

One-Dimensional Thermodynamic Analysis of a Novel Two-Stroke Aircraft Diesel Engine with an Integrated Cylinder–Dual-Role Crankshaft Architecture

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Abstract: *This study presents the one-dimensional thermodynamic assessment of a novel two-stroke, direct-injection aircraft Diesel engine incorporating two original mechanical elements, a dedicated cylinder architecture, and a dual-role crankshaft. The cylinder combines piston-controlled scavenging ports with a laterally integrated exhaust valve, while the crankshaft performs both its conventional power-transmission function and the direct mechanical actuation of the exhaust valve. This arrangement enables uniflow gas exchange without a conventional camshaft dedicated to exhaust-valve actuation. The engine further employs an analogue control architecture, wet-sump lubrication, turbocharging and charge-air cooling. The previously developed three-dimensional mechanical configuration is implemented in AVL BOOST and investigated using the AVL Mixing Controlled Combustion (AVL-MCC) model at a compressor pressure ratio of 3.0. Gas exchange, boosting, in-cylinder thermodynamics, combustion, heat transfer and indicated and brake performance are evaluated as coupled processes. The study establishes the functional relationship between the proposed cylinder–crankshaft architecture and the predicted thermodynamic response of the complete engine, providing a system-level numerical assessment of the proposed aircraft-engine concept.*

Key Words: *two-stroke aircraft Diesel engine, dual-role crankshaft, novel cylinder architecture, lateral exhaust valve, uniflow scavenging, direct injection, AVL BOOST, AVL-MCC*

1. INTRODUCTION

Compression-ignition piston engines remain attractive for aircraft propulsion because of their high thermal efficiency, fuel economy and compatibility with high-energy-density fuels. Two-stroke architectures additionally offer a favorable power-to-displacement potential, although their performance is strongly dependent on charge exchange, scavenging and boosting processes [1–4]. For aircraft applications, these interactions become particularly relevant because the boosting system must support cylinder charging under varying atmospheric conditions while maintaining acceptable combustion and thermal behavior [3, 4].

In uniflow two-stroke engines, fresh air typically enters through piston-controlled ports and displaces residual gases toward the exhaust-valve region. Consequently, scavenging-port geometry, valve timing and the pressure differential across the cylinder directly affect

scavenging efficiency, charge retention and the thermodynamic state preceding compression [5–10].

Experimental and numerical studies have demonstrated the strong coupling between scavenging characteristics, in-cylinder flow, subsequent combustion and overall engine performance [5–12].

The relevance of this architecture to aviation has also been demonstrated by investigations of two-stroke compression-ignition engines specifically developed for general aviation. Numerical studies have addressed scavenging effectiveness, port design and in-cylinder flow, whereas system-level investigations have emphasized the interaction between cylinder filling, scavenging and supercharging [3, 4, 13]. Nevertheless, these studies predominantly retain conventional functional separation between the crank mechanism and exhaust-valve actuation.

The engine investigated introduces two mechanically integrated design elements, a novel cylinder architecture and a dual-role crankshaft. The cylinder incorporates piston-controlled scavenging ports together with a lateral exhaust valve integrated into the cylinder structure. The crankshaft performs its primary function of transmitting piston work while simultaneously providing direct mechanical actuation of the exhaust valve. The resulting architecture integrates gas-exchange actuation into the crank mechanism and eliminates the requirement for a conventional camshaft dedicated to exhaust-valve operation. These elements constitute the principal mechanical novelty of the proposed engine.

The thermodynamic feasibility of such an architecture cannot, however, be inferred from its mechanical configuration alone. In a two-stroke direct-injection Diesel engine, scavenging, charge trapping, compression, combustion, expansion and exhaust processes are intrinsically coupled [5–12]. Moreover, turbocharging and charge-air cooling modify the boundary conditions governing these processes. A system-level numerical model is therefore required to determine whether the proposed mechanical architecture can sustain a thermodynamically consistent operating cycle.

Accordingly, the present study evaluates the proposed engine using a one-dimensional AVL BOOST model incorporating AVL Mixing Controlled Combustion (AVL-MCC) and a compressor pressure ratio of 3.0. The analysis addresses gas exchange, charge-air management, in-cylinder thermodynamics, combustion, heat transfer, turbocharger operation and indicated and brake performance.

The contribution of this work is the system-level thermodynamic assessment of the newly developed cylinder and dual-role crankshaft architecture, establishing the quantitative link between its mechanical implementation and the predicted functional behavior of the complete two-stroke aircraft Diesel engine.

2. ONE-DIMENSIONAL NUMERICAL MODEL AND SIMULATION METHODOLOGY

The thermodynamic assessment of the proposed engine concept was performed using a one-dimensional model developed in AVL BOOST, integrating the cylinders with the intake, exhaust, turbocharging, and charge-air cooling systems. This system-level formulation enables the coupled assessment of gas-exchange, combustion, and energy-conversion processes throughout the engine architecture.

AVL BOOST is a one-dimensional thermodynamic simulation environment developed for the system-level analysis of internal combustion engines. The framework enables the coupled representation of gas exchange, in-cylinder thermodynamics, combustion, fuel injection, turbocharging, and duct-flow processes within a unified numerical model [13, 14].

