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Modelling, Design and Optimisation of New Cooling Architectures for Fuel Cell, Hybrid-Electric Aircraft

By

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Declaration

I hereby declare that the contents and organisation of this dissertation constitute my own original work and do not compromise in any way the rights of third parties, including those relating to the security of personal data.

This dissertation has been carried out in collaboration with Leonardo S.p.A., co-funding the assigned PhD scholarship. All data and results reported in the manuscript are updated to the submission date. The data used in the present dissertation have been obtained from the available literature.

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Stefano Favre
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*In loving memory of my grandfather, Giuseppe Balbo,
who first nurtured my curiosity, inspired my interests,
and supported me throughout my life and academic journey.*

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Abstract

Recent advances in aircraft electrification and novel propulsion architectures introduce new challenges associated with substantially increased thermal loads, requiring innovative approaches to system design and integration to ensure effective onboard thermal management.

This dissertation addresses these challenges in the context of a hybrid-electric regional transport aircraft with a 20% fuel-cell power contribution, developed in collaboration with Leonardo Company at the Innovation Labs in Torino. A systems engineering approach is adopted to structure the investigation and ensure design traceability throughout the process. Within this framework, a system-oriented methodology for aircraft Thermal Management System (TMS) assessment is developed within the Industrial Systems Engineering and Design (ISED) group at Politecnico di Torino, integrating dynamic TMS models tailored to the selected industrial use case.

The developed methodology is applied to investigate emerging fuel-based TMS architectures (F-TMS) and evaluate their performance over representative mission profiles. The approach captures key transient effects, including fuel depletion, thermal inertia, and buoyancy-driven convection, to quantify the capabilities and limitations of fuel as a heat sink. Subsequently, air-based (A-TMS) configurations are developed and evaluated alongside a novel integrated air–fuel (AF-TMS) architecture. The AF-TMS exploits fuel thermal inertia for active thermal buffering under peak thermal loads, enabling reductions of up to $\sim 17\%$ in TMS mass and $\sim 30\%$ in air intake requirements.

At the component level, a physics-based modelling framework for compact, additively manufactured heat exchangers is developed within the Institute for Advanced Automotive Propulsion Systems (IAAPS) at the University of Bath. The framework integrates numerical models that are validated against literature data and extended to

account for manufacturing-induced geometric deviations, improving the accuracy of pressure-drop predictions. A genetic algorithm-based routine is then coupled with the developed simulation framework to support design optimisation while satisfying application-specific performance requirements, enabling the identification of lightweight, high-performance configurations with mass reductions of up to $\sim 78\%$.

The optimised heat exchangers are ultimately integrated into the dynamic simulation models, closing the design loop and linking component-level designs to system-level performance. This integration yields substantial improvements, including up to a $\sim 64\%$ reduction in TMS mass relative to early design assumptions, alongside reduced intake-induced drag for both A-TMS and AF-TMS configurations.

Overall, the dissertation provides a comprehensive framework for the early-stage design of TMSs for hybrid-electric aircraft, combining dynamic system modelling, component-level simulation, and multidisciplinary optimisation. The results define the operational limits of fuel-based cooling, quantify the benefits of integrated air–fuel architectures, and demonstrate the critical role of tightly coupled system–component design in advancing efficient thermal management for next-generation electrified aircraft.

Contents

List of Figures	xii
List of Tables	xxiv
Nomenclature	xxvi
1 Introduction	1
1.1 Batteries, fuel cells and supercapacitors	4
1.1.1 Batteries	4
1.1.2 Fuel cells	6
1.1.3 Supercapacitors	9
1.2 Research motivations and objectives	11
1.2.1 Case study	17
1.3 Original contribution of the research activity	22
1.4 Work organisation	27
2 Thermal management in aerospace	30
2.1 State of the art analysis	32
2.2 Classification of thermal management systems	37
2.3 Air-based Thermal Management Systems	44
2.3.1 Ram-air cooling	45

2.3.2	Bleed-air cooling	47
2.3.3	Surface HE/skin cooling	49
2.4	Fuel-based Thermal Management Systems	52
2.4.1	Non-recirculating architectures	54
2.4.2	Recirculating architectures	56
2.4.3	Fuel handling	61
2.5	Alternative Thermal Management Systems	65
2.5.1	Storage systems	65
2.5.2	Conversion systems	67
2.6	Summary and highlights	68
3	The Fuel-based Thermal Management System	71
3.1	State of the art	71
3.2	Scope	74
3.3	Methodology	75
3.3.1	The heat source	77
3.3.2	The coolant loop	80
3.3.3	The fuel system	83
3.3.4	The complete F-TMS model	97
3.4	Results	101
3.4.1	F-TMS mass and temperature dynamics	101
3.4.2	Influence of flight mission profile parameters	109
3.4.3	Maximum degree of thermally supported hybridisation	115
3.5	Summary and highlights	122
4	Integrated Air–Fuel TMS	126
4.1	State of the art	126

4.2	Scope	129
4.3	Methodology	130
4.3.1	The air system	133
4.3.2	The updated coolant system	137
4.3.3	The updated fuel system	139
4.3.4	The integrated AF-TMS model	144
4.4	Results	144
4.4.1	Air-based TMS dynamics	145
4.4.2	Integrated air–fuel TMS dynamics	152
4.4.3	Optimised A-TMS and AF-TMS comparison	162
4.5	Summary and highlights	165
5	Heat exchanger modelling	169
5.1	State of the art	170
5.2	Scope	173
5.3	Methodology	175
5.3.1	Thermal domain	176
5.3.2	Hydraulic domain	189
5.3.3	Volumetric and gravimetric domains	193
5.3.4	Integrated HE numerical model	194
5.4	Model validation	196
5.4.1	Reference validation case	196
5.4.2	Thermal length validation	199
5.4.3	Complete model validation	201
5.4.4	Corrected formulations for AM	203
5.5	Summary and highlights	207

6	Heat exchanger optimisation	212
6.1	State of the art	212
6.2	Scope	219
6.3	Methodology	220
6.3.1	The genetic optimisation routine	221
6.3.2	The characterisation and export routine	233
6.3.3	Integration with AF-TMS dynamic simulation model	235
6.4	Results	235
6.4.1	Methodology validation	236
6.4.2	A-TMS and AF-TMS HE optimisation	245
6.4.3	Refined A-TMS and AF-TMS simulation	248
6.5	Summary and highlights	250
7	Conclusions	255
7.1	Outcomes	255
7.2	Future work	260
	References	262

List of Figures

1.1	Major factors driving the research and development of improved and innovative Thermal Management Systems in aerospace.	2
1.2	Functional scheme of a copper-zinc battery galvanic cell. Reproduced from Winter and Brodd [1]. © 2004 American Chemical Society.	5
1.3	Functional scheme of a hydrogen PEM fuel cell. Reproduced from Winter and Brodd [1]. © 2004 American Chemical Society.	7
1.4	Functional scheme of a carbon-based supercapacitor. Reproduced from Winter and Brodd [1]. © 2004 American Chemical Society. . .	10
1.5	Energy density and power density trade-offs between major state-of-the-art electric energy storage devices and carriers. Reproduced from Winter and Brodd [1]. © 2004 American Chemical Society. . .	13
1.6	Drivers, topics, and technological enablers of aircraft decarbonisation and electrification.	14
1.7	Principal powertrain configurations in aerospace literature, including: conventional, all-electric, hybrid-electric, and turbo-electric propulsion. . .	15
1.8	Typical configurations and scales of major regional transport and commuter turboprop aircraft, based on dimensional data from Refs. [2–4].	18
1.9	Schematics of a reference parallel hybrid-electric aircraft configuration. Adapted from Cinar et al. [5]. ©2022 IEEE.	20

1.10	Power flow schematics for the reference parallel hybrid-electric powertrain. Arrow widths suggest relative power proportions but are not to scale.	21
2.1	Yearly publications on the topic of Thermal Management Systems in aerospace. Data retrieved from Scopus. ©2025 Elsevier B.V. . . .	34
2.2	Yearly publications on the topics of Thermal Management Systems, electrification, and decarbonisation-focused electrification in aerospace. Data retrieved from Scopus. ©2025 Elsevier B.V. . . .	36
2.3	Principal heat extraction mechanisms of fuel cells: a) cathode-flow cooling, b) separate-airflow cooling, c) single-phase liquid cooling (and boiling-based variants), d) evaporation cooling, and e) heat-pipe cooling.	38
2.4	Coolant loop representing the reference heat-extraction and heat-transport mechanisms adopted for fuel cell cooling in this dissertation.	42
2.5	Primary heat sinks in aerospace and associated heat-rejection mechanisms. Extended and refined from the author's previous work [6]. .	43
2.6	Yearly publications on the topics of general aircraft TMS and air-based aircraft TMS research in aerospace. Data retrieved from Scopus. ©2025 Elsevier B.V.	44
2.7	Functional schematic of ram-air cooling.	45
2.8	Functional schematic of compressor and fan bleed-air cooling. Adapted from Van Heerden et al. [7]. ©2021 Elsevier.	48
2.9	Functional schematic of surface heat exchangers and skin cooling (wing-mounted).	50
2.10	Yearly publications on the topics of general, air-based, and fuel-based TMS technology in aerospace. Data retrieved from Scopus. ©2025 Elsevier B.V.	53
2.11	Fuel-based TMS with submerged heat exchanger. Extended and refined from the author's previous work [6].	55
2.12	Fuel-based TMS with heat exchangers on the engine feed-line. Extended and refined from the author's previous work [6].	56

2.13	Fuel-based TMS with parallel heat exchanger recirculation. Extended and refined from the author's previous work [6].	57
2.14	Fuel-based TMS with series heat exchanger recirculation. Extended and refined from the author's previous work [6].	58
2.15	Dual-tank F-TMS with series heat exchanger recirculation. Extended and refined from the author's previous work [6].	59
2.16	Fuel-based TMS with hot/cold dual tank recirculation with drains. Based on concepts from Refs. [8, 9].	61
3.1	Functional scheme of a F-TMS for the cooling of a fuel cell. A recirculating architecture with a heat exchanger installed in series on the engine feed line is shown for reference.	76
3.2	Schematic representation of the Eulerian formulation for an open thermodynamic system.	76
3.3	Overview of the numerical model of the heat source component. . .	79
3.4	Overview of the numerical model of the coolant loop subsystem and outer HE structure elements connecting to the FC and fuel system. .	84
3.5	Overview of the numerical model of the fuel tank internal dynamics.	88
3.6	Convection promoting configurations in heated fuel tanks. (3.6a) submerged heat exchanger-induced convection. (3.6b) low-lying piccolo tube-induced convection.	92
3.7	Overview of the combined fuel tank internal and boundary numerical dynamics model.	94
3.8	Overview of the numerical model of the complete fuel subsystem. .	96
3.9	Overview of the complete F-TMS simulation model consisting of: heat source, coolant loop, fuel system, and the intermediate HE structure components connecting these elements, together with the additional mission profile and simulation options modules.	97
3.10	Schematic representation of the simulated flightpath and its main phases.	98

3.11	Altitude and combined propeller shaft power trends for a 400 km, 6000 m regional flight, followed by a 100 km, 4500 m alternate. . . .	102
3.12	Tank fuel mass dynamics for 0%, 10%, and 20% hybridized regional transport aircraft performing a 400 km, 6000 m main mission followed by a 200 km, 4500 m alternate flight.	102
3.13	Relevant temperature dynamics for a 5% hybrid-electric turboprop aircraft with <i>adiabatic</i> fuel tanks performing a 400 km, 6000 m main mission followed by a 200 km, 4500 m alternate flight. (3.13a) recirculating-parallel F-TMS configuration. (3.13b) recirculating-series F-TMS configuration.	103
3.14	Relevant temperature dynamics in a 5% hybrid-electric turboprop aircraft with natural convection in <i>diabatic</i> fuel tanks performing a 400 km, 6000 m main mission followed by a 200 km, 4500 m alternate flight. (3.14a) recirculating-parallel F-TMS configuration. (3.14b) recirculating-series F-TMS configuration.	105
3.15	Occurrence distribution of air and fuel convective heat-transfer coefficients at the fuel-tank interface over the reference mission profile (recirculating-series architecture for reference).	106
3.16	Thermal endurance of diverse F-TMS configurations for the cooling of a MT-PEM fuel cell supporting a 5% degree of aircraft hybridisation.	107
3.17	Relevant temperature dynamics in a 5% hybrid-electric turboprop aircraft with forced convection in <i>diabatic</i> fuel tanks performing a 400 km, 6000 m main mission followed by a 200 km, 4500 m alternate flight. (3.17a) recirculating-parallel F-TMS configuration. (3.17b) recirculating-series F-TMS configuration.	108
3.18	Significant mass and temperature trends for a series F-TMS performing a reference 400 km mission at diverse altitudes, followed by a 200 km, 4500 m alternate flight (5% aircraft electrification). (3.18a) tank-fuel mass evolution. (3.18b) heat source and tank-fuel temperature evolution.	110

3.19	Significant mass and temperature trends for a series F-TMS performing three different missions at a main cruise altitude of 6000 m, followed by a 200 km, 4500 m alternate flight (5% aircraft electrification). (3.19a) tank-fuel mass evolution. (3.19b) heat source and tank-fuel temperature evolution.	112
3.20	Significant temperature trends for a series F-TMS performing three different missions at a main cruise altitude of 6000 m, followed by a 200 km, 4500 m alternate flight (3% aircraft electrification).	113
3.21	Significant mass and temperature trends for a series F-TMS performing a 400 km, 6000 m, main cruise followed by different alternate flights. (3.21a) tank-fuel mass evolution. (3.21b) heat source and tank-fuel temperature evolution.	114
3.22	Maximum heat source temperature for a series F-TMS as a function of main mission range, alternate range, and target cruise altitude (4% aircraft electrification). Fuel-cell operating limit of 120 °C highlighted in red. The three subfigures correspond to different cruise altitudes.	117
3.23	Maximum tank-fuel temperature for a series F-TMS as a function of main mission range, alternate range, and target cruise altitude (4% aircraft electrification). The three subfigures correspond to different cruise altitudes.	118
4.1	Functional scheme of a ram-air-based A-TMS for the cooling of a fuel cell.	131
4.2	Functional scheme of a ram-air, recirculating-series single tank, AF-TMS for the cooling of a fuel cell.	131
4.3	Schematic representations of fuel-based thermal buffering for an AF-TMS: (4.3a) F-TMS cooling (absorbing heat from the coolant) and (4.3b) F-TMS recharging (releasing heat to the coolant).	133
4.4	Overview of the numerical model of the complete air subsystem. . .	138
4.5	Overview of the numerical model of the coolant loop subsystem and outer HE structure elements connecting to the fuel cell, air system, and fuel system.	140

4.6	Overview of the baseline (4.6a) and updated (4.6b) fuel-flow controllers for the AF-TMS.	143
4.7	Overview of the numerical model of the complete air–fuel thermal management system in Simulink.	144
4.8	Relevant temperature trends of an A-TMS with two 0.15 m^2 intakes. Cooling results for a 20% hybrid-electric turboprop aircraft performing a 400 km, 6000 m main mission followed by a 200 km, 4500 m alternate flight.	145
4.9	Relevant temperature trends of an A-TMS with two 0.10 m^2 , 0.15 m^2 , and 0.20 m^2 intakes. Cooling results for a 20% hybrid-electric turboprop aircraft performing a 400 km, 6000 m main mission followed by a 200 km, 4500 m alternate flight.	147
4.10	Relevant temperature trends of an A-TMS with two 0.15 m^2 intakes. Cooling results for a 20% hybrid-electric turboprop aircraft performing a 400 km main mission at different altitudes, followed by a 200 km, 4500 m alternate flight.	148
4.11	Relevant temperature trends of an A-TMS with two 0.15 m^2 intakes. Cooling results for a 20% hybrid-electric turboprop aircraft performing different mission profiles at a 6000 m main cruise altitude followed by a 4500 m alternate segment. (4.11a) comparison of results for 130 km, 400 km, and 1300 km main flight ranges. (4.11b) comparison of results for 50 km, 200 km, and 350 km alternate flight ranges.	150
4.12	Relevant temperature trends of an A-TMS with two 0.190 m^2 intakes. Cooling results for a 20% hybrid-electric turboprop aircraft performing a 400 km main mission at different altitudes, followed by a 200 km, 4500 m alternate flight.	152
4.13	Relevant temperature trends of an AF-TMS with two 0.15 m^2 intakes and a single tank recirculating-series arrangement. Cooling results for a 20% hybrid-electric turboprop aircraft performing a 400 km, 6000 m main mission followed by a 200 km, 4500 m alternate flight. (4.13a) baseline fuel-flow controller. (4.13b) updated fuel-flow controller.	153

4.14	Relevant temperature trends of an AF-TMS with two 0.10 m ² intakes and a single tank recirculating-series arrangement. Cooling results for a 20% hybrid-electric turboprop aircraft performing a 400 km main mission at different altitudes, followed by a 200 km, 4500 m alternate flight.	155
4.15	Relevant temperature trends of an AF-TMS with two 0.10 m ² intakes and a single tank recirculating-series arrangement. Cooling results for a 20% hybrid-electric turboprop aircraft performing different missions at a main cruise altitude of 6000 m, followed by a 200 km, 4500 m alternate flight.	156
4.16	Heat source temperature trends of an AF-TMS with two 0.10 m ² intakes and a single tank recirculating-series arrangement. Cooling results for a 20% hybrid-electric turboprop aircraft performing diverse missions at a main cruise altitude of 6000 m, followed by a 200 km, 4500 m alternate flight.	157
4.17	Relevant temperature trends of an AF-TMS with two 0.10 m ² intakes and a single tank recirculating-series arrangement. Cooling results for a 20% hybrid-electric turboprop aircraft performing a 400 km, 6000 m main cruise followed by different alternate flights.	158
4.18	Maximum heat source temperatures for an integrated AF-TMS with two 0.128 m ² intakes and a single tank recirculating-series arrangement in a 20% hybrid-electric aircraft configuration. Response surfaces of main and alternate mission ranges. The three subfigures correspond to the different main cruise altitudes.	161
5.1	Representation of the whole heat exchanger component and detail of a single plate module with the associated two-dimensional system of coordinates. Reproduced from the author's previous work [10], licensed under CC BY 4.0.	177

5.2	Representation of the plate control volume centred around the channels of one fluid, and extrapolation of the equivalent thermal resistance model. To ease visualization, the channels of both fluids are shown with cross-sections perpendicular to the reader, although the heat exchanger operates in a cross-flow configuration. Reproduced from the author's previous work [10], licensed under CC BY 4.0. . . .	181
5.3	Circular ducts approximation assumed to simplify fin efficiency calculation. Reproduced from the author's previous work [10], licensed under CC BY 4.0.	183
5.4	Comparison of the average computational costs (5.4a) and reference simulation outcomes (5.4b) for the proposed global ϕ -model and a finite difference approach, as a function of the temperature refinement tolerance. Global model results are provided for three minimum summation refinement thresholds in solving the ϕ equation: 1%, 0.01%, 0.0001%. The converged mesh sizes associated with each tolerance are also reported. Reproduced from the author's previous work [10], licensed under CC BY 4.0.	189
5.5	HE representation and subdivision according to the inlet, mid, and outlet pressure drop contributions adopted in the modelling approach. Reproduced from the author's previous work [10], licensed under CC BY 4.0.	190
5.6	Functional scheme of the integrated HE characterisation model, including detail of the thermal, hydraulic, and gravimetric/volumetric sub-models. Reproduced from the author's previous work [10], licensed under CC BY 4.0.	194
5.7	Representation of all geometrical inputs required by the model. A combined circular-rectangular channel design is provided, on the left, to evidence differences between parameters of the two shapes. Reproduced from the author's previous work [10], licensed under CC BY 4.0.	195
5.8	Literature-based test rig and sensor placement for the experimental measurement of pressure drops and temperatures. Adapted from da Silva et al. [11]. ©2021 Elsevier.	196

5.9	Photographs, geometries, and dimensions of the reference literature heat exchanger. Adapted from da Silva et al. [11]. ©2021 Elsevier. .	197
5.10	Representation of the heat exchanger core, headers, and manifolds with the corresponding sections used for pressure-drop evaluation according to the test rig in Ref. [11]. Reproduced from the author's previous work [10], licensed under CC BY 4.0.	198
5.11	Validation of the proposed thermal length definition against experimental data (black) and numerical predictions (blue) from Morteau et al. [12] (5.11a), Luo et al. [13] (5.11b), and Hein and Morteau [14] (5.11c). Present-model predictions (red) use either the reference heat transfer area (dashed) or the thermal length proposed by da Silva et al. [11] (solid). In (5.11c), the dotted and solid lines assume the thermal conductivities reported by Hein and Morteau [14] and in the EOS data sheet for sintered Polyamide 2200, respectively. Reproduced from the author's previous work [10] under CC BY 4.0.	201
5.12	Thermal-hydraulic results of the proposed integrated model (red line) compared to experimental data (black dotted line) and numerical predictions (black line) from Ref. [11]. (5.12a) Inlet and outlet air temperature measurements and numerical results. (5.12b) air pressure drop measurements and numerical results for the test rig. (5.12c) air pressure drop measurements and numerical results for the sole heat exchanger (core). Reproduced from the author's previous work [10], licensed under CC BY 4.0.	202
5.13	Functional scheme of the corrected channel cross-sections, accounting for the presence of a triangular-shaped laser melting dross at the top of the duct. Reproduced from the author's previous work [10], licensed under CC BY 4.0.	204
5.14	Hydraulic results of the dross-corrected model (red line) compared to experimental data (black dotted line) and simulation outcomes (black line) by da Silva et al. [11]. Detail of the air pressure drops of the heat exchanger core with respect to the fluid mass flow rate on a linear (5.14a) and logarithmic (5.14b) scales, and for the Reynolds numbers on a logarithmic scale (5.14c). Reproduced from the author's previous work [10], licensed under CC BY 4.0.	205

5.15	Hydraulic results of the K_p -corrected model (red line) compared to experimental data (black dotted line) and simulation outcomes (black line) from Ref. [11]. Detail of the air pressure drops of the heat exchanger core with respect to the fluid mass flow rate on a linear (5.15a) and logarithmic (5.15b) scales, and for the Reynolds numbers on a logarithmic scale (5.15c). Reproduced from the author's previous work [10], licensed under CC BY 4.0.	206
5.16	Comparison of results from the various models simulating the air pressure drops in the core HE from Ref. [11]. Detail of trends with respect to the air mass flow rates (5.16a) and experimental deviations (5.16b). Reproduced from the author's previous work [10], licensed under CC BY 4.0.	207
5.17	Thermal model results following the application of corrective formulations: detail of the outlet temperature trends (5.17a) and experimental deviations of the associated heat transfer rates (5.17b). Reproduced from the author's previous work [10], licensed under CC BY 4.0.	208
6.1	Integration of the thermo-hydraulic-gravimetric model from Ref. [10] into the proposed genetic optimisation framework, including comprehensive component characterisation and export to alternative simulation environments for dynamic system-level analysis (i.e. MATLAB/Simulink).	219
6.2	Early HE placement concept for a hybrid electric regional transport aircraft. Components are not to scale.	221
6.3	Functional scheme of the proposed HE design optimisation loop. The representation details the main steps of the process, highlighting the interfaces between the various sub-models and simulation environments.	222
6.4	Functional scheme of the genetic-algorithm optimisation loop. . . .	223

- 6.5 Representation and nomenclature of every chromosome gene employed for the evolutionary design optimisation of the HE. A combined circular-rectangular channel configuration is provided, on the left, to evidence differences between optimisation parameters of the two shapes. 224
- 6.6 Relevant temperature trends of optimised A-TMS and AF-TMS for a 20% hybrid-electric turboprop aircraft performing a 300 km, 7600 m main mission followed by a 50 km, 1000 m alternate flight. (6.6a) A-TMS configuration with two 0.190 m^2 intakes. (6.6b) AF-TMS configuration with two 0.128 m^2 intakes. 228
- 6.7 Updated intermediate heat exchangers modules in the AF-TMS model and detail of the look-up table implementing the thermal effectiveness response surface in HE_D (coolant-fuel). 235
- 6.8 Performance distribution of all chromosomes from the mass and heat transfer optimisation of the baseline HE geometry of Ref. [11] for: (6.8a) circular channel shape; (6.8b) square channel shape; (6.8c) rectangular channel shape, with a maximum aspect ratio of 1 : 5; (6.8d) rectangular channel shape, with an unbounded aspect ratio. The (1, 1) point indicates the baseline geometry performance. . . . 237
- 6.9 Baseline HE [11] mass and heat transfer rate comparison with the Pareto frontiers of non-dominated solutions featuring: circular, square, rectangular (maximum aspect ratio of 1 : 5), or unbounded rectangular channels, optimised within the same design space. . . . 238
- 6.10 Heat transfer rate per unit mass for the baseline HE [11] and the optimised non-dominated solutions featuring circular, square, rectangular (maximum aspect ratio of 1 : 5), and unconstrained rectangular channels. 239
- 6.11 Key relationships between the heat transfer performance of the optimised heat exchangers and their main dimensions for circular, square, rectangular (maximum aspect ratio of 1 : 5), and unconstrained rectangular channels. (6.11a) HE height. (6.11b) HE length (hot-side). (6.11c) HE width (cold-side). 241

- 6.12 Key relationships between the heat transfer performance of the optimised heat exchangers and their main dimensions for circular, square, rectangular (maximum aspect ratio of 1 : 5), and unconstrained rectangular channels. (6.12a) water channel height/diameter (hot-side). (6.12b) water channel base/diameter (hot-side). (6.12c) air channel height/diameter (cold-side). (6.12d) air channel base/diameter (cold-side) 243
- 6.13 Response surfaces of thermal effectiveness and heat transfer rates for the exported heat-exchanger optimised designs. (6.13a) coolant-air HE of the A-TMS. (6.13b) coolant-air HE of the AF-TMS. (6.13c) coolant-fuel HE of the AF-TMS. 247
- 6.14 Temperature evolution of the optimised A-TMS and AF-TMS configurations for a 20% hybrid-electric turboprop aircraft performing a 300 km, 7600 m main mission followed by a 50 km, 1000 m alternate flight. (6.14a) A-TMS configuration with two 0.190 m^2 intakes. (6.14b) AF-TMS configuration with two 0.128 m^2 intakes. 249

