allows the crankshaft to protrude and connect to the dynamometer, which is located outside the pressure vessel. An exhaust outlet is also provided; depending on the configuration, a nozzle, a Venturi tube, or an air extraction system driven by an auxiliary motor may be installed. The engine will be supplied through a calibrated, throttled tube and a carburetor. The carburetor, equipped with a mixing system, will blend ambient intake air with low-temperature air provided by a cryogenic system. These systems are like hospital oxygen systems, but in our case, they supply air rather than oxygen. The carburetor is feedback-controlled using thermal probes installed inside the pressure enclosure, which determine the required quantity of cold air needed to achieve the operator-set temperature. The orifice will be opened or closed to increase (when opened) or decrease (when closed) the pressure or depression inside the pressure enclosure. The test bench is thus highly simplified and designed for minimal cost. Exhaust backpressure is not regulated, as the entire system prioritizes simplicity and economy. In the following sections, we will examine several configurations and show that the only truly effective solution is the one based on a centrifugal vacuum pump for exhaust extraction. To illustrate the proposed test bench configuration, a single-rotor Wankel engine has been selected as a representative example (see Figure 1). This engine features a single-rotor architecture with a displacement of 360 cc, is liquid-cooled, and is equipped with electronic fuel injection. It delivers a maximum power output of 60 HP at 7500 rpm, with an exhaust mass flow rate of 60 g/s. The exhaust gases reach a maximum velocity of 500 m/s at operating temperature, and their peak temperature can reach 1000°C. To ensure adequate performance at high altitudes, the engine is also equipped with a turbocharging system, which compensates for the reduced air density encountered during flight. This engine serves as a representative case for evaluating the performance and feasibility of the proposed altitude test bench.

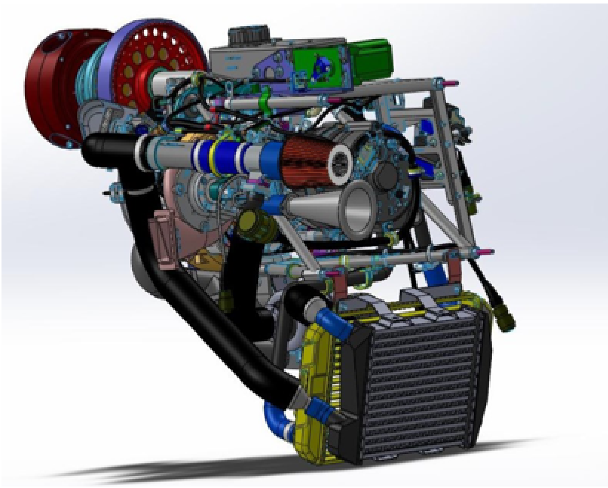

Figure 1. Schematic of the single-rotor Wankel engine used as an example. The figure illustrates the rotor, intake, exhaust, and turbocharging components

4 Ejector Exhaust System Design

The ejector exhaust system employed on the exhaust side of the test chamber is a well-established concept in aeronautics, historically used on aircraft and helicopters to increase the flow of fresh air over radiators, thereby enhancing propulsion system cooling. In the present application, the ejector is not coupled with a radiator; however, the underlying physical principle remains the same: the high-velocity, high-temperature exhaust gases are used to induce a substantial air flow that, when expanded through a divergent duct, generates a low-pressure region, drawing air from the intake zone. The design methodology follows the experimental investigation presented in the NACA-TR-818 [12], which studied a rectangular-section ejector applied to engine cooling systems. The ejector leverages the exhaust jet to pump cooling air from the rear of the engine to the atmosphere. This process reduces the static pressure downstream of the engine, thereby increasing the pressure differential available for airflow. The mechanism relies on momentum transfer between the high-speed exhaust jet and the low-speed ambient air in the mixing section. The inclusion of a diffuser at the ejector exit further converts kinetic energy into pressure, improving overall performance. The exhaust flow is inherently pulsating due to the cyclic variations in exhaust mass flow, velocity, and energy produced by the combustion process. Despite this, for preliminary design purposes, the flow can be treated as quasi-steady by considering an effective mean velocity, assuming complete mixing and negligible wall friction. In real systems, pressure recovery is slightly lower than theoretical predictions due to incomplete mixing and friction losses, both of which increase with ejector length. The uniform velocity assumption along the ejector cross-section is also an idealization; in practice, the primary exhaust jet accelerates adjacent air more efficiently than regions near the walls, and corner and wall resistance reduce the effective flow area. Consequently, theoretical cross-sections must be adjusted to account for these effects. The pressure increase generated by the ejector can be expressed as a function of several parameters, including the exhaust gas mass flow, the entrained air mass flow, the outlet area of the mixing section, gas density, specific gas constant, temperature, and empirical correction factors for nozzle contraction and di user effects. The mass flow ratio between air and exhaust gases $(\mathrm{m_a}/\mathrm{m_e})$ typically ranges from 6 to 9 for engines of interest. While some parameters, such as air temperature and density, are set by inlet conditions, others, like exhaust velocity and nozzle area, are determined by engine operating conditions. Optimal performance requires small nozzle areas to maximize exhaust momentum, constrained by engine operational limits. Ejector performance improves with the inclusion of a diffuser, which extends the operating range of the mass flow ratio and increases pressure recovery. The use of a flattened or elliptical nozzle shape enhances mixing and ejector efficiency compared to circular nozzles, though excessive flattening may reduce momentum and degrade performance. Ejector length also influences performance, with an optimum typically between six and seven times the diameter of the nozzle. Overall, the ejector design must balance the geometric parameters and engine operating conditions to achieve a sufficient pressure drop in the test chamber while maintaining high air entrainment efficiency.

