



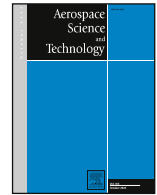
Performance and NO_x emission characteristics of intercooled composite cycle aero engines

Downloaded from: <https://research.chalmers.se>, 2026-10-26 08:30 UTC

Citation for the original published paper (version of record):

Johansson, A., Lundbladh, A., Grönstedt, T. et al (2027). Performance and NO_x emission characteristics of intercooled composite cycle aero engines. *Aerospace Science and Technology*, 180. <http://dx.doi.org/10.1016/j.ast.2026.113892>

N.B. When citing this work, cite the original published paper.



Original article

Performance and NO_x emission characteristics of intercooled composite cycle aero enginesAdam Johansson^{a,*}, Anders Lundblad^{a,b}, Tomas Grönstedt^a, Carlos Xisto^a^a Department of Mechanical Engineering, Chalmers University of Technology, Hörsalsvägen 7A, Göteborg, 41258, Sweden^b GKN Aerospace Sweden, Flygmotorvägen 1, Trollhättan, 46181, Sweden

ARTICLE INFO

Communicated by Dr Mehdi Ghoreysi

Keywords:

Composite cycle engine

NO_x emissions

Thermal efficiency

Power density

Parametric study

Multi-objective optimisation

ABSTRACT

The intercooled composite cycle engine (CCE) is a propulsion system concept with potential for fuel burn reduction exceeding 10% compared to a future geared turbofan. However, nitrogen oxides (NO_x) emissions are expected to exceed those of conventional combustion systems, and integrating a piston engine into the core introduces significant weight penalties. There is a lack of understanding of the underlying cycle design parameters that drive emissions and decrease core compactness, and how these can be traded off against efficiency for a more balanced design. In the present paper, we address this gap by systematically investigating the effect of key CCE cycle design parameters on thermal efficiency, power density, and NO_x emissions. The analysis demonstrates that most of the design changes that increase thermal efficiency will also lead to an increase in NO_x emissions, with clear trade-offs between specific power and efficiency. Multi-objective optimisations are conducted, generating Pareto fronts showing the efficiency-NO_x trade-off. A 13.4% improvement in thermal efficiency relative to a future geared turbofan is possible. This optimum is achieved at the maximum admissible cylinder pressure and by only adding heat in the piston engine. Higher constraints on specific power are found to shift the Pareto front towards higher NO_x and lower thermal efficiency. However, without the introduction of NO_x mitigation techniques, the NO_x emissions are estimated to be 5.3 times higher for the highest thermal efficiency design, highlighting the need for dedicated NO_x reduction strategies.

1. Introduction

Aviation contributes to approximately 3.5% of the net anthropogenic climate change, with non-CO₂ effects contributing to about two-thirds of the net radiative forcing [1]. Excluding contrails, the primary non-CO₂ warming effect is caused by nitrogen oxides (NO_x) emissions. There is both a warming and cooling effect caused by the emissions of NO_x, with a net-positive contribution to global warming. Several reduction targets have been set to curb the impact of aviation on the climate. An example is the targets set for the year 2050 by the European Strategic Research and Innovation Agenda. Reductions in 75% of CO₂ and 90% of NO_x emissions per passenger kilometre are expected relative to a year 2000 aircraft [2]. Exceptional improvements are needed in the propulsion system to meet such ambitious goals. In addition to increasing propulsive efficiency, the main driver for an efficient aero engine is the thermal efficiency of the engine core. However, for the current Brayton cycle, the projected improvements in thermal efficiency are expected to level off due to technological and material limitations [3]. To further increase the core efficiency, a new type of thermodynamic cycle is needed [4,5].

As a result of the EU project ULTIMATE [6], in which novel core engine architectures were investigated, the most promising way to increase the thermal efficiency of the thermodynamic cycle was to target the irreversibility of heat addition. This can be realised by replacing the isobaric heat addition of the Brayton cycle with a pressure-gain combustion system [7]. One concept for pressure-gain combustion, detonation engines [8–13], has been found to have the potential to generate thermal efficiency improvements of up to 8.5% compared to a year 2050 geared turbofan, but produces prohibitive amounts of NO_x with emission index between 100 g/kg and 200 g/kg. It also faces significant technological challenges before becoming practically viable [14,15]. The concept that was singled out to be closest to practical realisation was the composite cycle engine (CCE), operating a piston topping cycle [7].

The internal combustion engine was the primary power unit in aircraft propulsion up to the dawn of the jet age. Although gas turbines subsequently became the dominant technology, the pursuit of higher thermal efficiency and lower environmental impact has renewed interest in advanced engine concepts and combustion strategies [16,17]. In this context, the concept of combining a piston engine with the Brayton

* Corresponding author.

E-mail addresses: adam.l.johansson@chalmers.se (A. Johansson), anders.lundblad@gknaerospace.com (A. Lundblad), tomas.gronstedt@chalmers.se (T. Grönstedt), carlos.xisto@chalmers.se (C. Xisto).<https://doi.org/10.1016/j.ast.2026.113892>

Received 22 May 2026; Received in revised form 11 September 2026; Accepted 23 September 2026

Available online 29 September 2026

1270-9638/© 2026 The Author(s).

Published by Elsevier Masson SAS. This is an open access article under the CC BY license (<http://creativecommons.org/licenses/by/4.0/>).This is an open access article under the CC BY license (<http://creativecommons.org/licenses/by/4.0/>).

Nomenclature*Latin letters*

$(A/F)_{st}$	Stoichiometric air-fuel mass ratio
$\bar{v}_p$	Mean piston speed
$\dot{m}$	Rate of mass flow
$\dot{Q}$	Rate of heat transfer
$\dot{W}$	Power
$\dot{W}_{core,V_d}$	Core power per displacement volume
$\dot{W}_{ind}$	Indicated piston power
A	Area
A_{flow}	Valve flow area
c_{cool}	Turbine cooling technology parameter
c_D	Discharge coefficient
C_{fl}	Reaction zone flame temperature calibration factor
$C_{h,1}$	Heat transfer relation coefficient 1
$C_{h,2}$	Heat transfer relation coefficient 2
C_{min}	Minimum rate of heat capacity
c_p	Mass specific heat capacity at constant pressure
C_r	Reaction zone temperature coefficient
d	Cylinder bore
d_v	Valve diameter
f	Fuel-air ratio
f_s	Stoichiometric fuel-air ratio
G_r	Reaction zone shape function
h	Mass specific enthalpy
i	Cycles per revolution
k	Reaction rate coefficient
K_c	Equilibrium constant
L	HPC last stage blade height
l_{con}	Connecting rod length
l_v	Valve lift
m	Mass
m_w	Wiebe shape parameter
n	Revolutions per second
n_{var}	Number of optimisation variables
n_v	Number of valves
p	Pressure
p_{max}	Peak cylinder pressure
Q	Heat
$Q_{f,tot}$	Total added heat during one cycle
Q_{net}	Lower heating value
R	Specific gas constant
s	Stroke
T	Temperature
T_M	Metal temperature
T_{fl}	Flame temperature
U	Internal energy
V	Volume
v	Velocity
V_d	Displacement Volume
W	Work
w	Mass specific work

Greek letters

α	Heat transfer coefficient
χ_{PM}	Piston module NO _x fraction
$\Delta\theta_{CD,dp}$	Design point combustion duration
$\Delta\theta_{CD}$	Combustion duration
ϵ	Geometric compression ratio
$\eta_{0,HPC}$	Base level HPC polytropic efficiency
η_{cool}	Turbine cooling efficiency
η_o	Overall efficiency
η_p	Propulsive efficiency
η_{th}	Thermal efficiency

Λ	LPC-HPC compression split exponent
λ	Air-fuel-equivalence ratio
λ_0	Zone 2 air-fuel-equivalence ratio
λ_{dp}	Design point air-fuel-equivalence ratio
ϕ	Fuel-air-equivalence ratio
Π	Pressure ratio
τ_v	Relative valve opening angle
θ	Crank angle
ϵ	Heat exchanger effectiveness

Subscripts

c	Cold
$cool$	Turbine cooling flow
$core$	Engine core
d	Displacement
e	Exhaust
EVC	Exhaust valve closing
EVO	Exhaust valve opening
f	Fuel
gas	Hot gas
h	Hot
HEX	Heat exchanger
i	Intake
IC	Intercooler
id	Ideal
IVC	Intake valve closing
IVO	Intake valve opening
j	Station index
m	Motoring
max	Maximum
mb	Mean brake
mf	Mean friction
p	Piston
r	Reverse
SC	Start of combustion
VC	Valve closing
VO	Valve opening
$wall, k$	Wall surface k

Acronyms

ABDC	After Bottom Dead Centre
ATDC	After Top Dead Centre
BBDC	Before Bottom Dead Centre
BPR	Bypass Ratio
BTDC	Before Top Dead Centre
CCE	Composite Cycle Engine
EI_{NO_x}	Emission Index of NO _x
FPR	Fan Pressure Ratio
HEX	Heat Exchanger
HPC	High-Pressure Compressor
HPT	High-Pressure Turbine
IC	Intercooler
IMEP	Indicated Mean Effective Pressure
ISA	International Standard Atmosphere
LPC	Low-Pressure Compressor
LPT	Low-Pressure Turbine
NO_x	Nitrogen Oxides
OPR	Overall Pressure Ratio
PM	Piston Module
SAF	Sustainable Aviation Fuel
TDC	Top Dead Centre
ToC	Top-of-Climb
TRL	Technology Readiness Level
TSFC	Thrust Specific Fuel Consumption

cycle is not new [18], but in the last decade this concept has regained attention due to the increased interest in highly efficient aero engines. The 2013 patent for a free-double-cylinder design by MTU Aero Engines [19] sparked the latest wave of attention. Kaiser et al. presented a first study on CCE performance in 2016 [3], predicting an 18.3% improvement in efficiency compared to a year 2035 turbofan. Since then, a multitude of studies have confirmed the substantial reduction in fuel burn that is possible with CCE [20–24]. However, a more detailed study that included time resolved piston engine performance, and NO_x modelling, predicted that NO_x emissions will be eight times higher at cruise relative to a state-of-the-art turbofan, with an emission index of 80.5 g/kg [25]. The reason is the very high peak pressures and temperatures, combined with local nearly stoichiometric combustion. This would likely negate the climate benefit of the CCE. However, there are NO_x mitigation techniques that may reduce NO_x emissions to levels of a comparable turbofan. Before implementing them, learning how the cycle design variables affect NO_x emissions is crucial.

To the best of the authors' knowledge, no studies have addressed the trade-off between NO_x emissions, efficiency and specific power of the CCE. The purpose of this study is to fill this gap by presenting a systematic investigation of the effect and trends of key design parameters on performance and NO_x emissions. Since the lower power density of piston engines is expected to increase the engine weight and offset some of the expected thermodynamic benefits, the core power per displacement volume is also selected as one of the investigated metrics. A parametric study on the effect of the overall pressure ratio, turbine inlet temperature, turbomachinery compression split, geometric compression ratio, fuel-air ratio, pressure ratio over the piston module, amount of intercooling and combustion phasing inside the piston engine is established on the thermal efficiency, core power per displacement volume and NO_x emissions. Pareto fronts are established for designs with and without a secondary burner to determine optimal designs with respect to thermal efficiency and NO_x . The findings suggest that NO_x and thermal efficiency are positively correlated for most of the design variables investigated. Hence, to reduce NO_x emissions without compromising thermal efficiency, new low emission combustion technology needs to be developed.

The paper is structured in the following way. First, in Section 2, the CCE concept, working principle, and theoretical benefits are described in detail. Thereafter, in Section 3 the methods and assumptions used for the conceptual and system level evaluation are presented in detail. In Section 4, a parametric study of the effect of key engine design variables on performance and emissions is presented and discussed. The section is continued with a multi-objective optimisation study in which sets of optimal, non-dominated, designs are found with respect to performance and emissions, for architectures with and without secondary combustion, for different limitations on displacement volume specific core power. Finally, in Section 5 the conclusions and limitations are presented, and the future research outlook is proposed.

2. The composite cycle engine

In this section, a detailed description of the CCE is presented. First, we outline the theoretical motivation behind the concept. Thereafter, the studied CCE architecture and working principle are introduced and discussed.

2.1. Theoretical motivation

The rationale behind the CCE concept is that adding heat to the thermodynamic cycle is performed more efficiently in a piston engine than in a conventional constant pressure combustor. The reason is twofold. First, the combustion process inside a cylinder more closely approximates constant volume heat addition than the constant pressure process

of a conventional burner, which from a second law perspective is inherently more efficient. Secondly, higher cycle peak pressures and temperatures can be achieved as a result of the cyclic operation of the piston engine. The peak temperature in a conventional turbofan, operating with steady-state turbomachinery, is limited by the material and cooling capabilities of the high-pressure turbine (HPT) stator and rotor, which are constantly exposed to a steady flow of high temperature gas exiting the burner. However, inside the piston engine cylinder, the heat addition is cyclic, meaning that the cylinder walls are cooled with fresh charge between each combustion phase. This allows for higher cycle peak temperatures without damaging the cylinder and piston head.

Although piston engines can theoretically achieve higher thermal efficiencies than gas turbines, they are not used in civil aviation due to their inferior power-to-weight ratios. For the same power plant mass, gas turbines achieve substantially higher mass flows, allowing more fuel to be combusted and, thus, greater power output. The CCE combines the power density of the gas turbine with the thermal efficiency of the piston engine. Turbomachinery is used to handle large volume flows in the low-pressure part of the power plant, whereas the piston-topping cycle is located in the high-pressure part of the system to achieve peak pressures and temperatures unachievable by turbomachinery alone. Here, the gas density is high and the volume flows are small, allowing for a more compact piston engine at a given mass flow, reducing the engine weight and volume.

2.2. The intercooled crankshaft based CCE

The CCE concept investigated in this paper is illustrated in Fig. 1, and in Table 1 the station numbering is defined. It consists of an intercooled geared low pressure system with an axial-radial high-pressure compressor (HPC) unit driven by a crankshaft-based piston module (PM). The PM is comprised of a dual V12 engine, featuring 24 cylinders, which drives the axial-radial compressor via a reduction gearbox. Hence, the PM not only combusts fuel and increases the pressure of the working gas further, but also drives the HPC, thereby avoiding the need for an HPT. The piston-HPC system produces gas power for the low-pressure turbine, which in turn drives the low-pressure compressor (LPC) and the fan via a reduction gearbox.

An intercooler is placed between the LPC and the HPC, and uses a portion of the fan discharge air to cool the intermediate compressor air flow. This is beneficial to the cycle by increasing the density of the core air, reducing the compression work in the HPC, and at the same time reducing the size and weight of the piston engine. Another benefit is the reduction of NO_x production due to the reduced inlet combustion temperature and associated lower flame temperature. The intercooler

Table 1
Station numbering of the intercooled CCE.

Station	Number
Free stream	0
Inlet entry	1
Fan inlet	2
LPC inlet	21
Intercooler inlet	22
HPC inlet	25
HPC outlet	3
PM inlet	31
Aux compressor outlet	33
PM outlet	34
Burner inlet	35
Turbine inlet	4
Turbine outlet	5
Core nozzle outlet	8
Bypass stream inlet	13
Bypass nozzle outlet	18
Intercooler nozzle outlet	28
Oil cooler nozzle outlet	38

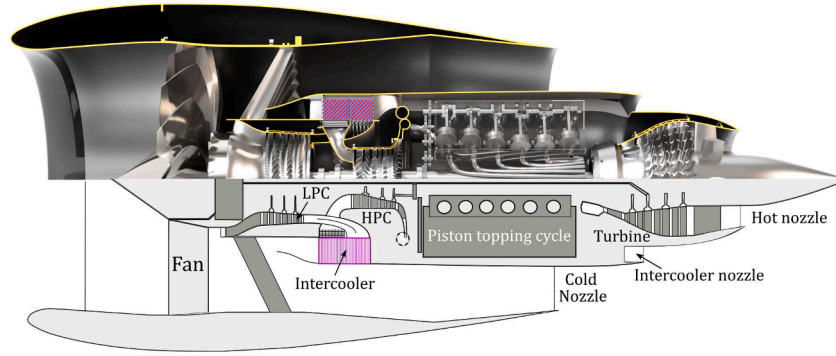


Fig. 1. The composite cycle engine (CCE) concept, with a rendering of the power plant in the upper half and a conceptual design drawing in the lower half. LPC - Low-pressure compressor; HPC - High-pressure compressor.

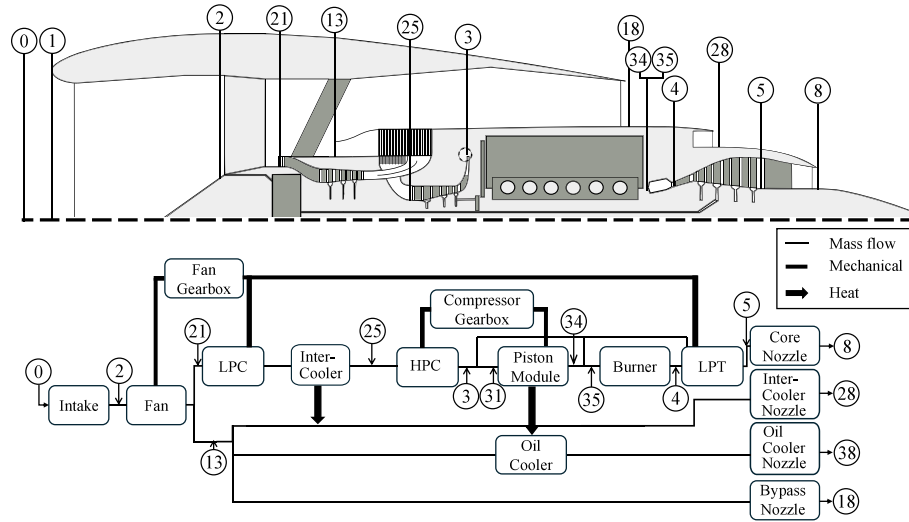


Fig. 2. The CCE with station numbering showing a conceptual design drawing in the upper half and a schematic defining the modules and their interconnections in the lower half. Neither the oil cooler nor its nozzle are shown in the upper figure.

cooling air flows in an auxiliary bypass duct and is ejected in a separate nozzle contributing to the generation of thrust. An oil-to-air heat exchanger is placed in an additional auxiliary duct. This heat exchanger is used as a heat sink for the PM lubrication/coolant oil. Therefore, thrust is generated by the core nozzle, the intercooler nozzle, the oil cooler nozzle (not shown in Fig. 1), and the bypass nozzle. Buffer volumes are placed between the steady-state turbomachinery and the piston inlet and outlet valves, to suppress pressure fluctuations that arise from the intermittent operation of the piston engine. A secondary combustor is placed after the piston engine, allowing for variability and increased power density at high power setting conditions, such as take-off.