Gas properties are evaluated locally according to the instantaneous thermodynamic state and gas composition, allowing the transient interaction between cylinders, connecting ducts, plenums, and boosting components to be resolved throughout the engine cycle [3]. Within the present study, AVL BOOST is employed to reproduce the functional architecture of the proposed two-stroke engine and to quantify the interaction between scavenging, exhaust, turbocharging, combustion, heat transfer, and energy conversion. The approach is particularly suited to system-level thermodynamic assessment because the individual engine components are solved within the same crank-angle-resolved computational framework [13, 14].

The one-dimensional formulation inherently imposes limitations on the spatial resolution of local flow phenomena. Accordingly, the present model is not intended to resolve three-dimensional in-cylinder structures, local scavenging-flow topology, spatial fuel-spray development, or detailed velocity and temperature fields. Such phenomena require multidimensional CFD methods, for which AVL provides dedicated three-dimensional flow, injection, mixture-formation, combustion, and thermal-load modelling capabilities [15].

Consequently, the AVL BOOST results are interpreted as a system-level thermodynamic assessment of the proposed architecture rather than as a spatially resolved characterization of the in-cylinder flow field. The investigated configuration corresponds to a four-cylinder, two-stroke, direct-injection Diesel engine employing uniflow gas exchange. The principal geometrical parameters and operating conditions implemented in the numerical model are summarized in Table 1.

Table 1. Main geometrical and operating parameters of the investigated engine

Parameter	Value
Number of cylinders	4
Bore	65 mm
Stroke	70 mm
Connecting rod length	147 mm
Swept volume	0.9291 L
Compression ratio	24:1
Engine speed	5500 rpm

The gas-exchange architecture reproduces the distinctive mechanical features of the proposed engine. Scavenging is achieved through cylinder ports controlled by piston motion, whereas exhaust flow is governed by a lateral exhaust valve integrated into the cylinder and directly actuated by the dual-role crankshaft. The numerical model therefore preserves the kinematic relationship between piston position, scavenging-port timing, and exhaust-valve actuation imposed by the proposed mechanical architecture.

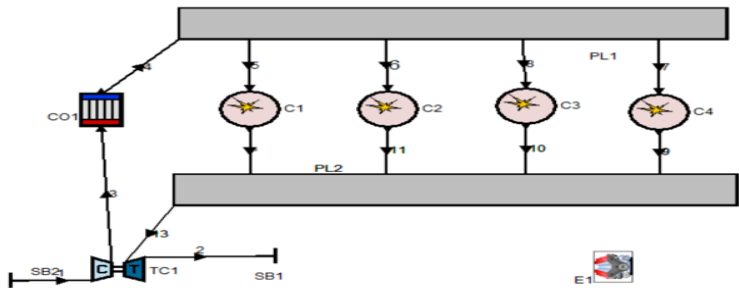


Fig. 1 One-dimensional AVL BOOST model of the proposed two-stroke aircraft Diesel engine

Figure 1 presents the one-dimensional architecture implemented in AVL BOOST. The four cylinders (C1–C4) are coupled to common intake and exhaust plenums (PL1 and PL2), while the gas-path model incorporates the turbocharger (TC1) and charge-air cooler (CO1). The resulting topology provides a system-level representation of the interactions between cylinder gas exchange, pressure-wave dynamics, turbocharging, and charge-air conditioning under the investigated operating condition. Combustion is described using the AVL Mixing Controlled Combustion (AVL-MCC) model. The present investigation is restricted to the configuration corresponding to a compressor pressure ratio of 3.0, thereby enabling the thermodynamic response of the proposed architecture to be assessed under a single, consistently defined operating condition. Consistent with two-stroke operation, the computational cycle spans 360° CA, and a multi-cycle calculation procedure is employed until numerical convergence is achieved. Within this framework, the 1D model provides a consistent basis for the coupled assessment of gas exchange, turbocharging, in-cylinder thermodynamics, combustion, heat transfer, and overall engine performance. The results are interpreted at the system level; three-dimensional flow structures, spatial fuel-jet development, and local thermofluid-field non-uniformities are not explicitly resolved. This distinction defines the scope of the numerical assessment and prevents extrapolation of the 1D results to phenomena requiring three-dimensional CFD analysis or experimental validation.

3. FUNCTIONAL AND THERMODYNAMIC ANALYSIS OF THE ENGINE CYCLE

The analysis addresses the functional and thermodynamic response of the proposed engine architecture over a complete 360° CA cycle for the numerical configuration defined in Section 2. The results are examined according to the physical sequence of the engine processes, from charge-air conditioning and admission to the discharge of the burned gases. This approach enables a direct correlation between the specific features of the cylinder and crankshaft architectures and the corresponding gas-exchange, compression, combustion, and expansion processes. The analysis is conducted for the selected AVL-MCC operating condition at a compressor pressure ratio of 3.0, using crank-angle-resolved results and the converged numerical values provided by the model. For the four-cylinder engine, the combustion events are phased at 0° , 90° , 180° , and 270° CA, resulting in a uniform 90° CA spacing over the 360° CA cycle. The results are interpreted at both cylinder and system levels to account for the interaction between the in-cylinder processes, the boosting system, and the gas-exchange path.

Particular attention is given to establishing whether the proposed architecture comprising piston-controlled scavenging ports, a lateral exhaust valve integrated into the cylinder, and its direct actuation by the dual-role crankshaft provides a thermodynamically consistent sequence of processes required for engine operation. Accordingly, the results are discussed in the physical order of charge-air admission and conditioning, scavenging and gas exchange, compression, injection and combustion, expansion, and exhaust, followed by an assessment of the resulting overall engine performance.