4.1 Ejector Sizing Based on the NACA Study

The sizing of the ejector was based on an experimental study conducted by NACA in the study [12]. This study was performed to guide the design of a rectangular-section ejector intended for cooling an aircraft engine. The experiment utilized a four-stroke, single-row radial Wright 1820-G gasoline engine, modified to operate as a single-cylinder unit, with an exhaust-gas velocity of 1625 ft/s (490 m/s) at a temperature of approximately 815°C (1500°F). The NACA ejector exhaust consists essentially of three elements. The first is a converging section whose presence, although not fundamentally significant for the operation of the ejector, serves to guide and center the exhaust-gas flow. This is followed by the mixing region, in which the exhaust gases from the engine mix with and entrain the surrounding air. Finally, the flow passes through a diverging section, where the entrained air flow decelerates, thereby generating a low-pressure zone that contributes to the overall ejector effect. Through several experimental tests in which parameters such as the cross-sectional area and the length of the mixing section, the final area, and the opening angle of the diverging section were varied, a set of preliminary guidelines for designing an efficient ejector was established. The experiments demonstrated that an ejector can induce a fresh-air mass flow equal to six times the exhaust-gas mass flow, i.e., $\mathrm{m_a}/\mathrm{m_{exh}} = 6$. It was also determined that the cross-sectional area of the mixing section must be ten times the exhaust-nozzle area, namely $\mathrm{A_{mix}} = 10\ \mathrm{A_e}$, and that its length should range between six and seven times the hydraulic diameter, expressed as $\mathrm{L_{mix}} = 6–7\ \mathrm{D_h}$. Furthermore, the ratio between the exit and inlet areas of the diffuser must satisfy $\mathrm{A_{out}}/\mathrm{A_{in}} = 1.87$, while the diffuser itself must expand with an opening angle not exceeding 12 degrees to avoid flow separation. The study also indicated that a pressure rise greater than 6 inches of water column (1494.533 Pa) must be achieved between the inlet and the outlet, and that operating without a diffuser result in an approximate 40% reduction in the obtainable pressure rise. Starting from the exhaust-pipe diameter of the Wankel engine, equal to $\mathrm{d_e} = 60$ mm, the cross-sectional area of the mixing section was first computed by applying Rule 2, which requires $\mathrm{A_{mix}} = 10\ \mathrm{A_e}$. Since the mixing section has a square geometry, its side length was obtained from the relation $\mathrm{A_{mix}} = \mathrm{L}^2_{\text{mix, side}}$. Because the cross-section is rectangular, the hydraulic diameter had to be evaluated, defined as $\mathrm{D_h} = 4\ A/P$, where, A denotes the cross-sectional area and P its perimeter. Once the mixing section had been dimensioned, attention was turned to the diverging section. By applying Rules 4 and 5, the exit area of the diffuser was calculated from the condition $\mathrm{A_{out}} = 1.87\ \mathrm{A_{in}}$, which also enabled the determination of the side length of the final square cross-section. Assuming an opening angle of 12 degrees, the corresponding length of the diverging section was then obtained through geometric considerations. Using these dimensions, an initial CAD model of the ejector exhaust system was subsequently generated (see Table 2). After sizing the various components, a final assembly was created by combining the individual CAD models. This enabled the execution of CFD simulations aimed at determining the performance and efficiency of the test bench developed with this first solution equipped with the ejector exhaust.