3. Methodology

To assess the thermodynamic performance and the NO_x emissions of the propulsion system, a thermodynamic system level model has been developed. A modular approach was chosen, where the different modules are connected to ensure the conservation of mass and energy, as illustrated in Fig. 2. This section describes the methodology used and assumptions made to simulate the components of the propulsion system. In addition, a model for the prediction of NO_x emissions arising from the piston module is presented, calibrated, and validated using experimental data. Finally, the whole engine system coupling and modelling approach is explained, with emphasis on the coupling between the reciprocating engine and the steady-state turbomachinery.

3.1. Gas properties

The working gas is modelled as an ideal gas mixture, with the composition varying between dry air and ideal combustion products, governed by the fuel-air equivalence ratio, $\phi = \frac{f}{f_s}$, where f and f_s are the fuel-air ratio and its stoichiometric counterpart, respectively. Dry air is assumed to have the following molar composition: 78.1% N_2 , 21.0% O_2 , 0.9% Ar. The fuel is modelled as $\text{C}_{12}\text{H}_{23}$, commonly used to approximate Jet-A, with a stoichiometric fuel-air ratio of $f_s = 0.0681$ and a lower heating value of $Q_{\text{net}} = 43.0 \text{ MJ/kg}$, evaluated using the NASA Glenn polynomials [26]. To represent sustainable aviation fuel (SAF), assumed to have a slightly higher lower heating value of 44.0 MJ/kg , the reported fuel flow is corrected by a factor of $43/44$. The individual species' heat capacity, enthalpy, and entropy are temperature dependent and represented by the NASA Glenn polynomials [26]. Hence, the properties of the working gas are dependent on temperature T and equivalence ratio ϕ . Throughout this work, the temperatures and pressures presented are total quantities.

By treating the working fluid as a semi-perfect gas, the gas pressure inside the cylinder is underpredicted. This is only relevant for the high-pressure phase inside the PM, as the pressure inside the steady-state turbomachinery is below 50 bar, where the real gas pressure is less than 1% higher than the ideal gas pressure for the relevant temperatures [27]. For the piston module, the higher pressure of the real gas would lead to a larger work output. A previous study on a similar CCE operating

at 250 bar peak pressure found that net fuel flow was reduced by 0.2% when accounting for real gas compressibility [25]. The present study focuses on the design conditions of the CCE, which limits the peak pressure at top-of-climb (ToC) to 150 bar. Therefore, for all evaluated points in this study the pressures will be below 150 bar and given the minor impact on thermal efficiency, we chose not to account for real gas effects. Still, this approach is on the conservative side with respect to thermal efficiency.

3.2. Steady-state turbomachinery

The axial and axial-radial compressors are modelled by specifying their individual pressure ratios and polytropic efficiencies. For a given mass flow rate, the compression power is found from the required increase in enthalpy. In the burner, the required injected fuel is calculated to reach a specified exit temperature. Additionally, since the burner is located downstream of the piston engine, the model accounts for vitiated air at its inlet. The intake and bypass duct are modelled adiabatically with specified total pressure losses. The thrust of the engine is computed by expanding the air or gas in separate, unmixed, convergent nozzles to ambient conditions, or to the throat conditions if choked. The nozzles are modelled by specifying the respective discharge and thrust coefficients.

The cooling of the low pressure turbine is modelled using an equivalent single stage cooling model, where the cooling air is taken from the HPC outlet. The nozzle guide vane cooling air is perfectly mixed and fully contributes to the turbine power, whereas the rotor cooling air does not contribute to the work performed on the rotor. The mass flow of cooling air, $\dot{m}_{cool}$, is found from the hot gas mass flow, $\dot{m}_{gas}$, and the cooling efficiency, η_{cool} , with the following expression:

$$\frac{\dot{m}_{cool}}{\dot{m}_{gas}} = c_{cool} \frac{\eta_{cool}}{1 - \eta_{cool}} \quad (1)$$

where c_{cool} is a technology parameter. The cooling efficiency is found from:

$$\eta_{cool} = \frac{T_{gas} - T_M}{T_{gas} - T_{cool}} \quad (2)$$

with T_{gas} being the hot gas temperature, which is T_4 for the stator and T_{41} for the rotor. Further, T_{cool} is the cooling air temperature, and T_M the specified metal temperature limit, assumed to be 1200 K for both stator and rotor [28]. By calibrating the single-stage model to the reference engine (see Table 3 for details), a value of $c_{cool} = 0.0875$ is found. The expansion across the rotor is modelled with a specified polytropic efficiency.

Scale corrections of the HPC efficiency are added to account for tip-clearance effects [13,29]. The correction uses the final stage blade height of the HPC and a base level efficiency, $\eta_{0,HPC}$, to obtain the correct polytropic efficiency:

$$\eta_{HPC} = \eta_{0,HPC} + \left(0.0532 - \frac{0.5547}{L} - \frac{1.7724}{L^2} \right) \quad (3)$$

where L is the last stage blade height in mm. Blade heights less than 13 mm penalise efficiency, and larger blade heights lead to small efficiency improvements. The blade height is calculated by assuming a hub/tip ratio of 0.925 and exit axial Mach number of 0.254. No correction is applied to the LPC, as the efficiency variation is expected to be less prominent with scaling [29].

3.3. Piston module performance

Thermodynamic performance of the PM is modelled with a spatially zero-dimensional, time resolved model, according to the approach outlined in [30]. The control volume of the system is taken to be the combustion chamber of the piston engine, which can be visualised in Fig. 3. The working medium is assumed to behave as a semi-perfect fluid governed by the ideal gas equation, which is differentiated with respect to

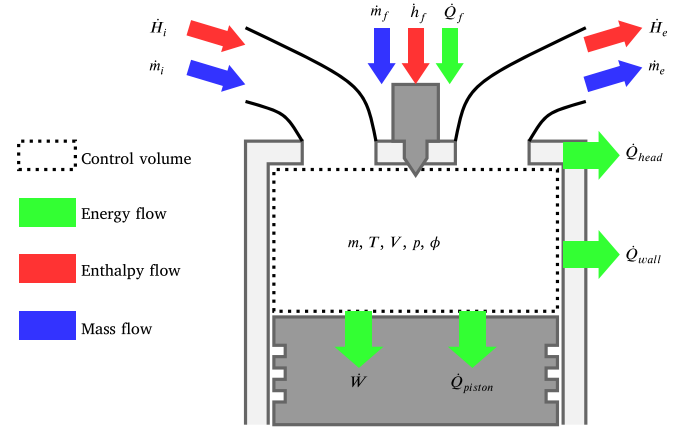


Fig. 3. Piston module performance model schematic showing the combustion chamber control volume with state variables mass m , temperature T , volume V , pressure p and equivalence ratio ϕ . Energy transfer through the system boundaries is shown with green arrows. Mass transfer with accompanying enthalpy transfer through fuel injector, inlet and exhaust valves is shown with blue and red arrows. (For interpretation of the references to colour in this figure legend, the reader is referred to the web version of this article.)

time to obtain the temporal evolution of the system variables:

$$p\dot{V} + V\dot{p} = mR\dot{T} + mT\dot{R} + RT\dot{m}, \quad (4)$$

where V is the volume of the combustion chamber and m represents the mass of gas within the system. Further, the conservation of energy is written as follows:

$$\dot{U} = \dot{Q} - \dot{W} + h_i\dot{m}_i - h_e\dot{m}_e + h_f\dot{m}_f, \quad (5)$$

where U represents the internal energy of the system, Q the heat supplied to the system, W the work performed by the system, h the specific enthalpy. The subscripts i , e and f denote the flows through the intake, exhaust, and fuel injector.

The change in system volume is driven by a prescribed reciprocating motion of the piston, which can be expressed as a function of the rotation angle of the crankshaft, θ . One revolution of the shaft results in one full piston volume sweep. The total cylinder volume variation with θ can therefore be expressed as [31]:

$$V(\theta) = V_d \left[\frac{1}{\epsilon - 1} + \frac{1 - \cos \theta}{2} + \frac{s}{16 \cdot l_{con}} (1 - \cos 2\theta) \right], \quad (6)$$

where ϵ is the geometric compression ratio, s is the stroke and l_{con} the length of the connecting rod. In this work a connecting rod ratio of $s/l_{con} = 0.56$ is used, which is a typical value for diesel engines [32].

The mass flow over the intake and exhaust valves is modelled as an isentropic gas expansion between two reservoirs with a prescribed pressure differential and flow area. The flow area is given by:

$$A_{flow} = n_v \cdot d_v \cdot c_D \cdot \pi \cdot l_v \quad (7)$$

where n_v is the number of valves. Two intake and two exhaust valves are used in this work, typical of high-performance engines. In addition, the valve diameter is assumed to be:

$$d_v = 0.75 \cdot (\sqrt{2} - 1) \cdot d \quad (8)$$

which is a 75% filling ratio of the piston head. Valve losses are prescribed by specifying a corresponding discharge coefficient, c_D , where 0.8 is used in this work in agreement with [30]. The variation of A_{flow} comes from the valve lift l_v , which is determined by the valve timings and the lift profile. The opening and closing of the valve takes 25% of

the total valve open duration:

$$l_v(\theta) = l_{v,\max} \cdot \begin{cases} \frac{1}{2} \left(1 - \cos \left(\frac{\pi \cdot \tau_v(\theta)}{0.25} \right) \right), & \tau_v(\theta) < 1/4 \\ 1, & 1/4 < \tau_v(\theta) < 3/4 \\ \frac{1}{2} \left(1 - \cos \left(\frac{\pi \cdot (1 - \tau_v(\theta))}{0.25} \right) \right), & 3/4 < \tau_v(\theta) \end{cases} \quad (9)$$

where

$$\tau_v(\theta) = \frac{\theta - \theta_{VO}}{\theta_{VC} - \theta_{VO}} \quad (10)$$

is the relative opening angle. In Eq. (10), θ_{VO} is the valve opening angle, and θ_{VC} is the valve closing angle. The maximum valve height is assumed to be 10% of the bore, $l_{v,\max} = 0.1 \cdot d$. In this work, the following valve timings have been used: $\theta_{IVO} = 1^\circ \text{BTDC}$, $\theta_{IVC} = 13.1^\circ \text{ABDC}$, $\theta_{EVO} = 27.5^\circ \text{BBDC}$ and $\theta_{EVC} = 10^\circ \text{ATDC}$ where subscripts *E* and *I* denote exhaust and intake. A and B stand for after and before, respectively, with BDC and TDC referring to bottom- and top-dead-centre. The timings were found by optimisation of the thermal efficiency of the piston engine for representative conditions.

The heat release from combustion is modelled to coincide with the fuel mass injection, simulating direct injection compression ignition, where the ignition delay is not modelled. The rate of heat release is prescribed by a single Wiebe function:

$$\frac{dQ_f}{d\theta} = \frac{Q_{f,\text{tot}}}{\Delta\theta_{CD}} \cdot a \cdot (m_w + 1) \cdot \tau_w^{m_w} \cdot \exp(-a \cdot \tau_w^{m_w+1}) \quad (11)$$

where

$$\tau_w = \frac{\theta - \theta_{SC}}{\Delta\theta_{CD}} \quad (12)$$

A combustion efficiency of 99.9% is prescribed, which results in a coefficient $a = 6.908$ [33]. The magnitude of the heat-release curve is given by the total added heat, $Q_{f,\text{tot}}$, obtained from a specified fuel-air ratio. In addition, the shape of the curve is defined by the shape factor, m_w , the duration of combustion, $\Delta\theta_{CD}$, and the start of combustion, θ_{SC} .

The gas composition inside the cylinder is governed by the fuel-air equivalence ratio ϕ , which varies both by mass transfer through the intake and exhaust valves and by fuel injection. Mathematically, the change of equivalence ratio with respect to crank-angle is written as:

$$\frac{d\phi}{d\theta} = \frac{1 + \phi \cdot f_s}{m} \left(\frac{\phi_i - \phi}{1 + \phi_i \cdot f_s} \dot{m}_i - \frac{\phi_e - \phi}{1 + \phi_e \cdot f_s} \dot{m}_e + \frac{\dot{m}_f}{f_s} \right), \quad (13)$$

where ϕ_i and ϕ_e are the equivalence ratios of the intake and exhaust manifolds, respectively. Equivalent calculations are performed in the intake and exhaust manifolds, to track the fuel-air ratio and correctly capture backflow across the valves.

Finally, the heat transfer rate between the gas and the walls, $\dot{Q}_{wall}$, is modelled using Newton's law of cooling, summed over each wall surface:

$$\dot{Q}_{wall} = \alpha \sum_k A_{wall,k} (T - T_{wall,k}), \quad (14)$$

where $A_{wall,k}$ and $T_{wall,k}$ denote the area and temperature of wall *k*, respectively. For the present concept *k* represents the piston crown, the cylinder wall and the cylinder head wall, see Fig. 3. The heat transfer coefficient is computed using a semi-empirical model proposed by Hohenberg [34]:

$$\alpha = C_{h,1} \cdot V^{-0.06} \cdot p^{0.8} \cdot T^{-0.4} \cdot (\bar{v}_p + C_{h,2})^{0.8}, \quad (15)$$

with coefficients $C_{h,1} = 130$, $C_{h,2} = 1.4 \text{ m/s}$ and $\bar{v}_p$ is the mean piston speed, defined as the average piston speed during one cycle. In this study, the temperature of the cylinder wall is set to 500 K, limited by the breakdown of the lubricating oil. The unlubricated surfaces, the cylinder head and piston top, are set to 600 K around the operating limit for aluminium alloys and in accordance with previous studies [25,35].

3.4. Averaged properties

The integration of the time-resolving piston model into a conventional turbomachinery steady-state performance model requires averaging. Since the temperature of the gas leaving the exhaust valve is varying, the mass average outlet enthalpy is calculated as:

$$h(T_{34}, \phi_{34}) - \frac{\int h_e dm_e}{\int dm_e} = 0, \quad (16)$$

where ϕ_{34} is the mass average equivalence ratio in the exhaust. The average exhaust outlet temperature is iterated until Eq. (16) is satisfied. In addition, the indicated piston power, $\dot{W}_{ind}$, is the total pressure work over the cycle divided by the cycle period, Δt_{cycle} :

$$\dot{W}_{ind} = \frac{\int p dV}{\Delta t_{cycle}} \quad (17)$$

Finally, the heat transfer through the walls is also time averaged:

$$\dot{Q}_{w,avg} = \frac{\int \dot{Q}_w dt}{\Delta t_{cycle}} \quad (18)$$

The resulting system of ordinary differential equations is numerically integrated in the crank-angle domain using the LSODA method [36,37], implemented in SciPy [38], with both relative and absolute tolerances set to 10^{-8} . Since LSODA is an adaptive-step solver, the crank-angle resolution varies automatically throughout the cycle to satisfy these tolerances, rather than using a fixed step size. The simulation is advanced over successive engine cycles until a periodic solution is reached, defined as the relative variation over three consecutive cycles of the in-cylinder properties and key performance integrals ($\dot{W}_{ind}$, T_{34} , $\dot{Q}_{w,avg}$, and mass flow) falling below 10^{-5} .

For a more detailed description and cross-validation of the model, see [39].

3.5. Mechanical losses

The calculated indicated piston power, $\dot{W}_{ind}$, does not account for any mechanical losses. Semi-empirical relations, proposed by Patton et al. [40], are used to model friction. First, it is convenient to normalise the indicated power with the cylinder displacement volume, obtaining the indicated mean effective pressure:

$$\text{IMEP} = \frac{\dot{W}_{ind}}{i \cdot n \cdot V_d} \quad (19)$$

where *i* is the number of cycles per revolution (0.5 for four-stroke operation and 1 for two-stroke), and *n* is revolutions per second. All friction values are expressed as the mean effective friction pressure, p_{mf} . The actual power available at the shaft is then given by the mean effective brake pressure:

$$p_{mb} = \text{IMEP} - p_{mf}, \quad (20)$$

which can be converted into power similarly to Eq. (19).

The total mean effective friction pressure is composed of the different friction contributions of all piston engine components. Correlations are given for crankshaft rotation, piston movement, valvetrain motion, and auxiliary losses. Each friction component accounts for the different types of friction, such as sealing friction, hydrodynamic friction, turbulent dissipation in the lubricating oil, piston ring friction, etc. For the full equations, see the Appendix A.