Air Supply, Intake-Plenum Dynamics, and Cylinder Charging

The functional cycle is characterized initially by analyzing the thermodynamic conditions imposed on the intake system. For the investigated configuration, the compressor operates at a pressure ratio of 3.0, while the intake plenum PL1 exhibits mean pressure and temperature values of 2.9100 bar and 358.63 K, respectively. These quantities define the thermodynamic state of the charge supplied to the cylinders during the scavenging process.

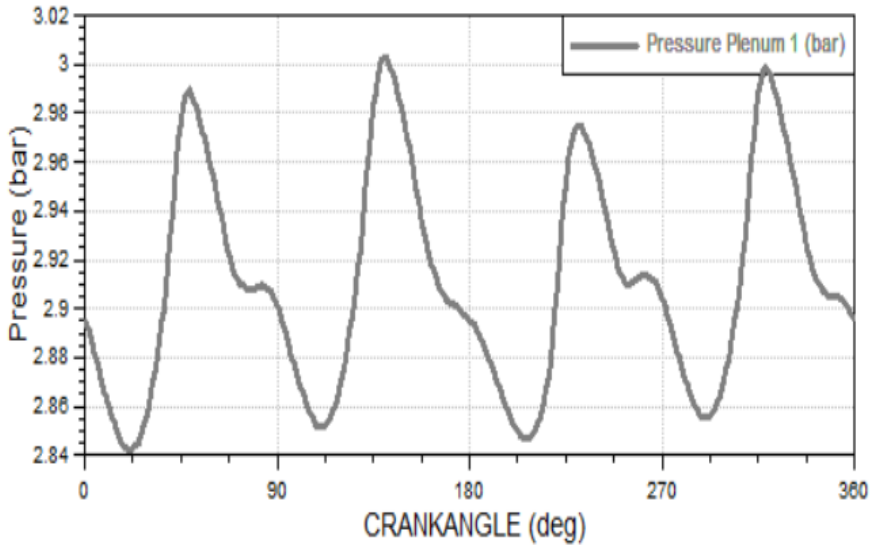


Fig. 2 Intake-plenum PL1 pressure variation as a function of crank angle

The pressure distribution in PL1 indicates a periodic gas-dynamic regime superimposed on the mean boost-pressure level. The sequence of pressure pulsations is correlated with the phase displacement between the cylinders, defined by firing angles of 0° , 90° , 180° , and 270° CA. This behavior highlights the role of the plenum as a common air-supply volume and, simultaneously, the propagation of unsteady interactions induced by the individual gas-exchange processes.

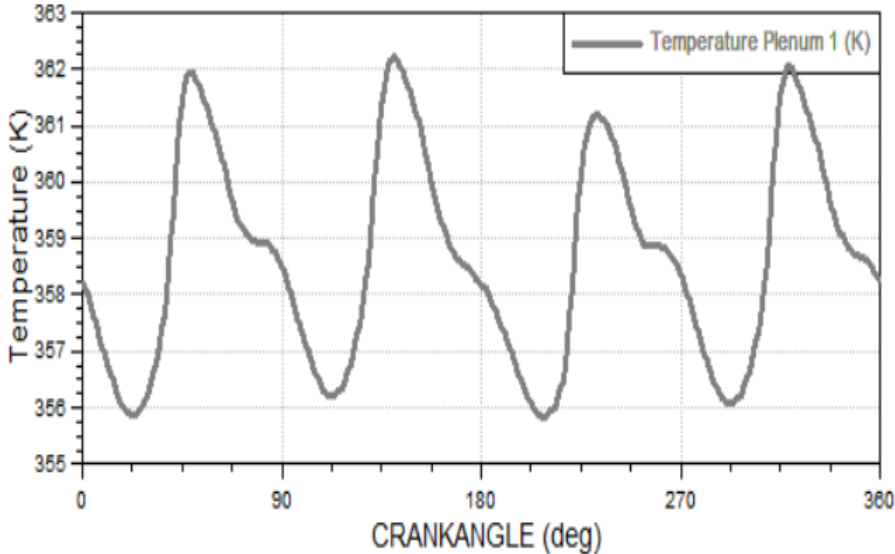


Fig. 3 Intake-plenum PL1 temperature variation as a function of crank angle

The thermodynamic state within PL1 is also characterized by a periodic temperature variation about the mean value of 358.63 K.

The temporal correlation between pressure and temperature oscillations confirms the unsteady nature of the charging process, even under globally converged operating conditions of the numerical model.

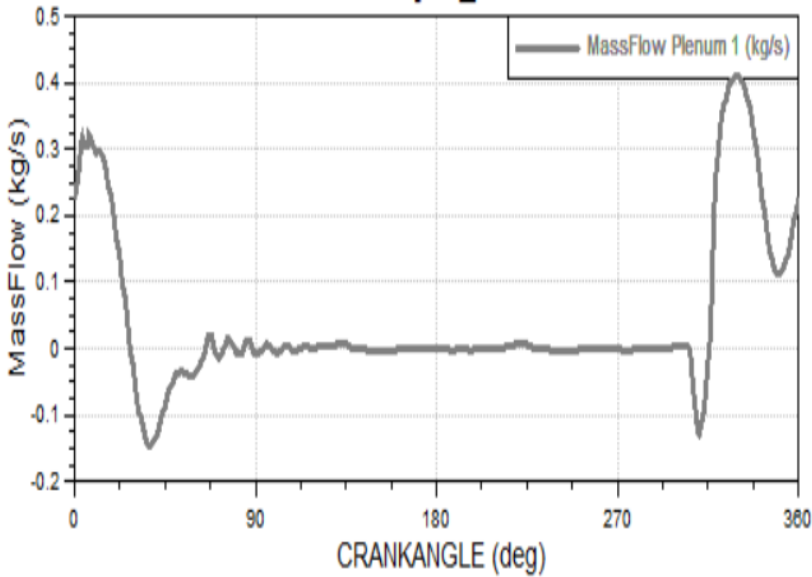


Fig. 4 Instantaneous mass-flow-rate variation in the intake plenum PL1 over one engine cycle