The following CFD simulations were performed using SolidWorks 2013, along with its Flow Simulation extension, which is specifically designed for analyzing fluid behavior within components (see Figure 2). The mesh resolution was initially set to 5 out of 8 to evaluate the general trends of key parameters, such as velocity and pressure, within relatively short computation times. Subsequently, for a more precise analysis of the optimized components, the mesh was refined to 8 out of 8, providing results as close as possible to the real conditions achievable with this software. The first simulations addressed the exhaust box equipped with a diffuser dimensioned according to the NACA study. The inlet parameters for the simulation were an exhaust gas velocity of 500 m/s, an exhaust gas temperature of 1050°C, and an external pressure of 101,000 Pa applied to the internal surface of the lids. The monitored output quantities were the static pressure in Pascals and the average velocity in meters per second. Using different intake valve openings, it was determined that the opening radius that maximized the pressure drop while maintaining the minimum required airflow for engine aspiration was 10 mm. In this configuration, the minimum pressure at the entrance of the mixing section reached approximately 91,000 Pa, while the pressure within the box was around 92,500 Pa, corresponding to an altitude of about 900 m, which is below the desired target. Further analysis of the velocity profile in the mixing and divergent sections revealed that excessively large opening angles resulted in flow separation. By progressively reducing the opening angle to 4 degrees, the flow separation was eliminated, allowing for the effective utilization of the divergent section. This optimization also resulted in the pressure at the mixing section inlet decreasing to 91,000 Pa, while the pressure inside the box reached approximately 92,500 Pa. Refining the mesh to the maximum resolution of 8 out of 8 yielded a box pressure of approximately 91,600 Pa, corresponding to an altitude near 1,000 m. Table 3 presents the boundary conditions, mesh settings, and key performance indicators employed in the CFD analysis conducted using SolidWorks Flow Simulation 2013.

Table 2. Summary of the numerical sizing of the ejector

Parameter	Value	Notes
Exhaust diameter $\mathrm{d_{exh}}$	60 mm	Given engine data
Exhaust area $\mathrm{A_{exh}}$	$2.827 \times 10^{-3}\ \mathrm{m}^2$	Circular section
Mixing-section area $\mathrm{A_{mix}}$	$2.827 \times 10^{-2}\ \mathrm{m}^2$	
Diffuser inlet area $\mathrm{A_{in}}$	$2.827 \times 10^{-2}\ \mathrm{m}^2$	Equal to Amiig
Diffuser exit area $\mathrm{A_{out}}$	$5.2869 \times 10^{-2}\ \mathrm{m}^2$	
Diffuser angle	4 deg	Final optimized value
Diffuser length $\mathrm{L_{di}}$	0.442 m	Geometric evaluation

Table 3. Simulation parameters and key CFD outcomes

Parameter	Value	Notes
CFD software	SolidWorks Flow Simulation 2013	Steady-state CFD solver
Mesh level (coarse)	5/8	Used for trend evaluation
Mesh level (fine)	8/8	Used for final accuracy
Inlet exhaust velocity	500 m/s	Boundary condition
Inlet exhaust temperature	1000°C	High-temperature gas
External pressure	101000 Pa	Applied to lid's inner surfaces
Monitored outputs	Pressure, velocity	Spatial averages and profiles
Intake-valve radius (optimal)	10 mm	Ensures required air flow
Min. pressure at mixing entry	approx. 91000 Pa	Achieved with an optimal intake-valve radius
Pressure in the exhaust box	approx. 92500 Pa	Equivalent to 1000 m altitude
Flow separation	None at 4 deg	Occurred at 12 deg