3.6. Heat exchangers

The intercooler and oil cooler heat-exchanger (HEX) performance is computed for a specified effectiveness, defined as:

$$\varepsilon = \frac{\dot{Q}_{HEX}}{\dot{Q}_{HEX,max}} \quad (21)$$

where $\dot{Q}_{HEX}$ is the HEX heat-transfer rate and $\dot{Q}_{HEX,max}$ is the theoretical maximum attainable heat-transfer rate for a given temperature difference between the hot and cold streams [41]. The heat transfer rate can be calculated as follows:

$$\dot{Q}_{HEX} = \varepsilon C_{min}(T_{h,i} - T_{c,i}), \quad (22)$$

where $C_{min} = \min(\dot{m}_c c_{p,c}, \dot{m}_h c_{p,h})$ is the minimum heat capacity rate of the hot and cold sides with input temperatures $T_{h,i}$ and $T_{c,i}$. For the engine oil cooler, a heat capacity ratio of one ($\dot{m}_c c_{p,c} = \dot{m}_h c_{p,h}$) is assumed. In the intercooler, the ratio between the cooling air and core air mass flows is a design variable. All of the core air is always passing the intercooler, whereas a variable amount of cooling air is used. The intercooling ratio can be understood as how big the intercooler is, with a larger ratio meaning more cooling air, greater temperature drop of the core air and larger pressure losses. The effectiveness is fixed as a design parameter for both heat-exchangers and set to $\varepsilon = 0.7$ for the air-to-air intercooler, the optimum value found in a previous CCE study [20]. For the oil cooler, a higher effectiveness is expected for a compact air-to-liquid heat-exchanger, hence $\varepsilon = 0.9$ is used. The pressure losses are modelled by prescribing a loss relative to the inlet total pressure. For the intercooler it varies linearly with the amount of intercooling between 0 and 7% on both sides of the intercooler. This value is deemed adequate for the type of air-to-air heat-exchanger that is assumed for the present work, and is based on earlier work on cross-flow intercoolers for aero engines [42] where pressure losses between 5% and 7.5% were reported. In the oil cooler, 5.7% pressure loss is assumed on the air side, according to previous work [25].

3.7. Mixing

Mixing of two separate gas flows occurs after the PM, before the secondary combustor, since air is allowed to bypass the PM if the mass flow through the cylinders does not match the core flow. The PM bypass stream, m_{33} , is compressed to the same pressure as the PM exhaust by the auxiliary compressor, which is powered by the PM. The losses that arise from the mixing of the two flows are included in the auxiliary compressor efficiency. Conservation of energy gives the temperature after mixing:

$$h_{35}(T_{35}, \phi_{35}) = \frac{m_{33}h_{33} + m_{34}h_{34}}{m_{35}}, \quad (23)$$

where T_{33} is the PM bypass stream temperature and T_{35} is found by iteration. ϕ_{35} is the fuel-air equivalence ratio based on the mixed mass stream m_{35} and is obtained from conservation of mass of the combustion end products. When the turbine is cooled, m_{35} is not equal to the entire core flow, since the cooling air is removed.

3.8. Nitrogen oxides emissions

Nitrogen oxides are produced both in the piston engine and in the secondary combustor. In the piston engine, NO_x production is modelled using chemical kinetics inside a reaction zone obtained from a semi-empirical two-zone model. For the conventional constant pressure combustor an empirical correlation that represents a future lean direct injection system [43] is used.

3.8.1. Two-zone model

Since NO_x production is sensitive to local temperature and oxygen content, the spatial variation of these properties must be captured to accurately predict NO_x emissions from the PM. Therefore, a two-zone model, developed by Heider et al. [44], has been implemented. The model captures the temperature and chemical composition behind the flame front, which differs from the spatially averaged values of the entire cylinder volume. Taking the pressure, temperature, and fuel-air ratio of the single-zone calculation as input, the model is implemented as a post-processing method. An illustrative representation of the model is

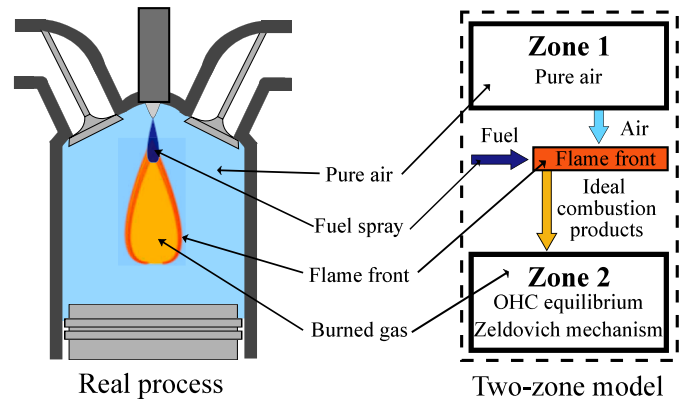


Fig. 4. The two-zone model used to obtain the mass and temperature of the NO_x reaction zone, Zone 2. Direct injection combustion shown to the left and a schematic of the model to the right. Mass transfers from Zone 1 to Zone 2 during combustion according to a specified stoichiometry.

provided in Fig. 4. Before fuel injection and combustion, the entire cylinder volume is occupied by an air-fuel mixture that defines the unburned region (zone 1). When the combustion process is initiated, the reaction zone is initialised and grows as the combustion progresses, which defines the burnt region (zone 2). As combustion progresses, the mass of the unburned region is transferred to the burnt zone. Locally in the flame region, it is assumed that the mixture is characterised by having an air-fuel equivalence ratio λ_0 . Depending on the type of engine, it could be different or similar to the cylinder average counterpart λ . The local λ_0 is used to calculate the mass of the burnt zone:

$$m_2(\theta) = (\lambda_0 \cdot (A/F)_{st} + 1) \cdot \int_{\theta_{sc}}^{\theta} \frac{dm_f(\theta)}{d\theta} d\theta, \quad (24)$$

here $(A/F)_{st}$ is the stoichiometric air-fuel mass ratio, where any vitiation of the gas prior to combustion is considered, which means that a larger mass of air is needed for stoichiometric combustion when the oxygen content is lower than that of pure air. θ_{sc} is the crank angle at which combustion begins, and m_f is the burnt fuel mass. For direct injection engines such as the one modelled here, Heider et al. [44] recommend $\lambda_0 = 1.0$ for diesel engines with a cylinder bore up to 240 mm. Since the bore in this work does not exceed 200 mm, this value is adopted. The difference in temperature between the zones is modelled as:

$$T_2(\theta) - T_1(\theta) = G_r(\theta)C_r, \quad (25)$$

with the coefficient C_r determining the temperature level of the reaction zone at the beginning of combustion,

$$C_r = \frac{1.2 + (\lambda - 1.2)^{0.15}}{2.2\lambda_0} \cdot C_{fl} \cdot T_{fl}, \quad (26)$$

where the flame temperature T_{fl} is calculated with NASA CEA [45] using Jet-A as fuel, and C_{fl} is a calibration factor. The shape of the temperature difference is based on the difference between the cylinder pressure and the motoring pressure p_m :

$$G_r(\theta) = 1 - \frac{\int_{\theta_{sc}}^{\theta} [p - p_m]m_2 d\theta}{\int_{\theta_{sc}}^{\theta_{EVO}} [p - p_m]m_2 d\theta}, \quad (27)$$

where the motoring pressure is the cylinder pressure during unfired operation and is approximated as a polytropic compression. The function $G_r(\theta)$ varies between 1 and 0, such that the temperature difference is the highest at the beginning of combustion, and decreases to zero at the exhaust valve opening θ_{EVO} . Finally, the pressure is uniform throughout the volume of the cylinder, and both zones are subject to the ideal gas law. During cross-validation, a calibration factor of $C_{fl} = 0.84$ was found to agree well with the empirical data and is used throughout this work. A similar value of $C_{fl} = 0.9$ was found to agree well with the experimental data in a previous publication [25], where the difference can

be explained by the variation of λ_0 in that publication. It should be noted that the calibration factor is an engine dependent parameter that might be used to represent a class of existing applications. This means that the NO_x predictions provided in Section 4.3 are not expected to accurately represent, in absolute terms, the operation of a future CCE for which no design exists. However, they can still provide useful insights on emission trends that result from the variation of engine design parameters, as well as reasonable estimates on the expected emission increase relative to state-of-the-art and future conventional turbofan applications.

3.8.2. Chemical kinetics

Nitrogen oxides, NO_x , consist of nitric oxide, NO, and nitrogen dioxide, NO_2 . In piston engines, the production of NO_x is dominated by NO [46], which in compression ignition engines constitutes 70–90% of the total NO_x [47]. Therefore, we will only model the reaction of NO. NO can be formed through different pathways: the prompt NO mechanism, from fuel-bound nitrogen and the thermal NO mechanism. Prompt NO is formed in the flame front itself, via the Fenimore mechanism. In typical engine combustion, only 5–10% [33] of the NO is formed through this mechanism. For combustion of fuels with chemically bound nitrogen, such as coal, this nitrogen is a principal source of NO_x emissions [48]. Diesel and kerosene, however, hardly contain any nitrogen, and NO from this pathway is expected to be even less than the prompt NO. Since both these pathways are of the order of magnitude smaller and fall within the minimum prediction uncertainty of thermal NO predictions, we neglect both fuel and prompt NO in accordance with literature [47].

The dominant pathway for NO creation in piston engines is the thermal pathway [35], which produces NO by oxidation of diatomic nitrogen. It can be formed via two mechanisms: the Zeldovich mechanism and the N_2O -intermediate pathway. The N_2O -pathway is mainly significant for lean fuel-air mixtures in combination with low temperatures that repress the formation of thermal NO through the Zeldovich mechanism [33]. However, it is a three-body reaction, which means it is pressure dependent [47] and has been found to make important contributions to the production of nitrogen oxides for pressures over 60 bar encountered inside diesel engines [49]. In this study, we have neglected the N_2O pathway, which probably leads to underprediction of NO emissions formed during the highest pressures inside the cylinder. We acknowledge this limitation to the study, but since all interesting CCE designs operate with peak pressures well above 60 bar, we believe the error will be systematic and that the trends will still be captured, since the NO emissions will be underpredicted for all designs.

Therefore, we only consider thermal NO formed by the extended Zeldovich mechanism:



Here, k_1 , k_2 and k_3 are the forward reaction rate coefficients, taken from the GRI-Mech 3.0 chemical kinetics mechanism [50]. The production of thermal NO is relatively slow, with the rate being limited by Eq. (28) due to the stable triple bond of N_2 . For the temperatures and pressures encountered inside the combustion chamber, the kinetics of the chemical reaction is slow relative to the physical time scales of the flow field, and hence chemical equilibrium cannot be assumed [51]. Since the forward reactions (29) and (30) are orders of magnitude faster than (28), the N produced in the first reaction is immediately converted in the second and third reaction, and the concentration of atomic nitrogen can therefore be assumed to be quasi-steady:

$$\frac{d[\text{N}]}{dt} \approx 0. \quad (31)$$

Combining reactions (28)–(30) with the quasi-steady approximation, one obtains the following equation for the change in nitric oxide con-

centration:

$$\frac{d[\text{NO}]}{dt} = 2k_1[\text{O}][\text{N}_2] - 2k_{1,r}[\text{NO}][\text{N}]. \quad (32)$$

Here, the concentration of atomic nitrogen is given by:

$$[\text{N}] = \frac{k_1[\text{O}][\text{N}_2] + k_{2,r}[\text{NO}][\text{O}] + k_{3,r}[\text{NO}][\text{H}]}{k_{1,r}[\text{NO}] + k_2[\text{O}_2] + k_3[\text{OH}]}, \quad (33)$$

where the reverse reaction rates have the subscript r . These are calculated with the equilibrium constant, K_c , determined by the difference in Gibbs free energy of the corresponding reactions [33]. Lastly, the concentration of oxygen atoms, $[\text{O}]$, is calculated from the OHC-system, Eqs. (34)–(38).



These reactions are orders of magnitude faster than the Zeldovich mechanism, and thus chemical equilibrium can be assumed [52], with the equilibrium concentrations obtained from Cantera [53].

To obtain the amount of NO produced in the piston engine, Eq. (32) is integrated in time using LSODA [36,37] with relative tolerance 10^{-12} and absolute tolerance 10^{-10} , inside the reaction zone of the two-zone model from the start of combustion until the opening of the exhaust valves. When calculating the mass flow of NO_x , it is assumed that all NO is in the form of NO_2 , therefore multiplying the mass flow of NO by 1.53, according to the definition of the emission index of NO_x [54].

3.8.3. Secondary combustor

The production of nitrogen oxides in the constant pressure combustor is estimated through an empirical correlation for a future lean direct-injection combustion system [43]. The correlation predicts the emission index of NO_x , defined as grams of NO_2 emitted per kilogram of fuel consumed [54]:

$$\text{EI}_{\text{NO}_x} = 7.55 \cdot 10^{-3} \cdot T_4 \cdot \left(\frac{p_{35}}{3027}\right)^{0.37} \exp\left(\frac{T_{35} - 817}{192}\right) \quad (39)$$

where T_{35} and p_{35} are the combustor inlet temperature (in degrees K) and the pressure (in kPa). T_4 is the temperature of the outlet of the combustor in degrees K. Note that this is an empirical correlation and is therefore not dimensionally homogeneous. The values of T_4 , T_{35} , and p_{35} must be given in the units specified above. Note also that the correlation was developed for fresh air entering the combustor. For CCEs, the inlet air will be vitiated, reducing NO_x emissions. Yet, the T_4 may be lower than normally associated with the correlation, which could artificially reduce the predicted NO_x , introducing another uncertainty regarding the accuracy of the model. Therefore, the effect of the secondary burner correlation is investigated in the sensitivity study in Section 4.5.

Further, the piston engine NO_x model was developed for conventional diesel engine combustion, whereas the NO_x correlation for the secondary combustor is valid for a future lean direct-injection combustion system. This comparison will be to the disadvantage of the PM since the NO_x technology levels of the two systems are not the same. Nevertheless, in the CCE cycles with high thermal efficiency, most of the fuel will be burned in the PM and the total NO_x production of the propulsion system is dominated by the PM. For both high-efficiency points on the Pareto front in the multi-objective optimisation in Section 4.4, more than 95% of the fuel is combusted in the PM. The high-efficiency part of the Pareto front is therefore not sensitive to the secondary burner NO_x correlation, which is why we use the correlation without any modification. Hence, we believe that the general trends of the CCE system are

Table 2

Main specifications of the experimental engine and heat release parameters used for validating the NO_x model.

Parameter	Value
Bore d (mm)	80.26
Stroke (mm)	88.90
Geometric compression ratio ϵ (-)	19.81
Revolutions per minute (1/min)	2500
Intake valve opening ($^{\circ}$ BTDC)	8
Intake valve closing ($^{\circ}$ ABDC)	42
Exhaust valve opening ($^{\circ}$ BBDC)	60
Exhaust valve closing ($^{\circ}$ ATDC)	12
Start of combustion (early injection) θ_{SC} ($^{\circ}$ BTDC)	10
Combustion duration (early injection) $\Delta\theta_{CD}$ ($^{\circ}$)	39.5
Shape parameter (early injection) m_w (-)	1.50
Start of combustion (late injection) θ_{SC} ($^{\circ}$ BTDC)	8
Combustion duration (late injection) $\Delta\theta_{CD}$ ($^{\circ}$)	40.25
Shape parameter (late injection) m_w (-)	2.50

captured, for a PM system without any advanced NO_x mitigation technologies. It does affect the comparison with the reference cycle, and the effect of an advanced low-NO_x combustion system would be interesting to see.

3.9. Validation of the NO_x model

The NO_x model is cross-validated and calibrated with publicly available experimental and simulation data from Rakopoulos et al. [55]. Exhaust NO emission data are presented from a single cylinder, four-stroke, water cooled experimental engine, operating as a naturally aspirated direct injection diesel engine. In Table 2, the main specifications of the experimental engine are presented together with the parameters used for the heat release profile.

The experimental engine has a smaller bore than the CCE PM and operates at lower intake pressures and temperatures. Further, it runs on diesel rather than kerosene. Despite this, the dataset was selected as the best available cross-validation case, since it provides time-resolved pressure, temperature, and NO measurements across multiple injection timings and loads. The major difference between kerosene and diesel is the lower cetane number of diesel, resulting in a longer ignition delay. However, at medium to high load, a similar combustion performance is expected to be achievable in compression ignition engines [56]. Further, the NO_x emissions are also not expected to deviate significantly between the two fuels. It was found in [57] that running a diesel engine on kerosene increased fuel consumption by 2% for the same NO_x emissions. Regarding the operating conditions, the CCE PM will operate at significantly higher inlet pressures and temperatures than the validation engine. However, the maximum temperatures will actually not exceed the peak temperature in the experimental data. The peak pressure reaches 150 bar in this study, compared to 90 bar in the validation data, which is a source of uncertainty. This connects to the discussion in Section 3.8.2, about the potential underprediction of NO_x due to the high pressures during combustion. Adding to that, the omission of real-gas effects will underestimate the highest pressures. Using a model calibrated at lower pressures than during CCE operation is therefore expected to result in an underestimation of NO_x emissions. Since all CCE designs of interest, see for example the three points on the Pareto front, operate at high peak pressures, the error is estimated to be systematic and the correct trends are anticipated to be obtained. To further assess the uncertainty of the NO_x model and the calibration factor, a sensitivity study is presented in Section 4.5.