The instantaneous mass-flow rate exhibits pronounced variations and changes in sign throughout the cycle. According to the sign convention adopted by the model, negative values correspond to local flow reversal. This behaviour highlights the gas-dynamic interaction between PL1 and the individual cylinder supply branches and supports the evaluation of the intake process using instantaneous quantities rather than exclusively cycle-averaged parameters.

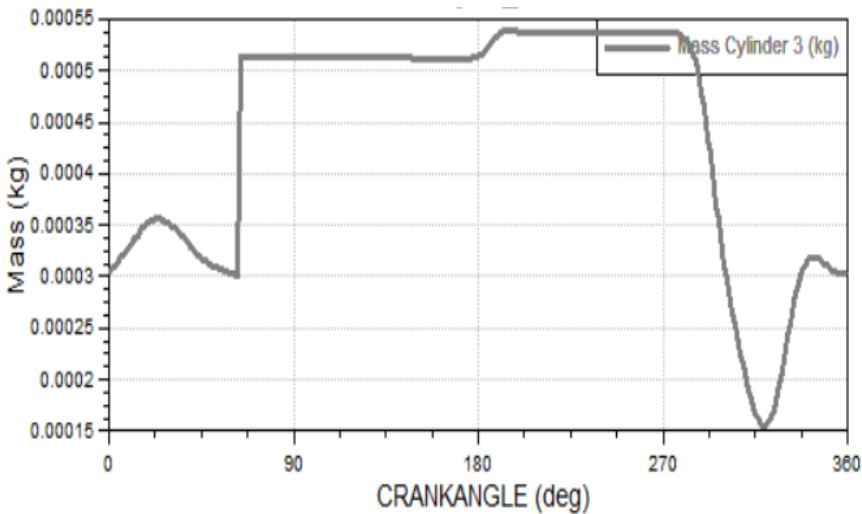


Fig. 5 Evolution of the charge mass in cylinder C3 as a function of crank angle

The cylinder response to the conditions imposed by the intake system is quantified through the evolution of the retained mass. For C3, the simulation yields an air mass of 0.4621 g at the start of the high-pressure phase and a volumetric efficiency of 1.7373. The mass evolution during gas exchange characterizes the sequence of exhaust, scavenging, and fresh-charge retention prior to cylinder isolation.

The combined analysis of PL1 pressure, temperature, and mass flow together with the retained mass in C3 establishes the relationship between the gas-dynamic response of the intake system and the initial condition of the closed-cycle phase. This state subsequently provides the basis for evaluating compression and the thermodynamic evolution preceding combustion.

Gas-Exchange Timing and Exhaust-Valve Gas-Dynamic Characteristics

The gas-exchange process of the investigated engine is governed by two distinct mechanisms synchronized with crankshaft position: the lateral exhaust valve and the piston-controlled scavenging ports. For the investigated numerical configuration, the exhaust valve opens at 95° CA, remains open for 120° CA, and closes at approximately 215° CA. A maximum valve lift of approximately 9 mm is reached during the first half of the opening interval, with the prescribed lift profile being asymmetric with respect to its maximum value.

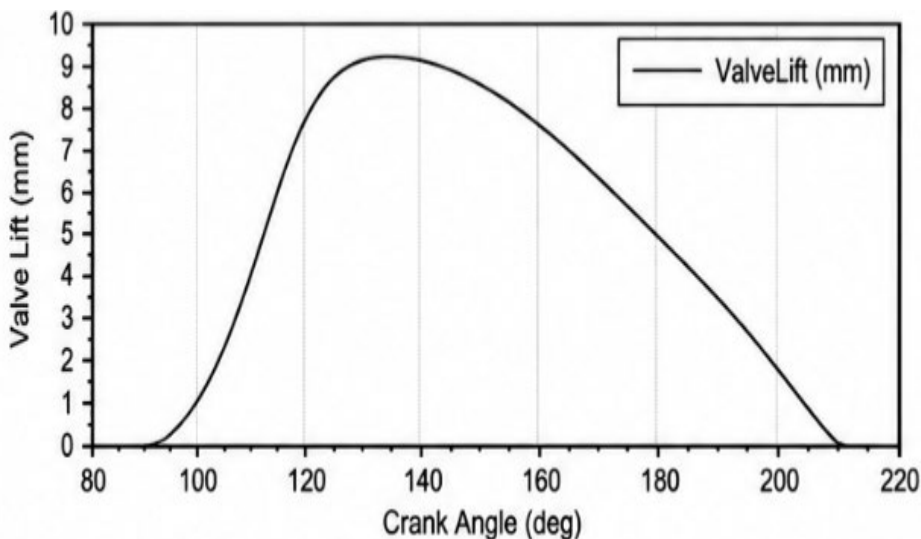


Fig. 6 Exhaust-valve lift profile as a function of crank angle

The phasing of the exhaust-valve event relative to the scavenging-port opening establishes the functional sequence of the gas-exchange process. Early exhaust-valve opening provides the interval required for cylinder blowdown before the scavenging ports develop a substantial effective flow area, whereas the subsequent overlap between the two flow paths defines the interval available for uniflow scavenging.