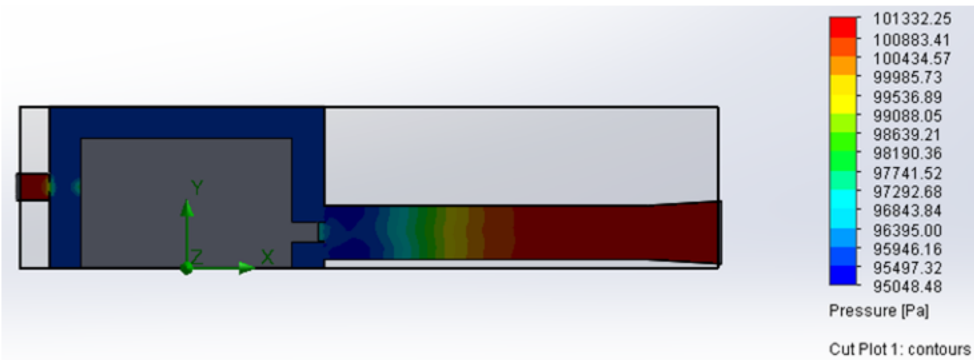

Figure 2. Pressure distribution within the test chamber using the optimized 4-degree diffuser angle and refined mesh

4.2 Auxiliary Engine Driven Venturi for Pressure Reduction

An alternative solution to the ejector-diffuser configuration was considered, involving the use of the exhaust from an auxiliary engine to supply a Venturi tube, thereby exploiting the pressure drop generated within it to reduce the pressure inside the test bench chamber. The auxiliary engine selected for this purpose was a diesel Iveco Cursor 13, commonly used in heavy-duty trucks. This engine features a six-cylinder architecture, a total displacement of 10,380 cm^3, and a maximum power output of approximately 450 HP. Its exhaust mass flow is sufficiently high to drive a Venturi tube of the dimensions required for the test bench. The Venturi effect, first studied by Giovanni Battista Venturi in 1797, describes the reduction of fluid pressure within a constricted section of a duct as the fluid velocity increases. In the present application, a reduced-section segment with a radius of 100 mm was inserted within the main exhaust conduit of the auxiliary engine, which has a radius of 130 mm. The contraction is connected to the main duct at both ends with an approximate angle of 20 degrees, forming a Venturi configuration. The mixing section of the test bench exhaust was modified to remove the divergent segment, enabling proper coupling with the Venturi tube. In this setup, the accelerated exhaust flow through the constriction produces a local pressure drop, which in turn reduces the pressure within the mixing section and the test chamber. For the CFD simulations, the boundary conditions were set as follows: atmospheric pressure (101,000 Pa) at the test bench intake valve, exhaust gas temperature of 1,343.15 K (1,070°C), and exhaust velocity of 510 m/s. The auxiliary engine exhaust was represented by a boundary with a velocity of 200 m/s and a pressure of 2 bar (200,000 Pa) at the Venturi inlet, while the outlet of the Venturi tube was set to atmospheric pressure (101,000 Pa). The simulations were performed with a refined mesh resolution of 8/8 to capture the pressure and velocity distributions within the components. The results indicate that the Venturi tube effectively reduces the pressure within the test bench (Figure 3). The minimum pressure achieved inside the box was approximately 76,000 Pa, corresponding to an altitude of about 2,500 m. This is significantly lower than the pressure obtained using the first ejector-based solution. Moreover, the exhaust gases entrain ambient air from the mixing section, and the Venturi-induced depression amplifies this effect, further lowering the chamber pressure. The flow acceleration and suction generated by the Venturi effect are clearly visible in the simulation's streamlines (Figure 4).

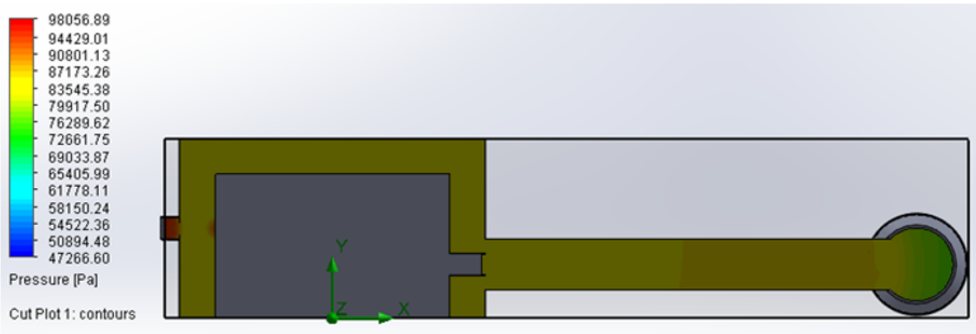

Figure 3. Pressure distribution in the test chamber with the Venturi tube connected to the auxiliary engine exhaust