Data from two measurement series for different injection timings, 20° and 15° crank angle before TDC, are used for cross-validation and calibration. Delaying the injection, relative to the timing for peak efficiency, leads to a decrease in peak pressure and temperature, with an associated reduction of NO_x. However, this also results in a reduction in thermal efficiency. The experimental pressure data for 80% load

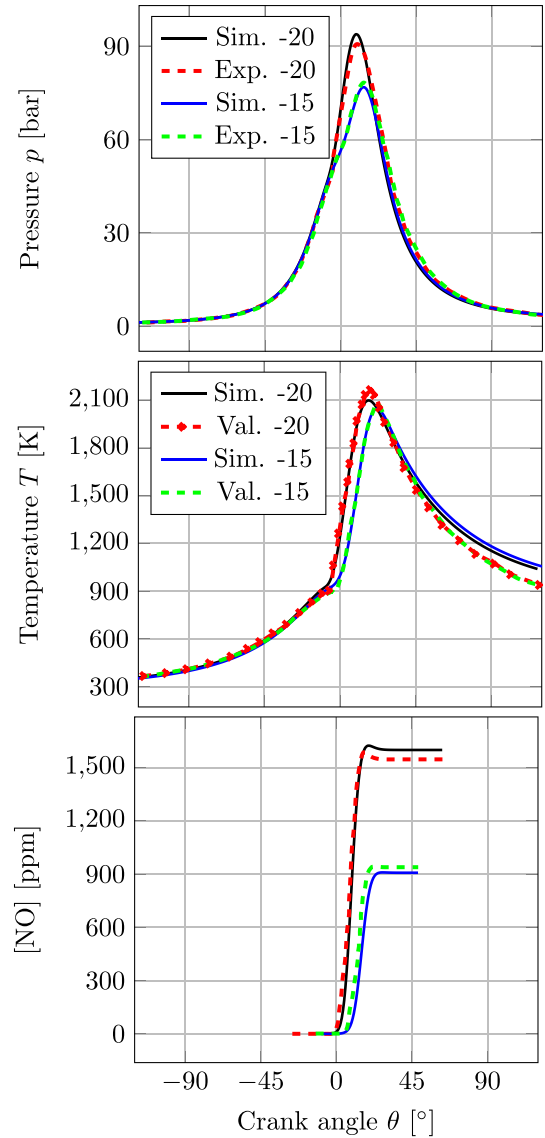


Fig. 5. Time resolved cylinder pressure, temperature and NO concentration for 20° and 15° BTDC injection timing at 80% load for the validation case. In the top figure the results from our model are shown in comparison to the experimental data. In the middle and bottom figures, our simulation data is compared with validation data from the reference paper [55].

are compared with the computed data in Fig. 5-top. Additionally, in Fig. 5-middle and bottom, the calculated mean temperature and concentration of NO are compared with simulation data available in the original paper [55]. The two-zone model flame temperature calibration factor was adjusted to match the NO concentration and was found to be $C_{ft} = 0.84$. An overall good agreement of all three properties can be observed, but with a noticeable difference in peak temperatures, where the maximum relative difference is about 3%. The reason for this difference is thought to be related to the heat transfer model that underpredicts the heat loss. In the present work, it was deemed more important to match the experimental pressure curve rather than their simulated data.

To cross-validate the variation of NO emissions with operating conditions, the exhaust concentration was validated at three load points for each injection timing. In each measurement sweep, the engine was operated at 40%, 60% and 80% load. To account for the variation in the air-fuel-equivalence ratio, the combustion duration was modified

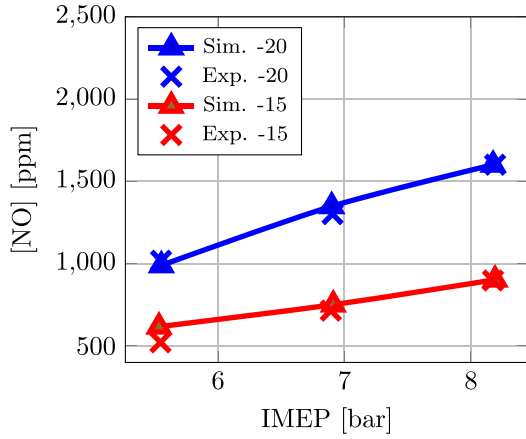


Fig. 6. Exhaust nitric oxide mass concentration of the validation case for three different loads, with the blue curve showing 20° injection timing and red curve 15° before TDC. Crosses are experimental data and triangles simulation data, from the reference [55]. (For interpretation of the references to colour in this figure legend, the reader is referred to the web version of this article.)

according to:

$$\Delta\theta_{CD} = \Delta\theta_{CD,dp} \cdot \left(\frac{\lambda_{dp}}{\lambda} \right)^{0.6} \quad (40)$$

where λ_{dp} is the air-fuel-equivalence ratio at the design point at 80% load. $\Delta\theta_{CD,dp}$ is the combustion duration at the design point and the 0.6 exponent was proposed by [33] for diesel-type engines. Note that this correction of the combustion duration is only applied during the validation, since it represents the change in the combustion duration when deviating from a known design point. During the parametric study and optimisation, we assume that the injection system can be designed to achieve the given heat release curve at each evaluated cycle. The different load cases are represented by their indicated mean effective pressure (see Eq. (19)) and the resulting predictions of the nitric oxide mass concentration of the exhaust are shown together with the experimental data in Fig. 6. As can be observed, good agreement is achieved, with a maximum deviation of 18% for the data point with the lowest NO value. Most importantly, the trend is captured in a satisfactory way.

3.10. CCE system model

The components described in the previous sections are combined in a system model for the entire propulsion system. A schematic of the system model was already presented in Fig. 2, where the lower half of the figure shows a wiring diagram of the engine model incorporating the different modules and their interconnections. Fig. 2 also defines the station numbering used throughout this study.

To cater for a straightforward comparison with the reference engine, the fan pressure ratio is set to 1.47 which is the same value as used for the reference engine [24]. The same fan diameter, 2.24 m, is also applied, such that the specific thrust of the fan stream is unchanged. To maintain the specific thrust, an iteration in the bypass ratio (BPR) of the CCE is performed. Furthermore, the pressure ratios over the LPC and the HPC are obtained from the specified OPR, the core fan pressure ratio, FPR_{core} , and the LPC-HPC compression split exponent, Λ , defined as:

$$\Pi_{LPC} = \left(\frac{OPR}{FPR_{core}} \right)^{\Lambda} \quad (41)$$

$$\Pi_{HPC} = \left(\frac{OPR}{FPR_{core}} \right)^{(1-\Lambda)} \quad (42)$$

with $0 \leq \Lambda \leq 1$. Note that a pressure gain from the piston engine inlet to its outlet is possible and that the OPR here only refers to the turbomachinery pressure ratio. The remaining design variable for the Joule-

Brayton cycle that is varied in the study is the turbine inlet temperature, T_4 .

The design parameters of the piston topping cycle are the fuel-air ratio, f , the compression ratio, ϵ , and the outlet-to-inlet pressure ratio, Π_p . The cylinder bore, d , is used as an iteration variable to match the power required to drive the high-pressure compressor. It is not required that the mass flow through the piston module matches the core flow. Therefore, some air is allowed to bypass the topping cycle, requiring additional compression by an auxiliary compressor if there is a pressure rise over the piston module. Since the mass flow through the piston module is an output of the performance calculations, the amount of piston bypass air is not known beforehand. The power requirement of the piston engine is set by the power needed to drive the HPC, the auxiliary compressor, the fuel and oil pumps, and to provide any required power off-take. The piston head and cylinder are cooled using lubrication oil, where the mass flow of the coolant is selected in accordance with a heat capacity ratio of one in the oil-to-air engine cooler.

Regarding the degradation of efficiency in HPC and LPT due to PM pressure fluctuations, we assume that efficiency losses can be kept below 0.05%, thus negligible, for both HPC and LPT using buffer volumes, according to assumptions in [25]. This assumption may be justified for this purely thermodynamic cycle study where large volumes can be assumed. For future studies where flow paths and buffer volume will be sized, this assumption may have to be revised.

3.10.1. Cycle performance metrics

To quantify the performance of the cycle and the different sub-systems, and allow for comparison and reproducibility, performance metrics must be well defined. We are primarily interested in improvements in thermal efficiency, i.e. the ability of the engine to convert chemical power in the fuel into kinetic power in the exhaust gases, defined as:

$$\eta_{th} = \frac{\frac{1}{2} \left(\sum_j \dot{m}_j v_{j,d}^2 - \dot{m}_0 v_0^2 \right)}{\dot{m}_f \cdot Q_{net}}, \quad (43)$$

where $\dot{m}_0$ is the intake mass flow, v_0 is the flight speed and the outlet kinetic power is summed over all nozzles, j , and is based on the ideal nozzle velocities, $v_{j,d}$. The thermal efficiency is a good metric for comparing the CCE with a turbofan since it is identically defined for both engines and captures the energy conversion efficiency of the engine core. The ideal velocity is obtained from the enthalpy difference between the total enthalpy in the nozzle j , $h_{j,t}$, and the static enthalpy $h_{j,s}$, which would be obtained after an isentropic expansion to ambient pressure:

$$v_{j,d} = \sqrt{2 \cdot w_{j,d}} = \sqrt{2 \cdot (h_{j,t} - h_{j,s})}, \quad (44)$$

where $w_{j,d}$ is the theoretical maximum specific work that could be obtained from the ideal expansion from the state $h_{j,t}$ to ambient conditions. The maximum extractable power from the state j is given by $\dot{m}_j \cdot w_{j,d}$, where $\dot{m}_j$ is the mass flow at the station with state j . The difference in maximum extractable power between the state *core* and the inlet is used to define the engine core power:

$$\dot{W}_{core} = (\dot{m}_{core} w_{core,d} - \dot{m}_{21} w_{0,d}), \quad (45)$$

where *core* is a theoretical state that is obtained in the LPT after only powering the core itself, i.e., the inner fan and the LPC. The core power is a metric of how much gas power is available to the low pressure system and the core nozzle. A very important metric of the CCE is the core power per total cylinder displacement volume, defined as:

$$\dot{W}_{core,V_d} = \frac{\dot{W}_{core}}{V_d}, \quad (46)$$

where V_d is the total displacement volume of the PM. $\dot{W}_{core,V_d}$ is a measure of displacement-based thermodynamic power density, used here as a proxy for the compactness of the engine core.

Table 3
Performance data of the reference year 2035
TRL6 geared turbofan engine at ToC.

Parameter	Value
BPR (-)	12.0
T_4 (K)	1704
OPR (-)	51.89
TSFC (mg/Ns)	13.74
Mass flow (kg/s)	249
Specific thrust (m/s)	109
$E_{I_{NO_x}}$ (g/kg fuel)	13.35
Thrust specific NO_x (mg/Ns)	0.184
$\dot{W}_{core}$ (MW)	9.37
$\dot{w}_{core}$ (kJ/kg)	489
η_{th} (%)	49.4
η_p (%)	79.1
η_o (%)	39.1
LPC pressure ratio (-)	2.8
HPC pressure ratio (-)	13.8
HPT cooling ratio (-)	0.200
LPT cooling ratio (-)	0.020
HPC last stage blade height (mm)	14.94

4. Results and discussion

In this section, the effect of principal cycle design parameters on engine performance and NO_x emissions is investigated. A parametric study is presented, where multiple variable sweeps have been performed around a baseline cycle design of a medium range engine at ToC. To put the results in context, a reference conventional geared turbofan engine, of similar technology level, is used for comparison. Details of the reference engine are presented in Section 4.1. The baseline configuration is presented in Section 4.2 and the results of the parameter sweeps are shown in Section 4.3. In Section 4.4, the results of a multi-objective optimisation are presented. To finalise the section, the effect of the main assumptions on the results is put forward in a sensitivity analysis in Section 4.5.

4.1. Reference gas turbine

The reference engine is a geared turbofan for a medium range aircraft, assumed to be available for a TRL6 year 2035 concept, used as a common reference in the EU project MINIMAL. It is based on the year 2050 engine from project ULTIMATE [58], with updated assumptions for year 2035 and is supported by historical trends and the consensus of the MINIMAL project industrial and academic partners. Performance data is listed in Table 3 for ToC. For comparison, the same component efficiencies are used for the reference and the CCE, and are presented in Table 4. Note that the NO_x value of the reference engine stems purely from the main combustor. For the reference engine and the baseline CCE, the thrust specific fuel consumption (TSFC) is reported along with the thermal efficiency to put the results in perspective.

4.2. Baseline composite cycle engine

During the parametric study, the parameters were varied around a baseline cycle, designed to be a representative CCE which gives rise to insightful trends. The baseline CCE is not intercooled, to isolate the effect of OPR variations from the temperature drop as a result of intercooling. Intercooling is instead investigated in a dedicated sweep. Therefore, the baseline cycle should not be viewed as an optimal cycle, but rather a cycle in which meaningful trends are observed when the design variables are varied.

At this early design stage, we limited the cycle evaluation to a single design point. ToC is chosen for this single-point evaluation since it is typically the most aero-thermally demanding condition of sustained operation. It is also representative of cruise, since it occurs at essentially the same altitude and flight conditions as the subsequent cruise

Table 4
CCE and reference geared turbofan technology parameters.

Parameter	Value
Mach (-)	0.78
Altitude (m)	10,668
ΔT ISA (K)	10
Net thrust (kN)	27.141
Specific thrust (m/s)	109.1
Power off-take (kW)	150
Intake pressure recovery (%)	99.7
Outer fan pressure ratio (-)	1.47
Inner fan pressure ratio (-)	1.34
Fan isentropic efficiency (%)	91.8
LPC polytropic efficiency (%)	89.3
Inter compressors pressure loss (%)	1.29
HPC polytropic efficiency (%)	90.1
HPC efficiency for 13 mm (%)	89.93
Combustion efficiency (%)	99.9
Burner total pressure ratio (-)	0.971
HPT polytropic efficiency (%)	89.35
LPT polytropic efficiency (%)	92.11
Bypass nozzle total pressure ratio (-)	0.990
Core nozzle total pressure ratio (-)	0.985
Nozzle thrust coefficients (-)	0.990
Nozzle discharge coefficients (-)	0.950
Shaft efficiency (%)	99.5
Gearbox efficiency (%)	99.5
Intercooled-CCE specific	
Intercooler hot-side pressure loss (%)	0.00–7.00
Intercooler cold-side pressure loss (%)	7.00
Intercooler effectiveness (-)	0.7
Oil cooler pressure loss cold side (%)	5.7
Oil cooler effectiveness (-)	0.9
Oil cooler nozzle pressure ratio (-)	0.990
Intercooler nozzle pressure ratio (-)	0.990
Auxiliary compressor efficiency (%)	85
Mean piston speed (m/s)	14
Bore-to-stroke ratio (-)	1.0

segment, just with a higher thrust demand. The feasibility of the engine at take-off is still ensured, as the limits on peak cylinder pressure and PM exhaust temperature at ToC are derived and scaled from take-off conditions, based on earlier work [25].

For a fair comparison, the same component efficiencies were applied to the CCE cycle and the reference geared turbofan cycle. The technology assumptions specific for the CCE are listed in Table 4 for the design point. One main difference between the reference and the CCE is the absence of an HPT in the CCE architecture. The specific work in the CCE LPT will be higher due to the smaller core mass flow. Comparing the cycle with the highest efficiency found from the multi-objective optimisation in Section 4.4, cycle C, with the reference, we see that the specific work is 702 kJ/kg compared to 532 kJ/kg. However, the total LPT power is actually lower, with 9.29 MW versus 10.18 MW. The 32% increase in specific work will lead to 1.2 more stages for a constant stage loading. During a new conceptual design, with the radii of the stages slightly increased, we can probably reduce the additional stages to one. The fact that we intend to keep the same stage loading, and that the CCE LPT is uncooled for the high-efficiency designs, makes it more comparable to the LPT of the reference rather than the HPT. This justifies the use of the same LPT efficiency as the reference. The effect of the additional stage will have to be further investigated in a future full mission fuel burn assessment, where first-order mass estimations and flow sizing should be carried out.

The mean piston speed is defined by the stroke length and rotational speed of the engine, with the highest value found for diesel engines in the literature being 14 m/s [35]. To obtain the highest output power per displacement volume, the maximum value was assumed for all simulation points. A fixed bore-to-stroke ratio of unity is as-

Table 5

Baseline design variables and performance metrics at ToC for the CCE. NO_x values are NO_2 -equivalent mass, see Section 3.8.2.

Design parameter	Value
OPR (-)	24
LPC-HPC compression split Λ (-)	0.25
Geometric compression ratio ϵ (-)	8
Fuel-air-ratio piston engine (%)	3.25
T_4 (K)	1250
PM pressure ratio Π_p (-)	1.1
Intercooling ratio (-)	0
Start of combustion θ_{sc} (°)	355
Output	Value
TSFC (mg/Ns)	12.58
η_{th} (%)	53.81
η_o (%)	42.70
$\dot{W}_{core}$ (MW)	8.90
$\dot{u}_{core}$ (kJ/kg)	758.18
$\dot{W}_{core,V_d}$ (kW/litre)	80.40
BPR (-)	20.20
Piston bore d (mm)	180
Number of cylinders (-)	24
Piston module rotational speed (1/min)	2327
Total piston module displacement (litres)	110.8
T_{34} (K)	1218.6
T_{35} (K)	1132.5
p_{max} (bar)	147.7
T_{max} (K)	2165.6
PM shaft power (MW)	4.19
PM heat loss (MW)	2.18
PM mechanical friction (kW)	366.9
Auxiliary compressor power (kW)	55.9
PM fuel fraction (-)	0.879
PM NO_x fraction (-)	0.939
PM flow fraction (-)	0.787
EI_{NO_x} (g/kg fuel)	95.39
Thrust specific NO_x (mg/Ns)	1.20
HPC last stage blade height (mm)	16.33

sumed, meaning that the stroke varies with the bore to match the power requirement on the high-pressure spool. To represent the heat addition inside the PM cylinders, typical direct injection combustion values of $m_w = 2$, $\theta_{SC} = 355^\circ$, and $\Delta\theta_{CD} = 50^\circ$ [35] were used.

The baseline CCE design parameters and performance are presented in Table 5. Note the almost 9% improvement in the thermal efficiency of the CCE compared to the reference geared turbofan, even though the CCE cycle was not optimised. However, this improvement is associated with a more than six-fold increase in NO_x emissions.