Consequently, the valve-lift profile represents not merely a kinematic boundary condition but a governing parameter of the effective exhaust flow area and, therefore, of the pressure differential driving the scavenging process.

The gas-dynamic behaviour of the exhaust valve is represented in the model by the discharge coefficient, specified as a function of normalized valve lift.

The pronounced initial increase in, followed by a progressively decreasing gradient at higher lifts, demonstrates the nonlinear relationship between valve lift and effective exhaust flow capacity.

At high normalized lifts, approaches approximately 0.85–0.90, indicating a reduced sensitivity of the effective flow capacity to further increases in valve lift.

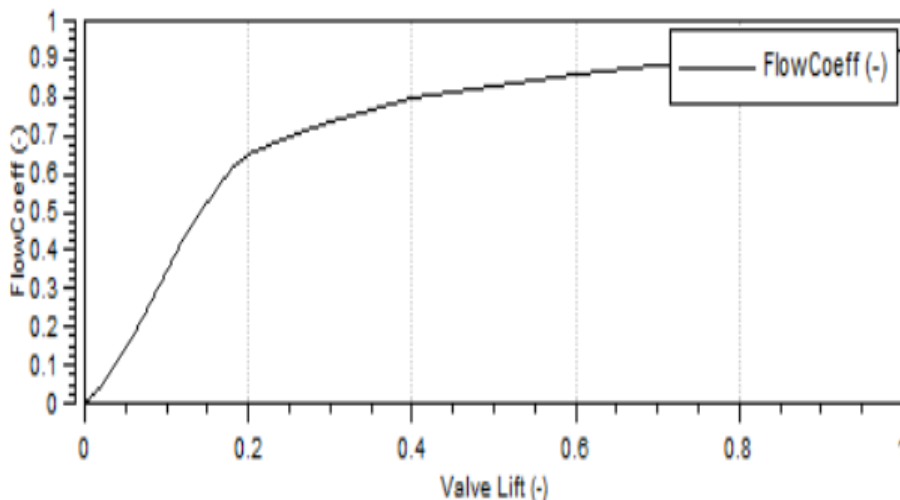


Fig. 7 Exhaust-valve discharge coefficient as a function of normalized valve lift

This nonlinear behaviour is relevant to the functional assessment of the proposed mechanism because the instantaneous exhaust flow cannot be characterized by the geometric valve-lift profile alone. The effective discharge capacity is determined by the coupled influence of valve lift, discharge coefficient, and the instantaneous upstream-to-downstream thermodynamic conditions. Accordingly, the combined implementation of the valve-lift profile and the characteristic provide a physically consistent gas-dynamic representation of the exhaust valve within the one-dimensional model, without directly equating geometric valve lift with effective flow area.

The scavenging ports constitute the second timing element governing the gas-exchange process. Their opening is controlled directly by piston position, and the available flow area increases from zero to approximately 2000 mm² in the vicinity of 180° CA, before progressively decreasing as the ports are covered during the subsequent piston motion.

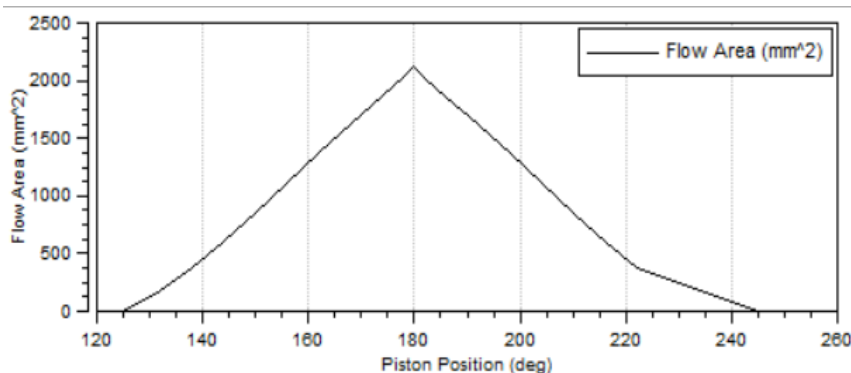


Fig. 8 Scavenging-port flow area as a function of crank angle

The temporal correlation between the scavenging-port flow area and the effective flow area associated with the exhaust valve defines the gas-exchange timing diagram of the investigated configuration. From a thermodynamic perspective, this overlap governs the intervals available for cylinder blowdown, scavenging, and subsequent charge trapping. The prescribed gas-exchange timing therefore establishes the direct functional link between the

specific architecture of the cylinder and the dual-role crankshaft and the resulting gas-dynamic response of the engine. This response is quantified in the following sections through the predicted in-cylinder pressure, trapped mass, and temperature histories.

In-Cylinder Thermodynamic Analysis

Characterisation of the in-cylinder thermodynamic process requires a correlated assessment of pressure and temperature evolution as a function of crank angle. These two state variables synthesize the response of the cylinder charge to compression, combustion-induced energy release, and gas expansion, thereby enabling the evaluation of combustion phasing relative to top dead centre (TDC). Within this framework, the in-cylinder pressure history predicted for cylinder is presented in Figure 9.