List of Tables

1.1	Summary of main state-of-the-art electric energy storage technologies and comparison with conventional jet fuel properties.	12
1.2	PW127F engine performance specifications at representative operating modes, based on data from Pratt & Whitney technical documentation and EASA certification records [15, 16].	19
1.3	Peak mechanical, electrical, and thermal powers for the reference hybrid-electric use case (nominal efficiencies: $\eta_{FC} = 0.50$, $\eta_{EM} = 0.95$, $\eta_{line} = 0.95$, $\eta_{gear} = 0.95$).	22
2.1	Scopus research string criteria for TMS literature analysis	33
2.2	Scopus research string criteria for electrification literature analysis	35
4.1	Operating modes of the updated air–fuel TMS control logic governing the fuel heat-exchanger mass flow rate.	141
4.2	Maximum heat source temperatures simulated during main flights of 130 km, 400 km, or 1300 km at an altitude of 6000 m, followed by a 200 km, 4500 m alternate flight.	156
4.3	System mass and maximum power comparison between A-TMS and AF-TMS architectures.	164
5.1	Significant modelling approaches for HE thermal characterisation, simulation, and design. Adapted from the author’s previous work [10], licensed under CC BY 4.0.	174

5.2	Convective heat transfer coefficients of fluids. Adapted and refined from the author's previous work [10], licensed under CC BY 4.0. . .	184
6.1	Principal methodologies applied to HE design optimisation.	218
6.2	Genetic algorithm chromosome genes.	225
6.3	GA optimisation boundary conditions and design targets.	230
6.4	GA optimisation hyper-parameters.	233
6.5	Optimised designs for HE_M and HE_D in the A-TMS and AF-TMS configurations.	247
6.6	Conservative optimised designs for HE_M and HE_D in the A-TMS and AF-TMS configurations.	249
6.7	System masses and maximum power penalties for A-TMS and AF-TMS architectures following the integration of refined HE characterisation.	251

Nomenclature

Roman Symbols

$\dot{m}$	Mass flow
$\dot{Q}$	Heat transfer rate
$\dot{W}$	Work rate (power)
$\vec{c}$	Velocity vector
$\vec{n}$	Normal vector
A	Area or surface
a_{vib}	Vibration amplitude
C	Thermal capacity
C_D	Coefficient of drag
c_p	Specific heat at constant pressure
c_v	Specific heat at constant volume
D_{hyd}	Hydraulic diameter
e_{ω}	Specific rotational energy
e_g	Specific gravitational energy
e_k	Specific kinetic energy
f	Objective function

f'	Normalised objective function
f_D	Darcy friction factor
G	Mass flux
g	Gravitational acceleration
Gs	Gershuni number
H	Height or thickness
h	Specific enthalpy
h_{conv}	Convective heat transfer coefficient
H_P	hybridisation degree (power)
k	Thermal conductivity (conductive heat transfer coefficient)
$K_{\dot{Q}}$	Heat normalization factor (objective function)
K_{contr}	Contraction loss factor
K_{exp}	Expansion loss factor
K_L	Loss coefficient (pressure drop in manifolds)
K_m	Mass normalization factor (objective function)
L	Length
m	Mass
Ma	Mach number
N	Number (of something)
Nu	Nusselt number
P	Perimeter
p	Pressure
P_{rad}	Solar irradiation incident on the upper tank surface

Pr	Prandtl number
Q	Heat
R	Resistance (thermal)
R_{air}	Ideal gas constant of air
Ra	Rayleigh number
Re	Reynolds number
S_{cool}	Activation signal (fuel cooling mode)
S_{rech}	Activation signal (fuel recharge mode)
S_1	Switching signal 1
S_2	Switching signal 2
SCF	Specific fuel consumption
T	Temperature
t	Time
U	Global heat transfer coefficient
u	Specific internal energy
V	Volume
W	Width
W_{int}	Internal work

Greek Symbols

α_{diff}	Thermal diffusivity
α_{rad}	Solar absorptance
β	Volumetric thermal expansion coefficient
δ_{bl}	Boundary layer thickness

ϵ_{rad}	Emissivity
η	Efficiency
η_0	Fin efficiency of a generalised channel
η_a	Fin efficiency of a rectangular geometry
γ	Heat capacity ratio
ν	Kinematic viscosity
Ω_{vib}	Vibration angular frequency
ϕ	Area-averaged dimensionless temperature difference between hot and cold fluids
ρ	Density
σ_{HE}	Heat exchanger porosity
σ_{rad}	Stefan–Boltzmann constant
θ	Local dimensionless temperature difference between hot and cold fluids
ϵ	Thermal effectiveness
ζ	Irreversibility coefficient (pressure drop at the headers)

Superscripts

gene	Gene (for optimisation)
target	Optimisation target

Subscripts

$\dot{Q}$	Heat-related
air	Air
base	Base (surface) of a rectangular channel
branch	Thermal resistance branch (in a parallel framework)

chan	Channel
char	Characteristic
chem	Chemical
Ch	Chebyshev
conv	Convection or convective
cool	Coolant
corr	Corrected
cs	Cross-section or cross-sectional
CV	Control volume
c	Cold
drag	Drag
drain	Drain or draining
dross	Dross
el	Electric/electrical
EM	Electric motor
eng	Engine
eq	Equivalent
FC	Fuel cell
fill	Fill or filling
fin	Fin (of a finned plate)
floor	Floor (of an enclosed space, tank, or channel)
FT	Fuel tank
fuel	Fuel

gear	Gearbox
HE _D	Downstream coolant heat exchanger related (coolant - fuel)
HE _M	Midstream coolant heat exchanger related (coolant - air)
HE _U	Upstream coolant heat exchanger related (heat source - coolant)
header	Header (connecting to the HE)
HE	Heat exchanger
h	Hot
intake	Intake (ram-air)
in	Inlet or input
ISA	International Standard Atmosphere
lam	Laminar
layers	Layer(s) of parallel channels sharing the same fluid
line	Electrical line
max	Maximum
mech	Mechanical
min	Minimum
ML	Minor losses
m	Mass-related
out	Outlet or output
PID	PID signal
plt	Plate
pump	Pump
PW127F	Pratt & Whitney PW127F (engine)

rad	Radiation or radiative
rbw	Regulation band width
recov	Recovery (air)
rec	Recirculating
sat	Saturation
shaft	Shaft (propeller)
stag	Stagnation or total (temperature)
s	Structure (solid part of the HE)
tot	Total
turb	Turbulent
WH	Waste heat
0	Upstream sensor (test rig)
1'	Upstream manifold-header interface (test rig)
1	Section just upstream the HE inlet
2	Section just downstream the HE inlet
3	Section just upstream the HE outlet
4'	Downstream manifold-header interface (test rig)
4	Section just downstream the HE outlet
5	Downstream sensor (test rig)
∞	Infinite distance (undisturbed)
i	Generic i -th element

Acronyms / Abbreviations

e^-	Electron
-------	----------

H^+	Hydrogen ion
H_2	Diatomic hydrogen
HE_D	Downstream heat exchanger (between coolant and fuel)
HE_M	Midstream heat exchanger (between coolant and air)
HE_U	Upstream heat exchanger (between heat source and coolant)
O_2	Diatomic oxygen
A-TMS	Air-based TMS
ACARE	Advisory Council for Aviation Research and Innovation
AF-TMS	Air–fuel TMS
ANN	Artificial Neural Network
ATR	Aerei da Trasporto Regionale (Regional Transport Aircraft)
CAAC	Civil Aviation Administration of China
CFD	Computational (thermo-)Fluid Dynamics (simulations)
CO_2	Carbon dioxide
DLR	Deutsches Zentrum für Luft- und Raumfahrt (German Aerospace Centre)
DoE	Design of Experiments
EAP	Electrified Aircraft Propulsion
EASA	European Union Aviation Safety Agency
ECS	Environmental Control System
F-TMS	Fuel-based TMS
FADEC	Full Authority Digital Engine Control
FLT	First Law of Thermodynamics
GA	Genetic Algorithm

HALE	High Altitude Long Endurance (aircraft)
HT-PEM	High-temperature (PEM) fuel cell
ICE	Internal combustion engine
IR	Infrared Radiation
ISA	International Standard Atmosphere
LCM	Law of Conservation of Mass
Li-ion	Lithium-ion (batteries)
LT-PEM	Low-temperature (PEM) fuel cell
MBSE	Model-Based System Engineering
MCFC	Molten carbonate fuel cell
MEA	More Electric Aircraft
MOGA	Multi-Objective Genetic Algorithm
MT-PEM	Medium-temperature (PEM) fuel cell
NASA	National Aeronautics and Space Administration (US)
NO _x	Nitrogen oxide(s)
PAX	Passengers
PCM	Phase-Change Material
PEM	Proton-Exchange Membrane
PID	Proportional-Integral-Derivative (controller)
RSM	Response Surface Methodology
SAF	Sustainable Aviation Fuel
SDGs	Sustainable Development Goal(s)
SFC	Specific fuel consumption

SI International System of Units

SOFC Solid oxide fuel cell

Tit.-Abs.-Key. Title-Abstract-Keyword

TMS Thermal Management System

TO Topology Optimisation

TRL Technology Readiness Level

UAV Unmanned Aerial Vehicle

Chapter 1

Introduction

The Thermal Management System (TMS) is a critical aircraft subsystem and a key consideration in aerospace design. It ensures the safe and reliable operation of the aircraft [7, 17, 18], maintaining the cabin, aisle, and onboard systems within optimal temperature ranges via the active supply and removal of heat. Traditionally regarded as a secondary system in aircraft design, the TMS has gained increasing prominence in recent years, to the point of influencing the overall feasibility of next-generation aircraft [19–22]. Advances in electronics, avionics, propulsion, and systems electrification have led to a substantial increase in onboard waste heat generation, posing unprecedented challenges to conventional cooling solutions and architectures. These challenges are well documented across all aerospace domains, including fixed-wing transport [23–28], rotorcraft [29], military platforms [30–33], supersonic and hypersonic aircraft [34–36], and even spacecraft [37, 38]. Fig. 1.1 highlights the emerging heat sources driving the recent development of next-generation TMSs across various aerospace domains.

A major factor contributing to the recent increase in aerospace waste heat generation is the growing commitment of civil aviation to decarbonisation and the reduction of its environmental impact, particularly through systems and propulsion electrification [39, 40]. Studies by Davis et al. [41] and Pietzcker et al. [42] highlight the urgency of cutting CO₂ and other pollutant emissions not only in transport but across all industrial sectors, promoting widespread electrification, broader access to carbon-neutral energy sources, and hydrogen as an energy carrier. Global initiatives such as the United Nations' 2030 Agenda support these objectives through the pro-

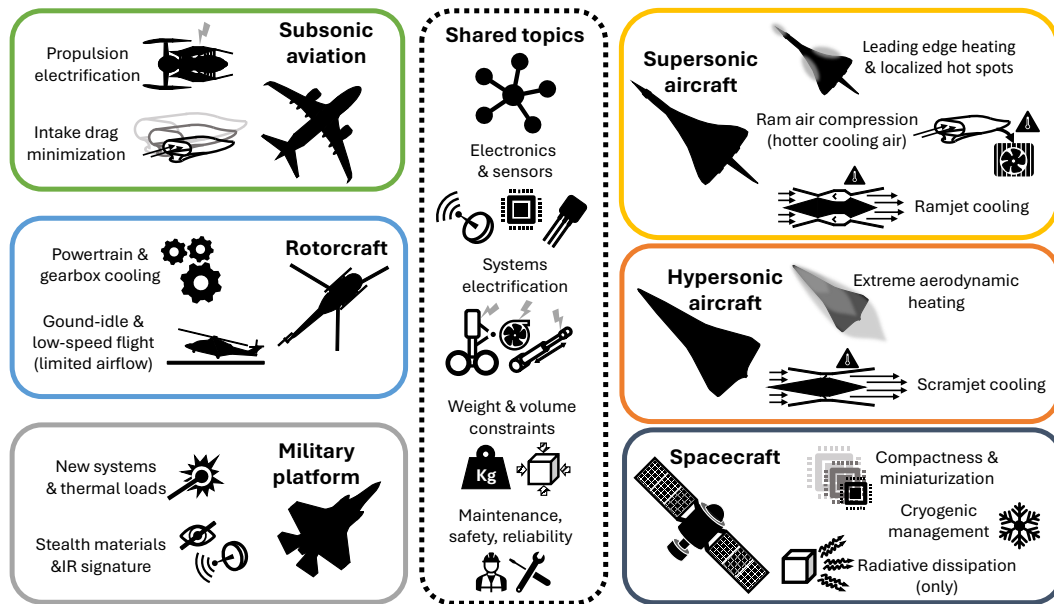


Fig. 1.1 Major factors driving the research and development of improved and innovative Thermal Management Systems in aerospace.

motion of the Sustainable Development Goals (SDGs) [43], which aim to mitigate global warming, minimise environmental impact, and foster a more efficient use of resources, while also encouraging effective reuse and recycling practices in support of a new circular economy.

National and international regulations translate this vision into concrete actions, defining clear sustainability targets and roadmaps for decarbonisation. Comparable commitments can be observed across all major transport domains, from automotive [44, 45] to marine [46–49]. In aerospace, the Flightpath 2050 roadmap by ACARE [50] sets ambitious environmental standards for future aviation in Europe, aligned with the broader Sustainable and Smart Mobility Strategy [51] of the European Green Deal. The roadmap targets a 75% reduction in CO₂ emissions per passenger kilometre and a 90% reduction in NO_x emissions by 2050 relative to state-of-the-art aircraft from the year 2000, together with emission-free taxiing and a 65% reduction in perceived noise. Concurrently, the National Blueprint for Transportation Decarbonization [52] establishes the strategic framework for green aviation research in the US, while the 2022 Strategic Plan by NASA [53] identifies the development of low-environmental-impact aircraft among the agency's key research priorities. For instance, the work by Jansen et al. [54] provides a comprehensive review of the NASA Electrified Aircraft Propulsion (EAP) research, summarising the major programmes,

demonstrators, and technological pathways currently under investigation. Likewise, the 2022 China Civil Aviation Green Development Policy and Action plan [55] and the Green Aviation Ecosystem Guideline [56] define new carbon-emission targets for the Civil Aviation Administration of China (CAAC), promoting Sustainable Aviation Fuel (SAF), electrification, hydrogen-powered aircraft, noise reduction, and airport CO₂ minimisation by 2035 [57].

The industry's commitment to comply with those programmes and the emerging vision for sustainable aircraft is demonstrated by initiatives such as the Clean Aviation Joint Undertaking [58], which brings together major European manufacturers and research centres to develop and demonstrate enabling technologies for low-emission flight. This partnership aims to deliver early market-ready, climate-neutral aircraft configurations by 2035 and fulfil the objectives set by Flightpath 2050.

A wide spectrum of solutions is currently being investigated to support aviation's net-zero transition, including the widespread adoption of sustainable fuels [59, 60], new designs and aerodynamic configurations [61–63], and next-generation turbo-machinery [64–66]. Among these options, propulsion and systems electrification emerge as the most promising yet ambitious pathways towards sustainable flight [67, 39]. A number of challenges still prevent electrified systems from effectively replacing conventional ones, especially in propulsion. In particular, beyond the well-known limitations concerning the energy density and specific energy of batteries and fuel cells, an equally critical issue lies in the effective cooling of these components, aggravated by the stringent safety and redundancy requirements of the sector. Enhanced thermal management is therefore essential for the successful integration of these components and the overall electrification of aircraft powertrains [68, 69].

This dissertation aims to address these emerging challenges in aviation, particularly with respect to the electrification of regional transport aircraft for civil applications. The study provides an overview of current TMS in aerospace, proposing and analysing refined cooling configurations for future hybrid-electric concepts. Ultimately, the main elements of the TMS, the heat exchangers, are modelled and optimised to provide an estimate of the real system performance and masses.

1.1 Batteries, fuel cells and supercapacitors

When considering partial or full electrification of aircraft systems and propulsion, several key elements are required to achieve an effective transition. These include motors and actuators, sensors, electronics, control systems, an electrical distribution line and a structural housing [70]. However, the most critical component for any off-the-grid electrification project is undoubtedly the energy storage device entrusted with supplying the required power to all connected subsystems [71]. At the present day, three primary electric energy storage technologies can be identified: batteries, fuel cells, and supercapacitors [1, 72].

1.1.1 Batteries

Historically, batteries have been the most widely adopted electric energy storage technology, with lithium-ion (Li-ion) batteries emerging as the dominant solution over the past few decades [73]. Their widespread diffusion is favoured by high efficiency, fast load response, effective scalability, and simpler architecture compared to alternative systems like fuel cells [74, 75]. These advantages allowed batteries to dominate the power supply market of portable applications, advancing the maturity of the technology and reducing manufacturing costs, eventually becoming the most common electric energy storage technology used in road transport today [73, 76].

From an engineering perspective, batteries are closed electrochemical systems that store chemical energy and convert it into electrical energy through redox reactions between an anode and a cathode, separated by an electrolyte [77, 1]. During discharge, the electrons flow from the anode to the cathode through an external circuit, while ions migrate through the electrolyte to maintain charge neutrality. In most batteries, this process is also reversible, allowing the cell to be recharged by applying an external voltage that drives the electrochemical reactions in the opposite direction. However, repeated cycling between charge and discharge leads to progressive chemical degradation and undesirable side reactions, which gradually reduce battery performance, ultimately shortening the device's lifespan [78, 79]. Fig. 1.2 illustrates the basic functioning of a galvanic (copper–zinc) cell, used here as a reference model for batteries.

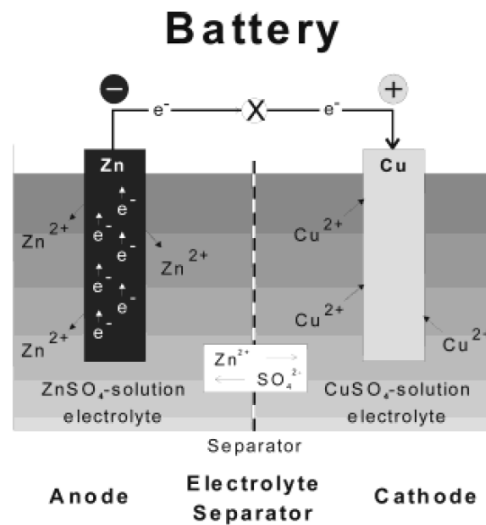


Fig. 1.2 Functional scheme of a copper-zinc battery galvanic cell. Reproduced from Winter and Brodd [1]. © 2004 American Chemical Society.

Batteries exhibit high round-trip efficiencies of approximately 80–90% [80]. Combined with the high efficiency of electric motors and powertrains ($\sim 90\%$), this ensures competitive propulsion system efficiencies compared to conventional architectures. Moreover, the batteries' high round-trip efficiency reduces energy losses and the associated heat generation, effectively reducing the cooling requirements. However, this advantage is generally offset by the narrow optimal operating temperature range of batteries, which typically lies between 15 °C to 35 °C [81]. These relatively low temperatures make dissipating even moderate heat loads challenging, due to the modest temperature difference between the heat source and the external environment. In hot ambient conditions exceeding 35 °C, dedicated refrigeration may even be required, adding system complexity, weight, and energy consumption [82]. An effective thermal management strategy is not only essential to maintain expected operational performance, but also to prolong component lifespan, as overheating accelerates chemical degradation [83]. In extreme cases, inadequate cooling can even trigger a thermal runaway, leading to cell destruction and posing a significant potential hazard to nearby personnel [84].

The primary limitation of current state-of-the-art batteries remains their low gravimetric energy density compared to conventional fuels. For instance, a typical Li-ion battery only stores about 250 Wh kg⁻¹ of energy [73, 85], whereas conventional jet fuels provide specific energies of approximately 12000 Wh kg⁻¹ [86]. This large dis-

crepancy represents a significant drawback for battery-driven aircraft electrification, as the sector's high power-to-weight ratios and stringent safety requirements [87, 88] severely constrain battery-electrification feasibility beyond the ultralight and general aviation classes. In practice, the low specific energy of Li-ion cells leads to increased aircraft mass, which in turn requires higher propulsive power and thus additional energy storage, establishing a self-reinforcing loop that can ultimately offset the environmental benefits of electrification [89]. Recent literature indicates that specific energies sufficient to break this cycle and enable the electrification of regional or larger aircraft are unlikely to be achieved before 2050 [67, 86], assuming current trends in battery capacity growth continue and target values of around 1500 Wh kg^{-1} are reached.

In addition to their limited specific energy, Li-ion cells also exhibit low volumetric energy densities of around 800 Wh L^{-1} [73], which further constrain their suitability for aerospace applications. Their specific power is, instead, competitive, typically exceeding 1000 W kg^{-1} .

1.1.2 Fuel cells

Fuel cells, particularly hydrogen-powered ones, emerge as a plausible alternative to lithium-ion batteries in transport electrification. Historically, their widespread adoption has been hindered in traditional markets, such as the automotive sector, by their high system complexity, elevated manufacturing costs, and the absence of a suitable refuelling infrastructure [90]. Nonetheless, hydrogen-based systems and fuel cells introduce compelling opportunities for the heavy-duty and aerospace sectors, where performance and endurance are prioritised over cost [91]. Moreover, the intrinsic centralisation of aviation around a limited number of regional and international hubs, can mitigate some of the infrastructure-related challenges that have constrained hydrogen usage in other transport domains [91, 92].

Similar to batteries, fuel cells are electrochemical devices that convert chemical energy into electricity through a series of redox reactions. However, unlike batteries, they do not store energy internally; instead, they operate as open systems, continuously generating electricity from a steady supply of reactants, typically hydrogen and oxygen [1]. Since the reactants are supplied externally, fuel cells can usually achieve longer endurance and faster refuelling, although their efficiency and operational

lifespan strongly depend on fuel processing, flow regulation, thermal management, and system integration [90].

Several kinds of fuel cells exist in the literature and industry [93], with the hydrogen-powered Proton Exchange Membrane (PEM) being the most prominent and mature configuration [69, 94]. In PEM systems, the anode oxidises the fuel (H_2), producing electrons (e^-) and protons, while the cathode reduces the oxidant (O_2) and combines it with hydrogen ions (H^+) to form water, while releasing heat as a by-product. Fig. 1.3 illustrates the basic functioning of a typical hydrogen PEM fuel cell.

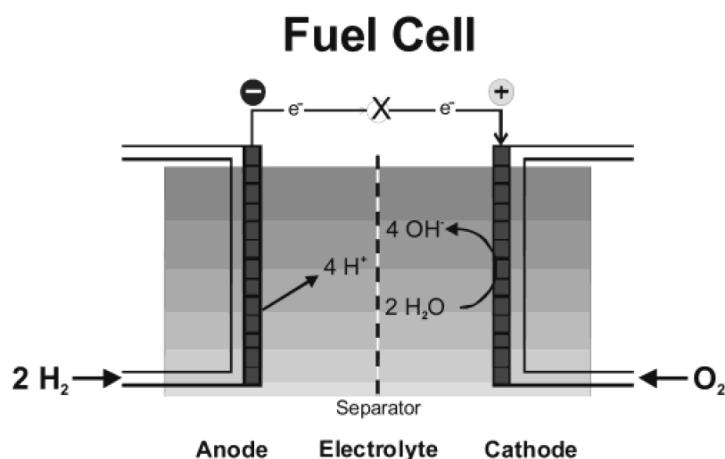


Fig. 1.3 Functional scheme of a hydrogen PEM fuel cell. Reproduced from Winter and Brodd [1]. © 2004 American Chemical Society.

On a theoretical level, hydrogen is an exceptional energy carrier, significantly outperforming Li-ion batteries with a specific energy of approximately $33.000 \text{ Wh kg}^{-1}$ [73], almost three times higher than conventional hydrocarbon jet fuels (12000 Wh kg^{-1}). However, H_2 is also highly volatile, resulting in modest volumetric energy densities and requiring advanced storage solutions. State-of-the-art systems rely either on compressed gaseous storage at extreme pressures up to 700 bar or on liquid storage at cryogenic temperatures of around -253°C [95, 90]. These configurations yield effective volumetric energy densities of about 1300 Wh L^{-1} and 2400 Wh L^{-1} , respectively [95, 96], both significantly lower than the nearly 10.000 Wh L^{-1} of conventional Jet-A fuel [97].

Moreover, the specific energy of the fuel cell stacks is substantially lower than that of pure hydrogen, ranging between 600 Wh kg^{-1} to 1200 Wh kg^{-1} [98]. Com-

plex tank designs incorporating multiple layers of advanced materials are then needed to withstand high H_2 storage pressures, provide thermal insulation, reduce boil-off of liquid hydrogen [99], and minimise gas permeation through the tank walls [100]. These design requirements reduce the available cabin and payload space and introduce additional masses, bringing the overall specific energy of the complete hydrogen–fuel cell system down to a minimum of 500 Wh kg^{-1} to 700 Wh kg^{-1} [101]. Nonetheless, these figures still surpass the specific energy of current battery technologies, with recent efforts targeting system-level, PEM gravimetric energy densities up to 1500 Wh kg^{-1} [102]. The specific power of PEM fuel cells, however, remains slightly lower than that of Li-ion batteries, rarely exceeding 1000 W kg^{-1} [73].

Additionally, fuel cells exhibit very slow responses to changing loads [103, 104], with PEMs typically requiring 10 s or more to reach steady-state conditions [105, 106]. This slow dynamic arises from the direct dependence of the electrical output on the reactant flow rates, which require seconds to stabilise. Moreover, these flows must be gradually and carefully regulated to maintain the proper balance between hydrogen and oxygen and prevent undesired side reactions, localised overheating, or membrane drying, that could damage the cell and significantly reduce its operational lifespan [107, 108].

A major drawback of fuel cell technology is its relatively low efficiency compared to battery systems ($\sim 90\%$), with most hydrogen-powered stacks rarely exceeding 50% efficiency during operation [109, 110, 93]. As a result, a significant portion of the chemical energy from the reactants is lost as waste heat, with roughly 1 kW of heat being generated for every kW of electrical power being produced. This makes effective cooling of high-power devices, such as propulsion systems, particularly challenging [111]. Proper stack thermal management is therefore essential to ensure safe and durable operation, as fuel cells, like batteries, are also susceptible to component degradation and thermal runaway when inadequately cooled [83, 112].

Compared to batteries, fuel cells benefit from higher operating temperatures, which ensure a positive temperature difference between the stack and the external environment, eliminating the need for a refrigeration cycle. However, most fuel cells' operating temperatures remain low compared to conventional combustion-based propulsion, which complicates heat dissipation. For instance, PEM fuel cells can be classified into low-temperature PEMs (LT-PEMs), operating between 60°C to 80°C [110], and high-temperature PEMs (HT-PEMs), working at 120°C to

200 °C [93]. However, both ranges remain significantly lower than the approximately 300 °C typical of internal combustion engine (ICE) blocks [113] and the 750 °C to 1000 °C of state-of-the-art turbine blades and elements [114, 115].