4.3. Parametric study

All design parameters in Table 5 were systematically varied in the parametric study, to investigate the effect on thermal efficiency η_{th} , thrust-specific NO_x emissions and core power per displacement volume $\dot{W}_{core,V_d}$. The results of the variation of the first seven variables are presented in Fig. 7 as carpet plots, where each line represents a constant design variable. On the x-axis, the core power per piston displacement volume, $\dot{W}_{core,V_d}$, is shown, and the y-axis indicates the thrust specific emissions of NO_x . The contour plot inside the carpet shows the thermal efficiency of the cycle, and the baseline CCE cycle is marked with a cross. Designs violating any constraints are shown with the grey areas. The effect of each variable is discussed in the following subsections. In each sweep, OPR was varied simultaneously with one other design parameter. Co-varying OPR with each parameter best highlights the trade-off between efficiency, emissions, and power density, as OPR strongly affects all of them.

The feasible design space was limited by three constraints on engine output parameters. Firstly, the peak pressure of the cylinder is limited by the mechanical strength of the piston engine, with the limit set

at 250 bar during take-off. This was assumed to be a reasonable upper bound based on peak pressures achieved in research engines [59]. With measures to reduce the peak pressure, p_{max} , at take-off, such as variable valve timing and injection delay, this corresponds to peak pressures of 150 bar at ToC [25]. There is an uncertainty regarding the transferability of this limit from a heavy-duty automotive diesel to a TR16 2035 aircraft engine, and therefore the impact of this limit on the CCE cycle is investigated in Section 4.5. Secondly, the exhaust temperature of the piston engine at take-off is limited by the temperature capabilities of the outlet valves. Nickel-based valves can operate up to around 1300 K [60] and in combination with ceramic coatings, an exhaust temperature limit of 1350 K was considered achievable. At ToC, this corresponds to 1250 K based on a previous study on a four-stroke CCE [25]. Finally, the bore was limited to 200 mm due to the structural constraints of the engine core. Note that the double V12 engine architecture is kept for all designs. The displacement volume only changes through variation of the bore and stroke.

4.3.1. Effect of turbine inlet temperature

In the first sweep, the turbine inlet temperature, T_4 , was varied between 1200 K and 1500 K, with the turbomachinery OPR swept from 12 to 26. The turbomachinery OPR is defined as the ratio between the total pressure after the HPC and before the fan, while T_4 is controlled by varying the amount of fuel burnt in the secondary combustor. The results of this study are presented in Fig. 7(a). The cycle is limited at high OPR by the peak cylinder pressure reaching 150 bar, due to the increased inlet pressure to the PM, indicated by the grey area in the top of the plot. In the lower left corner, for low OPR and T_4 , the cycle is limited by the bore size exceeding 200 mm. This is caused by the decrease of the specific PM power with a smaller OPR due to the lower inlet pressure, in combination with an increase in the power demand on the PM due to the lower T_4 . This is because a lower T_4 decreases the specific thrust of the core for a fixed OPR, leading to a smaller BPR to match the net thrust requirements. Smaller BPR means larger core mass flow, increasing the power required to drive the HPC.

The results show that turbomachinery OPR strongly correlates with thermal efficiency for a given T_4 , but also increases NO_x . This is caused by the higher inlet temperature and pressure of the piston engine, increasing the production of NO_x within the PM. In addition, decreasing T_4 at constant OPR increases efficiency as less fuel is injected into the secondary burner, causing a larger fraction of the fuel to burn in the PM, which occurs at higher pressures. The core power per displacement volume increases with OPR but increases even more with T_4 . Higher OPR increases the air density into the PM, producing more power per displacement volume. Increasing T_4 , on the other hand, reduces the core size due to the higher BPR, and hence reduces the PM size as well. At the same time, the specific core power increases from the increased T_4 , leading to a net increase in power per displacement volume.

4.3.2. Effect of piston module compression ratio

The second sweep consisted of varying the piston engine compression ratio ϵ from 6 to 12 and changing the OPR from 12 to 26. In a typical piston engine, ϵ is a strong driver of thermal efficiency, since it drives the pressure ratio of the cycle. In the CCE, only a portion of the total pressure increase of the cycle takes place in the cylinder during the compression stroke, leading to a more complex relation. In Fig. 7(b), the resulting carpet plot is presented. The lower limit for ϵ is caused by the bore exceeding 200 mm, as the specific power output decreases with a lower ϵ . At higher OPR, the PM inlet pressure increases, allowing for a lower ϵ . The upper limit on ϵ is set by the peak pressure within the PM, which means that the maximum allowed ϵ decreased for higher OPR.

As in the OPR- T_4 sweep, the OPR is the main driver of thermal efficiency. Thermal efficiency increases slightly with ϵ at lower compression ratios before decreasing at higher values. This is due to the decrease in the outlet temperature of the PM with increasing compression

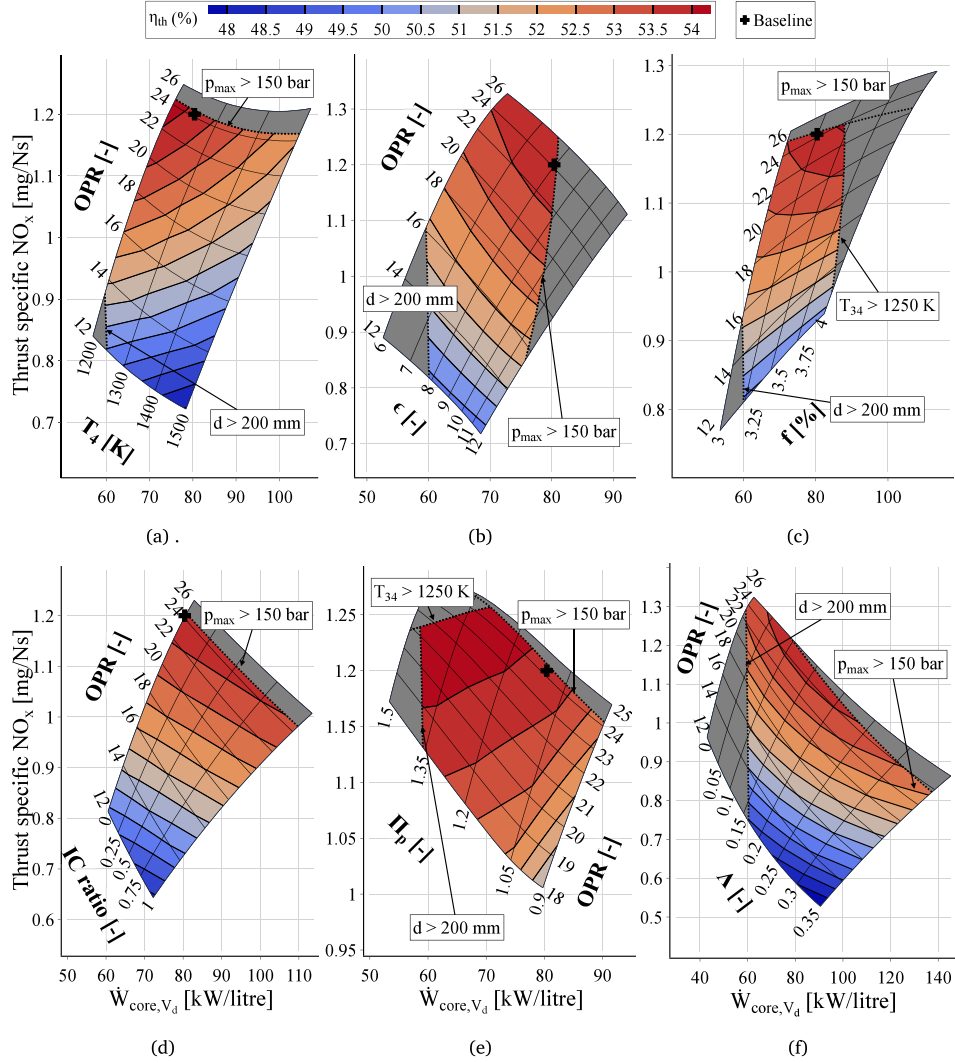


Fig. 7. Thermal efficiency, NO_x emissions (NO_2 -equivalent mass) and core power per displacement volume as function of OPR and six other design variables. (a) Effect of turbine inlet temperature, T_4 . (b) Effect of compression ratio, ϵ . (c) Effect of fuel-air-ratio f . (d) Effect of intercooling ratio. (e) Effect of piston pressure ratio, Π_p . (f) Effect of compression split exponent, Λ . The baseline cycle is represented with a black cross and the infeasible design space is shaded in grey.

ratio, which leads to more fuel being added to the secondary combustor to achieve the specified T_4 . The decrease in PM fuel consumption is outweighed by the increase in the constant pressure burner. However, NO_x emissions decrease and the core power per displacement volume increases with ϵ , which is beneficial. The reason is that the PM produces more power per displacement volume with increasing compression ratio, leading to smaller PM with accompanying lower mass flow. This also leads to lower NO_x emissions, as the PM mass flow decreases faster than the change in NO concentration in the exhaust.

4.3.3. Effect of piston module fuel-air-ratio

In the third sweep, the effect of the fuel-air ratio, f , inside the PM cylinder was investigated for different OPRs. In our simulations, f is a fixed quantity that governs the mass of fuel injected for a given air flow, directly affecting specific power, peak pressure, and maximum and outlet temperatures. For this study, f was varied between 3% and 4% and OPR was varied between 12 and 26. The results are presented in Fig. 7(c). In addition to the limits on peak pressure and bore size, the fuel-air ratio is also limited by the exhaust temperature of the PM, T_{34} , exceeding 1250 K. This is represented by the grey area on the right side of the plot.

At constant PM inlet conditions, an increase in f leads to a higher maximum temperature and pressure inside the cylinder. The effect of

this is clearly seen in the figure, since NO_x , η_{th} , and $\dot{W}_{core,V_d}$ all increase with higher f . For an OPR below 24, f is limited by the exhaust temperature of the PM. Since the inlet temperature to the PM increases with OPR the maximum allowable f decreases with OPR. However, for OPR equal to 24 and above, it is the peak pressure that is the constraint. The highest thermal efficiency is found right in the intersection of the peak pressure and exhaust temperature limit. A high T_{34} is beneficial for the present CCE setup, as less fuel must be added to the secondary burner to achieve the specified T_4 .

4.3.4. Effect of intercooling

Intercooling is an enabling technology for the CCE [20,61]. The synergy between intercooling and the CCE is obtained by cooling down the core stream, increasing the air density and the specific work of the PM, hence decreasing its size and weight. The intercooler is located after the LPC and the core-air is cooled by a fraction of the fan discharge air. The spent cooling air is ejected in a separate nozzle and is used to produce additional thrust. In the study, to vary the amount of intercooling, the ratio between the cooling mass flow and the core flow is varied, denoted IC ratio in Fig. 7(d), where the results are presented. A higher mass flow of cooling air leads to a greater temperature drop in the core for a fixed effectiveness. The cooling mass flow was swept such that the ratio was varied between 0 and 1. The OPR was again swept between

12 and 26, with the maximum temperature drop increasing with higher OPR, since the LPC outlet temperature increases and the fan discharge air temperature is constant.

The results show that intercooling increases the displacement volume specific power of the core, and at the same time reduces NO_x , for all OPR. This comes at the cost of thermal efficiency, since pressure losses are introduced and heat is removed from the higher pressure core to the lower pressure bypass duct. Intercooling becomes more beneficial at higher OPR from the increased temperature difference between cooling and core air, resulting in a larger density increase and a higher increase in core power per displacement volume.

4.3.5. Effect of outlet-to-inlet piston module pressure ratio

In the fifth study, the exit-to-inlet pressure ratio of the PM, Π_p , was varied from 0.9 to 1.5 for an OPR sweep between 18 and 25. Note that the OPR does not include Π_p . In Fig. 7(e) the results of this study are shown.

It is clear that an increase in Π_p leads to a higher thermal efficiency, since the pressure at the turbine inlet increases together with an increased T_{34} , reducing fuel injected in the secondary burner. This is because the gas is compressed inside the cylinder before being pushed out through the exhaust valves, increasing the effective exhaust temperature. Since the PM produces both shaft power and additional gas compression, the bore needs to increase to produce the required outputs. This limits the achievable Π_p due to the 200 mm bore constraint. For an OPR larger than 21.5, it is the exhaust temperature constraint of 1250 K that limits Π_p rather than the bore size. In addition, increasing Π_p reduces the core power per displacement volume due to the increased volume of the cylinder that is required to now provide shaft and additional gas compression power. NO_x is observed to increase both due to the larger PM size and due to the fact that more fuel is being added in the PM instead of the constant pressure burner.

4.3.6. Effect of LPC-HPC compression split

The sixth and final parametric sweep studied the compression split between the LPC and HPC, Λ . For $\Lambda = 0$ the turbomachinery compression is only performed by the HPC, while for increasing Λ more compression takes place in the LPC. It is noted that for this particular study we set the intercooling ratio to unity. Effectively, an increase in Λ also increases the amount of intercooling since the temperature difference between the cooling and core air increases for a higher pressure ratio in the LPC, hence Λ has a direct contribution to intercooling and this effect should also be investigated. The results of the present study are presented in Fig. 7(f). Note that, since intercooling was not considered for the baseline cycle, this point cannot be identified in the carpet plot.

Performing all the compression work in the HPC, without violating the 200 mm bore constraint, is only possible for an OPR of 23 and above, since the specific power of the PM decreases with decreasing OPR. At the same time, since the PM loading decreases with increasing Λ , the required displacement volume also decreases, which leads to an increase in core power per displacement volume $\dot{W}_{core,V_d}$. The second constraint of peak pressure exceeding 150 bar is reached for the combination of the highest OPR and Λ . This is because the inlet temperature to the PM decreases with increasing Λ due to intercooling, resulting in a lower initial temperature during heat addition. Since the heat capacity is lower at lower temperatures, the total increase in temperature inside the cylinder during combustion is greater for colder inlet air. For a partial isochoric heat addition process, the ratio between the peak and the initial temperature also increases, causing the peak pressure to increase. This, in addition to lower heat losses from the lower inlet temperature, leads to an optimum in thermal efficiency at a non-zero Λ . A counteracting effect is that a higher Λ loads the LPT more and the PM less, shifting the heat addition from the PM to the secondary burner. Regarding NO_x emissions, they increase with decreasing Λ since the higher HPC pressure ratio increases PM loading.

Table 6

Summary table of the effect of the design parameters on the cycle metrics.

Design parameter	η_{th}	NO_x	$\dot{W}_{core,V_d}$
OPR	↑	↑	↑
T_4	↓	↓	↑
ϵ	↑↓	↓	↑
f	↑	↑	↑
Intercooling ratio	↓	↓	↑
Π_p	↑	↑	↓
Λ	↓	↓	↑
θ_{SC}	↓	↑↓	↓

4.3.7. Effect of start of combustion

The Wiebe heat release curve is a simplification of real combustion, but can give insights into how one would want to design the heat release profile. In practise, the start of combustion, θ_{SC} , can be controlled by varying the injection timing. In reality, the combustion duration and shape would be affected by the start of combustion, but to isolate the effect of θ_{SC} , they were kept constant. The start of combustion was varied between 335° and 380° crank angle, i.e. from 25° before TDC to 20° after. In Fig. 8, the results of the sweep are presented. Note the very narrow feasible design space between the p_{max} limit on the left and the PM exhaust temperature T_{34} limit on the right. Earlier combustion leads to higher peak pressures and a retarded heat addition decreases the PM shaft work and increases the outlet temperature. Further to the right the bore limit is reached, since the PM power output per displacement volume decreases. We can see in Fig. 8(a) that the peak thermal efficiency is obtained for early combustion, even though T_{35} increases for later combustion, which decreases the fuel burn in the secondary burner. An optimised combustion strategy would probably try to reach a high peak pressure and outlet temperature simultaneously. A small peak for the displacement specific core power, $\dot{W}_{core,V_d}$, is seen at 340° and diminishing towards later heat addition, explained by the growing cylinder volume. The NO_x emissions on the other hand reach a maximum a bit later than peak efficiency, before decreasing sharply around 370°. This can be understood from the larger displacement volume and accompanying mass flow towards late combustion, increasing the total amount of NO_x even though the concentration in the PM exhaust actually decreases after 340°.

Finally, to summarise the parametric study, the direction of influence of each design variable is provided in Table 6. An upwards arrow indicates a positive correlation, a downwards arrow symbolises negative correlation, and both up and down mean the trend is not monotonous.