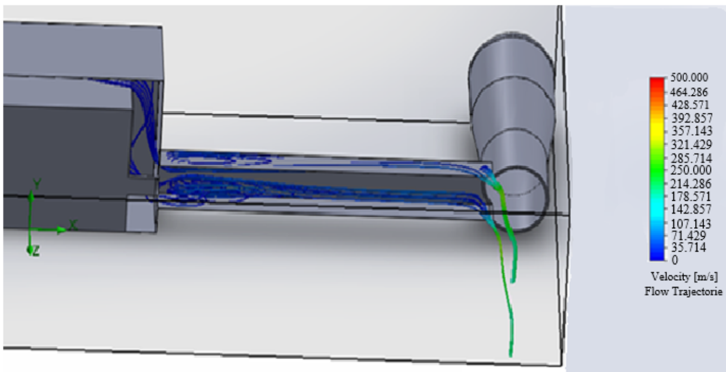

Figure 4. Streamlines illustrating the accelerated flow and entrainment caused by the Venturi tube

4.3 An Economical Alternative Solution

As an alternative, the ejector exhaust system can be replaced by a turbine driven by the exhaust energy of an external, auxiliary engine, which powers a turbocharger. In this configuration, the compressor of the turbocharger draws air from the pressurized chamber and discharges it to the atmosphere. This setup results in a modest reduction in auxiliary engine power and a slight increase in oil consumption, which exits through the compressor's discharge. However, it enables the generation of a vacuum at relatively low cost. The main limitation is that the largest commercially available automotive turbochargers, available at affordable prices, have an airflow capacity at atmospheric pressure of approximately 1 kg/s and a compression ratio of about 3. Consequently, such systems can recover altitudes of approximately 10,000 meters for small diesel and gasoline engines. Additionally, like in the previous test bench configuration, temperature and pressure inside the engine bay cannot be controlled independently of the intake air. In aircraft applications, the intake air is compressed through a convergent duct before entering the engine, allowing operation at even higher altitudes. Above 200 kW, the test bench inevitably becomes more complex and costly.

4.3.1 Third test bench configuration: Auxiliary engine coupled with a centrifugal turbocharger

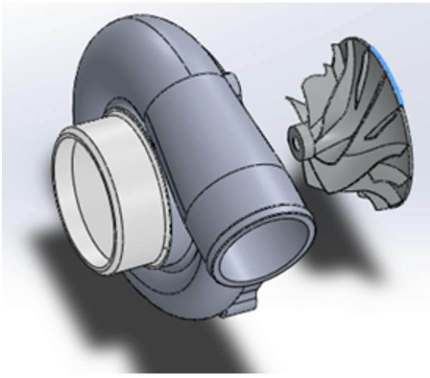

Figure 5. Assembly of the compressor wheel and volute

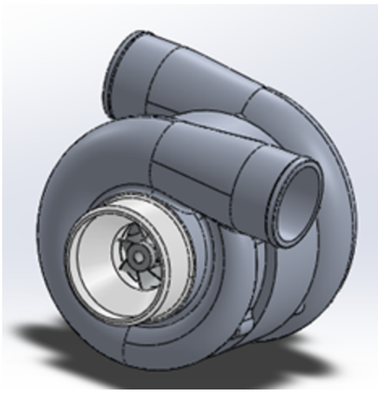

Figure 6. Fully assembled Holset HY55V turbocharger

The third solution adopted for the test bench consists of using the same auxiliary Cursor 13 engine as in the previous experiment, but now coupled, instead of to a Venturi tube, to a turbocharger. This configuration enables the evacuation of air from the interior of the test chamber to the outside through the action of the centrifugal compressor wheel, thereby generating the desired depression. A water-cooled Holset HY55V turbocharger with a titanium-alloy compressor wheel was selected for its availability, high thermal resilience, and suitability for integration with heavy-duty diesel engines (see Figure 5 and Figure 6). In fact, in standard series applications, the Cursor 13 engine is often supercharged using this model of turbocharger. After assembly, the turbocharger was connected to the test chamber via a duct. In this setup, there is no need for rigid and undeformable pipes, as was necessary for the ejector exhaust or the Venturi tube; a simple airtight duct connecting the test chamber to the intake of the centrifugal compressor is sufficient. The system operation begins with the exhaust gases from the auxiliary engine entering the turbine in a circumferential direction, driving the turbine wheel, and then exiting axially. The turbine wheel, mechanically connected to the compressor shaft, drives the compressor, which draws air from the test chamber through the intake duct. The air is compressed as it passes through the compressor and then expelled at a higher pressure in a circumferential direction. The pressure within the test chamber depends on both the opening of the up-stream intake valve and the rotational speed of the turbocharger. Higher rotational speeds result in greater air intake and, consequently, lower pressure inside the box. Subsequent fluid-dynamic simulations were performed to determine the pressure and velocity distributions in both the turbocharger and the test chamber.