As of late 2025, LT-PEMs remain the primary solution for hydrogen-powered aircraft electrification [116], and the only design currently tested and demonstrated in-flight [117–119]. Despite the intrinsic limitations associated with their lower operating temperatures, LT-PEMs are still favoured over HT-PEMs for their higher technological maturity, improved power density, faster start-up, and robustness [120]. Nonetheless, recent academic and industrial efforts are targeting the advancement of HT-PEM fuel cells [121], with ZeroAvia emerging as a major promoter of HT fuel cells in aviation as a mean to overcome the thermal management limitations of state-of-the-art LT-PEMs [122]. Contemporarily, innovations in PEM membrane materials are driving the development of new medium-temperature PEM fuel cells (MT-PEMs), capable of combining the main benefits of LT configurations while operating between 100 °C to 120 °C with negligible membrane degradation [123–125]. Notably, similar operating temperatures can also be achieved with more conventional LT-PEM assuming higher-than-ambient reactant pressures [109, 93], although, these require significant structural considerations to account for pressure differences and water management.

Alternative fuel cell technologies, such as Molten Carbonate Fuel Cells (MCFCs) and Solid Oxide Fuel Cells (SOFCs), allow stack operation at extreme temperatures of 600 °C to 1,000 °C. These elevated temperatures, comparable to those of turbine components [114, 115], facilitate heat dissipation, but introduce additional challenges with material durability, thermal control, and cold-start management. Combined with long start-up times and high manufacturing costs, these factors currently hinder the practical integration of such technologies into transport applications [93, 83].

1.1.3 Supercapacitors

The last major class of electric energy storage devices currently available on the market is represented by supercapacitors. These systems occupy a smaller market niche than batteries and fuel cells, being primarily employed in applications requiring rapid power delivery, frequent charge–discharge cycling, short-term energy buffering, or power stabilisation [126, 127]. In transport systems, their primary role is to sustain

fast power dynamics, buffer peak power demands, and support energy recovery through regenerative braking, start–stop systems, and power smoothing [128].

From an engineering perspective, supercapacitors, or ultracapacitors, are electrostatic energy storage devices that bridge the gap between conventional capacitors and batteries. Unlike batteries, they primarily store energy through physical rather than chemical processes. However, unlike conventional capacitors, they exploit extremely high surface-to-volume ratios and electrochemical phenomena to accumulate greater amounts of energy through electrostatic charge separation at the electrode–electrolyte interface, where an electric double layer is formed [1]. In some cases, this mechanism can be further enhanced by pseudocapacitance, a chemical phenomenon that leverages fast surface redox reactions to improve cell capacity. However, this effect is largely absent or negligible in most commercial ultracapacitors, including carbon-based types, which remain the predominant solution on the market today [126, 129]. Fig. 1.4 illustrates the basic functioning of a typical carbon-based supercapacitor.

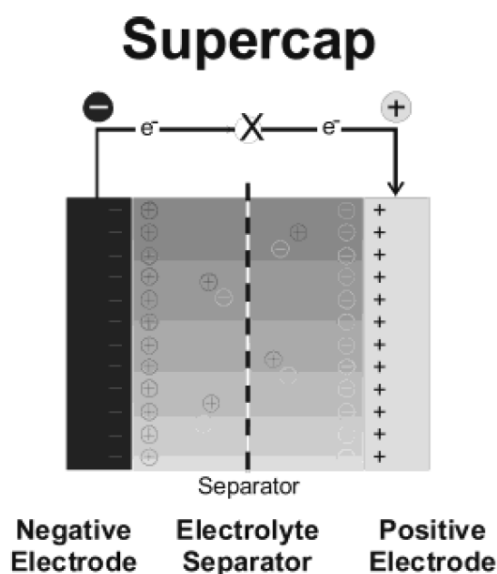


Fig. 1.4 Functional scheme of a carbon-based supercapacitor. Reproduced from Winter and Brodd [1]. © 2004 American Chemical Society.

By relying on physical rather than chemical storage mechanisms, supercapacitors offer exceptional load-response readiness and excellent durability under repeated cycling. They also achieve very high gravimetric power densities, typically around 10.000 W kg^{-1} or more, which far exceed those of both batteries and fuel

cells [129, 130]. Ultracapacitors are highly efficient, achieving round-trip efficiencies of 95% [131, 132] or higher, yet they exhibit noticeable self-discharge, resulting in significant charge losses within a few hours or days [126]. Combined with a relatively small energy capacity and a wide operating temperature range of -40°C to 70°C [133, 132] the thermal management of these components is comparatively straightforward [134].

Despite their advantages in power and durability, supercapacitors are limited by very low specific energies, significantly lower than those of batteries, with typical carbon-based solutions achieving between 5 Wh kg^{-1} to 10 Wh kg^{-1} [126, 132, 130]. These values prevent supercapacitors from being realistically considered as primary power sources for electrified vehicles [135]. Nonetheless, their rapid response introduces significant opportunities for integration with other energy storage systems. For instance, supercapacitors can act as small energy buffers, continuously cycling between charge and discharge to precisely follow the applied load profile, while batteries or fuel cells serve as the primary energy source, supplying the average power demand across a complete utilisation cycle [135, 126].

Concluding this subchapter, Tab. 1.1 summarises the main concepts presented in the previous sections, while Fig. 1.5 provides a qualitative representation of the energy and power density trade-offs between batteries, fuel cells, and supercapacitors [1, 73, 72].

1.2 Research motivations and objectives

Previous sections introduced key topics in current aerospace research and electric energy storage devices, addressing emerging challenges in thermal management, the growing effort towards civil transport decarbonisation, and the technological implications of increased aircraft electrification. These considerations frame the broader context of this dissertation and help clarify its underlying motivations. Building on this foundation, the following paragraphs narrow the focus to formalise the specific scope and main objectives of this research activity.

As anticipated in the introduction of Chap. 1, extensive literature highlights the need to reduce the environmental impact of air transport and meet near-future emission targets and regulatory constraints [41, 59]. Numerous strategies have

Table 1.1 Summary of main state-of-the-art electric energy storage technologies and comparison with conventional jet fuel properties.

Technology	Spec. energy [Whkg ⁻¹]	Energy density [WhL ⁻¹]	Spec. power [Wkg ⁻¹]	Temp. range [°C]
Li-ion batteries	200–300	~800	1 000+	15–35
PEM fuel cells	600–1 200	400–1 000	<1 000	60–80 (LT) 100–120 (MT) 120–200 (HT)
Hydrogen (pure)	~33 000	~1 300 (compr.) ~2 400 (liquid)	–	~20 (compr.) –253 (liquid)
Hydrogen (system)	500–700 (future: 1 500)	–	<1 000	depends on fuel cell
Supercap.	5–10	5–15	10 000+	–40 +70
Jet-A fuel	~12 000	~10 000	–	–

been proposed, including the introduction of sustainable aviation fuels [60], ultra-high-bypass turbofans [65], and disruptive airframe designs [61]. Among these, electrification emerges as the most ambitious pathway towards aviation decarbonisation. However, significant limitations still hinder its large-scale adoption in regional or larger aircraft, particularly regarding the achievement of competitive energy storage solutions through batteries, fuel cells, or supercapacitors [67, 89, 39], as discussed in Sect. 1.1. Fig. 1.6 summarises this research thread, illustrating the key technological enablers supporting aircraft decarbonisation and the electric transition and highlighting their principal features and major drawbacks.

Addressing the inferior gravimetric and volumetric energy densities of current electric energy storage devices remains the leading objective and primary challenge of ongoing research into More Electric Aircraft (MEA) development [39, 136], with mass minimisation being a fundamental requirement of aerospace design [67]. In this context, hydrogen fuel cells have emerged as the most promising technology, offering the best-in-class gravimetric energy density among the three primary candidates. Moreover, their enhanced durability over repeated cycling contributes to improved system reliability, a highly desirable feature given the stringent safety requirements of aviation. These assessments are strongly supported by the literature, with numerous studies advocating for hydrogen as the preferred energy carrier in large-scale aviation

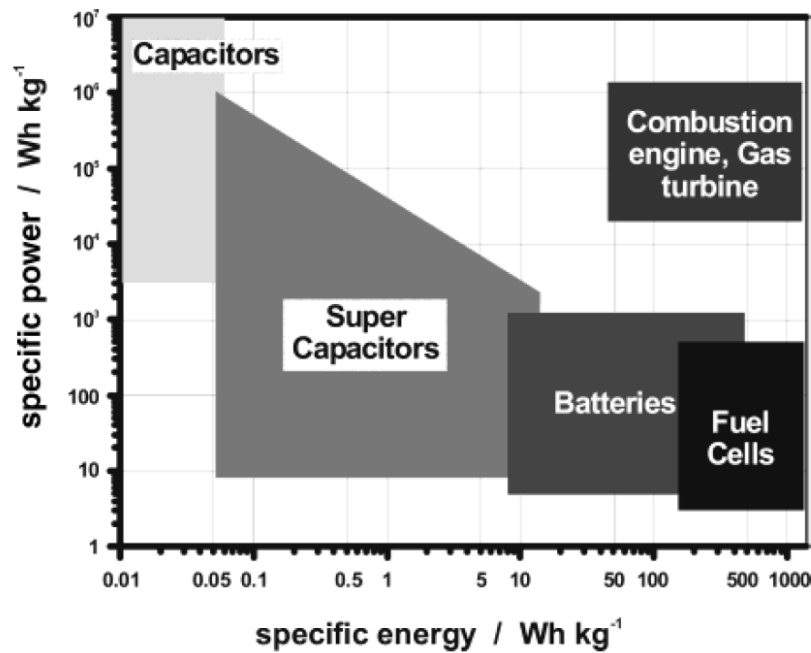


Fig. 1.5 Energy density and power density trade-offs between major state-of-the-art electric energy storage devices and carriers. Reproduced from Winter and Brodd [1]. © 2004 American Chemical Society.

electrification and investigating the airborne implementation of fuel cells [137–139]. Commitment from research institutions and the aerospace industry to develop and demonstrate hydrogen technologies in flight further consolidates this view [140–143], with numerous projects, experimental testbeds, and pre-commercial prototypes already pushing the boundaries of electrified aircraft. Notable examples include the 2008 Boeing Fuel Cell Demonstrator [117], early iterations of the NASA X-57 experimental concept [141], and the HY4 demonstrator by H2FLY (DLR) [118]. Broader initiatives include Airbus’s ZEROe programme [144], the retrofitted ATR-72 demonstrator by former Universal Hydrogen [145, 146], and next-generation hydrogen-electric powertrains from ZeroAvia [147], which also flight-tested a fuel-cell-powered Do228 small regional transport aircraft and commuter [119].

However, despite these advantageous features, and the promising results achieved in academic research and industrial demonstrators, the energy densities of fuel cells and hydrogen systems remain significantly lower than those of conventional jet fuel [138], and still require further developments to fully support a complete transition to electric aircraft propulsion. Nonetheless, immediate action is needed to

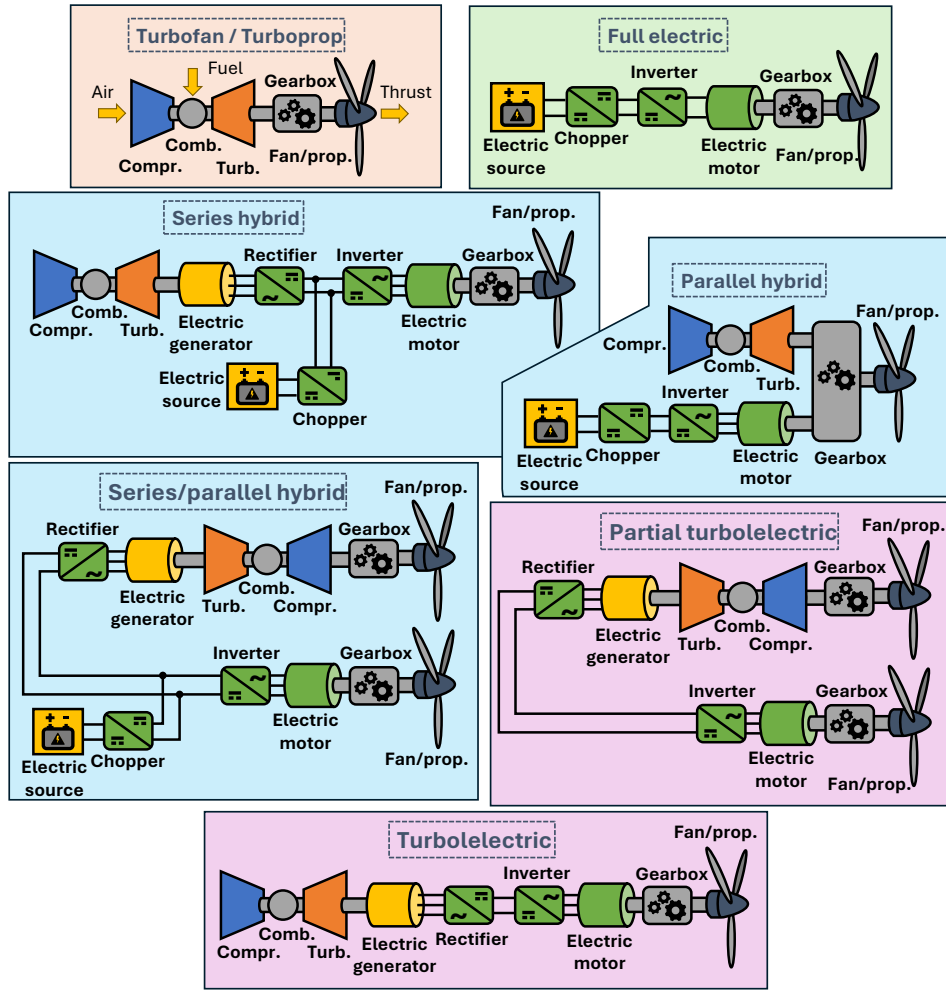


Fig. 1.7 Principal powertrain configurations in aerospace literature, including: conventional, all-electric, hybrid-electric, and turbo-electric propulsion.

to be harnessed while retaining the reliability and lower implementation costs of conventional engines. As of late 2025, a 10-30% hybridisation degree can reasonably be assumed for early MEA concepts in the regional transport market.

Hence, efforts to reduce the environmental impact of near-future aviation can be expected to rely primarily on two complementary approaches: hydrogen–fuel-cell powertrains and hybrid-electric configurations.

Nonetheless, two major drawbacks of fuel cells persist: their limited load response dynamics [103, 104] and relatively low operational efficiency compared to batteries and supercapacitors, which results in substantial waste heat generation [1, 83]. As anticipated in Sect. 1.1.3, integrating fuel cells with more responsive

devices like supercapacitors [135, 126] or batteries [154, 138] is a consolidated strategy to mitigate dynamic limitations, but requires careful system integration. Conversely, managing the thermal loads of fuel cell systems remains a major challenge for propulsion electrification, as the high power demands of most aircraft lead to considerable waste heat [155]. The low operating temperatures of fuel cells further complicate heat rejection, making conventional radiator-based cooling impractical due to the large air intakes required and the associated aerodynamic drag [156].

Numerous studies have addressed this topic [26, 17, 28], confirming the complexity and related challenges, proposing innovative solutions to enhance heat rejection without significantly affecting aircraft mass or aerodynamic drag [157]. Still, a pressing need remains for new and enhanced system-level strategies capable of managing the large heat fluxes of future high-power fuel cell systems efficiently. Most existing works remain at the conceptual stage, while current small-scale demonstrators rarely encounter the severe thermal loads anticipated for full-scale applications, leaving open questions regarding scalability, efficiency, and aerodynamic penalties [7]. Therefore, the development of new and optimised thermal management strategies for next-generation, fuel-cell-powered aircraft remains a key research priority [158].

Accordingly, this dissertation contributes to the ongoing effort towards air transport decarbonisation, by investigating recent advances in aerospace thermal management and proposing innovative concepts and system-level improvements for airborne fuel cell cooling. Additionally, this work seeks to leverage innovations in the design and manufacturing of heat exchangers to refine and optimise key components of the TMS, while maintaining a system-level and application-oriented perspective.

The thesis frames this contribution within a holistic, methodology-driven approach, aimed at demonstrating the effectiveness of the proposed models, design processes, and tools for early-stage system-level analyses and the design of future fuel-cell-powered aircraft.

A dedicated case study is introduced to consolidate and exemplify these objectives. The following subsection presents this case and its specifications, which serve as the reference framework for the remainder of this dissertation.

1.2.1 Case study

A regional transport aircraft is adopted as the reference application for this work. The choice is motivated by two main considerations. First, while the electrification of ultralight and general aviation aircraft has already reached in-flight demonstration, regional transport and commuter platforms remain largely confined to preliminary investigations, with only a few partial retrofits tested to date [146, 119]. As the next step in aircraft weight class, regional transports represent ideal candidates for assessing the scalability of concepts validated on smaller airframes and for developing new solutions that bridge the gap between light aviation and airliners. This explains the growing interest of recent literature in the electrification of regional aircraft [159]. Second, the development of a similar use case aligns with ongoing research initiatives by major industrial stakeholders, including Leonardo S.p.A. [160–162], and supports the relevance and practical applicability of the present study.

Principal representatives of the regional transport aircraft category include the Do228 NXT [2], Saab 340 [163], ATR 42-600 [3], the former Fokker F50 [164], Antonov An-24 and its derivatives [165], ATR 72-600 [166], and Dash 8-400 [4]. Typical characteristics of this class are as follows:

- Passenger capacity ranging from 15 to 90, depending on aircraft size and configuration;
- Twin-propeller layout, typically driven by geared turboshaft engines mounted on each wing;
- Combined shaft power at propellers comprised between 1.0 MW to 7.5 MW, depending on aircraft scale;
- Maximum operational range at full payload between 1000 km to 2000 km;
- Typical mission range of 300 km to 500 km under standard service conditions.

Fig. 1.8 illustrates representative configurations and scales of some major regional and commuter turboprop aircraft.

Larger platforms also exist, accommodating up to 100 passengers or more, including the Embraer 190 [167, 168], Embraer 195 [169], Bombardier CRJ-1000 [170], and the former Fokker 100 [171]. However, these aircraft are significantly heavier

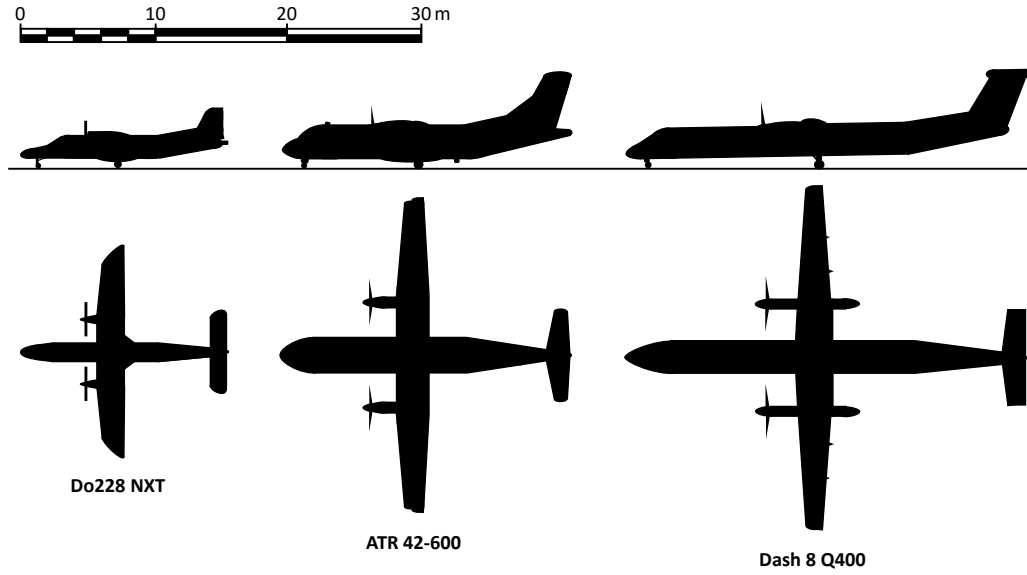


Fig. 1.8 Typical configurations and scales of major regional transport and commuter turboprop aircraft, based on dimensional data from Refs. [2–4].

than the aforementioned turboprops and designed for higher cruise speeds, requiring substantially greater propulsive power and therefore favouring turbofan propulsion over turboprops. As a result, the electrification of such configurations is considerably more challenging.

For the purpose of this study, a generic regional transport aircraft is considered with the following specifications:

- Maximum passenger capacity of approximately 90 passengers (PAX);
- Twin turboprop propulsion mounted under a high-wing configuration;
- Combined shaft power at propellers of approximately 4 MW;
- Maximum operational range at full payload of 1300 km;
- Typical service mission range of 400 km.

These parameters are consistent with those of typical ATR 72 [166] and Dash 8-400 [4] configurations. Accordingly, the PW127F [15] is adopted as the reference propulsion unit for a conventional benchmark design. The performance and specific fuel consumption (SFC) of a single engine are reported in Tab. 1.2.

Table 1.2 PW127F engine performance specifications at representative operating modes, based on data from Pratt & Whitney technical documentation and EASA certification records [15, 16].

Mode Power %	Power [kW]	Fuel flow [kg min ⁻¹]	SFC [g/kWh]	Energy efficiency (%)
Nominal idle (7%)	143	3.06	1282	6.6
Approach (30%)	615	5.15	502	16.8
Max cruise (78%)	1590	8.28	312	27.0
Max climb (80%)	1635	8.38	308	27.4
Max continuous (90%)	1846	9.22	300	28.1
Take-off (100%)	2050	9.90	290	29.1

The electrification of this reference aircraft defines the baseline case for this dissertation. Consistent with previous considerations on the feasibility of fully electric propulsion and in line with current trends in the literature, a parallel hybrid-electric architecture is assumed. A target hybridisation degree of 20% is selected to represent a realistic near-term implementation scenario, supplied by a PEM fuel cell system fed with liquid hydrogen. Next-generation MT-PEM fuel cells are considered for this purpose, offering improved robustness and efficiency while operating at moderately higher temperatures than conventional PEMs. Alternatively, regular LT-PEM fuel cells operating with higher-than-ambient fluid supply pressures could also be assumed [109, 93]. Fig. 1.9 illustrates the schematic of the adopted configuration.

A maximum propulsive power comparable to that of the original configuration is assumed. The corresponding mechanical power request at each propeller shaft ($\dot{W}_{\text{shaft}}$) is defined according to Eq. (1.1):

$$\dot{W}_{\text{shaft}} = \dot{W}_{\text{tot}} / N_{\text{shaft}} \quad (1.1)$$

where $\dot{W}_{\text{tot}}$ denotes the total propulsive power required by the aircraft and N_{shaft} the number of propeller shafts, each associated with a dedicated propulsion unit.

Each shaft is powered by a conventional turbine engine and an electric motor. The hybridisation degree H_p defines the mechanical power split between these two

demand at the fuel cell stack ($\dot{W}_{FC,el}$):

$$\dot{W}_{FC,el} = \frac{N_{EM} \cdot \dot{W}_{EM,mech}}{\eta_{EM} \eta_{line}} \quad (1.3)$$

where N_{EM} is the number of electric motors, here equal to the number of propeller shafts (N_{shaft}), and η_{EM} and η_{line} are the respective efficiencies of the electric motors and the electrical line. The electric power supplied to a single motor is instead $\dot{W}_{EM,el} = \dot{W}_{EM,mech} / \eta_{EM}$.

Ultimately, the fuel cell's own inefficiencies result in waste heat generation, with Eq. (1.4) returning the corresponding heat flow rate produced by the stack ($\dot{Q}_{FC}$):

$$\dot{Q}_{FC} = \dot{W}_{FC,el} \frac{1 - \eta_{FC}}{\eta_{FC}} \quad (1.4)$$

where η_{FC} is the electrical efficiency of the fuel cell stack. Fig. 1.10 details the presented power flow for the proposed parallel hybrid-electric configuration.

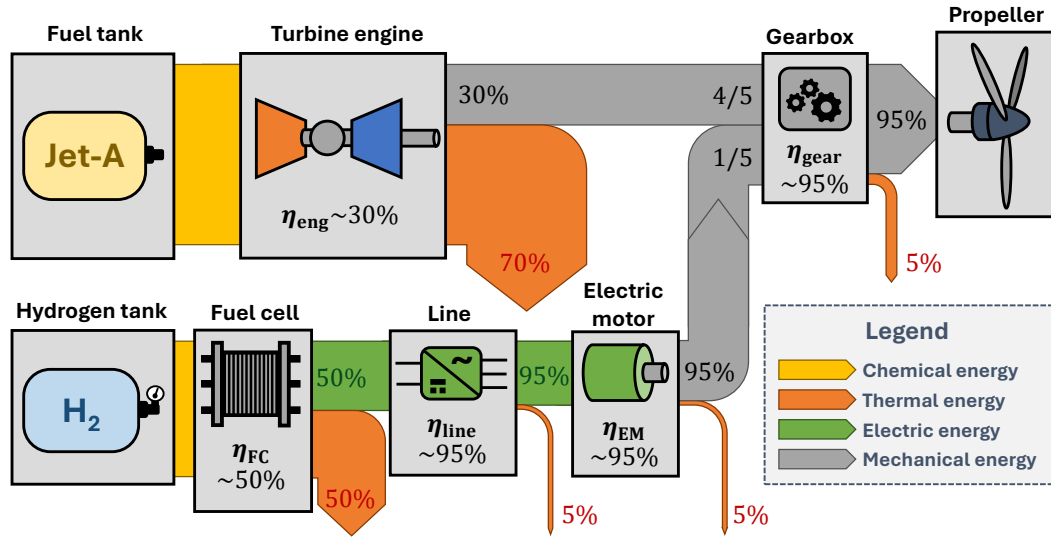


Fig. 1.10 Power flow schematics for the reference parallel hybrid-electric powertrain. Arrow widths suggest relative power proportions but are not to scale.

The effective management of this final waste heat ($\dot{Q}_{FC}$) constitutes the primary objective of this dissertation. The goal is to ensure that the fuel cell system remains

within its optimal operating temperature range and that the generated heat is efficiently dissipated. The resulting TMS must achieve this with minimal impact on aircraft performance by reducing component mass and limiting induced aerodynamic drag. Therefore, assuming nominal efficiencies of 50% for the fuel cell, 95% for the electric motors, 95% for the electrical line, and 95% for the gearbox, the system powers and heat flows reported in Tab. 1.3 can be regarded as the principal performance requirements for the investigated TMS and for this dissertation.

Table 1.3 Peak mechanical, electrical, and thermal powers for the reference hybrid-electric use case (nominal efficiencies: $\eta_{FC} = 0.50$, $\eta_{EM} = 0.95$, $\eta_{line} = 0.95$, $\eta_{gear} = 0.95$).