4.4. Multi-objective optimisation

The design variables and insights from the parametric study were used to define a multi-objective optimisation problem. Here, the propulsion system is optimised with respect to thermal efficiency and thrust specific NO_x emissions to obtain a set of optimal, non-dominated solutions, which define a Pareto front. This was performed with the NSGA-II genetic algorithm [62] available in the pymoo framework [63], using pymoo's default operators: simulated binary crossover (distribution index 15, probability 0.9) and polynomial mutation (distribution index 20, probability $1/n_{var}$), with n_{var} being the number of optimisation variables. The optimisation variables and their ranges are presented in Table 7, together with the optimisation constraints. In addition to the design variables investigated in the variable sweeps, the combustion duration $\Delta\theta_{CD}$ and combustion shape parameter m_w were added to the optimisation variables. All variable limits have been set so that the high-efficiency designs do not result in design variables on the bounds. The exception is the intercooling ratio, whose bounds are set by physical limits. Since the effectiveness is fixed for the intercooler and both media are air, the heat transfer is limited by the core mass flow when the cooling mass flow exceeds it. Therefore, no more intercooling is achieved for

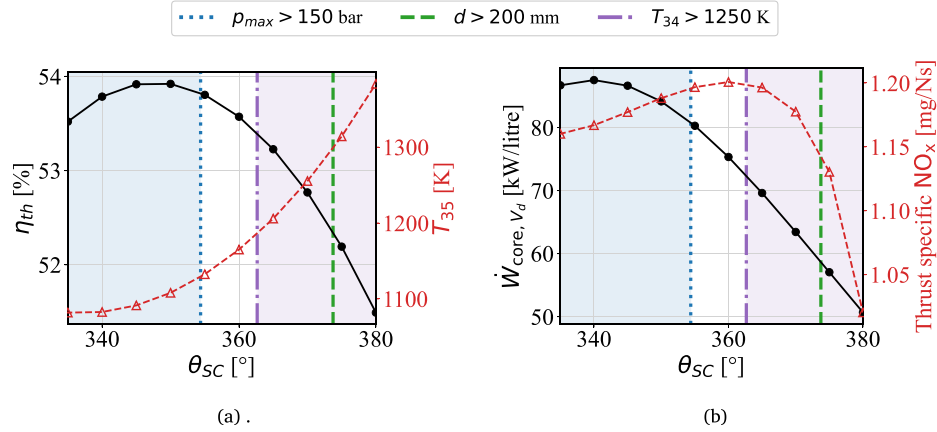


Fig. 8. Variation of start of combustion θ_{SC} around the baseline cycle. (a) Thermal efficiency η_{th} and secondary burner inlet temperature T_{35} . (b) Displacement specific core power $\dot{W}_{core,V_d}$ and thrust specific NO_x (NO_2 -equivalent mass). The limits on peak pressure, bore and PM outlet temperature are shown with the vertical lines and shaded areas.

Table 7
Multi-objective optimisation constraints and variable ranges.

Design parameter	Range
OPR (-)	10–30
LPC-HPC compression split Λ (-)	0.0 - 0.5
Compression ratio ϵ (-)	4 - 15
f piston engine (%)	2 - 5
T_4 (K)	1000 - 1600
PM pressure ratio Π_p (-)	0.9 - 2.0
Intercooling ratio (-)	0.0 - 1.0
Start of combustion θ_{sc} (°)	340 - 390
Combustion duration $\Delta\theta_{CD}$ (°)	20 - 100
Combustion shape parameter m_w (-)	0.5 - 5.0
Constraint	Condition
p_{max} (bar)	< 150
T_{34} (K)	< 1250
d (mm)	< 200
Output	Target
η_{th} (%)	Maximise
Thrust specific NO_x (mg/Ns)	Minimise

IC ratios above 1, setting the upper bound. Constraints are set on the piston peak pressure, piston exhaust temperature and cylinder bore, all enforced as hard limits with no tolerance band.

The hypervolume of the solution was used as the convergence indicator, the criterion being an increase of less than 0.1% over 10 generations. The hypervolume reference point was set to $(-\eta_{th}, NO_x) = (-0.35, 1.5)$, corresponding to a thermal efficiency of 35% and a thrust-specific NO_x of 1.5 mg/Ns, chosen to be worse than every converged, feasible design evaluated during the optimisation. For three different seeds, convergence to the same solution surface was reached after approximately 250 generations for a population size of 200. The converged hypervolumes for the different seeds were: 0.283, 0.281 and 0.282.

In Fig. 9 the thermal efficiency and NO_x emissions of all evaluated data points are shown, with the points coloured by the piston fuel split, which gives an indication of the fuel fraction used in the PM, which is a proxy for the PM load. The Pareto front is coloured red, indicating the optimal solutions. Infeasible solutions, where the constraints in Table 7 were violated, were rejected and are shown with greyed-out crosses. If the cycle simulation failed to converge for a given candidate design, that point was marked as constraint-infeasible and assigned a thermal efficiency of 0% and a thrust-specific NO_x of 999 mg/Ns, ensuring that it was excluded from the Pareto front. Additionally, the reference geared turbofan is shown with a yellow cross for comparison.

Table 8
Three points of interest on the Pareto front.

Parameter	A	B	C
OPR (-)	10.00	18.90	19.56
Λ (-)	0.50	0.50	0.319
Compression ratio ϵ (-)	14.99	10.64	9.55
f piston engine (%)	3.07	3.25	3.62
T_4 (K)	1173.7	1131.2	1100.6
PM pressure ratio Π_p (-)	1.00	1.80	1.88
Intercooling ratio (-)	1.00	1.00	0.99
θ_{SC} (°)	363.1	356.6	356.3
$\Delta\theta_{CD}$ (°)	99.9	94.5	53.2
m_w (-)	2.27	2.05	1.55
η_{th} (%)	42.47	52.26	55.63
Thrust specific NO_x (mg/Ns)	0.039	0.184	1.052
$\dot{W}_{core,V_d}$ (kW/litre)	59.51	60.0	66.06
BPR (-)	12.90	16.56	17.79
Bore (mm)	199.77	197.4	191.6
Displacement volume (litres)	150.27	145.0	132.55
T_{34} (K)	1122.1	1244.9	1248.7
T_{35} (K)	730.9	1096.9	1093.1
p_{max} (bar)	126.12	146.46	149.6
T_{max} (K)	1596.9	1714.7	2105.6
PM shaft power (MW)	2.25	2.99	3.90
PM heat loss (MW)	0.966	1.84	2.11
PM mechanical friction (kW)	351.1	424.4	379.9
Auxiliary compressor power (kW)	3.57	417.5	547.5
PM fuel fraction (-)	0.487	0.959	0.992
PM NO_x fraction (-)	0.469	0.927	0.998
PM flow fraction (-)	0.382	0.731	0.683

The NO_x emissions increase non-linearly with efficiency improvements, with a significant increase seen above 52% efficiency. A clear trend with higher piston fuel split for higher thermal efficiency is also seen. Three points were selected in the Pareto front for the purpose of comparison: **A** is the one with the lowest NO_x , **B** the cycle with the same NO_x as the reference and **C** is the highest η_{th} point. For these cycles, the design variables and performance metrics are presented in Table 8.

One general trend that can be observed from the three points is that OPR, f and Π_p increase with higher η_{th} . Further, Λ , ϵ , T_4 , θ_{SC} , $\Delta\theta_{CD}$ and m_w decrease with increasing η_{th} . The higher efficiency designs seem to achieve peak pressures close to the 150 bar limit and T_{34} approaching the 1250 K limit. This leads to designs where T_{35} is almost equal to T_4 , meaning the secondary burner is essentially not used and all heat addition takes place in the PM. This is also seen from the larger PM shaft power for the high-efficiency designs. The change in η_{th} is very large, with the range spanning between just over 42% and almost 56%. This

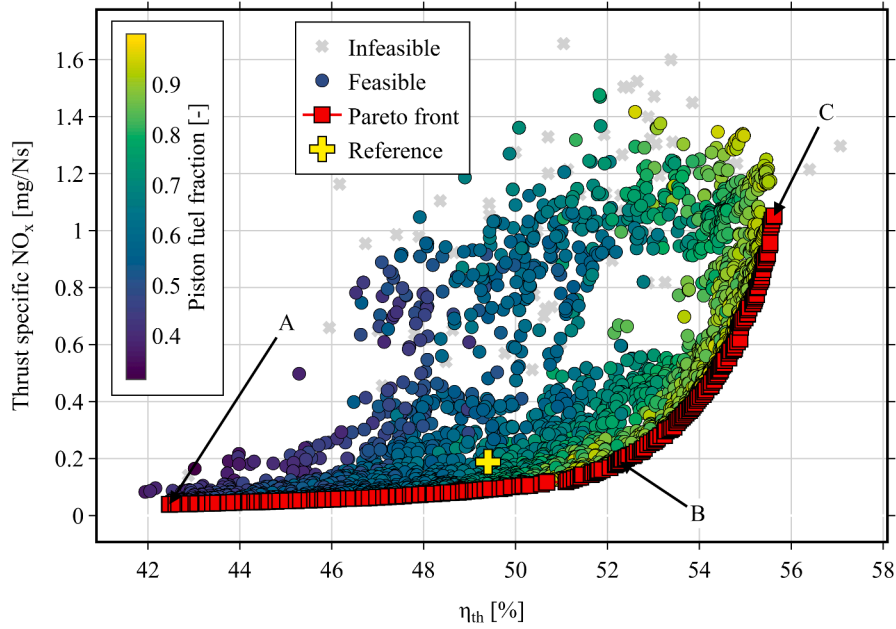


Fig. 9. NO_x (NO_2 -equivalent mass) versus thermal efficiency at ToC. All data points from the multi-objective optimisation, with the Pareto front shown in red. The data points are coloured by the fraction of total fuel burned inside the piston engine. Reference turbopump marked with a yellow cross. (For interpretation of the references to colour in this figure legend, the reader is referred to the web version of this article.)

large variation is explained by the low OPR, low peak pressure and temperature in the PM in combination with no pressure rise over the PM, leading to a very low η_{th} . The highest efficiency design, however, combusts essentially all fuel in the PM, with high OPR, peak pressure and temperature, resulting in the big improvement.

Interestingly, the compression ratio ϵ is at the upper limit for the lowest NO_x cycle. That can be understood by the fact that the cycle strives for the lowest possible OPR and T_{35} to reduce the amount of NO_x created in the secondary burner, since over 50% of the total NO_x is produced there. Since Δ is at its upper limit, the PM load cannot be reduced any further. For the PM to still generate enough power for the HPC, either a high f or a high ϵ is needed. Higher ϵ leads to lower T_{35} , which is why it is preferable over higher f , which also increases the exhaust temperature.

For point C, no design variable except the intercooling ratio is at any limit, indicating that an optimal design has been found. The intercooling ratio cannot exceed 1, since that would not bring any benefit to the cycle, since the temperature drop would not increase in the core stream. Quite surprisingly, intercooling is essentially at its maximum even for the high-efficiency designs. This is because the efficiency improvement from the decreased PM inlet temperature dominated the increased pressure loss from the intercooler. A lower PM inlet temperature releases the bore constraints, reduces wall heat losses and results in a higher peak pressure from the same compression ratio and added heat, as discussed in Section 4.3.6. This allows other variables to move in a favourable direction, an effect that was not captured in the parametric study. A larger intercooler could of course have a greater effectiveness, but that has not been investigated in the present study. Additionally, a very large pressure increase over the PM is observed for both points B and C, which leads to a large piston size and possibly increased weight. Both effects can only be quantified in a full-scope fuel burn mission analysis accounting for engine installation effects and size and weight of the intercooler.

Regarding the heat addition, a longer combustion duration with a larger shape parameter, m_{w^*} , in combination with later start of combustion, seems to correlate with lower NO_x emissions. A larger m_{w^*} means the heat addition is spread out more evenly during the combustion duration. This all leads to an overall smoother heat addition profile with lower peak pressure and temperature. For high efficiency, an earlier

start of combustion with a short duration and sharper heat release shape seems to be preferable. This leads to more heat being added close to the top-dead-centre, where the pressure is higher and more similar to constant volume combustion.

Comparing the Pareto front with the reference geared turbopump, we see that the CCE achieves a higher η_{th} for the same amount of NO_x . However, we need to acknowledge that the uncertainty regarding the NO_x emissions is high and that a higher η_{th} does not equal a lower mission fuel burn. However, we see that significantly higher η_{th} is achievable with the CCE compared to the reference turbopump, but at NO_x levels that would likely be unacceptable without mitigation. This clearly indicates that in order to produce an efficient and competitive CCE, additional NO_x mitigation technology is required.

4.4.1. Optimisation without secondary combustor

To further study the high-efficiency designs, a second optimisation was performed on a CCE without the secondary burner, since this was found to result in the highest η_{th} . This means that T_4 is no longer a design variable, but is set to $T_4 = T_{35}$. The same constraints and design variable limits as in Table 7, except for T_4 , were applied.

The result of this optimisation is presented in Fig. 10(a), with the design points coloured by the displacement specific core power, $\dot{W}_{core,V_d}$. The feasible design space is smaller and this architecture only results in systems with high efficiency. All feasible designs are between 49.3% and 56.0% thermal efficiency and thrust specific NO_x between 0.07 mg/Ns and 0.98 mg/Ns, compared to the lower limits of 42.5% and 0.04 mg/Ns with the secondary combustor. The slightly higher maximum thermal efficiency comes from the reduction in pressure loss from having the secondary combustor turned off, in combination with $T_{35} = T_4$ which eased the optimisation.

The $\dot{W}_{core,V_d}$ data in Fig. 10(a) show some scatter, but reveal a trend with an increase in $\dot{W}_{core,V_d}$ further away from the Pareto front. Hence, an optimal design from a mission fuel burn analysis probably lies somewhere above the front. The influence of the displacement specific core power was therefore assessed by performing multiple optimisations with constraints on the displacement specific core power, as can be seen in Fig. 10(b). The curves represent the Pareto fronts for different minimum constraints on $\dot{W}_{core,V_d}$, compared to the unconstrained curve. A higher

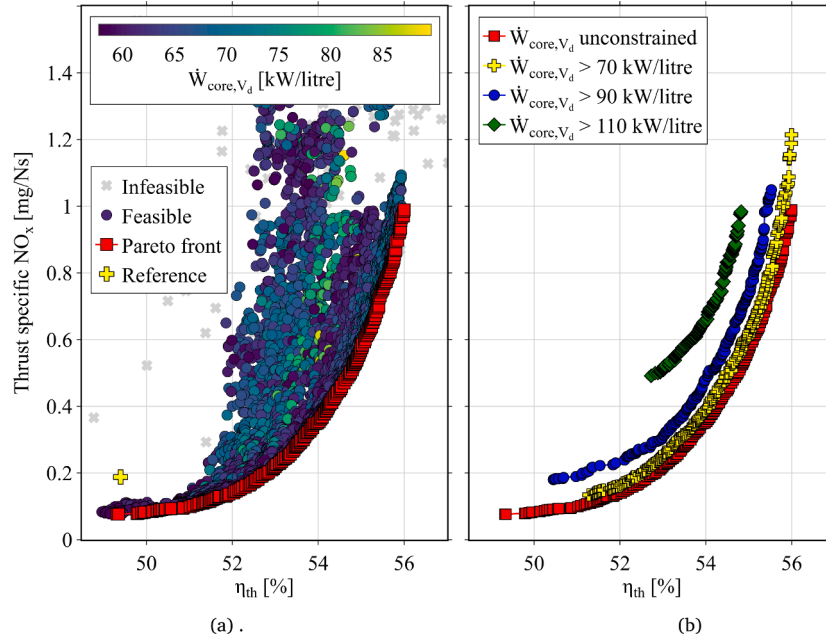


Fig. 10. NO_x (NO₂-equivalent mass) versus thermal efficiency at ToC without a secondary combustor. (a) Data points coloured by the core power per displacement volume, $\dot{W}_{core,V_d}$, with the Pareto front shown in red. (b) Pareto fronts for different lower limits on $\dot{W}_{core,V_d}$.

minimum specific power results in a shift of the front towards worse efficiency and higher NO_x emissions. It can be noted that the Pareto front gets noticeably smaller for $\dot{W}_{core,V_d} > 110$ kW/litre, indicating that the design space shrinks for higher requirements on specific power and that even the lowest efficiency designs result in substantially more NO_x. The unconstrained Pareto optimal results favoured lower displacement specific power. A higher specific power must be traded for the other objective metrics, which indicates that the mission fuel burn optimal design probably lies outside of the original Pareto front.

4.5. Sensitivity study

Since there are uncertainties and assumptions connected with the variables used in the modelling, a sensitivity study has been performed to assess the effect of changes to the main modelling parameters. The NO_x model has been calibrated against a diesel engine operating at different pressures, temperatures and bore from the CCE engine. Further, NO_x production is known to be very sensitive to temperature and fuel-air-ratio. To address the uncertainty in the NO_x modelling, a sensitivity study was carried out. The flame-temperature calibration factor C_{fl} , the air-fuel ratio of the reaction-zone λ_0 and the secondary burner NO_x correlation were varied for the three cycles found from the Pareto front: **A**, **B** and **C**. In Table 9 the percentage change in total NO_x emissions is shown.

We notice that the NO_x production is highly non-linear and that an increase in C_{fl} has the largest impact on point **A**. Remember that C_{fl} directly scales the temperature of the reaction zone in the two-zone model. In **A**, which is the lowest NO_x point, temperatures in the reaction zone are below the threshold for NO_x production, which is exponentially temperature dependent. When C_{fl} is increased, the temperature surpasses the threshold. For the highest efficiency point **C** on the other hand, the effect of increased C_{fl} is more modest. Nevertheless, we can clearly see the importance of C_{fl} on NO_x. For **B**, a 20% reduction of C_{fl} would actually reduce the NO_x by an order of magnitude. We conclude by acknowledging the importance of C_{fl} , and seeing that for $\pm 10\%$ the NO_x stays on the same order of magnitude. The air-fuel equivalence ratio of the reaction zone, λ_0 , turns out to be less important at **A**, with a small increase of nitrogen oxides at a slightly lean reaction zone, but then the

Table 9

Sensitivity analysis of the NO_x modelling parameters. Cycles **A**, **B** and **C** from the Pareto front. The parameter values are given in column one, except for the secondary burner NO_x correlation, where the multiplication factor is shown.

		A	B	C
C_{fl}	ΔC_{fl} (%)	ΔNO_x (%)	ΔNO_x (%)	ΔNO_x (%)
0.672	−20	−44.5	−84.1	−40.8
0.756	−10	−36.0	−62.1	−17.3
0.84	0	0	0	0
0.924	+10	+126.2	+116.5	+11.5
1.008	+20	+435.2	+251.5	+19.8
λ_0	$\Delta \lambda_0$ (%)	ΔNO_x (%)	ΔNO_x (%)	ΔNO_x (%)
1.0	0	0	0	0
1.01	+1	+1.7	+4.7	+11.5
1.05	+5	−7.9	−3.1	+45.0
1.10	+10	−26.3	−35.7	+59.6
Secondary burner	Δ Correlation (%)	ΔNO_x (%)	ΔNO_x (%)	ΔNO_x (%)
0.5	−50	−26.6	±0.0	±0.0
1.0	0	0	0	0
1.5	+50	+26.6	±0.0	±0.0

lowered flame temperature decreases the produced NO_x. For point **C**, the temperature is already so high, that an increasingly lean reaction zone produces more nitrogen oxides from the more available oxygen. Yet, the variation does not change the order of magnitude of the results. Finally, scaling the secondary burner NO_x correlation only affects point **A**, with around half of the change on the correlation, since around half of the NO_x is produced there. For the high-efficiency cycles, this is negligible.