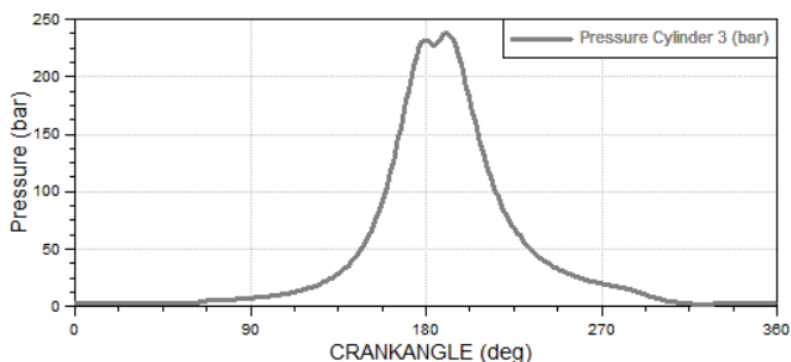


Fig. 9 In-cylinder pressure evolution as a function of crank angle for Cylinder 3

The pressure evolution delineates the transition from charge compression to a regime dominated by combustion-induced energy release. In the proximity of TDC, the energy released during combustion produces a pronounced acceleration of the pressure rise beyond the contribution associated solely with the reduction in cylinder volume.

For Cylinder 3, the calculation yields a peak firing pressure of 238.06 bar, located at 10.76° after TDC. The occurrence of the pressure maximum during the initial stage of the expansion stroke constitutes a direct indicator of combustion phasing, demonstrating that the maximum pressure development occurs after piston motion has reversed and the instantaneous cylinder volume has begun to increase. A more rigorous characterisation of the process is obtained by examining the pressure gradient with respect to crank angle. The calculated maximum value of 9.56 bar/deg occurs at 4.56° before TDC. The angular separation between the maximum pressure gradient and the absolute pressure maximum reflects the transient nature of the combustion energy-release process.

Accordingly, the maximum identifies the interval of most rapid pressure development, whereas the pressure maximum is displaced towards the expansion stroke as a consequence of the instantaneous balance between combustion-induced energy addition and the pressure-reducing effect associated with the increasing cylinder volume. From this perspective, peak pressure magnitude alone does not constitute a sufficient descriptor of the combustion process. Its angular location and the maximum pressure-rise rate represent complementary parameters for assessing combustion phasing and intensity.

The thermodynamic conditions prevailing at the onset of combustion further support this interpretation. For Cylinder 3, the calculated state at the start of combustion is characterized by a pressure of 193.9542 bar and a temperature of 1526.10 K. These levels are associated with compression of the cylinder charge under a configuration characterized by a geometric

compression ratio of 24.00 and a dynamic compression ratio of 18.48. Consequently, combustion is initiated under conditions of high pressure and elevated thermal state, while the subsequent pressure evolution results from the superposition of residual compression effects, combustion-induced energy release, and the rapid variation in cylinder volume in the TDC region. For a comprehensive interpretation of the thermodynamic response, the pressure analysis must be correlated with the evolution of the in-cylinder gas temperature. In contrast to pressure, whose behaviour in the proximity of TDC is strongly influenced by the instantaneous variation in cylinder volume, temperature provides a direct indicator of the energetic state of the charge and of the competition between combustion-induced energy release, expansion work, and heat transfer to the cylinder boundaries. The calculated temperature evolution for Cylinder 3 is presented in Figure 10.

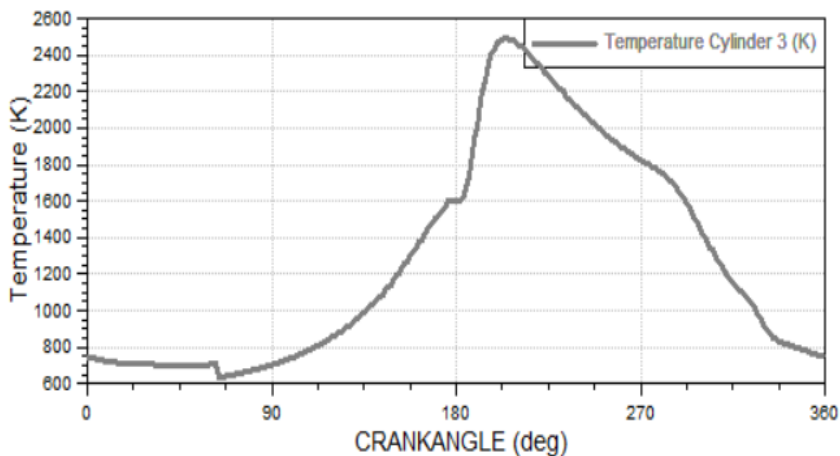


Fig. 10 In-cylinder gas temperature evolution as a function of crank angle for Cylinder 3

The calculated peak firing temperature for Cylinder 3 is 2488.84 K, attained at 24.45° after TDC. Comparison of the angular positions of the thermodynamic extrema reveals a fundamental feature of the process. The temperature maximum is displaced by 13.69° relative to the pressure maximum, the latter occurring at 10.76° after TDC. This angular offset does not merely represent a geometrical feature of the two profiles, rather, it expresses the difference between the mechanisms governing the instantaneous evolution of pressure and temperature. Following the attainment of peak pressure, the cylinder volume continues to increase, thereby limiting any further pressure rise. Simultaneously, combustion-related energy release continues to influence the internal energy of the gases, resulting in the thermal maximum being reached at a later crank-angle position.