System	Power reference	Energy form	Notation	Power [kW]
Aircraft	total	mechanical	$\dot{W}_{tot}$	4000.00
Propeller	per shaft	mechanical	$\dot{W}_{shaft}$	2000.00
Engine	per shaft	mechanical	$\dot{W}_{eng}$	1684.21
Electric motor	per shaft, supplied	mechanical	$\dot{W}_{EM,mech}$	421.05
	per shaft, received	electrical	$\dot{W}_{EM,el}$	443.21
	total, received	electrical	$n_{EM}\dot{W}_{EM,el}$	886.43
Fuel cell	total, supplied	electrical	$\dot{W}_{FC,el}$	933.08
	total, received	chemical	$\dot{W}_{FC,chem}$	1866.16
	total, lost	thermal	$\dot{Q}_{FC}$	933.08

1.3 Original contribution of the research activity

The following points summarise the original contributions of this dissertation, which extend and integrate previously published results from the relevant scientific literature.

- **Systematic review of thermal management practices in aerospace.** The dissertation illustrates and discusses recent key trends in the scientific literature on thermal management, electrification, and associated technical solutions. It provides a comprehensive evaluation of Thermal Management System design in aerospace, particularly for the cooling of onboard heat sources, and analyses the primary heat extraction and transport methods for fuel cells. It then

classifies the principal aircraft heat sinks, examining their features, limitations, associated heat-rejection mechanisms, and applications.

- **Analysis of primary thermal management solutions for air- and fuel-based cooling in aerospace.** The discussion addresses air- and fuel-based cooling as primary candidates for the effective thermal management of fuel cells in aerospace applications. Based on Model-Based Systems Engineering (MBSE) methodologies, it derives the main requirements of a cooling solution employing either or both heat sinks. These requirements are translated into system-level compliance criteria for the design verification and validation of the proposed solutions, and used to define their functional and logical architecture, including the identification and characterisation of key subsystems, components, and relevant parameters.
- **System-level modelling and simulation of a fuel-based Thermal Management System for fuel cell cooling in aerospace.** The analysis assesses the feasibility of fuel-based Thermal Management Systems (F-TMS) as an innovative solution for fuel cell thermal control in hybrid-electric regional transport aircraft. The methodology is developed from a dynamic, system-level perspective, comparing single-tank recirculating-parallel and recirculating-series configurations, and demonstrating up to a 35% improvement in thermal endurance for the series layout.
- The dissertation advances the dynamic modelling of aircraft fuel-tank thermal behaviour through the integration of computationally efficient passive heat-rejection models from the skin-cooling literature, achieving improvements of 62% or more in overall system thermal endurance compared to adiabatic baseline models. The adopted modelling assumptions are validated against literature data, corroborating recent findings on the non-negligible effects of passive heat rejection at the system scale, even under high levels of modelling abstraction.
- The study further extends the wing tank heat transfer literature through the explicit inclusion of internal fuel-side convection, non-negligible thermal inertia, and transient temperature evolution, resulting in improved system-level performance compared to existing steady-state frameworks. The analysis investigates and compares different fuel convection modelling assumptions, in-

cluding adiabatic tank walls, purely conductive heat transfer, buoyancy-driven convection, vibration-induced convection, approximate forced convection, and idealised heat transfer conditions.

- Finally, the work assesses F-TMS performance across a range of mission profiles varying in altitude, range, and alternate destination. It identifies the key dependencies governing the most demanding operating scenarios and determines the maximum achievable level of fuel-cell electrification supported by a fuel-based cooling architecture.
- **System-level modelling and simulation of an integrated air–fuel Thermal Management System for fuel cell cooling in aerospace.** The analysis develops and evaluates a novel integrated air–fuel Thermal Management System (AF-TMS) architecture for hybrid-electric regional transport aircraft, combining conventional ram-air cooling with fuel-based thermal buffering. The framework extends the baseline F-TMS formulation into a generalised architecture capable of representing air-based (A-TMS), fuel-based (F-TMS), and integrated AF-TMS configurations.
- The dissertation advances current research on aircraft thermal management modelling by integrating air- and fuel-based cooling practices within a unified dynamic simulation framework. A dual-mode fuel control strategy is introduced, enabling fuel-based cooling during high thermal load conditions while promoting fuel thermal recovery during lower-demand phases to restore the system’s thermal inertia. Compared with baseline control strategies, the proposed approach reduces peak heat-source temperatures by 0.4 °C and decreases the maximum tank fuel temperature by 8.8 °C.
- The analysis further explores the role of fuel thermal inertia in supporting ram-air cooling during thermally critical mission phases, demonstrating that fuel-assisted thermal buffering can significantly reduce sizing requirements for the air system. Preliminary optimisation of a two-intake architecture indicates that fuel buffering reduces the required inlet area by approximately 30%.
- As per the F-TMS configuration, the work assesses the performance of the A-TMS and AF-TMS across a range of mission profiles varying in altitude, range, and alternate destination, to identify and characterise the most demanding operating scenarios.

- Finally, the work evaluates the system-level trade-offs associated with AF-TMS integration, achieving up to a 17.4% reduction in total system mass compared to conventional A-TMS configurations, at the expense of a modest increase in fuel pumping power consumption. The associated 30% air intake area reduction translates instead into ram-induced drag savings exceeding 100 kW, highlighting the potential of fuel-assisted cooling architectures to improve the efficiency of hybrid-electric aircraft thermal management systems.
- **Numerical modelling and simulation of compact heat exchangers compatible with additive manufacturing.** The study presents and validates a robust and computationally efficient modelling methodology for the simulation of compact cross-flow heat exchangers, suitable for additively manufactured geometries. It develops an integrated framework combining thermal, hydraulic, and gravimetric correlations within a unified numerical environment, solving the thermo-hydraulic problem using a reduced set of input variables: inlet temperatures, outlet pressures, and mass flow rates.
- The dissertation extends existing ϕ -based modelling approaches to a broader range of heat exchanger geometries and configurations, including arrangements with circular channels and uneven numbers of layers between the two fluid streams. Additionally, the study introduces and validates the thermal length definition proposed by da Silva et al. [11] within a plate-based modelling framework based on the averaged dimensionless temperature difference ϕ , demonstrating consistency across alternative modelling formulations.
- The work also advances the modelling of additively manufactured heat exchangers by addressing hydraulic prediction discrepancies with experimental data. Two corrective formulations are proposed: a geometry-based correction of the hydraulic diameter that accounts for AM deformations, and a conservative empirical uncertainty factor calibrated against experimental measurements from the literature, resulting in reduced systematic underestimation of the pressure drops.
- The numerical implementation of the framework is also carefully designed to ensure robustness and computational stability, enabling the seamless integration of the tool into automated optimisation workflows and additive manufacturing design pipelines.

- **Numerical optimisation of compact heat exchanger geometries compatible with additive manufacturing.** The study develops a heat exchanger design optimisation framework based on validated simulation models tailored to applied engineering design. It introduces corrective terms within the objective function formulations aimed at ensuring convergence towards application-specific requirements. Moreover, the methodology further advances existing heat exchanger optimisation methods by enabling both the refinement of internal geometric features and the continuous adjustment of the main geometric dimensions within a predefined design space.
- The optimisation framework is validated against benchmark geometries from the literature, demonstrating significant performance improvements, including weight reductions of up to 55% and heat transfer enhancements of up to 35%, with additional mass savings of approximately 11% when rectangular channels are adopted instead of circular ones. Beyond validation, the study presents a systematic characterisation of Pareto-optimal design trends, showing that global dimensions are reduced to minimise mass and increased to enhance heat transfer, while internal features exhibit opposing behaviour, with larger cavities favouring mass reduction and smaller features improving thermal effectiveness. Fin widths and wall thicknesses are also addressed and constrained according to additive manufacturing limitations, highlighting manufacturing-driven bounds on optimal design.
- Ultimately, the dissertation implements an early integration of the proposed component-level design optimisation within the system-oriented thermal management analysis. A combined simulation loop is developed, coupling the dynamic system model with the optimised geometries generated by the optimisation framework. The reference air heat exchangers of the A-TMS and AF-TMS configurations are refined, achieving mass reductions between 42–78.5%, depending on the design assumptions. In contrast, inconsistencies arising from overly optimistic early assumptions at the fuel heat exchanger are identified and resolved, underscoring the importance of integrated system- and component-level assessments. The resulting co-simulation approach demonstrates improved system performance compared with initial assumptions and enables a second iteration of air intake refinement, resulting in approximately 9–17% reductions in intake area and the associated aerodynamic drag.

1.4 Work organisation

The following chapters of this dissertation are structured as follows:

- **Chapter 2** introduces fundamental knowledge on major topics and recent advances in thermal management for the aerospace sector. The chapter analyses emerging trends in the literature, presenting the state of the art in airborne Thermal Management Systems for the extraction, transport, and dissipation of heat. The principal designs are detailed for air, fuel, and alternative heat sinks, illustrating the main benefits and drawbacks associated with each solution. Relevant applications, use cases, and demonstrators are provided, where applicable, along with relevant studies and methodologies.
- **Chapter 3** presents the complete system-level modelling and simulation of a fuel-based TMS, applied to the cooling of a fuel-cell-powered electrified propulsion system. The analysis addresses relevant mass and temperature dynamics for the main heat source (the fuel cell), coolant loop, and fuel system, particularly the tanks, comparing parallel and series heat exchanger recirculation strategies for a single-tank F-TMS layout. The study models and validates the dominant heat transfer phenomena at the fuel tank–environment interface, comparing adiabatic, purely conductive, buoyancy-driven, and vibration-induced heat transfer mechanisms. Subsequently, a detailed investigation of the effects of altitude, range, and alternate flight conditions on mission performance identifies the most challenging scenarios for F-TMS operation, and determines the associated maximum degree of electrification thermally supported by the system. Final considerations on emerging research directions and preliminary considerations for F-TMS integration with traditional air-based cooling architectures are also presented.
- **Chapter 4** extends the modelling rationale of Chap. 3 to the analysis of conventional air-based TMS and combines them into a new integrated air–fuel TMS framework. The complete system-level modelling and simulation of the A- and AF-TMS is addressed and applied to the cooling of a fuel-cell-powered 20% hybrid-electric regional transport aircraft. The performance of the two solutions is assessed and compared in terms of relevant temperature dynamics for the heat source and all relevant heat exchangers and components belonging

to the coolant loop, air supply, and fuel system. The study addresses the preliminary modelling and sizing of the air intakes to provide the required cooling airflow. The performance of both TMSs is then assessed across varying mission altitudes, ranges, and alternate flight conditions to identify critical profile segments and scenarios, and derive the minimum intake areas required to ensure safe system operation across the complete simulation envelope. Ultimately, the A- and AF-TMS are compared in terms of added system mass, parasitic pumping power, and induced intake aerodynamic drag. Some considerations are also introduced regarding emerging research directions and future improvements to the presented modelling approach.

- **Chapter 5** addresses the numerical modelling of heat exchangers, developing a robust simulation framework for the early design of compact cross-flow configurations compatible with additive manufacturing. The study formulates an integrated methodology that captures the thermal, hydraulic, and gravimetric behaviour of the components, prioritising physics-based relationships over empirical correlations to preserve modelling flexibility. A lumped thermal formulation based on the averaged dimensionless temperature difference ϕ is implemented to enable rapid solving while maintaining accuracy and adaptability. The study details the numerical implementation and integration of each physical domain, orchestrating their execution in a unified simulation environment to fully characterise the component from a limited set of thermo-hydraulic inputs. The thermal and hydraulic methodologies are subsequently validated against experimental datasets from the literature. Numerical discrepancies in experimental pressure-drop predictions are later addressed through alternative corrective strategies accounting for undesired droplet formation in AM components, resulting in significantly improved accuracy. Finally, the chapter acknowledges the current limitations of the adopted modelling assumptions and outlines potential research directions, including the extension to additional heat exchanger geometries, the introduction of preliminary structural considerations, and the integration with applied engineering workflows and tools.
- **Chapter 6** extends the modelling framework of Chap. 5, addressing the design optimisation of heat exchangers through a dedicated genetic algorithm procedure that leverages the previously validated simulation model to refine the

component's main geometric parameters. The methodology defines and details all key optimisation elements, including genes, boundary conditions, objective functions, and algorithm hyper-parameters. Moreover, the approach is further enhanced through a series of corrective formulations aimed at ensuring consistent geometry generation, supporting both single- and multi-objective optimisation, and promoting convergence towards application-driven requirements, such as meeting a prescribed heat transfer target or remaining within allowable pressure-drop thresholds. The optimised designs are then exported through the definition of dedicated thermal effectiveness response surfaces, which are subsequently integrated into the Simulink dynamic model of Chap. 4. The approach is tested and verified against a benchmark geometry from the literature, yielding simultaneous mass minimisation and heat transfer maximisation, while also extracting key geometrical trends from the non-dominated solutions of the Pareto front. Finally, the optimisation routine is applied to the reference heat exchangers of the A-TMS and AF-TMS configurations, and the resulting geometries are integrated into the dynamic simulation model, which is then rerun to assess system-level performance with the updated designs.

Chapter 2

Thermal management in aerospace

As stated in Chap. 1, the Thermal Management System is a key element in supporting safe and enduring aircraft operation. Its primary task is to maintain the crew, passengers, and onboard systems within their nominal temperature ranges by extracting, distributing, and dissipating heat. Through these functions, the TMS supports high system efficiency and reliability, prolonging the expected lifespan of critical components, and ensuring adequate thermal protection and comfort for the crew and passengers [7, 17, 18].

At the dawn of aviation, when piston engines represented the only reliable source of propulsion, the primary focus of TMSs was the effective cooling of the power units, including the engine block, pistons, and oil [172]. The substantial amount of waste heat released by combustion demanded careful sizing of cooling architectures, driving the parallel development of both air- and liquid-cooled configurations [173]. Accordingly, radiator performance maximisation, intake duct optimisation, and cowl geometry refinement became central themes in aircraft design, particularly during the later stages of the Schneider Cup seaplane-racing era [174, 175] and in the subsequent development of high-performance wartime piston aircraft in the 1930s and 1940s [176, 177]. In both instances, the primary objective remained to achieve sufficient heat rejection while minimising aerodynamic penalties, as cooling systems often constituted decisive factors in determining effective aircraft performance and endurance [178].

The introduction and subsequent widespread adoption of the jet engine in the post-war period redefined the nature of aircraft thermal management, as a larger

fraction of the combustion heat was now directly expelled through the exhaust stream [179]. Accordingly, the primary focus of TMSs shifted from system-level engine cooling to the localised protection of turbine components from combusted gases reaching temperatures of 1000 °C to 1500 °C [115, 180, 181]. The development of advanced high-temperature materials and superalloys was pivotal to addressing this challenge [182, 183]. At the same time, increasingly sophisticated internal blade geometries and advances in film-cooling design further enhanced thermal protection by exploiting relatively cold bleed air extracted from the compressor stage [184–186]. As a result, large-scale heat rejection at the engine level became less critical, with heat exchangers being employed primarily to cool lubricants and hydraulic fluids and to support modern Environmental Control Systems (ECSs) in maintaining cabin and avionics temperatures even at high altitudes [187, 7].

Subsequent developments in avionics, electronics, and radar technologies progressively led to a renewed increase in system-level waste heat generation, reintroducing the need for enhanced heat rejection and more substantial cooling capacity. This trend first emerged in military aviation, where the integration of high-power radars and advanced electronic systems demanded improved thermal management strategies [188]. Early contributions on this topic include the works of Hudson and Levin [189] and Harrison et al. [190], which document the innovative approaches developed during the 1980s to address the growing thermal constraints of 4th-generation fighter aircraft such as the F-15, F-16, and F/A-18. Petkus and Gallington [191], instead, provide an overview of the primary cooling technologies suitable for the thermal management of the avionics and power plant in a new High Altitude Long Endurance aircraft (HALE) concept. Eventually, the demand for more advanced TMSs intensified even further as onboard electrical loads continued to rise, driven by increasing system complexity, more demanding performance requirements, and higher power densities [192, 193]. This trend persisted into the development of recent military platforms, including the F-22 and F-35 [30, 194], and is expected to become even more critical for upcoming 6th-generation fighter aircraft [195, 33].

Over time, similar challenges also extended to civil aviation, driven by the widespread adoption of advanced electronics and the progressive electrification of aircraft components and subsystems [196]. The Boeing 787, Airbus A380, and Airbus A350 are notable examples in which next-generation avionics and more-electric architectures substantially increased waste heat generation, necessitating the implementation of advanced closed-loop vapour cycles to regulate the temperatures

of galley equipment and avionic bays [7]. As in military aviation, this trend continues to this day, and is expected to intensify in future platforms, as increasingly sophisticated electronic systems and growing electrification introduce unprecedented thermal challenges and stricter operating temperature windows. As discussed in Sect. 1.1, the projected integration of electric energy storage devices, such as batteries [68, 197] and fuel cells [69, 198], poses significant challenges to current TMS capabilities [28, 156], especially for configurations involving propulsion electrification of any kind. Fuel cells, in particular, present notable integration difficulties due to their substantial low-grade waste heat generation, far exceeding typical values associated with traditional aircraft heat sources such as electronics [199] or ECS packs [187]. Accordingly, the effective design of next-generation TMSs has become a crucial enabler of modern aerospace research and civil aviation decarbonisation [17].

Parallel to system-level cooling, current research on TMSs also spans several other areas that are worth mentioning. These include system [200] and surface cooling [201] for supersonic aircraft, infrared (IR) signature mitigation [202] and advanced component thermal control [203] for military platforms, radiation-based cooling [204] and cryogenic management [205] for spacecraft, and advanced engine and airframe thermal protection for future reusable hypersonic vehicles [204]. These themes are briefly summarised in Fig. 1.1 from Chap. 1, and collectively represent the major technological drivers of recent advances in TMS research.

The following sections present a structured state-of-the-art analysis of aviation thermal management technologies, with a specific focus on electric components' heat dissipation and its enabling solutions. First, the methodological approach adopted for the literature review is outlined. Subsequently, the primary cooling strategies are identified and classified according to their defining characteristics, and the most relevant architectures within each category are discussed. Ultimately, a preliminary assessment of the feasibility of these systems is conducted with respect to the reference use case presented in Sect. 1.2.1.

2.1 State of the art analysis

A literature survey is first conducted to identify relevant publications and research trends on TMSs and cooling solutions, using the Scopus-indexed abstract and citation database. To this end, a search string consistent with the criteria listed in Tab. 2.1 is

defined and applied to retrieve the complete body of literature on aerospace thermal management, temperature control, and cooling technologies.

Table 2.1 Scopus research string criteria for TMS literature analysis

Criteria	Specification	Purpose
Tit.-Abs.-Key.	"thermal management" OR "temperature control" OR "cool*"	Target key functions of Thermal Management Systems
Tit.-Abs.-Key.	"aircraft" OR "airplane" OR "aeroplane" OR "aviation" OR "aerospace"	Target aerospace applications
Tit.-Abs.-Key.	"heat sink" OR "heat absorb*" OR "heat exchang*" OR "heat reject*" OR "heat dissip*"	Focus on heat transfer management processes
Tit.-Abs.-Key.	AND NOT ("manufacturing" OR "machining" OR "thermomechanical treatment" OR "forming")	Avoid manufacturing- or production-focused studies
Subject area	Engineering	Target engineering applications
Language	English	/
Database	Scopus	/

where Tit.-Abs.-Key. denotes the Title-Abstract-Keyword criteria for the search query.

As of late 2025, the applied search string returns a total of 1 629 publications, indicating a substantial and well-established body of literature confirming the relevance of thermal management technologies in aerospace. Fig. 2.1 illustrates the temporal distribution of these publications, showing an early concentration of studies in the mid-1980s, followed by a sustained rise in research output starting in the 1990s and continuing to this day.

A comparative literature analysis is also conducted for the research string listed in Tab. 2.2, addressing the full body of literature on aerospace electrification and its principal energy storage technologies.

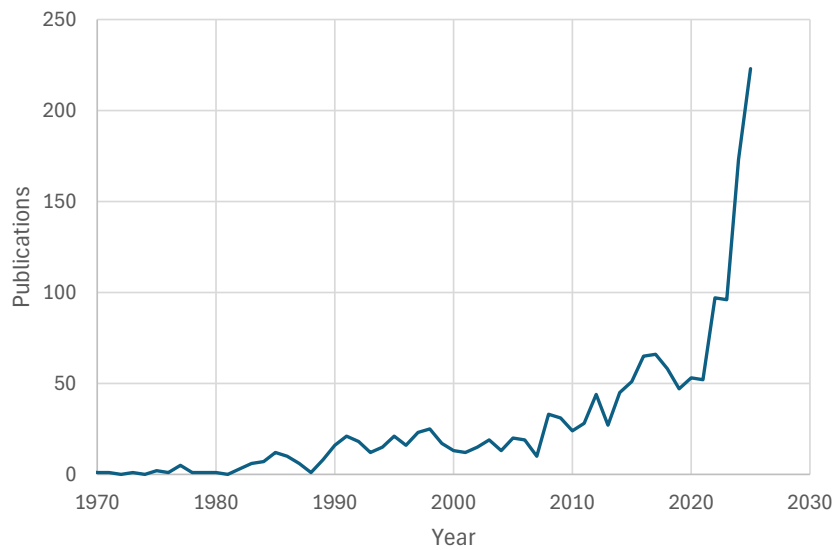


Fig. 2.1 Yearly publications on the topic of Thermal Management Systems in aerospace. Data retrieved from Scopus. ©2025 Elsevier B.V.

Ultimately, a supplementary command string is derived from that of Tab. 2.2 by introducing a fourth search criterion:

- Tit.-Abs.-Key.: "decarbonization" OR "carbon footprint" OR "emission*" OR "environmental impact"

This additional refinement narrows the previous literature survey to focus exclusively on studies that address electrification in the context of environmental considerations, decarbonisation and emission reduction.

As of late 2025, the literature survey based on the search string reported in Tab. 2.2 yields a total of 3872 publications, whereas the refined, decarbonisation-focused analysis comprises 745 studies. Fig. 2.2 illustrates the temporal distribution of publications retrieved using both strings, comparing them with the results obtained from the search line of Tab. 2.1.

Fig. 2.2 confirms the steady growth in yearly publications on TMSs, electrification, and decarbonisation, highlighting these as three key topics in current aerospace research. Compared to the previous analysis of publication distribution, studies on aerospace electrification appear to slightly precede those on thermal management, with notable publication activity emerging prior to the 1990s. Although not directly quantifiable, the data suggest that the publication trends for TMS generally follow

Table 2.2 Scopus research string criteria for electrification literature analysis

Criteria	Specification	Purpose
Tit.-Abs.-Key.	"electrification" OR "electrified" OR "electric"	Target research on electrification
Tit.-Abs.-Key.	"aircraft" OR "airplane" OR "aeroplane" OR "aviation" OR "aerospace"	Target aerospace applications
Tit.-Abs.-Key.	"battery" OR "batteries" OR "fuel cell" OR "fuel cells" OR "supercap*"	Focus on key energy storage technologies
Subject area	Engineering	Target engineering applications
Language	English	/
Database	Scopus	/

those of aircraft electrification with an estimated lag of 5 to 10 years, likely reflecting how new developments in system electrification generate subsequent demand for enhanced cooling solutions. This interpretation is consistent with the role of TMS as pivotal yet auxiliary aircraft systems, typically developed and integrated to support the functioning of other devices rather than preceding them.

In contrast, the refined, decarbonisation-focused analysis shows few publications prior to the 2000s, followed by moderate activity over the next 15 years. From 2015 onward, a sharp increase in publication numbers is observed. This surge is likely associated with the introduction and international adoption of the United Nations 2030 Agenda for Sustainable Development that same year, which emphasized decarbonisation and environmental impact considerations across all sectors, including aviation. Nonetheless, decarbonisation-oriented aircraft electrification and TMS development have both emerged as increasingly relevant themes since 2020.

A collection of notable literature reviews provides structured insights into the major findings, dominant research themes, and remaining gaps within the broad publication pool retrieved through the search string of Tab. 2.1. The review by Van Heerden et al. [7] offers a particularly comprehensive overview of the principal topics addressed in recent literature, outlining the historical background of each solution and its primary applications. The study identifies five fundamental elements of

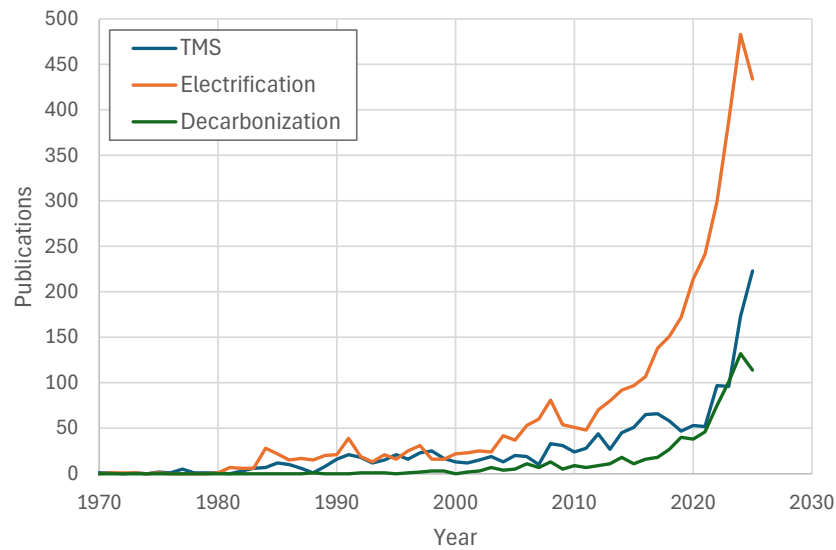


Fig. 2.2 Yearly publications on the topics of Thermal Management Systems, electrification, and decarbonisation-focused electrification in aerospace. Data retrieved from Scopus. ©2025 Elsevier B.V.

thermal management systems: heat sources (although these are not part of the actual TMS), heat acquisition mechanisms, thermal transport, heat rejection technologies, and heat sinks, introducing the main representatives of each category alongside their defining characteristics. Of particular relevance, the study highlights emerging cooling challenges posed by aircraft electrification, ECS development, supersonic thermal management, and future ultra-high bypass geared turbofans. A consistent perspective is also found in the review by Wang et al. [206], which examines recent developments in energy-optimised aerospace vehicles in China, including an in-depth analysis of advancements for spacecraft applications. Acknowledging the substantial thermal challenges associated with more-electric and all-electric aircraft concepts, the study ultimately advocates for expanded research on phase-change cooling approaches, the advancement of fuel-based thermal-management systems (F-TMS) in aviation, and the urgent definition of dedicated fuel-utilisation scheduling strategies.

The review by Li et al. [8] provides a focused analysis of fuel-based TMS technologies, outlining state-of-the-art strategies for fuel flow-rate control and describing the principal architectures employed for fuel-driven cooling. Complementarily, the study by Jia et al. [207] examines the key chemical processes associated with jet-fuel

heating, primarily oxidation, deposition, and pyrolysis, discussing their implications on fuel-based cooling of both airframe and engines in supersonic aircraft.