Two of the most important limits are $p_{max} < 150$ bar and $T_{34} < 1250$ K, which were extrapolated from $p_{max} < 250$ bar and $T_{34} < 1350$ K at take-off. This extrapolation was taken from previous studies on a particular design, and may differ for other cycles with different OPR, intercooling, pressure split, etc. Since the high-efficiency designs lie on these constraints, they directly impact the Pareto front. To study this impact, four optimisations were carried out with different constraints. Two optimisations were carried out with higher limits on peak pressure

Table 10

Sensitivity analysis of piston module performance modelling parameters on point B from Pareto front of the multi-objective optimisation. All parameters were multiplied by the factors in column 2.

Parameter	Factor	$\Delta\eta_{th}$ (%)	ΔNO_x (%)	$\Delta \dot{W}_{core,V_d}$ (%)	Δp_{max} (%)	ΔT_{34} (%)
α	0.8	+1.89	+7.75	+1.73	+0.56	+1.68
α	0.9	+0.93	+3.76	+0.86	+0.28	+0.83
α	1.1	-0.90	-3.59	-0.86	-0.27	-0.81
α	1.2	-1.77	-6.97	-1.71	-0.54	-1.61
T_{wall}	0.8	-1.49	-12.03	+0.87	-1.20	-1.66
T_{wall}	0.9	-0.75	-6.18	+0.43	-0.60	-0.83
T_{wall}	1.1	+0.77	+6.55	-0.43	+0.59	+0.83
T_{wall}	1.2	+1.55	+13.40	-0.86	+1.18	+1.67
p_{mf}	0.8	+0.72	-1.01	+3.22	-	-
p_{mf}	0.9	+0.36	-0.49	+1.60	-	-
p_{mf}	1.1	-0.36	+0.46	-1.59	-	-
p_{mf}	1.2	-0.73	+0.81	-3.18	-	-
ϵ	0.9	-0.20	+8.50	-9.20	-0.63	+0.96
ϵ	0.95	-0.01	+4.35	-4.74	-0.32	+0.48
ϵ	1.05	+0.09	-4.39	+5.04	+0.32	-0.48
ϵ	1.1	+0.17	-8.78	+10.41	+0.65	-0.96
Δp_{IC}	0.5	+1.22	-1.28	+7.08	+3.80	± 0.00
Δp_{IC}	0.75	+0.61	-0.69	+3.52	+1.90	± 0.00
Δp_{IC}	1.25	-0.63	+0.68	-3.47	-1.90	± 0.00
Δp_{IC}	1.5	-1.27	+1.45	-6.90	-3.81	± 0.00

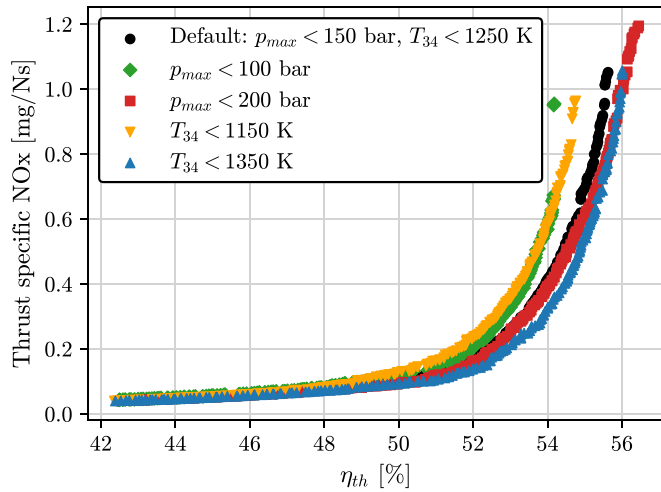


Fig. 11. Pareto fronts for NO_x (NO_2 -equivalent mass, see Section 3.8.2) versus thermal efficiency at ToC, for different limits on peak pressure, p_{max} and piston module outlet temperature, T_{34} . The limit not mentioned in the legend was kept at its default value.

and PM exhaust temperature and two with lower limits. The results can be seen in Fig. 11. The constraints shown in the legend are the limits at ToC. Note that the different limits only seem to affect the Pareto front for higher η_{th} , which is reasonable since it is only then the limits are reached. For intermediate η_{th} , the limits on T_{34} shift the front the most. However, for the maximum efficiency designs, the p_{max} limit seems to have the larger impact. Varying $p_{max} \pm 50$ bar changes maximum η_{th} by (+1.48%, -2.60%) and $T_{34} \pm 100$ K changes maximum η_{th} by (+0.69%, -1.57%).

A final sensitivity study was undertaken to quantify the effect of the PM and intercooler parameters on the cycle performance of point B. Five parameters were scaled, one at a time, to study the effect on η_{th} , NO_x , $\dot{W}_{core,V_d}$, p_{max} and T_{34} . The results and the scaling factors are presented in Table 10, with all results showing the difference from the unperturbed cycle in percentage. Five parameters were investigated: wall heat transfer coefficient α , cylinder wall temperatures, PM mechanical friction p_{mf} , intercooler effectiveness ϵ , and intercooler pressure drop Δp_{IC} . For

η_{th} , the cylinder heat losses were the most important, governed by α and $T_{wall,s}$, followed by IC pressure drop. This indicates the importance of the assumed wall temperatures and pressure drop. For NO_x , heat losses and the effectiveness of the IC were important since they directly impact the peak temperature during combustion. For specific core power, the intercooler has the greatest effect, followed by the mechanical losses. More losses lead to a larger displacement for the same power output. The impact on p_{max} and T_{34} was small, except for the IC pressure drop on peak pressure. To summarise, the effect in general from these parameters was smaller than the effect of NO_x model parameters on the prediction of the emissions.

Of these parameters, the most uncertain are α and the mechanical friction. The first because heat losses are hard in general to predict, and the second because the friction correlation was developed for automotive engines. However, the results show that the effect on the total system thermal efficiency is at least within $\pm 1\%$ for $\pm 20\%$ change in friction losses.

5. Conclusion and outlook

The present paper investigated the impact of the intercooled CCE cycle design parameters on NO_x emissions, thermal efficiency, and core power per displacement volume. The effects of varying the OPR, geometric compression ratio, turbine inlet temperature, piston pressure ratio, LPC-HPC compression split, fuel-air ratio, intercooling and start of combustion have been studied and discussed. The results show that NO_x and η_{th} correlate positively for most of the design variables investigated, except for ϵ and Λ , for which there are optima with respect to η_{th} . Clear trade-offs exist between η_{th} and $\dot{W}_{core,V_d}$ for T_4 , intercooling and Π_p , essentially trading core compactness for efficiency. OPR and f are found to increase both η_{th} and $\dot{W}_{core,V_d}$, at the cost of NO_x emissions.

From the optimisation it was found that for high thermal efficiency, the piston-topping cycle should be highly loaded, meaning that all heat addition should take place in the piston module, with the secondary combustor turned off. In practise, this means that T_4 should be matched to T_{35} during cruise, and that the secondary burner should only be ignited for high power conditions, such as take-off. For the lower NO_x design points, more heat addition is shifted towards the secondary burner, which is detrimental to thermal efficiency and, therefore, not desirable. Furthermore, for high thermal efficiency, the fuel-air ratio, the compression ratio, the OPR, the compression split exponent, PM pressure ratio and heat release shape should be chosen so that the maximum permissible peak pressure and the exhaust temperature of the piston module are reached. A 12.6% higher thermal efficiency relative to the reference turbofan was found from multi-objective optimisation for the following set of design variables: $f = 3.62\%$, $\epsilon = 9.55$, $OPR = 19.56$, $\Lambda = 0.319$, $\Pi_p = 1.88$, $\theta_{SC} = 356.3^\circ$, $\Delta\theta_{CD} = 53.2^\circ$ and $m_w = 1.55$, with $T_4 = 1100.6$ K. Furthermore, for the assumed pressure drop and effectiveness, maximum intercooling should be applied. For the same NO_x emissions as the reference, a 5.8% improvement of thermal efficiency was achieved. It should be noted that an increase in thermal efficiency does not directly translate to an equivalent reduction in mission fuel burn, since fuel burn also depends on installed engine weight and mission-level integration effects, which are outside the scope of this thermodynamic assessment. Under the current modelling assumptions, a 5.7-fold increase in NO_x emissions is estimated for the highest efficiency cycle, highlighting the need for mitigation technology to make the CCE viable. It should be emphasised that the absolute value of the NO_x result should be regarded as an estimation rather than a quantitative prediction, and the main results are the trends.

Since all high-efficiency CCE cycles combusted virtually all fuel in the PM, a subsequent optimisation was carried out with the secondary combustor turned off. The most efficient design was 13.4% more thermally efficient than the reference, with 5.3 times more NO_x . It was found that cycles with higher core power per displacement volume were further away from the Pareto front. Further optimisations were performed

with different lower bounds on $\dot{W}_{core,V_d}$, which showed that higher limits resulted in Pareto fronts shifted towards worse designs. Limiting $\dot{W}_{core,V_d}$ to above 110 kW/litre reduced η_{th} by 2.1% for the most efficient cycle and increased NO_x by 162% for the design with lowest emissions.

The study was built on many assumptions regarding modelling parameters and physical limits, which lead to a sensitivity study. Largest uncertainty was found for the predicted NO_x emissions, which showed a high sensitivity to the parameters of the two-zone model. For point B on the Pareto front, NO_x changed (+116.5%, −62.1%) when the flame temperature calibration factor C_{fl} was varied $\pm 10\%$. Of the piston engine and intercooling performance parameters, the cylinder wall temperature and heat transfer coefficient had the largest impact, where a variation of $\pm 10\%$ resulted in $\approx \pm 0.75\%$ and $\approx \pm 0.9\%$ change in thermal efficiency, respectively. Upper bounds on peak cylinder pressure and exhaust temperature were found to shift the Pareto front, with $p_{max} \pm 50$ bar changing maximum η_{th} by (+1.48%, −2.60%) and $T_{34} \pm 100$ K changing maximum η_{th} by (+0.69%, −1.57%).

5.1. Limitations

This study is limited to a thermodynamic assessment of the inter-cooled CCE cycle at top-of-climb. Installation effects, off-design performance, and mission-level integration are outside its scope. Consequently, the multi-objective optimisation identifies thermodynamic Pareto-optimal solutions, and these should not be interpreted as aircraft-level or mission-level optimal designs. No component mass estimates are included for the piston structure, heat exchangers, buffer volumes, gearbox, auxiliary compressor, or ducting. Core power per displacement volume, $\dot{W}_{core,V_d}$, is therefore used strictly as a measure of thermodynamic power density and not as a proxy for installed engine weight. Similarly, buffer volumes and other flow volumes are not physically sized in this work. Their assumed effect on turbomachinery efficiency losses is instead based on prior assumptions [25]. The study also excludes LTO-cycle emissions, particulate emissions and contrail effects. Any mechanical challenges regarding lubrication or power transmission associated with the 24-cylinder piston module have not been studied nor accounted for.

In the parametric study, T_4 was held constant except for the study on T_4 . This forced more combustion into the secondary burner when T_{35} decreased, from e.g. higher compression ratios or increased intercooling, which deteriorated the cycle efficiency. This setting limited the parametric study from revealing all the trends and optima that were later found in the optimisation. However, obtaining any meaningful results from parametric sweeps without the secondary burner is very hard since the design space is so limited. The parametric study with fixed T_4 still proved to generate meaningful CCE trends, in which the key effects of individual parameters were found.

5.2. Outlook

NO_x mitigation strategies for the CCE represent an important direction for future research. Potential methods include exhaust gas recirculation, steam injection, or lean combustion techniques such as homogeneous charge compression ignition. However, each comes with additional losses and challenges: exhaust gas recirculation requires an additional heat exchanger, homogeneous charge compression ignition limits OPR due to premature ignition, and steam injection requires the on-board storage of water or a complex and heavy water extraction system for the engine exhaust.

Although significant challenges remain for the CCE, particularly regarding NO_x emissions, the study confirms the thermodynamic advantage of the engine, as well as how efficiency and power density can be traded against emissions if the NO_x mitigation techniques prove to be insufficient. The insights from the present study allowed us to fully characterise the cycle behaviour and how an efficient engine could be configured. The subsequent step should be to use such insights to investigate

the climate impact potential of CCE technology, while accounting for installation effects and the incorporation of NO_x mitigation pathways into a representative aircraft mission and fleet-level emissions assessment. This should be supported by a first-order mass model and physical sizing of the excluded components (piston structure, heat exchangers, buffer volumes, gearbox, ducting), enabling a full trade-off between efficiency and installed weight.

CRediT authorship contribution statement

Adam Johansson: Writing – original draft, Visualization, Software, Methodology, Investigation; **Anders Lundblad:** Writing – review & editing, Supervision, Conceptualization; **Tomas Grönstedt:** Writing – review & editing; **Carlos Xisto:** Writing – review & editing, Supervision, Funding acquisition, Conceptualization.

Funding

The work was performed in project MINIMAL - Minimum environmental impact ultra-efficient cores for aircraft propulsion. The MINIMAL project is co-funded by the European Union's Horizon Europe Programme under the grant agreement n° 101056863 and by the UK Research and Innovation (UKRI) funding guarantee under the project reference n° 10040930, 10053292 and 10039071.

Declaration of competing interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

Data availability

Data will be made available on request.

Appendix A. Piston engine friction correlations

Note that the units used in these correlations are mm and kPa. Crankshaft friction is composed of sealing friction, bearing hydrodynamic friction, and turbulent dissipation in the lubricating oil:

$$P_{mf,crankshaft} = P_{mf,seal} + P_{mf,bear} + P_{mf,turb}, \quad (A.1)$$

where the sealing friction is estimated as:

$$P_{mf,seal} = 1.22 \cdot 10^5 \cdot \frac{d_b}{n_c \cdot s \cdot d^2}, \quad (A.2)$$

with bearing diameter, d_b and number of cylinders, n_c . Our engine concept assumes that two V12-engines are used. For V-engines with two connecting rods per pin, the following relations are given: $d_b = 0.62 \cdot d$ and $l_b = 0.4 \cdot d_b$, where l_b is the bearing length. The hydrodynamic friction in the bearing is given by:

$$P_{mf,bear} = 3.03 \cdot 10^{-4} \cdot \frac{N \cdot d_b^3 \cdot l_b \cdot n_b}{d^2 \cdot s \cdot n_c}, \quad (A.3)$$

where N is the engine speed in revolutions per minute and n_b is the number of bearings. The turbulent dissipation of oil in the crankshaft is estimated to be:

$$P_{mf,turb} = 1.35 \cdot 10^{-10} \cdot \frac{d_b^2 \cdot N^2 \cdot n_b}{n_c}. \quad (A.4)$$

The engines also experience losses due to the oscillating motion of the piston, which is broken down into friction from the piston skirt, piston ring friction, ring friction from gas pressure, and connecting rod friction:

$$P_{mf,piston} = P_{mf,skirt} + P_{mf,ring} + P_{mf,gas} + P_{mf,conrod}. \quad (A.5)$$

The friction of the piston skirt is given by the following:

$$p_{mf,skirt} = 2.94 \cdot 10^2 \cdot \frac{\bar{v}_p}{d}, \quad (A.6)$$

piston ring friction:

$$p_{mf,ring} = 4.06 \cdot 10^4 \cdot \left(1 + \frac{1000}{N}\right) \frac{1}{d^2}, \quad (A.7)$$

and the increase in ring friction due to the gas pressure:

$$p_{mf,gas} = 6.89 \frac{p_i}{p_a} \left(0.088\epsilon + 0.182\epsilon^{(1.33-K\bar{v}_p)}\right), \quad (A.8)$$

with $K = 2.38 \times 10^{-2}$ s/m and p_i is the intake pressure. The friction of the connecting rod bearing is again given by Eq. (A.3), with the bearing dimensions given by: $d_b = 0.67 \cdot d$ and $l_b = 0.39 \cdot d_b$.

The friction of the valvetrain arises from the camshaft, cam follower, and valve actuation mechanism:

$$p_{mf,valvetrain} = p_{mf,bearing} + p_{mf,follower} + p_{mf,hydro} + p_{mf,mixed}. \quad (A.9)$$

The camshaft bearing friction is composed of one hydrodynamic term which varies with N and one constant term due to the sealing friction:

$$p_{mf,bearing} = 2.44 \cdot 10^2 \cdot \frac{N \cdot n_b}{d^2 \cdot s \cdot n_c} + 4.12, \quad (A.10)$$

with the number of camshaft bearings being equal to the number of main bearings multiplied by the number of crankshafts. In this work, a double overhead camshaft is assumed. For roller cam followers, the friction comes from the rolling resistance and is estimated as:

$$p_{mf,follower} = 0.005 \cdot \frac{N \cdot n_v}{s \cdot n_c}, \quad (A.11)$$

where n_v is the total number of valves. In addition, the friction between all oscillating parts in the valvetrain is represented by one hydrodynamic term for the friction between two surfaces in relative motion:

$$p_{mf,hydro} = 0.5 \cdot \frac{l_{v,max}^{1.5} \cdot n_v \cdot \sqrt{N}}{d \cdot s \cdot n_c} \quad (A.12)$$

and one term for when the relative velocity of the surfaces approaches zero:

$$p_{mf,mixed} = 10.7 \cdot \left(1 + \frac{1000}{N}\right) \frac{l_{v,max} \cdot n_v}{s \cdot n_c}, \quad (A.13)$$

where $l_{v,max}$ is the maximum valve lift.