4.3.2 CFD Analysis of the full experimental test bench featuring a centrifugal compressor operating in pump mode

To simplify the model, the fluid-dynamic simulations were performed considering only the compressor, excluding the centrifugal turbine, as including it would have increased the simulation complexity beyond the software's and computational resources' capabilities. This simplification is justified, as the operating regime is assumed to be above the minimum useful rotational speed of the turbocharger, sufficiently high to allow the compressor wheel to compress the intake air. Accordingly, the simulations were conducted with rotational speeds ranging from approximately 50,000 rpm to 100,000 rpm, consistent with the characteristic data of the Holset HY55V turbocharger. After connecting the centrifugal compressor to the test bench via the intake duct, several simulations were carried out varying both the opening of the intake valve upstream of the box and the rotational speed of the compressor wheel, within the admissible limits of the Holset HY55V. Boundary conditions were set with atmospheric pressure at both the upstream valve of the test bench and the outlet of the compressor duct. Initially, a mesh resolution of 4/8 was used to obtain approximate trends efficiently, followed by higher-resolution simulations with a mesh of 7/8 for more precise results. With an intake valve opening of 30 mm and a rotational speed of 80,000 rpm, the outlet pressure from the compressor was approximately 2.7 bar, while the pressure inside the test chamber stabilized at around 48,000 Pa. Subsequently, the minimum achievable pressure inside the test chamber was determined by optimizing the intake valve opening and the compressor rotational speed. With the turbocharger operating at its maximum speed of 100,000 rpm and the valve opening set to 26 mm, the compressor outlet pressure reached approximately 3.5 bar, while the pressure inside the test chamber stabilized at around 15,000 Pa. This corresponds to an altitude of approximately 14,000 meters, covering almost the entire operational flight envelope of light aircraft. These results confirm that the test bench configuration can reproduce the target operational pressures for the Wankel engine, providing a solid basis for further performance analysis and system optimization (see Figure 7, Figure 8, and Figure 9).

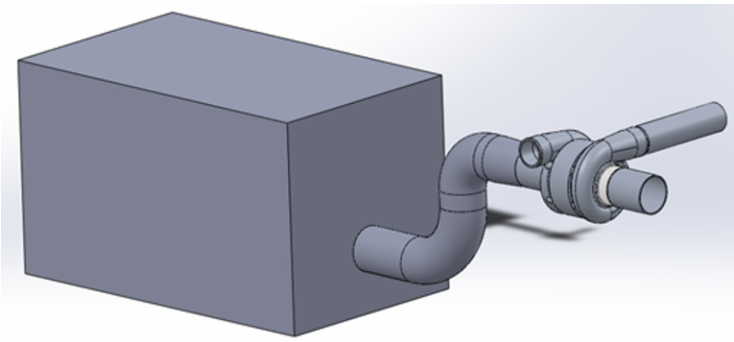

Figure 7. Complete test bench with connections to the intake and exhaust ducts

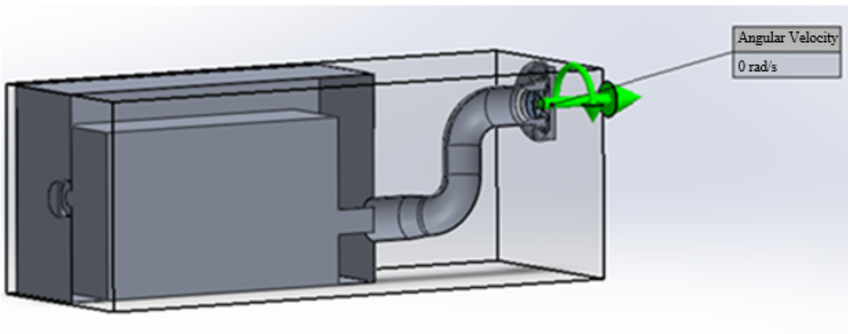

Figure 8. Control volume of the complete test bench, including the centrifugal compressor