Coutinho et al. [17] and Asli et al. [28] also provide comprehensive assessments of recent advances in aerospace TMS technologies, specifically addressing solutions tailored to the development of fully electric and hybrid-electric aircraft. Ref. [17] examines the principal thermal management approaches with a strong emphasis on system-level analyses and architectural considerations, identifying future opportunities in skin heat exchanger design, F-TMS development, and the use of AM for enhanced heat exchanger geometries. In contrast, Ref. [28] adopts a more component-oriented perspective, addressing technological advancements in cryogenic cooling and the integration of Phase-Change Materials (PCMs) for battery thermal management, also supporting improved HE performance through additive manufacturing techniques.

More in-depth analyses of thermal management for electrified aircraft can be found in Refs. [6, 208, 198]. The works of Favre et al. [6] and Srinath et al. [198] detail key considerations for integrating F-TMS into hybrid-electric aircraft development, whereas the study by Konovalov et al. [208] addresses recent developments in electric-motor cooling for aviation applications. Finally, Lv et al. [209] provide deeper insight into state-of-the-art heat acquisition mechanisms and their advancements, particularly in the context of electronic component cooling.

2.2 Classification of thermal management systems

A number of different approaches emerge from the literature for the cooling and thermal management of aerospace systems and components. Following the framework proposed by Van Heerden et al. [7], these can be distinguished and classified according to five main elements:

- The **heat source**, which introduces the thermal load needing to be dissipated or transferred.
- The **heat-extraction mechanism**, which removes the heat from the source and keeps it within nominal temperature ranges.

- The **thermal transport system**, which carries the extracted heat towards components in need of heating or to the dissipation systems.
- The **heat-rejection mechanism**, which discharges the transported thermal load from the TMS into another component or into a designated heat sink.
- The **terminal heat sink**, which receives and absorbs the rejected heat from the system. It may either be part of the TMS or external to the aircraft.

Applying these criteria to the reference case study of this dissertation (Sect. 1.2.1), the heat source is known and fixed. The analysis focuses on the cooling of a hydrogen MT-PEM fuel cell for a hybrid-electric regional transport aircraft, whereas other heat-generating subsystems, such as electronics, avionics, lubrication systems, and hydraulics, fall outside the scope of the present work.

Accordingly, only heat-extraction mechanisms suitable for fuel cell cooling shall be considered. The review by Rashidi et al. [83] provides a comprehensive overview of these methods, their defining characteristics, and recent technological developments. These are illustrated in Fig. 2.3, and summarised in the following.

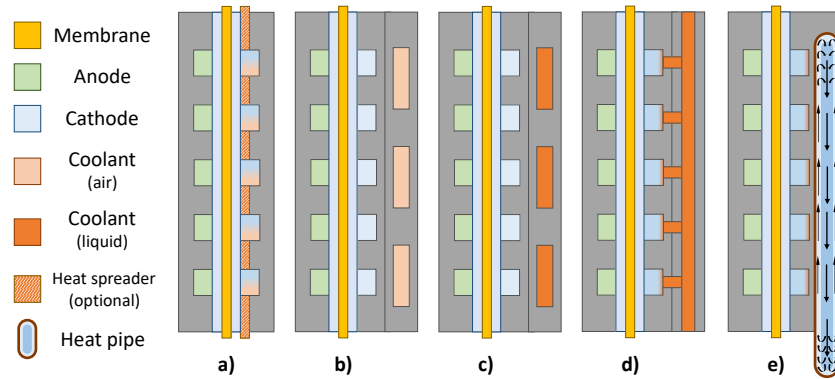


Fig. 2.3 Principal heat extraction mechanisms of fuel cells: a) cathode-flow cooling, b) separate-airflow cooling, c) single-phase liquid cooling (and boiling-based variants), d) evaporation cooling, and e) heat-pipe cooling.

Cooling with reactant flow This method uses one of the fuel cell reactants as the primary stack coolant, sustaining both the electrochemical reaction and heat removal in the cell. In PEM fuel cells, this role is typically performed by the cathode: air. The approach reduces overall system complexity, but requires larger cathode channels

and higher airflow rates, which increase the risk of membrane drying and hinder component compactness. These considerations limit the applicability of reactant flow-based cooling to systems below 0.1 kW.

Cooling with a separate airflow In this case, the reactant flow is decoupled from the airflow used for cooling the stack. This separation offers greater control over heat removal and allows enhanced cooling without risking membrane dehydration. However, it introduces additional complexity, requiring dedicated cooling plates alongside the cathode supply channels. Moreover, the method only supports moderate heat dissipation and is therefore typically applied to fuel cells in the range of 0.2 kW to 2 kW.

Cooling with a single-phase liquid coolant In this approach, a liquid coolant replaces air, leveraging its superior thermal properties to enhance heat extraction from the stack. Considerable research is currently devoted to further improving this solution performance, by refining coolant-plate geometries or incorporating heat spreaders between the cathode and the plates. Thanks to this high cooling capacity and the precise control it offers, liquid cooling is presently the most widely adopted method for fuel cells exceeding 10 kW of electrical output. The main drawback of the approach is the increased system complexity, which requires a complete recirculating loop, including pumps, a coolant reservoir, heat exchangers, and a valve regulation framework. Common coolant fluids include deionised water and water–glycol mixtures. However, these fluids are more prone to parasitic currents compared to air, necessitating the use of deionising filters within the loop.

Cooling with boiling phase change This concept enhances traditional liquid cooling by exploiting latent heat absorption of the coolant as it undergoes phase change. However, it remains largely conceptual due to two-phase flow instabilities and the requirement for a boiling point below the stack operating temperature.

Cooling with evaporation phase change A more mature approach compared to the previous method is evaporation-based cooling, which uses pure water, supplied directly to the cathode channels, to hydrate the membrane and absorb heat through evaporation. This method is highly effective, enhancing cooling performance at

reduced mass flow rates compared to traditional liquid cooling, being applied to fuel cells generating up to 75 kW. However, the method is susceptible to cathode flooding and, although it eliminates the need for a closed coolant loop, it is prone to complete coolant consumption, as the water is expelled through the fuel-cell exhaust. Consequently, the system requires larger water reserves, more frequent refilling, or the integration of water separators or condensers in the exhaust stream to recover water.

Cooling with heat pipes This emerging technique leverages the rapidly growing technology of heat pipes to passively extract substantial amounts of heat from the stack. However, while promising, this approach remains limited to small-scale applications (10 W to 100 W) due to challenges with the scalability of the heat pipes and their manufacturing and integration within the stack architecture.

Given the heat source and its suitable heat-extraction methods, the associated heat-transport mechanisms are determined, as the characteristics of the heat-extraction process dictate the applicable heat-transport approaches. For cathode-flow cooling and separate airflow cooling, no dedicated heat-transport mechanism is required, as the airflow also serves directly as the system heat sink. Accordingly, the heat-extraction method and the heat-rejection mechanism coincide in these configurations. Similar considerations can also be made for evaporative cooling, unless water separators or condensers are employed, causing the water–steam mixture to become the system heat-transport medium.

In contrast, for liquid cooling and boiling-based cooling, the recirculating coolant loop represents the textbook example of a dedicated heat-transport mechanism. In this context, three primary fluids are typically considered:

- **Pure, deionized water** is the most straightforward coolant option. It offers excellent thermal performance due to its high specific heat capacity, and is also typically associated with modest parasitic currents. Its main drawback is the water's susceptibility to freezing at temperatures below 0 °C, which limits its applicability in cold environments.
- **Water–ethylene glycol mixtures** represent the most widespread coolant class across all transport applications. Their primary advantage over pure water is the substantially reduced freezing point, which decreases to approximately

–50 °C. However, while increasing the glycol content improves resistance to freezing, it also leads to higher parasitic currents and a reduction in the specific heat capacity, diminishing thermal performance.

- **Nanofluids** constitute a more recent innovation, in which solid nanoparticles are dispersed in a base fluid, typically water, to enhance its heat transfer performance and potentially lower the freezing point. However, these particles also markedly increase fluid viscosity, leading to raised pumping power requirements, while also aggravating parasitic current generation and exacerbating clogging risks. As a result, nanofluids have not yet been adopted in any commercial PEMFC system and remain confined to research studies and analyses.

In aerospace applications, the high power density required from fuel cell stacks renders air-based cooling impractical. Liquid cooling remains the primary option for managing both the thermal loads and compact packaging constraints, particularly in larger aircraft applications involving megawatt-class propulsion systems. Among the available coolants, water–glycol mixtures represent the only practical choice, as pure deionised water is unsuitable for the low ambient temperatures encountered at altitude, while nanofluid technologies still lack the operational validation required for aviation applications.

Accordingly, a liquid-cooling-based heat-extraction architecture is assumed for the MT-PEM system considered in this dissertation. The associated heat-transport mechanism consists of a closed recirculating loop driven by an ideal pump, a dedicated heat exchanger for heat dissipation, and an expansion tank accommodating the thermally induced volume changes in the coolant. A deionization filter is also included to maintain low ionic conductivity within the loop, limit parasitic currents, and reduce the risk of electrical short circuits. Fig. 2.4 illustrates this base configuration, bounded to the reference heat source of this dissertation.

Unlike the heat-extraction and heat-transport mechanisms, the heat-rejection strategy and terminal heat sink are largely independent of the heat source, enabling greater design flexibility across several solutions. In fuel cell systems, the heat-extraction and heat-transport stages are typically defined by the manufacturer, whereas the selection and integration of the terminal heat sink and heat-rejection strategy remain primarily aircraft design choices. Accordingly, this dissertation focuses on the

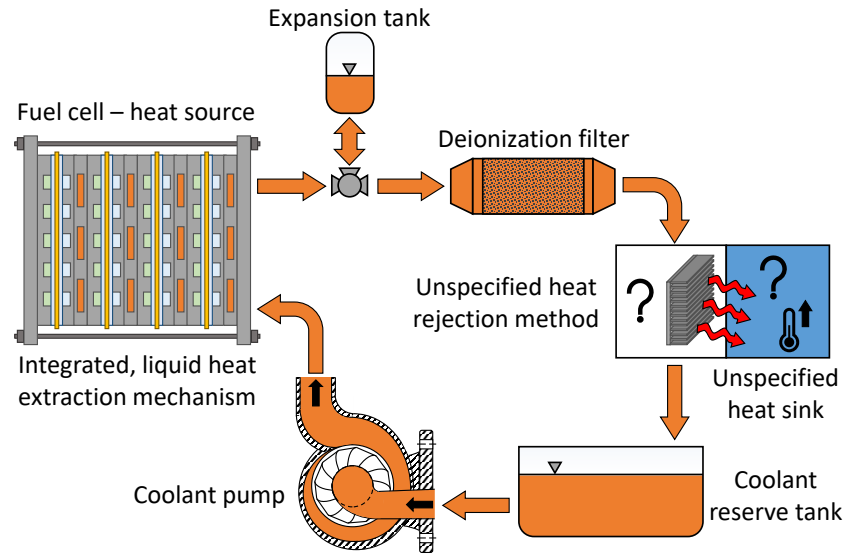


Fig. 2.4 Coolant loop representing the reference heat-extraction and heat-transport mechanisms adopted for fuel cell cooling in this dissertation.

identification, modelling, and simulation of the principal heat-rejection mechanisms applicable to fuel cell heat loads in aircraft.

The primary discriminator when addressing the dissipation of any thermal load is the determination of the intended terminal heat sink, as it strongly influences the viable heat-rejection mechanisms. In aerospace applications, three primary heat sinks can be found:

- **Air.** Typically provided by the atmosphere surrounding the aircraft, it is the most straightforward and widespread heat sink, offering continuous and readily accessible cooling. However, limited control over its temperature and mass flow significantly reduces its effectiveness during taxi and idle, while inertial compression heating increasingly degrades cooling performance when approaching supersonic speeds. When entering high supersonic and hypersonic flight regimes, the external air may even transition from functioning as a heat sink to behaving as a heat source [7].
- **Fuel.** Typically Jet-A or JP-8, fuels are the second most common heat sink, offering improved thermal performance through high specific heat and greater control over temperatures and mass flows. Their use promotes deeper integration between the propulsion system and the TMS, maximising the utilisation of onboard resources without affecting the aircraft aerodynamics. However, their

total heat-absorption capability is ultimately limited by the available fuel quantity, and any heating of combustible fluids introduces safety considerations that must be carefully managed.

- Alternative **storage** and **conversion heat sinks**. These represent a broad and less commonly adopted class of solutions, including structural heat sinks, phase-change materials, thermoelectric generators, and others. They offer unconventional and innovative strategies for managing thermal loads, but their practical application remains limited due to high added mass, restricted heat capacity, low Technology Readiness Level (TRL), or a combination of these factors.

Each of these heat sinks supports several suitable heat-rejection mechanisms, as shown in Fig. 2.5, which may differ both at the system-architecture level and in the individual component layout. The following sections detail the principal heat-rejection mechanisms associated with each of the reference heat sinks introduced above.

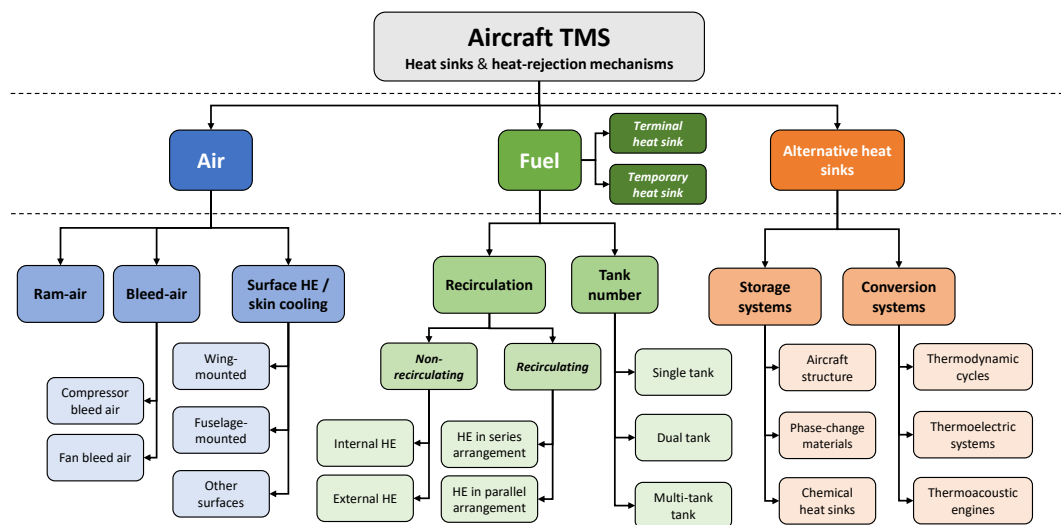


Fig. 2.5 Primary heat sinks in aerospace and associated heat-rejection mechanisms. Extended and refined from the author's previous work [6].

2.3 Air-based Thermal Management Systems

Air is currently the most widespread terminal heat sink in aviation, with air-based TMS emerging as the dominant solution since the early days of aircraft development. This trend is reflected in Fig. 2.6, which compares publication numbers for air-based TMSs with those for general aircraft TMS technology since the 1970s. The comparison is performed by refining the research string reported in Tab. 2.1 through the introduction of the following criteria:

- Tit.-Abs.-Key.: "air" OR "airflow" OR "pneumatic" OR "ram air" OR "ram-air" OR "intake" OR "bleed air" OR "bleed-air" OR "fan air" OR "fan-air"

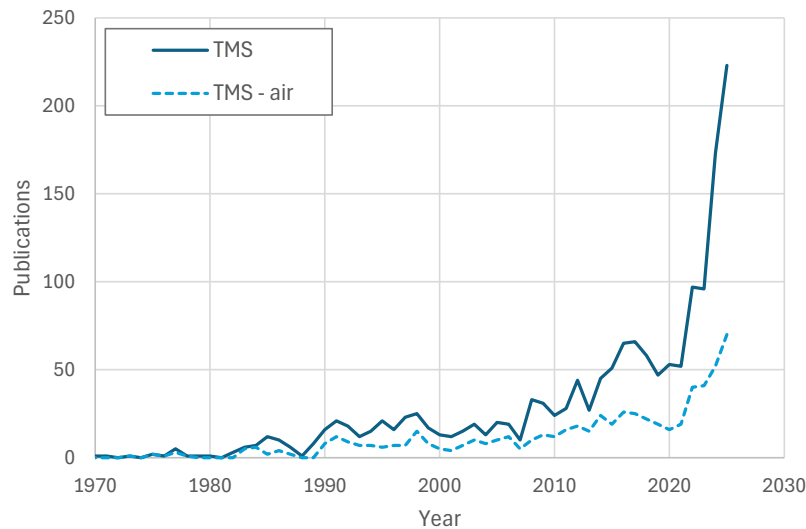


Fig. 2.6 Yearly publications on the topics of general aircraft TMS and air-based aircraft TMS research in aerospace. Data retrieved from Scopus. ©2025 Elsevier B.V.

The analysis returns a total of 657 publications, showing steady publication activity since 1990 followed by a pronounced growth after 2020. The relevant literature identifies three primary heat-rejection mechanisms in air-based thermal management: ram-air cooling, bleed-air cooling, and surface heat exchanger/skin cooling.

2.3.1 Ram-air cooling

Ram-air cooling is the prevalent solution. It consists of an airframe-mounted intake that supplies ambient air to an internal heat exchanger and an outlet nozzle discharging the heated flow. Intakes and outlets are typically located along the fuselage and shaped to minimise aerodynamic penalties, as ram-air operation introduces drag that increases with the intake cross-section and aircraft speed. The aircraft's forward motion supplies the airflow, yielding higher mass flow rates at increased speeds. Accordingly, outlet-mounted puller fans are commonly employed to ensure sufficient mass flow is provided during ground operation and low-speed flight. However, as aircraft speed increases, the airflow entering the ram intake experiences greater inertial compression heating, which raises local air temperatures, reducing the cooling effectiveness of the heat sink.

Fig. 2.7 illustrates a typical ram-air cooling configuration.

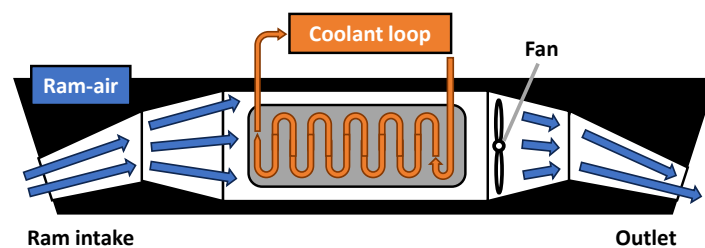


Fig. 2.7 Functional schematic of ram-air cooling.

A substantial body of literature addresses the design and development of ram-air cooling systems for aircraft. Eisfeld and Ruetten [210] present a representative study on fuselage-mounted ram-air configurations for commercial airliners, proposing an integrated pre-design methodology for optimised intake and outlet placements for drag minimisation. In contrast, Russ and Drela [211] investigate a high-altitude (25 km) subsonic aircraft, proposing an under-wing heat-exchanger duct optimised for very low-density flow. Kellermann et al. [212] model a complete ram-air cooling system for a hybrid-electric 180 PAX airliner, including a diffuser, heat exchanger, nozzle, coolant loop, and cold plates for the powertrain generator, power-electronics, and motor. Using global analytical models, the study shows that increasing component temperatures up to 400 K enables drag recovery through the Meredith effect [177], whereas lower operating temperatures lead to increased drag

and additional fuel burn. Despite these three applications being very different, drag minimisation at both intake and outlet is confirmed as a key design driver.

Intake design is therefore critical in limiting ram-air system penalties. This aspect has been extensively studied in the UAV literature, where ram-air typically supplies propulsion rather than thermal management and cooling loops. Tan and Guo [213] provide early design and wind tunnel validations of classical UAV scoop intakes, while Grois et al. [214] numerically compare different intake geometries and placements using CFD analyses. Findings evidence that front-mounted, top-placed configurations are the most efficient, with flush (NACA-type) intakes exhibiting greater stability at high angles of attack compared with scoops.

Ram-air cooling has also been investigated for fuel-cell-powered aircraft. Romeo et al. [215, 216] present two early studies, applied to a general aviation platform. The first focuses on intake CFD modelling and design accounting for propeller effects [215], while the second extends the analysis to a complete fuel-cell thermal management and water system model validated against experimental data [216]. Sain et al. [217], instead, investigate ram-air fuel-cell cooling for a regional transport aircraft featuring a fully integrated nacelle housing the propeller motor, power electronics, fuel cell, heat exchanger, and ram-air intake. The study compares and optimises axial and annular heat-exchanger layouts, dissipating approximately 34 kW to 67 kW of heat. However, all configurations exhibit non-negligible drag penalties and added mass, particularly for the most thermally efficient designs.

System mass implications are further examined by Potamiti et al. [197], who compare centralised and decentralised ram-air TMS architectures for a series hybrid-electric aircraft powered with batteries. Heat exchangers and pumps dominate the system mass in both cases, with heat exchangers accounting for more than 60% of the total mass, reaching up to 80% in centralised systems. Decentralised architectures are shown to improve temperature control at the cost of higher coolant flow rates and a 32% increase in total mass. Similarly, Link et al. [218] conduct a comprehensive trade-off analysis for HT-PEM fuel cells associated with ram-air TMS, showing that higher operating temperatures, despite reducing stack efficiencies, can effectively relax thermal constraints to reduce heat-exchanger size and coolant flows, ultimately lowering the combined system mass by approximately 20%. These results highlight the benefits of high-temperature fuel-cell operation at the aircraft and system level analyses.

Finally, recent studies also explore the integration of advanced thermal management solutions for low-temperature component cooling in adverse environmental conditions. Pal and Severson [219] compare multiple air-cooling solutions for power electronics, combining ram-air with vapour-cycle systems for extreme hot-day conditions up to 60 °C to 70 °C. Adler et al. [220] extend these approaches to a battery-hybrid 180-PAX airliner with 1 MW-class electric motors, integrating ram-air cooling with battery-dedicated vapour cycles and thermoacoustic technologies, achieving fuel savings of up to 3.3% and 1.69%, respectively.

2.3.2 Bleed-air cooling

Bleed-air cooling is the second most common heat-rejection approach. It relies on internally mounted heat exchangers supplied with air extracted from the engines rather than externally mounted intakes, ensuring that cooling is available whenever the turbines are operating, including during ground operation. While this configuration avoids external aerodynamic penalties, it can significantly degrade engine performance and increase fuel consumption when large bleed mass flows are required [221]. In addition, engine compression raises bleed-air temperatures, reducing the effective cooling potential of the heat sink also in this case.

Bleed air can be extracted from an engine compressor or fan. Compressor bleed air, available in most turbine architectures, provides a high-pressure flow that is suitable for pressurised systems, such as the aircraft cabin and engine blades and chambers. However, the elevated temperatures associated with substantial compression limit its viability for use with thermally sensitive components, thus confining its use to higher-temperature applications such as turbine blades, engine casings, and lubricating oils. For this reason, compressor bleed-air requires additional cooling and conditioning before being supplied to the cabin, to ensure safety and comfort to the crew and passengers. By contrast, fan bleed air experiences lower compression heating, as it is extracted downstream of the engine fan. This makes it better suited for cooling low-temperature components such as electronics. However, effective airflow extraction typically requires a ducted-fan architecture, rendering this application impractical for most turboprop configurations. In some designs, the associated heat exchanger may even be integrated directly within the fan nacelle.

Fig. 2.8 illustrates some typical compressor and fan bleed-air cooling configurations for a turbofan layout.

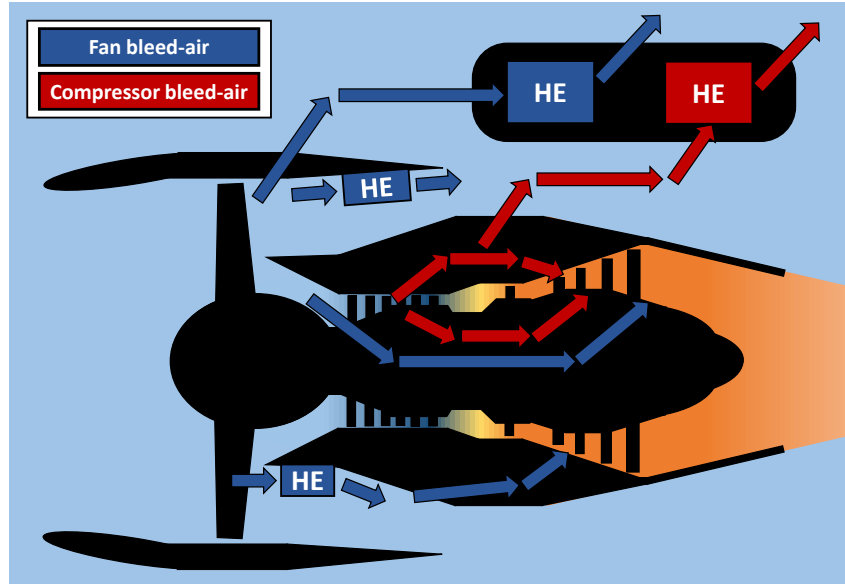


Fig. 2.8 Functional schematic of compressor and fan bleed-air cooling. Adapted from Van Heerden et al. [7]. ©2021 Elsevier.

Most studies on bleed-air cooling focus on the conditioning of air supplied to the aircraft ECS. Early work by Perez et al. [222] analyses a traditional bootstrap cycle for cabin pressurisation and the optimisation of its heat exchangers, highlighting how simultaneously minimising system mass and entropy generation is crucial to avoid identifying false optima characterised by excessive dimensions or reduced thermal effectiveness. Vargas et al. [223] further develop heat exchanger modelling and optimisation for ECSs, while Shang and Liu [224] extend these efforts by addressing performance degradation due to fouling and its identification through experimentally validated detection methods.

Puterbaugh et al. [221] examine the coupling between bleed-air extraction and propulsion performance by integrating lumped-parameter models of the ECS and engines. Their analysis quantifies penalties in engine specific fuel consumption as a function of extracted bleed flow, showing SFC increases of approximately 2% per percentage point of core compressor bleed airflow. These penalties motivated the exploration of alternative ECS architectures to reduce reliance on compressor bleed air. Saito et al. [225] compare conventional bleed-air ECS with electrically driven, ram-air-based systems for more-electric aircraft, demonstrating improved power

balance and highlighting the benefits of vapour-cycle integration. Parrilla [226] further extends this comparison to hybrid architectures employing combined ram air and bleed air cooling with electric compression, achieving fuel savings of up to 1.33% relative to conventional bleed-air ECSs.