This temporal sequence maximum at 4.56°, peak pressure at 10.76°, and peak temperature at 24.45° provides a coherent representation of the thermodynamic evolution in the TDC region. It delineates the transition from the stage of rapid pressure development to the attainment of peak cylinder pressure and, subsequently, to the maximum thermal state of the working gases. The simultaneous interpretation of these three characteristic locations is more informative than an independent assessment of the respective maximum values, since it enables the phasing of the thermodynamic response of the cylinder charge to be characterised relative to piston kinematics.

Following the thermal maximum, the progressive decrease in gas temperature is governed by the increasing predominance of expansion and wall heat-transfer effects over the residual energy addition associated with combustion. This stage should not be considered

independently of the subsequent gas-exchange process. The thermodynamic state established towards the end of expansion defines the initial conditions for burned-gas discharge, while the pressure and temperature differentials prevailing at exhaust opening govern the intensity of the blowdown process and, subsequently, the interaction between exhaust discharge and scavenging. Consequently, the results presented in Figures 9 and 10 establish the thermodynamic basis for the analysis of the gas-exchange process and scavenging effectiveness addressed in the following section.

Pressure-Rise Characteristics

The pressure-rise rate provides a differential characterization of the in-cylinder pressure evolution and enables identification of the crank-angle interval over which the gas loading develops most rapidly. For Cylinder 3, the calculated maximum pressure-rise rate is 9.56 bar/deg, occurring at -4.56° CA relative to TDC. The occurrence of this maximum before TDC, whereas the peak cylinder pressure of 238.06 bar is reached at 10.76° CA after TDC, establishes a clear angular separation between the maximum rate of pressure development and the attainment of the absolute pressure maximum.

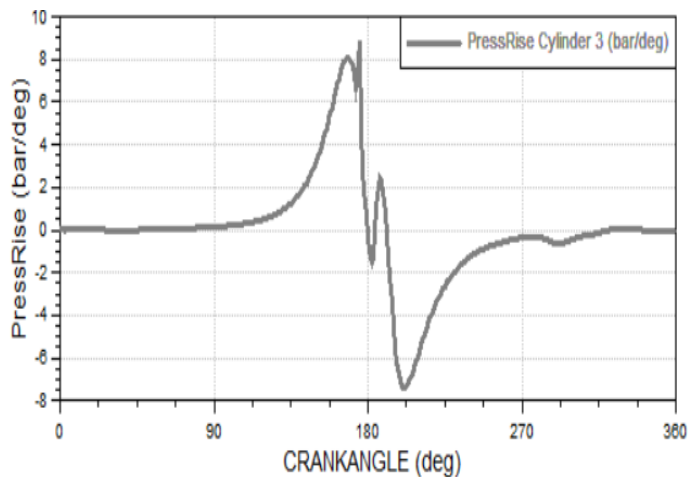


Fig. 11 In-cylinder pressure-rise rate as a function of crank angle for Cylinder 3

The profile presented in Fig. 11 exhibits a pronounced transition from positive to negative pressure-gradient values across the TDC region. This reversal delineates the transition from the pressure-build-up regime to the stage in which expansion increasingly governs the instantaneous pressure evolution. From the standpoint of mechanical loading, the maximum pressure-rise rate provides a complementary metric to peak cylinder pressure, as it characterizes the transient intensity of the loading transmitted through the piston–connecting-rod–crankshaft assembly.

Wall Heat-Transfer Characteristics

Wall heat transfer constitutes an irreversible energy-transfer mechanism that directly reduces the fraction of the working-gas energy available for conversion into indicated work. The crank-angle-resolved response calculated for Cylinder 3 exhibits a strongly transient behaviour, with the intensification of wall heat transfer associated with the interval of highest thermal loading of the in-cylinder gases. The phasing of this region relative to the pressure and temperature maxima confirms the dependence of wall heat transfer on the instantaneous thermodynamic state of the charge and on the convective conditions established within the cylinder.

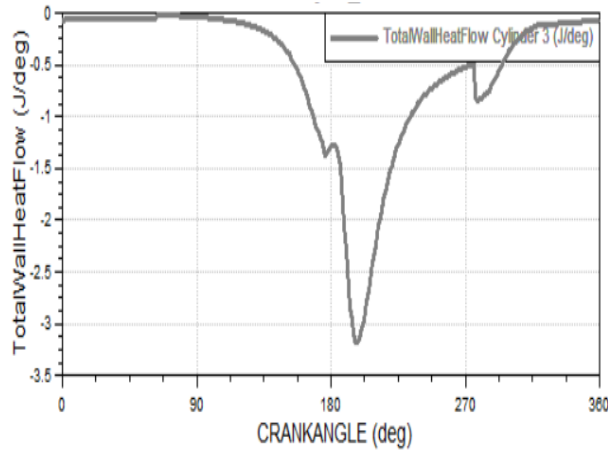


Fig. 12 Total wall heat flow as a function of crank angle for Cylinder 3

The subsequent decrease in wall heat flow reflects the progressive reduction in the thermodynamic potential for energy transfer during the expansion process. For Cylinder 3, integration over the complete cycle yields a net wall heat transfer of -0.19665 kJ/cycle, corresponding to 15.32% of the fuel energy input. The magnitude of this fraction establishes wall heat transfer as a significant contribution to the cycle energy balance and quantifies the energy rendered unavailable for conversion into indicated work. Accordingly, this result provides a relevant basis for correlating the thermal response of the cylinder with the overall performance characteristics of the investigated engine configuration.