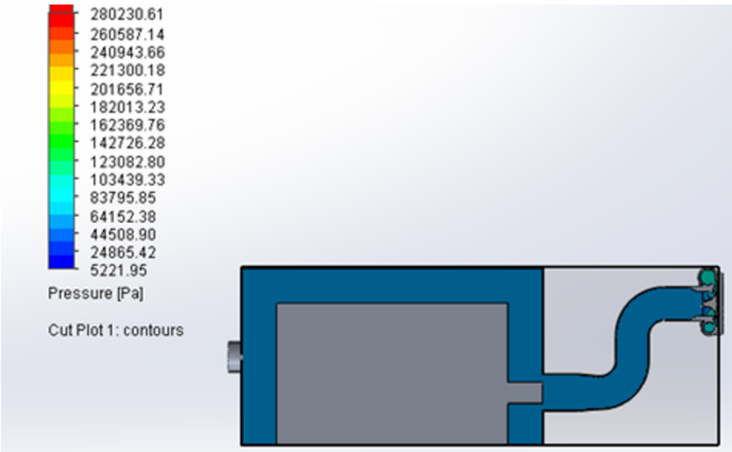

Figure 9. Pressure distribution within the test chamber at 100,000 rpm with an intake valve opening of 26 mm

5 Conclusions

This study investigated three different configurations for generating controlled depressions inside an altitude engine test bench, focusing on engines with a power output of up to approximately 200 kW. The first configuration utilized an ejector-diffuser system based on NACA design principles, which provided moderate pressure reduction but was prone to flow separation in the diffuser at higher angles. The second configuration utilized the exhaust of an auxiliary Cursor 13 engine, coupled to a Venturi tube, which exploited the Venturi effect to further reduce the pressure inside the test chamber. This approach achieved a minimum chamber pressure of approximately 76,000 Pa, corresponding to an altitude of about 2,500 m, and enhanced air entrainment in the mixing section. The third configuration combined the same auxiliary engine with a centrifugal turbocharger (Holset HY55V). By adjusting the compressor rotational speed and the intake valve opening, pressures inside the test chamber could be reduced to approximately 15,000 Pa, equivalent to an altitude of 14,000 m. This configuration demonstrated the highest flexibility and capability, covering virtually the entire flight envelope of the aircraft engine. CFD simulations confirmed that this solution enables precise control of airflow and pressure inside the test chamber, with minimal influence from geometric simplifications and mesh resolution. Altitude engine test benches represent highly integrated systems with stringent operational constraints, elevated installation costs, and demanding reliability requirements. Propeller aircraft propulsion is predominantly based on piston engines, which exhibit relatively low dependency on ram-air cooling, simplifying their operation under simulated altitude conditions. Reproducing altitude environments requires significant variations in static pressure and temperature, ranging from sea-level conditions (approximately 35°C and 100,000 Pa) to extreme altitudes (about -56.5°C and 14,101.8 Pa at 14,000 m). The performance of an altitude test bench is significantly influenced by the management of energy input, airflow rates, and thermal control. Piston engines with a power output of up to 200 HP typically operate with air flow rates below 2 kg/s; however, achieving the target vacuum in the test chamber requires careful coordination between the air-supply system, exhaust evacuation, and compressor action. Minimizing the volume of the pressurized enclosure improves efficiency and reduces the energy required to maintain target conditions. Transient operation constitutes a critical performance metric, as engines experience rapid variations in ambient state variables during climb or other phases of flight. The third configuration demonstrated that coupling an auxiliary engine with a turbocharger allows rapid and controlled evacuation of air from the test chamber, overcoming the limitations of self-depressurization and micro-leak sensitivity inherent to simpler configurations. This ensures both accurate steady-state pressures and reliable transient response, making the bench suitable for extended endurance tests and operational simulations. In conclusion, the study demonstrates that a combination of physical principles, careful definition of boundary conditions, and component selection enables effective altitude simulation for piston engines with a power rating of up to 200 kW. The turbocharger-based configuration provides the most reliable, flexible, and energy-efficient solution, capable of simulating the full flight envelope and supporting advanced experimental investigations of engine performance under controlled altitude conditions.

Data Availability

The data used to support the research findings are available from the corresponding author upon request.

Conflicts of Interest

The author declares no conflicts of interest.

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