Beyond ECS applications, compressor bleed air has been investigated for direct cooling of aircraft systems. Herring and Heister [227] review its use in turbine blade thermal protection via film cooling and discuss emerging opportunities enabled by fuel pre-coolers and advanced manufacturing techniques, including metal foams, micro-tubes, and diffusion bonding. Williams et al. [228] further develop these concepts through the design and optimisation of microchannel geometries for air–fuel heat exchangers, while Kasim et al. [229] propose advanced additively manufactured plate-fin designs for engine pylon and embedded nacelle pre-coolers.

Fan bleed-air cooling has also been extensively studied. Villafañe and Paniagua [230] experimentally investigate an oil fan-air cooler integrated within a turbo-fan bypass duct, developing dedicated acquisition systems and test benches. Their results show that continuous-fin configurations outperform interrupted designs by reducing flow distortion in the downstream wake, particularly in transonic regimes. DeSomma et al. [231] extend this approach to a small-bypass, double-shaft turbofan, examining heat rejection through an internally mounted fan-air heat exchanger in the low-pressure stage, demonstrating modest thrust recovery via the Meredith effect.

Ultimately, the optimal choice between ram-air and bleed-air cooling depends on multiple parameters: operating temperatures, heat loads, airflow requirements, and flight conditions. Rong et al. [232] compare both approaches for a high-speed aircraft across a broad range of cruising speeds, selecting between configurations to minimise fuel-weight penalties. Their findings indicate that fan and compressor bleed air are preferable below Mach ~ 2 , whereas ram-air cooling becomes more advantageous at higher speeds, particularly above Mach ~ 3.4 .

2.3.3 Surface HE/skin cooling

Surface heat exchangers and skin cooling represent a more niche solution. Unlike ram-air or bleed-air systems, they dissipate heat directly through the aircraft wings, fuselage, or both, typically via embedded heat exchangers or serpentine channels underneath the aircraft skin. By avoiding intake ducts, these configurations preserve

aerodynamic performance. However, careful integration is required to prevent local heating from triggering boundary layer separation or adverse turbulence, which can negatively affect lift and drag [233]. Additionally, heat dissipation depends entirely on aircraft speed, which significantly limits skin cooling effectiveness during ground operation, while the progressive substitution of aluminium airframes with composites and polymers may significantly hinder skin heat rejection. Finally, the distributed layout can lead to increased system mass compared with more compact architectures, potentially offsetting the aerodynamic benefits [234, 235].

Fig. 2.9 illustrates a typical surface heat-exchanger/skin-cooling configuration.

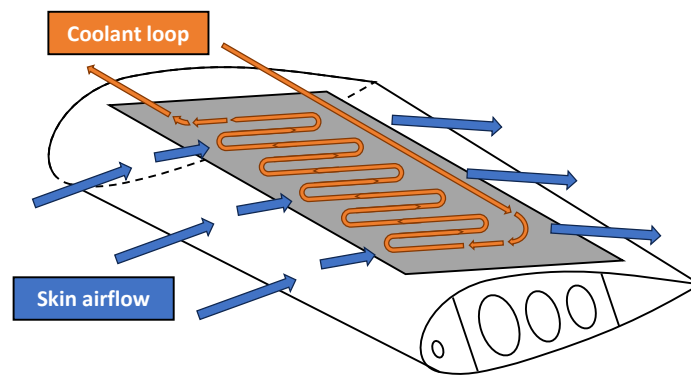


Fig. 2.9 Functional schematic of surface heat exchangers and skin cooling (wing-mounted).

Early concepts of wing-surface radiators were introduced by Curtiss as early as 1922 [236] and gained increasing prominence within the high-speed seaplanes developed for the Schneider and Pulitzer Trophy races, where minimising intake drag was critical to aerodynamic performance. These concepts culminated in the Supermarine S.6b, winner of the final Schneider Trophy in 1931, and the Macchi M.C.72, holder of the seaplane speed record of 709 km h^{-1} from 1934 to this day. Both aircraft relied on extensive integration of skin radiators over the wing surfaces, fuselage sides, and float structures to manage extreme engine cooling requirements.

In recent decades, surface heat exchangers have re-emerged in the context of aircraft electrification. Kellermann et al. [237] present a comprehensive assessment of skin cooling on a battery hybrid-electric Airbus A320, considering wings, fuselage, engine nacelles, and tail surfaces. Findings demonstrate up to 7 MW of heat rejection during cruise with an average surface temperature of 360 K, while hot-day take-off is identified as the most critical condition, dissipating only 0.25 MW. The study also

observes a strong coupling between heat transfer and aerodynamics, with increased skin-friction drag in heated laminar flow regions and reduced drag under fully turbulent conditions.

Several studies address skin cooling with a specific focus towards wing-mounted layouts. Kallath et al. [233] compare top and bottom surface installations at the wing leading edge, mid-chord, and trailing edge, showing that adverse conditions can degrade aerodynamic performance due to flow separation, particularly on the upper surface and at low Reynolds numbers, high heat loads, and elevated angles of attack. Nevertheless, favourable configurations can yield aerodynamic benefits, with lower leading-edge heating reducing drag by 1.6 % and lower trailing-edge heating increasing lift by 2.5 %. Habermann et al. [238] extend these analyses by accounting for propeller slipstream effects in RANS simulations of NACA 45018 wing surface heat exchangers for a turboprop configuration. The study demonstrates that propeller effects are negligible during cruise, but can increase heat rejection by up to 12% during low-speed ground operation. Consistently, increasing skin-to-ambient temperature difference from 25 K to 300 K is shown to enhance heat transfer but degrade lift-to-drag ratio by approximately 15% and the average heat transfer coefficient by 23%. Schaefer et al. [234] analyse the feasibility of wing-mounted surface heat exchangers for a 3.5 MW fully electric aircraft with distributed fuel cell propulsion. The study combines both ram-air and skin-air cooling designs, highlighting that the aerodynamic advantages of surface HEs can be offset by their lower heat-transfer capability, as the added mass required to support high propulsion thermal loads outweighs the benefits associated with reduced induced drag.

Wing-mounted skin heat exchangers also offer opportunities for functional integration with other thermal management subsystems. In particular, integrating surface cooling with wing de-icing and anti-icing systems, which require a sustained heat supply during high-altitude operation [239, 240], may enable system-level efficiency improvements. Alternatively, integration with fuel-based thermal management can allow enhanced passive heat rejection through the tank walls. Roland et al. [241] model such a concept for a C-130 Hercules, demonstrating that simplified flat isothermal surface correlations provide good agreement with higher-fidelity methods, achieving approximately 1 MW of heat dissipation for a 400 K wing-tank surface at Mach 0.5.

Studies on fuselage-mounted skin heat exchangers are comparatively rarer. Pang et al. [242] investigate a typical air-to-air surface heat exchanger concept for elec-

tronic bay cooling, in which cabin and bay air is recirculated through finned channels underneath the fuselage skin. Experimentally validated under ground-operating conditions, the numerical analysis identifies nose-belly arrangements as optimal for maximising cruise heat transfer.

A more recent approach by Schnulo et al. [235] embeds a serpentine coolant loop within the aircraft fibreglass skin to reject heat through the external surface, comparing different installations and their aerodynamic impact. Bottom-mounted layouts in the aft fuselage region yield the highest lift-to-drag ratios, whereas wing and nacelle panels provide the greatest heat rejection capability. The study concludes that, with appropriate placement, aerodynamic performance can largely be preserved, albeit at the cost of increased system weight.

Given the reference use case of cooling a fuel cell supplying the electric powertrain of a hybrid-electric turboprop configuration, only a subset of the available air-based heat-rejection mechanisms is suitable. Fan bleed-air solutions are largely impractical due to the twin-turboprop architecture, while the high-temperature compressor bleed-air is mostly unsuitable for both LT-PEM and MT-PEM fuel cells. Surface heat exchangers and skin-cooling concepts present interesting opportunities for the reference application; however, a study by Schaefer et al. [234] indicates that the resulting system mass penalties may render these solutions impractical for substantial regional transport aircraft electrification.

Accordingly, ram-air cooling remains the most viable solution, as reflected by the extensive literature addressing the design, development, and optimisation of intake geometries and integrated cooling layouts. Nevertheless, significant challenges remain, particularly with the induced aerodynamic drag associated with high cooling mass flow rates, and the capability to ensure adequate heat rejection during low-speed and ground operation.

2.4 Fuel-based Thermal Management Systems

Initially adopted in military and supersonic aircraft during the early 1950s, fuel rapidly emerges as the second most common heat sink after air, with its application progressively expanding to a broader range of aircraft and becoming a major research topic. This trend is reflected in Fig. 2.10, which compares publication counts for fuel-

based TMSs with the corresponding trends for general aerospace TMS technology and air-based TMSs since the 1970s. The comparison is performed by refining the research string reported in Tab. 2.1 through the introduction of the following criteria, returning a total of 509 publications:

- Tit.-Abs.-Key.: "fuel" OR "jet-A" OR "JP-8" OR "kerosene" OR "tank"

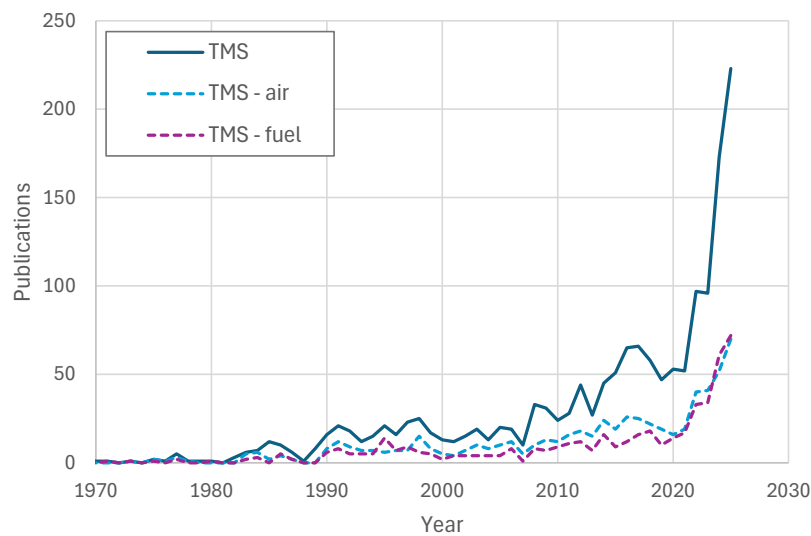


Fig. 2.10 Yearly publications on the topics of general, air-based, and fuel-based TMS technology in aerospace. Data retrieved from Scopus. ©2025 Elsevier B.V.

The analysis shows comparable publication activity to that of air-based solutions up to 2020, albeit with slightly lower publication numbers. Subsequently, publications on fuel-based thermal management systems exhibit a marked increase, reaching and surpassing those on air-based TMS, ultimately confirming the growing relevance of fuel-based solutions in contemporary aerospace thermal management research.

Depending on the configuration, fuel can act either as the system's terminal heat sink or as a secondary (temporary) heat sink. Fuel behaves as a terminal heat sink when the tanks are effectively adiabatic with respect to the external environment and no significant fuel cooling occurs after heat absorption. Conversely, fuel acts as a secondary heat sink when it continuously absorbs heat from onboard sources and subsequently rejects it to other sinks, typically the external environment, through passive tank wall dissipation or active cooling of a recirculated fuel mass flow. This second configuration is more typical in applications requiring substantial cooling,

while terminal fuel heat sinks are better suited for managing modest heat loads or operating conditions where external air cooling is impractical, such as during sustained supersonic flight. Nonetheless, in all cases, the absorbed heat is ultimately rejected to the atmosphere through engine exhaust.

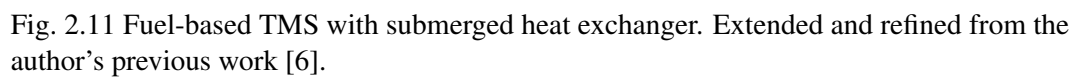
The literature details several fuel-based TMS architectures, which can be distinguished based on three main factors: the placement of the coolant–fuel heat exchanger, the presence of a dedicated fuel recirculation circuit, and the number of fuel tanks involved. Among these, the presence or absence of a dedicated fuel recirculation loop represents the most fundamental architectural distinction. Although all commercially available fuel systems include an engine overflow line returning excess fuel to the tanks, this is solely governed by engine demand and differs conceptually from a dedicated F-TMS recirculation circuit. In a dedicated loop, the extracted and recirculated fuel flows are governed not only by propulsion requirements but also by the thermal demands of the heat sources, enabling substantially higher recirculation rates and greater heat-extraction capability compared with conventional overflow systems.

2.4.1 Non-recirculating architectures

Focusing on the simpler, non-recirculating architectures, these can be further distinguished according to the location of the coolant–fuel heat exchanger, which may either be external or internal to the fuel tank.

Submerged HE

When the heat exchanger is located inside the fuel tank, the thermal management system is referred to as a *Fuel-based TMS with submerged heat exchanger*, since the HE must remain fully submerged throughout the mission to ensure effective heat transfer from the coolant to the fuel, as shown in Fig. 2.11. Heat transfer occurs via natural convection from the HE walls to the surrounding fuel, which limits the overall effectiveness. Nevertheless, the gradual increase in fuel temperature within the tank allows this configuration to support passive heat rejection through the tank walls, as discussed by Kellermann et al. [243].



HE on feed-line

The main advantages of this layout are its straightforward design, consistently high fuel temperatures prior to combustion, and minimal in-tank fuel heating or vaporization leading to reduced safety risks. Its primary limitation lies in the constrained maximum fuel mass flow, which is dictated by the engine feed demand and the limited overflow, often necessitating auxiliary and backup cooling methods, such

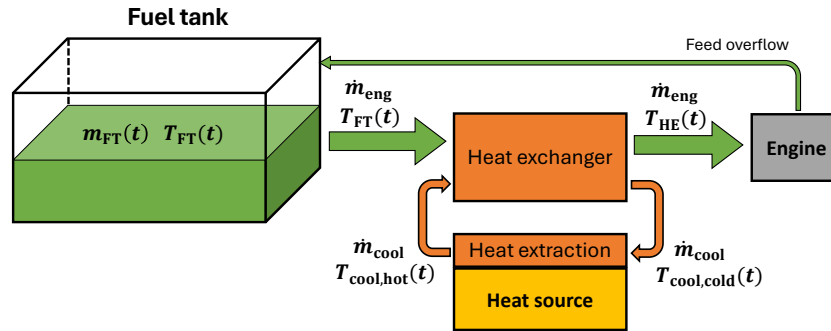


Fig. 2.12 Fuel-based TMS with heat exchangers on the engine feed-line. Extended and refined from the author's previous work [6].

as ram air, to dissipate excess heat. Accordingly, non-recirculating F-TMSs with feed-line heat exchangers are best suited for handling modest heat loads located near the engines.

2.4.2 Recirculating architectures

More advanced configurations retain externally mounted heat exchangers while incorporating dedicated recirculation loops that return substantial amounts of fuel back to the tanks. This ensures the fuel flow can be decoupled from engine demand and independently controlled to meet the heat source needs. Achieving this requires more complex architectures, including a dedicated control loop for the cooling fuel mass flow. Recirculating architectures can be further classified according to their flow arrangement, which may involve one or more fuel tanks, a heat exchanger installed in series with the engine feed line, or a dedicated cooling loop arranged on a parallel line.

HE on a parallel line

Implementing a dedicated fuel recirculation line for cooling, parallel to the engine feed line, is the most straightforward approach to decouple engine fuel supply from TMS cooling requirements. This configuration, referred to as a *fuel-based TMS with parallel heat exchanger recirculation*, relies on separate pumps and controls for the engine feed and cooling circuits. In this arrangement, fuel flowing through the feed line is all consumed by the engines, while the cooling flow is separately

extracted and fully recirculated back to the tank, as shown in Fig. 2.13. Owing to the complete separation of the two pipelines, this recirculating-parallel layout is particularly attractive for retrofitting existing fuel systems with integrated thermal management capabilities.

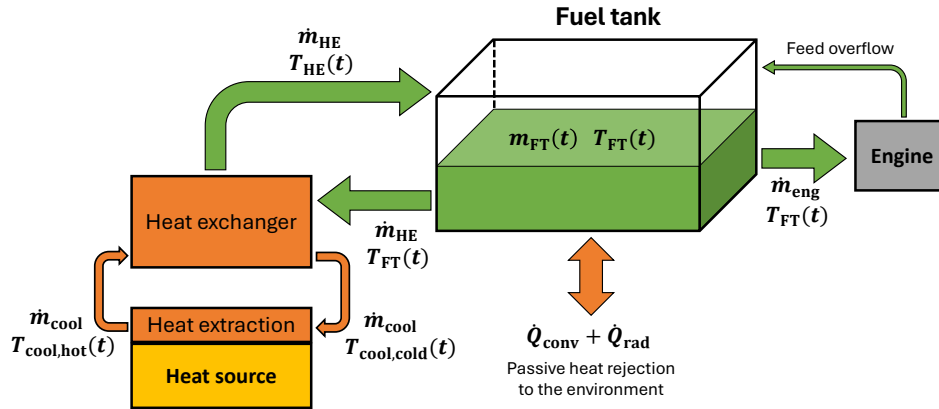


Fig. 2.13 Fuel-based TMS with parallel heat exchanger recirculation. Extended and refined from the author's previous work [6].

The main advantages of this configuration over alternative F-TMS architectures include straightforward layout, high recirculation rates (increasing the maximum heat extraction potential), fully decoupled control of cooling and engine feed flows, ease of retrofit, and viability for passive heat rejection and use as secondary heat sink. Primary limitations include a doubled amount of pumps, sensors, and controls, low fuel temperatures during early operation (reducing combustion efficiency and underutilising thermal inertia), and progressive tank fuel evaporation and heating raising flammability concerns. Accordingly, recirculating F-TMS with parallel-line heat exchangers are best suited for managing moderate heat loads, particularly for components operating independently of engine regimes or exhibiting intermittent cooling demands, such as avionics and certain electronic systems.

HE in series (single tank)

Alternatively, the fuel recirculation loop can be integrated with the engine feed line by installing the heat exchanger in series upstream of the combustor, increasing the fuel supply flow beyond the engine demand, and introducing a flow-regulation valve downstream of the HE to return excess fuel to the tank, as shown in Fig. 2.14. This

configuration is referred to as a *fuel-based TMS with series heat exchanger recirculation*. Compared with parallel architectures, this layout reduces the number of required pumps while retaining dedicated control loops for the feed and cooling functions; however, it also introduces additional complexity in managing the mass-flow split. Accordingly, only a fraction of the heated fuel exiting the HE is recirculated back to the tank, while the remainder is consumed by the engines. Some configurations even integrate auxiliary ram-air heat exchangers to pre-cool recirculated fuel before returning it to the tank, restoring part of its thermal capacity for future use. The main advantages of this configuration include a reduced number of pumps

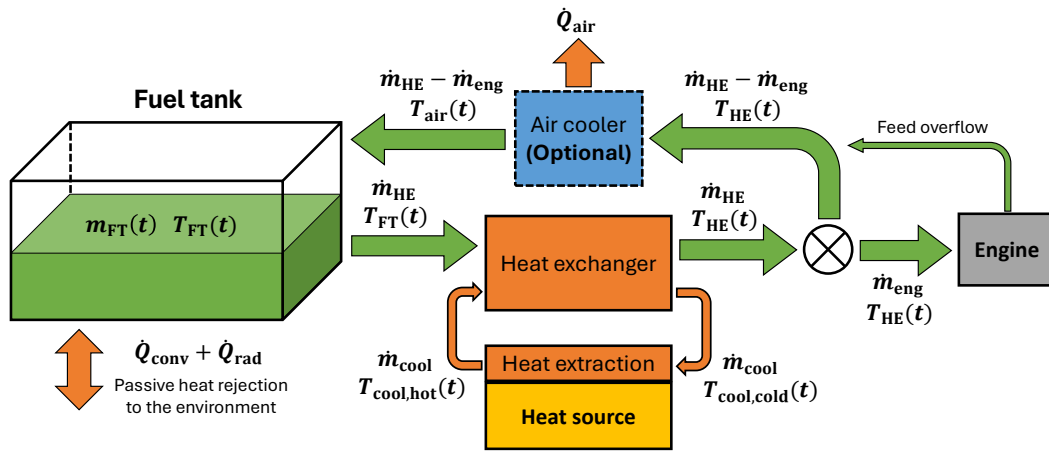


Fig. 2.14 Fuel-based TMS with series heat exchanger recirculation. Extended and refined from the author's previous work [6].

and pipes compared to parallel arrangements, high recirculation rates (increasing the maximum heat extraction potential), decoupled control of cooling and engine feed, consistently high fuel temperatures before the combustion, and viability for passive heat rejection and use as a secondary heat sink. Its limitations include increased system and control complexity, an integrated layout that discourages retrofits, and progressive fuel evaporation and heating in the tank raising flammability concerns. Accordingly, the integrated design of recirculating F-TMS with series heat exchangers is better suited for advanced aircraft concepts with moderate to high heat loads, and should be addressed from the early stages of aircraft product development.

More advanced layouts incorporate complex fuel tank and piping architectures. A notable example is the dual-tank F-TMS concept introduced by Doman et al. [9], which enhances the previous recirculating-series configuration by splitting the main tank into two separate capacities, as shown in Fig. 2.15. This layout is referred to as a *dual-tank F-TMS with series heat exchanger recirculation*. The arrangement consists of a reservoir tank, storing a larger fuel volume with moderate or no recirculation, and a smaller recirculation tank, which receives fuel from the main HE and remains completely full during operation. In this configuration, the reservoir tank experiences minimal to no heating, while the recirculation tank sees a rapid increase in fuel temperature. This ensures that low-temperature fuel is always available to cool more sensitive components, such as the FADEC (Full Authority Digital Engine Control), via the reservoir tank. Fuel from the recirculation tank can be mixed with reservoir fuel to satisfy engine demand and cool higher-temperature heat sources instead. This way, the recirculation tank can exceed the temperature limits of more sensitive components, increasing the system's thermal capacity without compromising their cooling. An additional benefit is that the recirculation tank reaches elevated temperatures more rapidly, enabling earlier and more effective passive heat rejection through its walls.

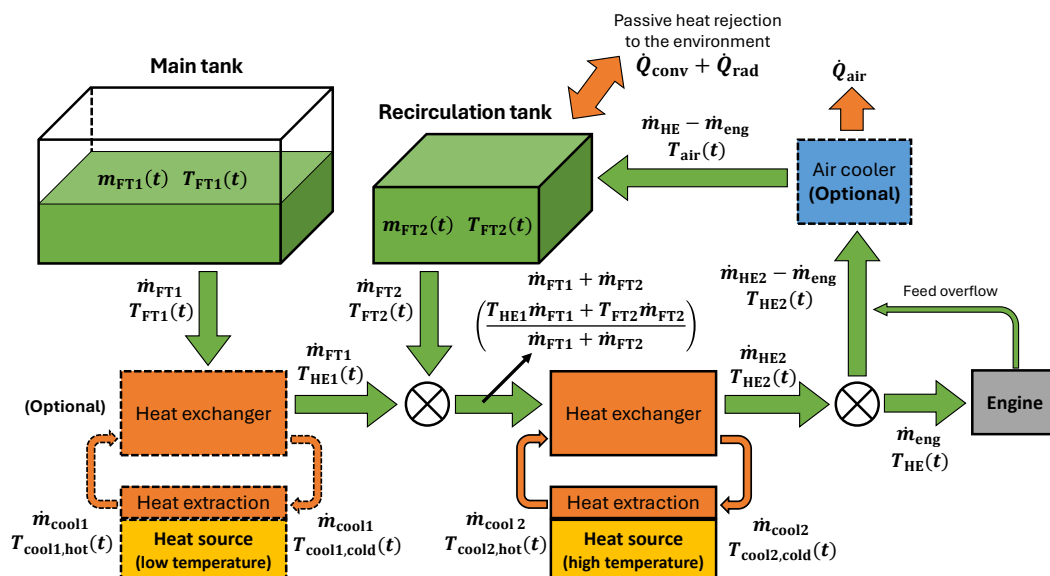


Fig. 2.15 Dual-tank F-TMS with series heat exchanger recirculation. Extended and refined from the author's previous work [6].

The main advantages of this configuration include high recirculation rates (increasing cooling potential), sustained access to a low-temperature fuel reservoir, higher maximum fuel temperature in the recirculation tank, early and effective heat rejection from the recirculation tank, decoupled cooling and engine feed control, maximised fuel temperature before combustion, viability for use as a secondary heat sink, and reduced flammability risk due to the consistently full recirculation tank. Its main limitations are substantial architecture and control complexity (weight, cost, reliability), integrated design that discourages retrofits, and progressive fuel heating in tanks raising flammability and safety concerns. Moreover, architecture benefits stemming from the separate tank temperatures are only fully harnessed when multiple heat sources are present and operate at different temperatures. Accordingly, recirculating-series F-TMS architectures with a dual-tank arrangement are best suited for highly advanced aircraft featuring significant thermal loads and multiple heat sources, and their design should be addressed from the early stages of concept development.

HE in series (multi-tank)

Ultimately, even more complex designs may be considered, involving the management and control of multiple interconnected fuel tanks. These configurations can be classified as *multi-tank F-TMS with series heat exchanger recirculation*. In most cases, however, these layouts represent just a refined implementation of the dual-tank concept, featuring one or few recirculation tanks supplied by an array of interconnected reservoir tanks. These representations primarily support centre-of-mass analyses associated with fuel transfer and tank-emptying routines, rather than assessing the F-TMS performance.

A notable exception is the *hot/cold dual-tank recirculation with drains* architecture proposed by Doman et al. [9] for addressing high-intensity, short-duration peak heat loads in military aircraft. This configuration assumes two relatively small tanks, designated as a hot tank and a cold tank, insulated from the main fuel system, and dedicated exclusively to cooling an intermittent high-energy heat source, as shown in Fig. 2.16. During operation, the two tanks are initially filled with fuel from the main system. When the heat source is activated, a vapour cycle transfers the waste heat to the tanks, heating a recirculating flow from the cold tank to the hot tank. As the fuel absorbs heat, temperature in the hot tank starts to rise, and the stored

fuel is sent back to the cold tank after being partially cooled through a ram-air HE. This loop is maintained for the full heat-source operation, resulting in a progressive rise in the average fuel temperature within the two tanks. Following heat-source deactivation, the fuel continues to circulate between the hot and cold tanks until sufficiently cooled, and is later drained back to the main fuel system.