Finally, the losses of the auxiliary components are the oil pump, the water pump, and the alternator friction, estimated as:

$$p_{mf,aux} = 6.23 + 5.44 \cdot 10^{-3} \cdot N - 1.79 \cdot 10^{-7} \cdot N^2. \quad (A.14)$$

In our application, oil is assumed to act as a coolant as well, but the total pumping power is deemed to be similar.

The total friction mean effective pressure to be subtracted from the indicated mean effective pressure is given by:

$$p_{mf} = p_{mf,crankshaft} + p_{mf,piston} + p_{mf,valvetrain} + p_{mf,aux}. \quad (A.15)$$

References

- [1] D.S. Lee, D.W. Fahey, A. Skowron, M.R. Allen, U. Burkhardt, Q. Chen, S.J. Doherty, S. Freeman, P.M. Forster, J. Fuglestedt, A. Gettelman, R.R. De León, L.L. Lim, M.T. Lund, R.J. Millar, B. Owen, J.E. Penner, G. Pitari, M.J. Prather, R. Sausen, L.J. Wilcox, The contribution of global aviation to anthropogenic climate forcing for 2000 to 2018, *Atmos. Environ.* 244 (2021) 117834. <https://doi.org/10.1016/j.atmosenv.2020.117834>
- [2] European Commission, Directorate-General for Mobility, Transport, Directorate-General for Research, and Innovation, Flightpath 2050—Europe's Vision for Aviation—Maintaining Global Leadership and Serving Society's Needs, Publications Office, 2012. <https://doi.org/10.2777/15458>
- [3] S. Kaiser, A. Seitz, P. Vratny, M. Hornung, Unified thermodynamic evaluation of radical aero engine cycles, in: *Proceedings of ASME Turbo Expo 2016, GT2016-56313*, Seoul, South Korea, 2016.
- [4] J. Kurzke, Achieving maximum thermal efficiency with the simple gas turbine cycle, 2003, (Google Scholar). 9th CEAS European Propulsion Forum: Virtual Engine—A Challenge for Integrated Computer Modelling, 15–17 October 2003, Rome, Italy.

- [5] F. Herbertsson, X. Zhao, A. Lundblad, T. Grönstedt, Performance assessment of the water enhanced turbofan engine, *Aerosp. Sci. Technol.* 178 (2026) 112389. <https://doi.org/10.1016/j.ast.2026.112389>
- [6] T. Grönstedt, C. Xisto, V. Sethi, A. Rolt, N. García Rosa, A. Seitz, K. Yakinthos, S. Donnerhack, P. Newton, N. Tantot, et al., Ultra low emission technology innovations for mid-century aircraft turbine engines, in: *Turbo Expo: Power for Land, Sea, and Air*, Vol. 49743, American Society of Mechanical Engineers, 2016, pp. V003T06A001.
- [7] T. Grönstedt, C. Xisto, V. Sethi, A. Rolt, N.G. Roas, A. Seitz, D. Misirlis, J. Whurr, N. Tantot, M. Dietz, et al., Conceptual design of ultra-efficient cores for mid-century aircraft turbine engines, in: *Proceedings of the 24th ISABE Conference*, Canberra, Australia, 2019, pp. 22–27.
- [8] L. Su, F. Wen, S. Wang, Z. Wang, Analysis of energy saving and thrust characteristics of rotating detonation turbine engine, *Aerosp. Sci. Technol.* 124 (2022) 107555.
- [9] W.H. Heiser, D.T. Pratt, Thermodynamic cycle analysis of pulse detonation engines, *J. Propuls. Power* 18 (1) (2002) 68–76.
- [10] W. Chen, W. Fan, H. Qiu, H. Qin, C. Yan, Thermodynamic performance analysis of turbofan engine with a pulse detonation duct heater, *Aerosp. Sci. Technol.* 23 (1) (2012) 206–212.
- [11] C. Xisto, F. Ali, O. Petit, T. Grönstedt, A. Rolt, A. Lundblad, Analytical model for the performance estimation of pre-cooled pulse detonation turbofan engines, in: *Turbo Expo: Power for Land, Sea, and Air*, vol. 50770, American Society of Mechanical Engineers, 2017, pp. V001T01A016.
- [12] C. Xisto, O. Petit, T. Grönstedt, A. Rolt, A. Lundblad, G. Paniagua, The efficiency of a pulsed detonation combustor—axial turbine integration, *Aerosp. Sci. Technol.* 82 (2018) 80–91.
- [13] C. Xisto, O. Petit, T. Grönstedt, A. Lundblad, Assessment of CO₂ and NO_x emissions in intercooled pulsed detonation turbofan engines, *J. Eng. Gas Turbines Power* 141 (1) (2019) 011016.
- [14] F.K. Lu, E.M. Braun, Rotating detonation wave propulsion: experimental challenges, modeling, and engine concepts, *J. Propuls. Power* 30 (5) (2014) 1125–1142.
- [15] A.K. Thakur, K. Debbarma, A. Pallela, R. Kumar, R. Singh, S. Srivastava, Advances in pulse detonation engine technology: optimization strategies for next-generation aerospace applications, *Aerosp. Syst.* (2025) 1–29.
- [16] Z. Yang, G. Jia, Y. Du, Z. Fang, X. Gao, G. He, Z. Wang, Investigation of high-tumble chamber of ammonia-hydrogen fueled elliptical rotary engine based on turbulence and combustion characteristics, *Fuel* 400 (2025) 135834. <https://doi.org/10.1016/j.fuel.2025.135834>
- [17] Y. Du, Z. Zhang, Y. Pan, Z. Yang, X. Gao, G. He, Ignition strategy optimization of novel n-butanol doping kerosene elliptical rotary engine using air-assisted direct injection: combustion characteristics, thermodynamic and environmental performance improvement evaluation, *Energy* 344 (2026) 139931. <https://doi.org/10.1016/j.energy.2026.139931>
- [18] H. Sammons, Napier nomad aircraft diesel engine, *J. R. Aeronaut. Soc.* 59 (1955) 211–233.
- [19] H. Klingels, Wärmekraftmaschine Mit Freikolbenverdichter, *European Patent EP2650510 A 2* (2013) 2013.
- [20] S. Kaiser, M. Nickl, C. Salpingidou, Z. Vlahostergios, S. Donnerhack, H. Klingels, Investigations of the synergy of composite cycle and intercooled recuperation, *Aeronaut. J.* 122 (1252) (2018) 869–888.
- [21] S. Kaiser, H. Kellermann, M. Nickl, A. Seitz, A composite cycle engine concept for year 2050, in: *Proceedings of the 31st Congress of the International Council of the Aeronautical Sciences*, Belo Horizonte, Brazil, 2018, pp. 9–14.
- [22] S. Kaiser, O. Schmitz, H. Klingels, Aero engine concepts beyond 2030: Part 2—the free-piston composite cycle engine, in: *Turbo Expo: Power for Land, Sea, and Air*, Vol. 84140, American Society of Mechanical Engineers, 2020, pp. V005T06A018.
- [23] F. Klein, S. Staudacher, Cycle optimization potential of composite cycle and turbo-compound aero-engines, *J. Eng. Gas Turbines Power* 142 (6) (2020) 061007.
- [24] Y.-H. Lin, G. Uhl, F. Winter, A. Lessis, F. Witzgall, A. Seitz, Engine design study for free double piston integrated composite cycle engine, *Aerospace* 13 (4) (2026) 354.
- [25] S. Kaiser, Multidisciplinary Design of Aeronautical Composite Cycle Engines, Ph.D. thesis, Technische Universität München, 2020.
- [26] B.J. McBride, M.J. Zehe, S. Gordon, NASA Glenn Coefficients for Calculating Thermodynamic Properties of Individual Species, Technical Report, Glenn Research Center, Cleveland, Ohio, 2002.
- [27] E. Wolgast, Ein Beitrag zur Prozessrechnung an Verbrennungsmotoren unter Berücksichtigung der realen Gaseigenschaften, Ph.D. thesis, Helmut-Schmidt-Universität / Universität der Bundeswehr Hamburg, 2014.
- [28] J.-C. Han, Advanced cooling in gas turbines 2016 max jakob memorial award paper, *J. Heat Transf.* 140 (11) (2018) 113001. <https://doi.org/10.1115/1.4039644>
- [29] A. Rolt, V. Sethi, F. Jacob, J. Sebastianpillai, C. Xisto, T. Grönstedt, L. Raffaelli, Scale effects on conventional and intercooled turbofan engine performance, *Aeronaut. J.* 121 (1242) (2017) 1162–1185. <https://doi.org/10.1017/aer.2017.38>
- [30] J.H. VanGerpen, A Two-Stroke Diesel Engine Simulation Program, Technical Report, Engine Research Institute, Iowa State University, Ames, Iowa, 1990.
- [31] G.P. Merker, R. Teichmann, et al., *Grundlagen Verbrennungsmotoren*, Springer, 2014.
- [32] R. Van Basshuysen, F. Schäfer, et al., *Handbuch Verbrennungsmotor*, Springer Fachmedien Wiesbaden Wiesbaden, 2015.
- [33] G.P. Merker, C. Schwarz, G. Stiesch, F. Otto, *Simulating Combustion: Simulation of Combustion and Pollutant Formation for Engine-Development*, Springer Berlin, Heidelberg, 1st ed., 2006.
- [34] G.F. Hohenberg, Advanced approaches for heat transfer calculations, *SAE Trans.* 88 (1979) 2788–2806. <http://www.jstor.org/stable/44699090>
- [35] J. Heywood, *Internal Combustion Engine Fundamentals*, McGraw-Hill Education, 2018.

- [36] A.C. Hindmarsh, ODEPACK, A systemized collection of ODE solvers, *Sci. Comput.* 2 (4) (1983) 24–30.
- [37] L. Petzold, Automatic selection of methods for solving stiff and nonstiff systems of ordinary differential equations, *SIAM J. Sci. Stat. Comput.* 4 (1) (1983) 136–148.
- [38] P. Virtanen, R. Gommers, T.E. Oliphant, M. Haberland, T. Reddy, D. Cournapeau, E. Burovski, P. Peterson, W. Weckesser, J. Bright, S.J. van der Walt, M. Brett, J. Wilson, K. Jarrod Millman, N. Mayorov, A.R.J. Nelson, E. Jones, R. Kern, E. Larson, C.J. Carey, Í. Polat, Y. Feng, E.W. Moore, J. VanderPlas, D. Laxalde, J. Perktold, R. Cimrman, I. Henriksen, E.A. Quintero, C.R. Harris, A.M. Archibald, A.H. Ribeiro, F. Pedregosa, P. van Mulbregt, SciPy 1.0 Contributors, SciPy 1.0: fundamental algorithms for scientific computing in Python, *Nat. Methods* 17 (2020) 261–272. <https://doi.org/10.1038/s41592-019-0686-2>
- [39] A. Johansson, P. Miltén, A. Lundblad, C. Xisto, Modelling a hydrogen fuelled composite cycle aeroengine, in: Proceedings of the 34th Congress of the International Council of the Aeronautical Sciences, Florence, Italy, 2024.
- [40] K.J. Patton, R.G. Nitschke, J.B. Heywood, Development and evaluation of a friction model for spark-ignition engines, *SAE Trans.* 98 (1989) 1441–1461. <http://www.jstor.org/stable/44581035>.
- [41] F.P. Incropera, Fundamentals of Heat and Mass Transfer, John Wiley & Sons, Inc., Hoboken, NJ, USA, 2006.
- [42] X. Zhao, T. Grönstedt, Conceptual design of a two-pass cross-flow aeroengine intercooler, *Proc. Inst. Mech. Eng. G: J. Aerosp. Eng.* 229 (11) (2015) 2006–2023. <https://doi.org/10.1177/0954410014563587>
- [43] N.E. Antoine, I.M. Kroo, Framework for aircraft conceptual design and environmental performance studies, *AIAA J.* 43 (10) (2005) 2100–2109.
- [44] G. Heider, G. Woschni, K. Zeilinger, 2-Zonen Rechenmodell zur Vorausrechnung der NO-Emission von Dieselmotoren, *MTZ - Mot. Z.* 59 (11) (1998) 770–775. <https://doi.org/10.1007/BF03226479>
- [45] S. Gordon, B.J. McBride, Computer Program for Calculation of Complex Chemical Equilibrium Compositions and Applications, Reference Publication RP-1311, NASA, Cleveland, OH, 1994. Lewis Research Center.
- [46] P. Eyzat, J.-C. Guibet, A new look at nitrogen oxides formation in internal combustion engines, *SAE Trans.* 77 (1968) 481–500.
- [47] G.A. Weisser, Modelling of Combustion and Nitric Oxide Formation for Medium-Speed DI Diesel Engines: A Comparative Evaluation of Zero-and Three-Dimensional Approaches, Ph.D. thesis, ETH Zurich, 2001.
- [48] J.A. Miller, C.T. Bowman, Mechanism and modeling of nitrogen chemistry in combustion, *Prog. Energy Combust. Sci.* 15 (4) (1989) 287–338.
- [49] A.M. Mellor, J.P. Mello, K.P. Duffy, W.L. Easley, J.C. Faulkner, Skeletal mechanism for NOx chemistry in diesel engines, in: International Spring Fuels and Lubricants Meeting and Exposition, 1998.
- [50] G.P. Smith, D.M. Golden, M. Frenklach, N.W. Moriarty, B. Eiteneer, M. Goldenberg, C. Thomas Bowman, R.K. Hanson, S. Song, W.C. Gardiner, V.V. Lissianski, Z. Qin, GRI-Mech 3.0, 2000, (http://www.me.berkeley.edu/gri_mech/). Gas Research Institute.
- [51] S.M. Shahed, H.K. Newhall, Kinetics of nitric oxide formation in propane-air and hydrogen-air-diluent flames, *Combust. Flame* 17 (2) (1971) 131–137.
- [52] H.K. Newhall, S.M. Shahed, Kinetics of nitric oxide formation in high-pressure flames, *Symp. Combust.* 13 (1) (1971) 381–389. Thirteenth symposium (International) on Combustion, [https://doi.org/10.1016/S0082-0784\(71\)80041-3](https://doi.org/10.1016/S0082-0784(71)80041-3)
- [53] D.G. Goodwin, H.K. Moffat, I. Schoegl, R.L. Speth, B.W. Weber, Cantera: an object-oriented software toolkit for chemical kinetics, thermodynamics, and transport processes, 2025, (<https://www.cantera.org>). Version 3.2.0. <https://doi.org/10.5281/zenodo.17620923>
- [54] T. Harlass, R. Dischl, S. Kaufmann, R. Märkl, D. Sauer, M. Scheibe, P. Stock, T. Bräuer, A. Dörnbrack, A. Roiger, H. Schlager, U. Schumann, M. Pühl, T. Schripp, T. Grein, L. Bondorf, C. Renard, M. Gauthier, M. Johnson, D. Luff, P. Madden, P. Swann, D. Ahrens, R. Sallinen, C. Voigt, Measurement report: in-flight and ground-based measurements of nitrogen oxide emissions from latest-generation jet engines and 100 % sustainable aviation fuel, *Atmos. Chem. Phys.* 24 (20) (2024) 11807–11822. <https://doi.org/10.5194/acp-24-11807-2024>
- [55] C.D. Rakopoulos, D.C. Rakopoulos, D.C. Kyritsis, Development and validation of a comprehensive two-zone model for combustion and emissions formation in a di diesel engine, *Int. J. Energy Res.* 27 (14) (2003) 1221–1249.
- [56] S. Wei, L. Sun, L. Wu, Z. Yu, Z. Zhang, Study of combustion characteristics of diesel, kerosene (RP-3) and kerosene-ethanol blends in a compression ignition engine, *Fuel* 317 (2022) 123468. <https://doi.org/10.1016/j.fuel.2022.123468>
- [57] P. Bergstrand, Effects on combustion by using kerosene or MK1 diesel, in: *Fuels & Emissions Conference*, 2007.
- [58] F.S. Mastropierro, J. Sebastianpillai, F. Jacob, A. Rolt, Modeling geared turbofan and open rotor engine performance for year-2050 long-range and short-range aircraft, *J. Eng. Gas Turbines Power* 142 (4) (2020) 041016.
- [59] M. Megel, B. Westmoreland, G. Jones, F. Phillips, D. Eberle, M. Tussing, N. Yeomans, Development of a structurally optimized heavy duty diesel cylinder head design capable of 250 bar peak cylinder pressure operation, *SAE Int. J. Engines* 4 (3) (2011) 2736–2755. <http://www.jstor.org/stable/26278332>.
- [60] N.K. Raghuwanshi, A. Pandey, R.K. Mandloi, Failure analysis of internal combustion engine valves: a review, *Int. J. Innov. Res. Sci. Eng. Technol.* 1 (2) (2012) 173–181.
- [61] A. Rolt, Enhancing Aero Engine Performance through Synergistic Combinations of Advanced Technologies, Phd thesis, Cranfield University, 2019. <https://dspace.lib.cranfield.ac.uk/items/25d60c1d-df89-446a-92e9-9edb32357642>.
- [62] K. Deb, A. Pratap, S. Agarwal, T.A.M.T. Meyarivan, A fast and elitist multiobjective genetic algorithm: NSGA-II, *IEEE Trans. Evol. Comput.* 6 (2) (2002) 182–197.
- [63] J. Blank, K. Deb, pymoo: multi-objective optimization in python, *IEEE Access* 8 (2020) 89497–89509.