Brake Performance Characteristics

The converged engine response is characterized by an effective torque of 290.15 N·m, an effective power of 167.11 kW, and a BMEP of 19.62 bar, indicating a consistent conversion of the in-cylinder work into useful shaft output. The corresponding BSFC of 247.30 g/kWh defines the fuel requirement associated with this performance level, while the mechanical efficiency approaching 0.90 indicates comparatively limited mechanical losses. The simultaneous stabilization of these parameters confirms the establishment of a numerically converged operating regime and supports the thermodynamic consistency of the predicted engine performance. The convergence of the effective torque and power towards their stabilized values is illustrated in Fig. 13.

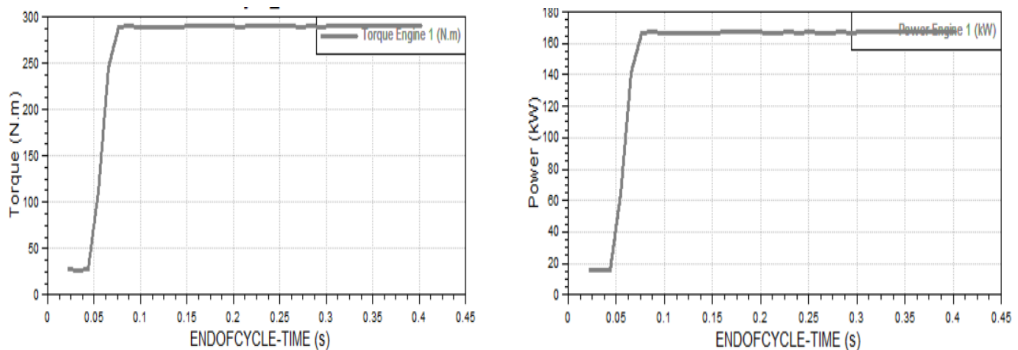


Fig. 13 Temporal evolution of the engine brake performance: (a) effective torque and (b) effective power, showing the transition from the initial transient response to the converged operating regime

The evolution of mechanical efficiency and BMEP is presented in Fig. 14. Following the initial transient, both parameters converge rapidly towards stable levels of approximately 0.90 and 19.62 bar, respectively. Their concurrent stabilization indicates limited mechanical losses and a consistent conversion of the indicated cycle work into effective engine output.

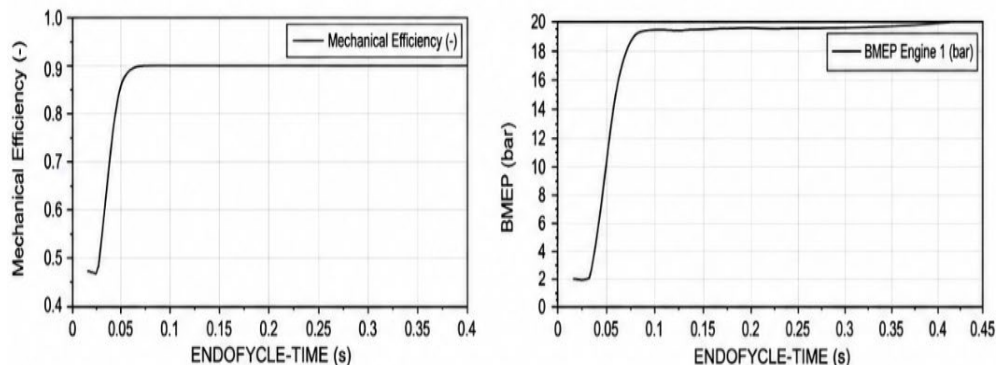


Fig. 14 Temporal evolution of (a) mechanical efficiency and (b) brake mean effective pressure (BMEP), illustrating convergence towards the stabilized engine operating regime

Overall, the numerical results establish a consistent relationship between the in-cylinder thermodynamic processes and the resulting engine performance. The predicted combustion and heat-transfer behaviour is reflected in the stabilized brake response, characterized by an effective torque of 290.15 Nm, an effective power of 167.11 kW, a BMEP of 19.62 bar, and a BSFC of 247.30 g/kWh. The convergence of the principal performance indicators further supports the numerical consistency of the developed model and provides the basis for the subsequent assessment of the investigated engine configuration.

4. CONCLUSIONS

The developed thermodynamic model provides a coherent representation of the investigated engine configuration by integrating gas exchange, compression, combustion, heat transfer, and the conversion of indicated work into effective engine performance. The numerical results demonstrate the functional compatibility between the proposed cylinder architecture, incorporating scavenging ports and a lateral exhaust valve, and the dual-function crankshaft, which additionally provides direct actuation of the exhaust valve.

At the investigated operating point, the engine configuration delivers an effective torque of 290.15 Nm, an effective power of 167.11 kW, and a BMEP of 19.62 bar, with a corresponding BSFC of 247.30 g/kWh. The correlation of these performance parameters with the predicted pressure, temperature, pressure-rise rate, and wall heat-transfer characteristics demonstrates the thermodynamic consistency of the numerical response and indicates the capability of the proposed architecture to sustain operation at a high specific power density.

The principal contribution of this study lies in the integrated one-dimensional numerical assessment of a novel engine architecture based on two distinctive design elements: a cylinder incorporating scavenging ports and a lateral exhaust valve, and a crankshaft with the additional function of directly actuating the exhaust valve. The results establish a numerical basis for assessing the functional and thermodynamic behaviour of the proposed concept, while not substituting for experimental validation. Future work should therefore focus on experimental validation of the numerical model and on evaluating the proposed architecture over an extended range of engine speeds and loads.

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