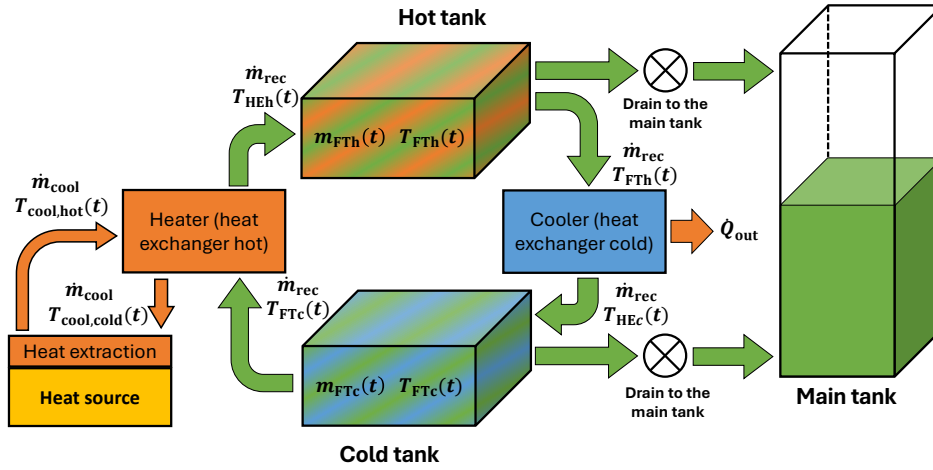


Fig. 2.16 Fuel-based TMS with hot/cold dual tank recirculation with drains. Based on concepts from Refs. [8, 9].

This configuration enables a relatively small fuel mass to rapidly absorb large heat fluxes and reject them to the external environment over an extended time frame, effectively acting as a thermal buffer. Moreover, by remaining insulated and completely filled with fuel, the architecture mitigates flammability risks associated with fuel evaporation.

2.4.3 Fuel handling

Special care shall be taken when handling fuels due to their high chemical reactivity, particularly when heated. Three primary phenomena must be considered in the context of fuel-based thermal management: flammability, coking, and pyrolysis.

Flammability Flammability represents the primary concern in fuel management, as it determines the potential risks of combustion, deflagration, and explosion resulting from energy inputs such as heating, electrical discharges, or sparks. This risk is especially critical in aerospace applications, where fuel is inevitably stored

and circulated in proximity to heat sources and electrical equipment. As a result, stringent regulatory frameworks issued by national and international authorities govern the assessment and mitigation of flammability hazards. A notable example is the CS-25 regulation for large civil aircraft [244] by the European Union Aviation Safety Agency (EASA), which defines specific requirements for fuel system design, allowable fuel temperatures, tank ventilation, and ignition prevention.

In fuel-based TMS architectures, flammability risks are further increased by fuel heating, recirculation, and the presence of partially filled tanks, which favour vapour formation and the development of flammable ullage mixtures. Accordingly, architectures involving elevated tank temperatures or extensive recirculation require dedicated measures, such as inerting systems, controlled tank-filling strategies, insulation, or active fuel cooling, to maintain compliance with certification constraints.

Two primary threshold temperatures are commonly used to assess fuel flammability: the flash point, defined as the lowest temperature at which a liquid fuel generates sufficient vapour to form an ignitable mixture in the presence of air and an ignition source; and the autoignition temperature, corresponding to the minimum temperature at which spontaneous ignition occurs without external ignition. These values depend on fuel composition, ambient pressure, and environmental conditions. For conventional kerosene-based jet fuels at ambient pressure, a flash point of approximately 38 °C and an autoignition temperature of about 210 °C are typically assumed. However, while the flash point itself expresses an intrinsic fuel property, the associated flammability risk can effectively be reduced by insulating the fuel from ignition sources and inerting the ullage through oxygen concentration reduction.

Representative constraints for flammability mitigation from EASA CS-25 regulations include:

- The bulk fuel temperature shall remain below the fuel autoignition temperature, reduced by an appropriate safety margin (typically $\simeq 50$ °C).
- Fuel-wetted surfaces shall not exceed the fuel autoignition temperature, reduced by an appropriate safety margin (typically $\simeq 30$ °C).
- Fuel systems shall be electrically insulated and adequately separated from electric components and cabling. In fuel cell aircraft, this constraint precludes direct fuel cooling of the stack, requiring the adoption of an intermediate coolant loop with reliable deionization.

- The fuel system shall preclude internal filaments or loose metallic particles to minimise the risk of spark generation.
- Fuel flow velocities in pipes shall be limited to below 6 m s^{-1} to reduce electrostatic charge accumulation.
- Fuel system pumps shall always remain fully submerged during operation to prevent spark formation at the impeller.

Coking At temperatures exceeding 150°C , kerosene fuels start to undergo chemical degradation, which become increasingly intense as the fuel temperature rises, leading to the formation of solid precipitates that attach strongly to wetted surfaces. This phenomenon takes the name of coking and can lead to severe fouling of pipes and heat exchangers and clogging of sensitive components such as valves, injectors, and nozzles.

Today, coking is most commonly observed in gas turbine engines, where it primarily affects the fuel injectors and spray nozzles of the combustor. As a result, extensive literature addresses both coke formation mechanisms and mitigation strategies, since deposited carbon layers are difficult to remove once established. Studies indicate that reducing sulphur content, metal contaminants, and fuel additives concentrations can significantly mitigate coking, as these species tend to promote reaction initiation. Literature shows that early coke formations can be effectively removed through controlled mechanical vibrations. However, no reliable chemical, thermal, or mechanical removal technique exists once the deposited layer exceeds thicknesses of approximately 10 nm, typically forcing component replacement or extensive maintenance. Other fuel degradation mechanisms and deposits may occur, but these are generally less frequent and much less detrimental than coking in aerospace fuel systems.

Pyrolysis The last major chemical transformation affecting aviation fuels is pyrolysis, a process occurring when organic compounds are exposed to very high temperatures under oxygen-deficient conditions. In the absence of oxygen, combustion is inhibited and endothermic reactions take place instead, causing the thermal cracking of fuel molecules into smaller, simpler species. For kerosene-based aerospace fuels, pyrolysis becomes a thermodynamically and kinetically significant at temperatures

exceeding 500 °C. This process has been widely investigated in the literature, with some studies proposing the exploitation of pyrolysis as a plausible means of enhancing the thermal endurance of F-TMS by combining conventional fuel heating with endothermic reactions acting as chemical heat sinks.

These approaches have been primarily explored for the thermal management of hypersonic aircraft operating above Mach 5, where the intense aerodynamic heating renders surrounding air ineffective for cooling the cabin, structures, and engines. Under these conditions, fuel becomes the preferred heat sink, cooling the most temperature-sensitive components first, followed by the airframe surfaces and finally the propulsion units, which typically tolerate higher operating temperatures. For these high-supersonic and hypersonic configurations, ramjet or scramjet engines are typically employed. These engines abandon conventional compressor–turbine assemblies, relying instead on the kinetic compression of high-speed airflow rather than mechanical compression, resulting in extremely high inlet and combustion temperatures. Such conditions allow fuel to reach temperatures well beyond conventional limits, enabling the system to harness both the physical and chemical heat sink capacities of kerosene, provided that the heated fuel is not recirculated back to the tanks, where contact with air would create severe flammability hazards.

Accordingly, similar concepts could be considered for the cooling of very high-temperature fuel cell systems, such as molten carbonate or solid oxide fuel cells. However, despite the promising results reported in the literature, practical implementation of chemical heat sinks in F-TMS remains a challenge, as fuel degradation, irreversible changes in chemical composition, and elevated operating temperatures significantly complicate fuel recirculation and overflow management. Moreover, coking phenomena remain a critical concern, exacerbated by the extreme temperatures required to activate pyrolysis.

For the reference use case of a hybrid-electric aircraft powered by MT-PEM fuel cells, the adoption of a fuel-based TMS represents a viable option. Owing to the relatively low heat-source operating temperatures of approximately 120 °C, concerns regarding coking can be relaxed, while fuel pyrolysis is excluded as a viable means of increasing the system thermal capacity. Flammability remains a critical consideration, although the expected operating temperatures remain well below the reference kerosene autoignition temperature, even when accounting for an additional

50 °C safety margin. Therefore, fuel recirculation into partially filled tanks can be considered feasible, provided that adequate ullage inerting is ensured.

However, due to the high levels of waste heat involved, only recirculating architectures are realistically of interest for the reference application, while the specific arrangement of heat exchangers and tanks warrants further investigation.

2.5 Alternative Thermal Management Systems

Alternative thermal management systems encompass a range of unconventional cooling solutions that do not rely on air or fuel as the primary heat sink. Despite exploiting different physical mechanisms, these systems can be distinguished into two main categories: non-conventional energy storage systems and non-conventional energy conversion systems.

2.5.1 Storage systems

Storage systems act as pure thermal capacities, temporarily absorbing waste heat and releasing it at a later stage. They are typically employed as secondary heat sinks, assisting the primary cooling system during critical operation by buffering peak thermal load and enabling its gradual dissipation.

Aircraft structure The first and most straightforward alternative thermal storage medium is the aircraft structure itself. Structural heat sinks exploit the components, airframe, and skin surrounding the source to absorb and store thermal energy through conduction. They provide passive thermal management by buffering peak heat loads; however, their thermal capacity is inherently limited, restricting their application to modest or intermittent heat loads. Moreover, their effectiveness strongly depends on the thermal properties of the adopted materials, with modern polymer- and fibreglass-based structures generally exhibiting lower thermal conductivity than traditional aluminium airframes. Accordingly, a reassessment of traditional skin-based heat-rejection strategies is needed, while current research investigates composite structures with enhanced thermal conductivity to improve their effectiveness both as structural heat sinks and as passive heat-rejection surfaces. These considerations

are particularly relevant for elongated structural components, such as wing fuel tanks, which require detailed thermo-structural analyses to mitigate structural risks, including thermally induced buckling.

Phase-change materials Phase-change materials are a rapidly emerging class of alternative thermal storage heat sinks. They comprise a range of chemical compounds characterised by high latent heats and melting temperatures close to the operating temperatures of common heat sources. Accordingly, PCMs can absorb large amounts of thermal energy while maintaining an approximately constant temperature during the phase transition. Since the phase transition is typically reversible, PCMs are well suited to managing intermittent thermal loads by buffering peak heat generation and enabling the downsizing of primary thermal management systems.

However, the use of PCMs in aircraft applications remains currently under investigation, as their potential benefits must be balanced against penalties in mass and volume. Additional concerns related to material cost, toxicity, and performance degradation under repeated thermal cycling shall also be considered. Despite these challenges, interest in PCMs for aviation continues to grow, with proposed applications ranging from small-scale energy harvesting for sensors to the thermal management of electrified propulsion systems. Current technology readiness levels (TRLs) are generally estimated to lie between 4 and 6, reflecting ongoing development in both academic research and patented technologies.

Chemical heat sinks Chemical heat sinks represent the final category of alternative heat storage systems, absorbing thermal energy through endothermic chemical reactions. Unlike structural and phase-change solutions, chemical heat sinks typically rely on irreversible reactions. Consequently, their thermal storage capacity cannot be restored through passive heat dissipation over time but requires replenishment or replacement after each use. A representative example of a chemical heat sink is the previously mentioned fuel pyrolysis, in which thermal energy is absorbed by breaking kerosene molecules into smaller compounds. Alternative concepts involve dedicated consumable chemicals stored in modular units and replaced after each flight. While such solutions are impractical for providing full-flight cooling, they may offer valuable supplementary capability for peak heat-load management or

emergency cooling. However, as with phase change heat sinks, challenges related to material cost, toxicity, and disposal remain significant.

2.5.2 Conversion systems

Conversion systems, on the other hand, recover useful work from waste heat prior to its rejection to the environment. Accordingly, these systems are better suited for sustained operation rather than intermittent heat load cycling. However, while they improve overall system efficiency by reducing losses, they are not capable of dissipating the entire thermal load independently. Hence, alternative conversion systems may only function as secondary heat sinks supporting a primary TMS and heat sink.

Thermodynamic cycles Thermodynamic cycles represent the primary class of alternative energy-conversion approaches for thermal management. They rely on the well-established concept of converting waste heat into useful mechanical or cooling power, yet have seen limited adoption in aircraft applications to date. The main barriers limiting widespread implementation are the additional system mass, increased layout complexity, potential impacts on reliability and safety, and higher costs. Nonetheless, Rankine cycles can sometimes be employed to recover high-temperature waste heat and convert it into shaft power, provided that suitable high-temperature heat sources are available. Alternatively, vapour absorption cycles can be introduced to provide refrigeration and localised cooling for temperature-sensitive components.

Thermoelectric heat sinks Thermoelectric devices constitute a second category of alternative energy-conversion systems for TMSs. They exploit direct heat–electricity interactions to generate electrical power from temperature gradients through the Seebeck effect, or to pump heat when an external voltage is applied through the Peltier effect. Although several aircraft applications have already been explored, the adoption of thermoelectric devices remains limited due to their low power density, which typically lies around 0.05 kW kg^{-1} . Nevertheless, ongoing research on thermoelectric materials and novel installation architectures promises significant improvements in their viability for large-scale applications. For instance, studies have

shown that positioning thermoelectric generators between a gas turbine's hot core and cooler bypass flows can effectively reduce aircraft specific fuel consumption for electric power generation. However, despite their potential, the technology readiness level of thermoelectric systems for aviation remains low, typically around 3–4.

Thermoacoustic engines Thermoacoustic engines constitute the final category of alternative cooling systems in advanced aerospace TMS. They are distinguished into, thermoacoustic generators, which convert temperature gradients within working gases into acoustic waves to produce useful work, and thermoacoustic refrigerators, which employ acoustic energy to transfer heat from a cold region to a hot one. Although still at an early stage of development, these systems offer a novel approach to heat-driven cooling and energy conversion, free from conventional moving mechanical components, with some concepts already being explored for aircraft thermal management.

Given their primary role as auxiliary heat sinks within the TMS, alternative thermal storage and conversion methods are unsuited to fully support the cooling requirements of the reference fuel-cell-powered hybrid-electric aircraft. However, they could be integrated into established concepts to enhance overall system feasibility. In particular, owing to the low operating temperatures of the MT-PEM heat source, phase-change materials may be considered for managing peak thermal loads, for instance, during take-off, while structural heat capacity should be included in advanced analyses in integration with fuel-based TMS architectures.

2.6 Summary and highlights

Recent trends in the scientific literature indicate a growing interest in aircraft thermal management, electrification, and the associated enabling technologies. Within this analysis, suitable heat extraction and transport methods for fuel cell heat sources are presented and discussed, highlighting their fundamental operating principles, benefits, and limitations. A fuel-cell cooling loop concept is proposed for the reference use case of this dissertation, based on integrated liquid heat extraction within the stack and deionized coolant recirculation to transfer the thermal load to an appropriate heat sink.

A comprehensive review of the primary heat sinks and heat-rejection methods reported in the aerospace literature is presented, identifying all viable candidates for completing the proposed fuel-cell cooling loop. These solutions can be broadly classified into three main categories depending on the adopted heat sink:

- *Air-based TMS*, which leverage external atmospheric air for component cooling. These systems are the most widespread and thoroughly investigated, offering relatively simple, safe, and effective cooling solutions through the widely available surrounding air.
- *Fuel-based TMS*, which exploit the intrinsic thermal inertia of the onboard stored fuel to absorb heat from a thermal source. Although less prominent than air-based solutions, these concepts are well established in the aerospace industry, particularly for cooling components located near the propulsion systems. They benefit from improved utilisation of the fuel resource, favourable thermal properties of the working fluid, and enhanced control over mass flow rates and temperatures.
- *Alternative heat-sink TMS*, which collect a diverse range of concepts that rely on different physical phenomena to absorb or reject heat. Depending on their working principles, these systems may be further divided into thermal storage and energy conversion approaches, but remain largely confined to auxiliary roles and niche applications.

Among air-based solutions, three primary subclasses can be identified: ram-air, bleed-air (from either the compressor or fan bypass), and skin-air cooling. The main challenges associated with these concepts include the induced aerodynamic drag penalties, particularly for ram-air designs, the added system mass, especially critical for surface-mounted heat exchangers, and the engine performance degradation of bleed-air concepts. For the reference use case of MT-PEM fuel cell cooling, only ram-air and, potentially, skin-air cooling could realistically be considered, owing to the high waste heat generation involved.

For fuel-based solutions, multiple architectures are reported in the literature, with five to six primary designs commonly identified, distinguished by heat exchanger placement, recirculation arrangements, and fuel tank compartmentalisation. Compared to air-based designs, fuel-based heat sinks are primarily limited by their finite

thermal capacity, which is constrained by the onboard fuel mass and continuously reduced by fuel consumption during operation. This limitation introduces the concept of aircraft *thermal endurance*, defined as the maximum duration over which the TMS can ensure adequate system cooling, potentially imposing a constraint on the achievable mission range. Furthermore, fuel heating raises important concerns related to flammability hazards and chemical degradation phenomena, including coking and, at higher temperatures, pyrolysis. These issues must be carefully managed through appropriate system layout design, tank venting, inerting strategies, and adequate electrical insulation and protection.

For the reference MT-PEM fuel cell cooling application, a fuel-based TMS may be considered owing to the adopted hybrid-electric propulsion architecture. However, an intermediate coolant loop should be implemented between the fuel cell and the fuel system, incorporating reliable deionisation, to ensure adequate electrical insulation between the two systems. While the selection of an appropriate F-TMS architecture still requires further investigation, attention should be focused on layouts featuring substantial recirculation, due to the high heat loads involved.

Among alternative heat sink concepts, no solution is realistically capable of independently supporting the complete cooling of the reference MT-PEM hybrid-electric aircraft. However, approaches such as structural heat sinks and phase-change materials may be considered as complementary solutions for future improvements to established TMS architectures, compatibly with the low operating temperatures of fuel cells.

Ultimately, the findings of this survey indicate that conventional ram-air and fuel-based systems represent the two most promising candidates for the cooling of fuel-cell-powered, hybrid-electric aircraft. Among these, air-based designs have been extensively explored in the literature, evidencing challenges with the management of system-induced aerodynamic drag. Fuel-based architectures have also received significant attention; however, their exploration in supporting fuel cell cooling remains an open research area. Accordingly, the following chapter of this dissertation shall address the application and thermal endurance of fuel-based TMS for fuel cell cooling within hybrid-electric aircraft configurations.

Chapter 3

The Fuel-based Thermal Management System

Chapter 1 highlighted the challenges of aircraft cooling in the context of aerospace electrification, with particular emphasis on next-generation hybrid-electric regional transport aircraft powered by fuel cells. Chapter 2 reviewed the main trends in the scientific literature on aircraft thermal management systems, comparing available heat sinks and associated heat rejection mechanisms. From the analysis, fuel emerges as a highly relevant candidate, extensively investigated due to its high specific heat capacity, favourable heat-transfer characteristics, and greater thermal-control capability compared to air.

3.1 State of the art

Recent reviews by Van Heerden et al. [7] and Coutinho et al. [17] provide comprehensive overviews of state-of-the-art fuel-based thermal management systems in aerospace applications. Early studies by Hudson and Levin [189] and Harrison et al. [190] introduce fuel-based TMS concepts to address increasing thermal loads in fourth-generation U.S. fighter aircraft, including the F-15, F-16, and F/A-18. These studies employ lumped-parameter models to simulate fuel-tank temperature evolution, focusing respectively on integrated control strategies for heat loads up to 88 kW and experimental investigations of fuel fouling effects.

Building on these foundations, Doman et al. investigate multiple F-TMS architectures for next-generation military aircraft, highlighting the importance of flow topology and introducing an innovative dual-tank recirculating configuration. Their studies employ lumped-parameter models of fuel tanks, heat sources, and propulsion systems to track fuel temperature evolution across different stages of system development, including conceptual architecture design [9, 245], detailed analytical investigations [246–249, 21, 22], and experimental validation [250]. Collectively, these works demonstrate improved thermal endurance for dual-tank configurations compared with single-tank layouts. Experimental evidence [250] further shows that passive heat rejection through tank walls can significantly influence thermal performance, a finding later confirmed through parametric analyses by Manna et al. [251]. More complex tank architectures have also been explored [252] for dynamic simulations; however, passive heat rejection mechanisms are still generally represented using simplified system-level models.

Concurrently, parallel research efforts have been addressing the detailed modelling of heat rejection at the fuel–air interface of the tank under steady-state conditions. In the work of Roland and Rumpfkeil [241], simplified isothermal flat-surface assumptions are compared with detailed simulations of NACA airfoils from a C-130 aircraft, demonstrating that the simplified approach provides acceptable accuracy at the system level. Their findings indicate that cumulative heat dissipation on the order of 1 MW may be achievable when considering all tanks; however, the analysis does not account for fuel-side temperature dynamics, recirculation effects, or heat transfer coefficients. Liu et al. [253] address these limitations by combining dynamic system-level models with detailed simulations of internal fuel motion and tank-wall heat transfer. The study relies on a CFD-based approach to capture the complex tank geometries of the reference fighter aircraft; however, this results in substantially higher computational costs, particularly for long-duration transient simulations.

However, the technology transfer of F-TMS concepts from military aviation to civil applications is explored in a comparatively smaller body of work. Li et al. [254] present an early example, developing a lumped-parameter model of a submerged fuel tank heat exchanger (HE) for oil cooling in a conventional airliner, effectively managing moderate heat loads of around 20 kW. Van Heerden et al. [25] subsequently extend these concepts to emerging electrical components and lubrication systems for more-electric civil aviation, achieving good agreement with experimental data from an Airbus A310 and addressing heat loads up to 100 kW. These approaches have

also been adapted to hybrid-electric configurations. Albomali et al. [255] integrate fuel- and air-based cooling systems for a long-endurance, battery-assisted aircraft, providing cooling capability for thermal loads of approximately 30 kW while also minimising added mass compared to solely air-cooled configurations. The study adopts a lumped-parameter approach but retains simplified passive heat rejection models. Roumeliotis et al. [29] extend this concept to the cooling of a hybrid-electric rotorcraft, managing combined oil and battery heat loads of 20 kW for a 20% hybrid configuration.

In contrast, large-scale electrification introduces substantially greater thermal management challenges. Kim et al. [256] analyse combined fuel–air TMS designs for a delta-wing aircraft with megawatt-class heat loads, showing that such concepts remain unfeasible with current technologies due to extreme power requirements and integration constraints. Kellermann et al. [243] propose an alternative assessment methodology for fuel-based cooling in a conventional airframe with 20% battery hybridisation, evaluating steady-state tank-interface models across different flight phases. The study compares a submerged tank heat exchanger configuration against two wing-surface fuel recirculation strategies, managing peak heat loads of up to 126 kW and demonstrating system feasibility throughout the mission, except during taxi phases. However, despite a high computational efficiency, the approach remains also limited to steady-state analyses and does not capture transient fuel temperature evolution or thermal inertia effects.

Overall, three main research directions and associated modelling strategies emerge from the presented F-TMS literature. The first and most prevalent direction involves the system-level modelling of fuel tanks and the dynamic simulation of temperature transients through lumped-parameter frameworks, as demonstrated by Doman et al. [9, 245–249, 21, 22, 250], Manna et al. [251], Albomali et al. [255], and Kim et al. [256]. The second research direction focuses on computationally efficient steady-state representations of passive heat rejection through tank walls, often based on nodal and discretised models derived from the surface-HE and skin-cooling literature [238], as demonstrated by Roland and Rumpfkeil [241] and Kellermann et al. [243]. The third and final direction involves high-fidelity three-dimensional CFD simulations of internal fuel motion and convective heat transfer [253], capturing complex tank behaviour at the expense of increased computational costs, particularly when coupled with prolonged system-level and transient simulations.

However, despite extensive research in the field, relatively few studies have investigated F-TMS integration for hybrid-electric aircraft propulsion, with most analyses focusing on battery-driven configurations [255, 29, 243], rather than fuel cells. This highlights an ongoing research gap concerning the development and assessment of F-TMS architectures for emerging fuel-cell-based hybrid-electric aircraft concepts.

3.2 Scope

Within this chapter, a numerical model is developed to predict the performance of fuel-based thermal management systems, addressing key gaps identified in the literature. The methodology is applied to the reference hybrid-electric regional transport aircraft considered in this dissertation, powered by MT-PEM fuel cells and evaluated across varying degrees of propulsion electrification. The approach integrates elements from the first and second F-TMS research directions, accounting for tank thermal inertia, fuel temperature dynamics, and passive wall heat dissipation to assess system thermal endurance. For the remainder of this dissertation, the thermal endurance shall be defined as the system's ability to provide adequate component cooling until heat-source or fuel temperature limits are reached.

The model simulates the evolution of stored fuel mass and temperature, capturing the sustained fuel depletion and its effects on the progressive temperature rise. A simplified representation of heat rejection through the tank walls is included, comparing conductive heat transfer within the fuel with buoyancy-driven and vibration-induced convective mechanisms. In addition, a comparative analysis is conducted between the recirculating-parallel and recirculating-series F-TMS configurations shown in Figs. 3.6a and 3.6b.

The performance is evaluated across a set of representative mission profiles to assess the influence of mission altitude, range, and alternate-flight parameters on the aircraft thermal endurance. This approach facilitates the identification of critical operating scenarios and the determination of the maximum level of fuel-cell propulsion electrification that can be consistently sustained by the F-TMS. As the analysis aims to evaluate the feasibility and limitations of fuel cell cooling using current F-TMS technologies at a functional level, detailed component design and failure-mode analyses are considered outside the scope of this work.

For the reference use case considered in this dissertation, two identical wing fuel tanks are assumed for the simulation, consistent with typical regional transport aircraft configurations, with no additional fuselage-mounted tanks. Modelled MT-PEM fuel cells retain the nominal efficiency of 50% and the maximum operating temperature of 120 °C. A deionisation filter is assumed to treat the coolant prior to entering the coolant–fuel heat exchanger; however, owing to previous considerations, its modelling and sizing are not addressed in the present analysis.

To mitigate fuel degradation and coking, a maximum allowable fuel temperature of 150 °C is imposed. This limit is inherently satisfied, as the 120 °C maximum operating temperature constraint of the fuel cell already restricts fuel temperatures to lower values. Accordingly, the kerosene autoignition temperature of approximately 227 °C is also not restrictive, as it remains well beyond the analysed operating range, even when considering the 50 °C safety margin required by CS-25 regulations for fuel tank safety and flammability assessments. Nevertheless, since fuel temperatures within the tanks are expected to exceed the kerosene flash point of approximately 37 °C, a dedicated fuel tank inerting system is assumed for safety reasons, yet its modelling and sizing remain outside the scope of this work.

3.3 Methodology

A system-level model is developed to simulate F-TMS performance for fuel cell cooling in MATLAB/Simulink. The architecture comprises three primary subsystems: the fuel cell heat source, the coolant heat extraction and transport loop, and the fuel sink and heat rejection system. Two heat exchanger elements connect these subsystems through the coolant loop, enabling heat transfer from the source to the sink, as shown in Fig. 3.1.

A lumped-parameter approach is adopted, representing each major component as a control volume under the assumptions of homogeneous, isotropic, and spatially uniform properties, in line with the established literature [246, 247, 251]. The Eulerian formulation for open systems, illustrated in Fig. 3.2, is employed to describe mass and energy exchanges at the control volume boundaries and to compute the time evolution of the associated state variables.

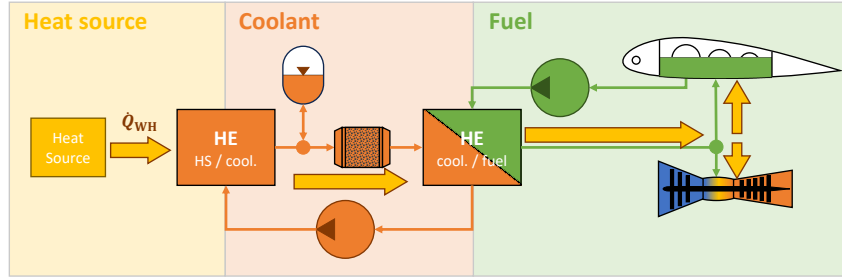


Fig. 3.1 Functional scheme of a F-TMS for the cooling of a fuel cell. A recirculating architecture with a heat exchanger installed in series on the engine feed line is shown for reference.

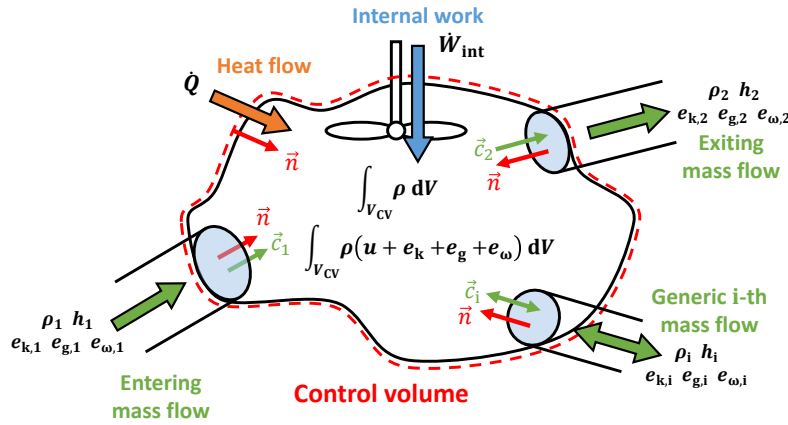


Fig. 3.2 Schematic representation of the Eulerian formulation for an open thermodynamic system.

Accordingly, the Eulerian Law of Conservation of Mass (LCM) for open systems is employed to compute the mass balance at the control volume boundaries, as given in Eq. (3.1):

$$\frac{d}{dt} \int_{V_{CV}} \rho dV = \sum_{i=1}^n \int_{A_i} \rho_i \vec{c}_i \cdot \vec{n}_i dA \quad (3.1)$$

where $\frac{d}{dt} \int_{V_{CV}} \rho dV$ is the mass variation rate within the control volume CV , and $\int_{A_i} \rho_i \vec{c}_i \cdot \vec{n}_i dA$ is the i -th mass flow rate $\dot{m}_i$ crossing the boundary section A_i , with a flow density ρ_i and an average velocity $\vec{c}_i$. The sign convention is defined according to the reference system shown in Fig. 3.2, which assumes normal vectors $\vec{n}_i$ pointing inside the control volume. Under this assumption, mass fluxes entering the control

volume produce a positive contribution to the mass variation, whereas fluxes exiting the control volume result in a negative term.

The Eulerian form of the First Law of Thermodynamics (FLT) for open systems is then used to model energy transfer, as expressed in Eq. (3.2):

$$\frac{d}{dt} \int_{V_{CV}} \rho (u + e_k + e_g + e_\omega) dV = \dot{Q} + \dot{W}_{int} + \sum_{i=1}^n \left[\int_{A_i} \rho_i (h_i + e_{k,i} + e_{g,i} + e_{\omega,i}) \vec{c}_i \cdot \vec{n}_i dA \right] \quad (3.2)$$

where $\int_{V_{CV}} \rho (u + e_k + e_g + e_\omega) dV$ is the total energy stored within the control volume, $\dot{Q}$ and $\dot{W}_{int}$ are the heat transfer and internal work rates applied to the control volume, and $\int_{A_i} \rho_i (h_i + e_{k,i} + e_{g,i} + e_{\omega,i}) \vec{c}_i \cdot \vec{n}_i dA$ represents the energy content transported by a mass flow traversing the control volume boundary. The u , e_k , e_g , and e_ω terms denote the specific internal, kinetic, gravitational, and rotational energies of the control volume or its fluxes, while h expresses the corresponding specific enthalpy. Consistent with the Law of Conservation of Mass, the sign convention for the FLT is defined according to the reference system from Fig. 3.2. Under this assumption, heat transfers, internal work, and mass flows entering the control volume contribute positively to its energy balance, whereas corresponding exiting fluxes contribute negatively to the control volume balance.

In the following sections, both governing equations are applied to each relevant element of the F-TMS and specialized for numerical implementation and solving.

3.3.1 The heat source

The thermal load managed by the F-TMS originates from the heat source, here represented by the fuel cell stack. In typical configurations, heat is transferred from the stack to the cooling system without any mass exchange. Accordingly, from the perspective of the TMS, the heat source can be modelled as a diabatic, closed system.

Under these assumptions, the surface-integral terms can be omitted from the LCM and FLT, so that the LCM reduces to Eq. (3.3):

$$\frac{d}{dt} \int_{V_{FC}} \rho dV = 0 \quad (3.3)$$

whereas the energy balance equation simplifies to Eq. (3.4) by neglecting the internal work rate:

$$\frac{d}{dt} \int_{V_{FC}} \rho (u + e_k + e_g + e_\omega) dV = \dot{Q}_{FC} \quad (3.4)$$

where $\dot{Q}_{FC}$ is the net heat transfer rate at the FC boundary, defined as the difference between the thermal waste generated by the FC, $\dot{Q}_{WH}$, and the heat extraction rate imposed by the upstream coolant HE, $\dot{Q}_{HE,U}$. The kinetic, gravitational, and rotational energy variations are also negligible for the present analysis, leading to the simplified expression of Eq. (3.5):

$$\frac{d}{dt} \int_{V_{FC}} \rho u dV = \dot{Q}_{WH} - \dot{Q}_{HE,U} \quad (3.5)$$

Provided the specific internal energy approximation of $u = c_v T$, the fuel cell energy balance reduces to Eq. (3.6):

$$\frac{d(\rho V c_v T)_{FC}}{dt} = \dot{Q}_{WH} - \dot{Q}_{HE,U} \quad (3.6)$$

This formulation can finally be expressed as an Ordinary Differential Equation (ODE) for the fuel cell temperature and solved according to Eq. (3.7):

$$\dot{T}_{FC}(t) = \frac{\dot{Q}_{WH}(t) - \dot{Q}_{HE,U}(t)}{(\rho V c_v)_{FC}} \quad (3.7)$$

where $(\rho V c_v)_{FC}$ is the thermal inertia of the fuel cell stack and $T_{FC}(t)$ its average temperature. The fuel cell thermal inertia is here modelled proportionally to the stack's maximum power output, $\dot{W}_{FC,max}$, according to Eq. (3.8):

$$(\rho V c_v)_{FC} = \frac{\dot{W}_{FC,max}}{\left(\frac{P}{m}\right)_{FC}} c_{v,FC} \quad (3.8)$$

where $\left(\frac{P}{m}\right)_{FC}$ and $c_{v,FC}$ are the specific power and average specific heat of the fuel. For the present analysis, a nominal specific power of 0.5 kW kg^{-1} is assumed, together with a reference specific heat of $1.2 \text{ kJ kg}^{-1} \text{ K}^{-1}$ [257].

Inside the model, a single digital component represents the fuel cell heat source, implementing Eq. (3.7) as illustrated in Fig. 3.3. Within this module, the waste heat generation rate $\dot{Q}_{WH}$ is computed from the aircraft mechanical power demand at the electric-powertrain. The conversion follows the efficiency assumptions introduced in Sect. 1.2.1 and is here condensed to Eq. (3.9):

$$\dot{Q}_{WH} = \frac{1 - \eta_{FC}}{\eta_{FC} \eta_{line} \eta_{EM}} N_{EM} \cdot \dot{W}_{EM,mech} \quad (3.9)$$

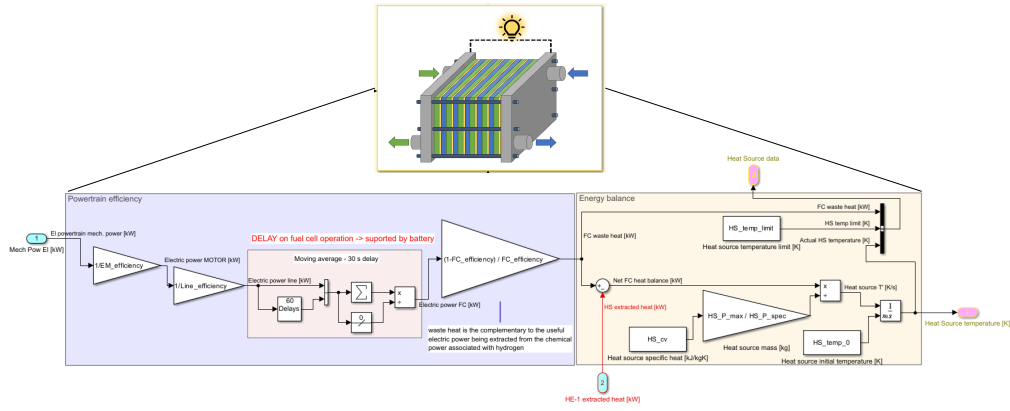


Fig. 3.3 Overview of the numerical model of the heat source component.

where η_{FC} , η_{line} , and η_{EM} denote the fuel cell, power line, and electric motor efficiencies, and $N_{EM}\dot{W}_{EM,mech}$ is the total mechanical power delivered by the electric powertrain. To reflect the inherently slow thermal response of the stack, a 30 s moving average is applied to the power-to-heat transformation. Higher-frequency power fluctuations are assumed to be buffered by an ideal battery, whose explicit modelling remains beyond the scope of the present analysis.

Finally, Eq. (3.7) is numerically integrated to obtain the transient fuel cell temperature $T_{FC}(t)$, provided an initial condition $T_{FC}(0)$. The resulting temperature history is subsequently propagated to the coolant-loop subsystem to determine and regulate the heat extraction term $\dot{Q}_{HEU}(t)$.

3.3.2 The coolant loop

The coolant loop is responsible for extracting the waste heat generated by the heat source and transferring it to the heat sink, here represented by the fuel system. A complete loop typically includes one or more pumps, a control system, a deionization filter, an expansion tank, a reserve tank, and two heat exchangers: one transferring the heat from the source to the coolant (HE_U), and another rejecting it from the coolant to the sink (HE_D). A simplified representation of the coolant system is adopted, consistent with the current level of abstraction of this functional analysis. It considers only the principal components, relying on lumped-parameter models of the two heat exchangers and an integrated controller–pump element.

Heat exchangers can conceptually be decomposed into three main elements: a hot side, a cold side, and an intermediate structure. Each element can be associated with a reference control volume and analysed via the corresponding governing equations for mass and energy conservation. Focusing on the coolant, it serves as the cold fluid in the upstream heat exchanger (HE_U) and as the hot fluid in the downstream heat exchanger (HE_D). Accordingly, the fuel cell constitutes the hot side in the upstream heat exchanger, while the fuel system acts as the cold side in the downstream heat exchanger.

The two heat exchangers are implemented as separate digital components within the numerical model, but remain governed by the same underlying mass and energy formulations. Focusing on the coolant side of both elements, the transient term of Eq. (3.1) can be neglected, as the heat exchangers are supposed to remain filled throughout the entire operation, without net changes to the mass contained within the control volumes.. The surface integrals are retained, so that the LCM for the coolant side of a heat exchanger is reduced to Eq. (3.10), under the assumption of a single inlet and outlet port:

$$\int_{A_{HE,cool,in}} \rho_{in} \vec{c}_{in} \cdot \vec{n}_{in} dA = \int_{A_{HE,cool,out}} \rho_{out} \vec{c}_{out} \cdot \vec{n}_{out} dA \quad (3.10)$$

where the left-hand side of the equation corresponds to the coolant mass flow rate entering a generic HE $(\dot{m}_{HE,in})_{cool}$, and the right-hand side represents the associated outlet flow, $(\dot{m}_{HE,out})_{cool}$. Assuming that the two heat exchangers are arranged

in series without any bypasses, a single mass flow rate $\dot{m}_{\text{cool}} = (\dot{m}_{\text{HEU,in}})_{\text{cool}} = (\dot{m}_{\text{HED,out}})_{\text{cool}}$ can be adopted for the entire coolant loop.

A digital pump sub-model regulates the coolant mass flow rate via a saturated PID (Proportional-Integral-Derivative) controller driven by the upstream fuel cell temperature signal. The controller linearly increases the coolant flow rate from zero to the maximum capacity, $\dot{m}_{\text{cool,max}}$, as the heat source temperature increases from 100 °C to 115 °C, where the maximum flow rate is reached. This upper temperature threshold is set just below the fuel cell's maximum operating temperature of 120 °C. If the required cooling flow exceeds the maximum rate $\dot{m}_{\text{cool,max}}$, heat removal becomes insufficient, leading to a progressive rise in heat source temperature. The model allows the fuel cell temperature to exceed the nominal 120 °C threshold, facilitating the identification of critical thermal conditions in the simulation, under the assumption that no active safety measures are implemented to prevent thermal runaway and that fuel cell operation is not degraded by elevated temperatures.

Eq. (3.11) provides the governing equation for the coolant-loop flow control:

$$\dot{m}_{\text{cool}}(t) = \text{sat}_{[0, \dot{m}_{\text{cool,max}}]} \left(\frac{\dot{m}_{\text{cool,max}}}{\Delta T_{\text{cool,rbw}}} [T_{\text{HS}}(t) - (T_{\text{cool,sat}} - \Delta T_{\text{cool,rbw}})] \right) \quad (3.11)$$

where $\frac{\dot{m}_{\text{cool,max}}}{\Delta T_{\text{cool,rbw}}}$ is the proportional gain of the controller, and $\dot{m}_{\text{cool,max}}$ is the saturated coolant mass flow rate. $T_{\text{cool,sat}}$ is the heat source temperature at which the coolant flow rate reaches saturation (115 °C in this case), whereas $\Delta T_{\text{cool,rbw}}$ is the controller regulation bandwidth (15 °C).

The FLT on the coolant side of the heat exchangers simplifies to Eq. (3.12) by neglecting the internal work rate and the specific kinetic, potential, and rotational energy terms:

$$\begin{aligned} \dot{Q}_{\text{HE}} + \int_{A_{\text{HE,cool,in}}} \rho_{\text{in}} h_{\text{in}} \vec{c}_{\text{in}} \cdot \vec{n}_{\text{in}} dA - \int_{A_{\text{HE,cool,out}}} \rho_{\text{out}} h_{\text{out}} \vec{c}_{\text{out}} \cdot \vec{n}_{\text{out}} dA \\ + \frac{d}{dt} \int_{V_{\text{HE,cool}}} \rho u dV = \end{aligned} \quad (3.12)$$

Given the previously defined mass flow rate, the $u = c_v T$ assumption, and the corresponding $h = c_p T$ formulation, the energy balance reduces to Eq. (3.13). Omitting the cool subscript for clarity, the governing equation for the coolant side of

a generic HE is:

$$\frac{d(mc_v T)_{\text{HE}}}{dt} = \dot{Q}_{\text{HE}} + \dot{m} c_{p,\text{HE},\text{in}} T_{\text{HE},\text{in}} - \dot{m} c_{p,\text{HE},\text{out}} T_{\text{HE},\text{out}} \quad (3.13)$$

If the coolant remains in the liquid phase across the entire operating range, the specific heats at constant volume and pressure can be considered approximately equal, $c_v \approx c_p$. This assumption is consistent with the behaviour of pressurized water–glycol mixtures, which remain liquid up to 120 °C. Additionally, the moderate temperature differences between the HE inlets and outlets justify treating the specific heat as uniform $c_{p,\text{cool}} \approx (c_{p,\text{HE},\text{in}})_{\text{cool}} \approx (c_{p,\text{HE},\text{out}})_{\text{cool}}$, leading to the formulation in Eq. (3.14), where the "cool" subscript is omitted hereafter for clarity:

$$\frac{dT_{\text{HE}}}{dt} = \frac{1}{m_{\text{HE}}} \left(\frac{\dot{Q}_{\text{HE}}}{c_p} + \dot{m} (T_{\text{HE},\text{in}} - T_{\text{HE},\text{out}}) \right) \quad (3.14)$$

where m_{HE} is the coolant mass contained within the heat exchanger and T_{HE} is its average temperature. The coolant is assumed to be a 50%-50% water–glycol mixture with an approximate specific heat capacity, $c_{p,\text{cool}}$, of 3.73 kJ kg⁻¹ K⁻¹ at 100 °C.

In lumped parameter practice, mass flows leaving the control volume are assumed at the control volume's average temperature, $T_{\text{HE},\text{out}} = T_{\text{HE}}$. Under this assumption, the energy balance for the coolant side of a generic heat exchanger can be written as Eq. (3.15), which represents the first-order ODE governing the bulk temperature of the HE coolant:

$$\dot{T}_{\text{HE}}(t) = \frac{1}{m_{\text{HE}}} \left(\frac{\dot{Q}_{\text{HE}}(t)}{c_p} + \dot{m}(t) (T_{\text{HE},\text{in}}(t) - T_{\text{HE}}(t)) \right) \quad (3.15)$$

with the heat transfer rate $\dot{Q}_{\text{HE}}(t)$ typically assuming a positive sign for the coolant in the upstream HE (HE_U), and a negative sign in the downstream HE (HE_D). The equation is solved numerically within the model, given an initial coolant temperature $T_{\text{HE},\text{cool}}(0)$. The resulting temperature derivative is then integrated to determine the corresponding temperature history $T_{\text{HE},\text{cool}}(t)$ of the selected HE.

The heat transfer rate $\dot{Q}_{\text{HE}}(t)$ is computed within intermediate HE structure sub-models, connecting the systems acting as the hot and cold sides. In principle,

both LCM and FLT also apply to the HE structure; however, as a first approximation, the thermal dynamics of the solid medium can be neglected and the heat transfer simplified by assuming that all heat released by the hot side is instantaneously absorbed by the cold one $\dot{Q}_{\text{HE}} = \dot{Q}_{\text{HE,h}} = \dot{Q}_{\text{HE,c}}$.

The heat exchanger thermal effectiveness formulation of Eq. (3.16) is adopted to approximate this unified heat transfer rate:

$$\dot{Q}_{\text{HE}} = \varepsilon_{\text{HE}} C_{\min} (T_{\text{HE,h,in}} - T_{\text{HE,c,in}}) \quad (3.16)$$

where ε_{HE} is the heat exchanger thermal effectiveness, $T_{\text{HE,h,in}}$ and $T_{\text{HE,c,in}}$ are the inlet temperatures of the hot and cold fluids, and $C_{\min}$ is the minimum heat capacity rate between the two streams.

For the upstream coolant heat exchanger (HE_{U}), this minimum capacity rate reduces to Eq. (3.17):

$$C_{\min} = c_{\text{p,cool}} \dot{m}_{\text{cool}} \quad (3.17)$$

whereas for the downstream coolant heat exchanger (HE_{D}), it is given by Eq. (3.18):

$$C_{\min} = \min(c_{\text{p,cool}} \dot{m}_{\text{cool}}, c_{\text{p,fuel}} \dot{m}_{\text{HE,fuel}}) \quad (3.18)$$

Within the numerical model, Eq. (3.16) is solved in dedicated HE-structure modules connecting the fuel cell to the coolant loop, and the coolant loop the fuel system, as illustrated in Fig. 3.4. A constant heat exchanger effectiveness of $\varepsilon_{\text{HE}} = 0.7$ is assumed for both components to reflect realistic operating conditions, while detailed geometric characterisation and effectiveness mapping remain beyond the scope of the present analysis.

3.3.3 The fuel system

The fuel system receives the heat transported by the coolant loop through the downstream coolant heat exchanger (HE_{D}), recirculating the heated fuel back to the tanks where the thermal load is ultimately stored. In addition to thermal management, the fuel system is primarily responsible for delivering the required fuel supply to the

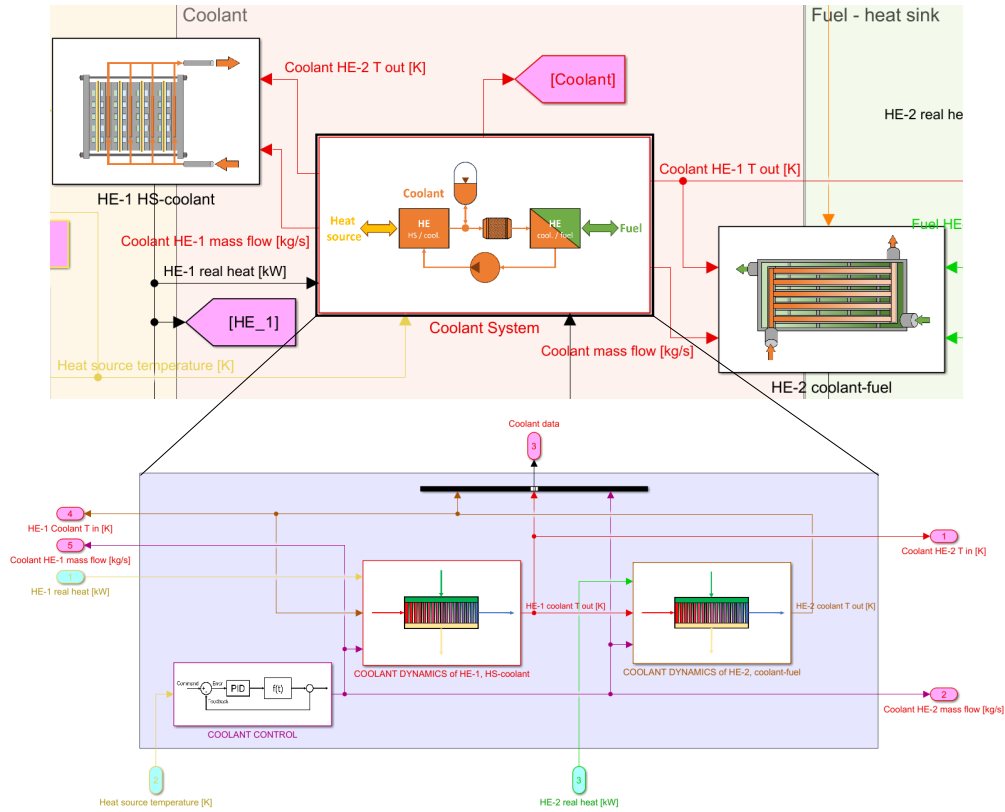


Fig. 3.4 Overview of the numerical model of the coolant loop subsystem and outer HE structure elements connecting to the FC and fuel system.

engines. Reflecting this dual purpose, the fuel system model comprises several key elements, including: the fuel side of HE_D , a virtual controller-pump, the engines, and all fuel tanks.

Fuel heat exchanger

The fuel-side HE sub-model is the first of these elements. It receives the heat transfer signal from the intermediate structure module of HE_D and applies it to the fuel. The same considerations adopted for the coolant side of the HE in Sect. 3.3.2 are applied here, adjusted to reflect the thermophysical properties of standard jet-A fuel. Accordingly, the mass and energy balances for the fuel side of the downstream HE can be expressed as adapted versions of Eqs. (3.10) and (3.15).

However, in contrast to the continuous coolant loop, the fuel system includes multiple branches, bypasses, and diverse users. As a result, the mass flow passing through the fuel-side heat exchanger cannot be generalised to a universal flow rate, $\dot{m}_{\text{HE}_D, \text{fuel}} \neq \dot{m}_{\text{fuel}}$, as differences exist between the tank-entering, tank-exiting, recirculated, and consumed fuel mass flows.

Also in this case, a digital controller-pump element regulates the fuel mass flow through the heat exchanger $\dot{m}_{\text{HE}_D, \text{fuel}}$ using a saturated PID scheme, driven by the HE_D coolant inlet temperature signal. The controller linearly increases the fuel flow rate from zero to $\dot{m}_{\text{fuel}, \text{max}}$, activating once the coolant inlet temperature reaches 110°C and saturating when it exceeds 118°C . For the present configuration, this inlet coolant temperature at HE_D is effectively equal to the upstream HE outlet temperature, $T_{\text{HE}_U, \text{cool}}(t)$.

Similarly to the coolant case, Eq. (3.19) provides the governing equation for the fuel HE cooling controller:

$$\dot{m}_{\text{HE}_D, \text{fuel}}(t) = \text{sat}_{[0, \dot{m}_{\text{fuel}, \text{max}}]} \left(\frac{\dot{m}_{\text{fuel}, \text{max}}}{\Delta T_{\text{fuel}, \text{rbw}}} [T_{\text{HE}_U, \text{cool}}(t) - (T_{\text{fuel}, \text{sat}} - \Delta T_{\text{fuel}, \text{rbw}})] \right) \quad (3.19)$$

with the proportional gain and other terms opportunely updated to $T_{\text{fuel}, \text{sat}} = 118^\circ\text{C}$, and $\Delta T_{\text{fuel}, \text{rbw}} = 8^\circ\text{C}$.

Also in this case, if the required cooling surpasses $\dot{m}_{\text{fuel}, \text{max}}$, heat removal becomes insufficient and leads to progressive heating of the heat source beyond the 120°C upper operating temperature limit.

Fuel tank - internal dynamics

The fuel tank is responsible for storing the fuel mass and supplying it on demand to the heat exchangers and the engines. Depending on the considered F-TMS architecture, fuel tanks may also receive substantial recirculated fuel flows, exchange heat with the external environment, or absorb thermal loads from internally mounted components, such as a submerged heat exchanger. Under these assumptions, the fuel tank behaves as an open, potentially diabatic system, and its dynamics can be described using the Eulerian mass (LCM) and energy balance (FLT) lumped-parameter equations. Consistent with the previously stated assumptions and with the relevant literature on lumped-parameter F-TMS models, physical properties

are considered homogeneous, isotropic, and spatially uniform within each tank. Assuming that all fuel tanks share the same average fuel temperature, and that no distinction between hot and cold tanks is made in the reference architecture, the mass balance for the combined fuel tanks control volume can be expressed as Eq. (3.20):

$$\frac{d}{dt} \int_{V_{FT}} \rho dV = \sum_{i=1}^n \int_{A_{FT,in,i}} \rho_i c_i dA - \sum_{j=1}^n \int_{A_{FT,out,j}} \rho_j c_j dA \quad (3.20)$$

where $\frac{d}{dt} \int_{V_{FT}} \rho dV$ represents the rate of change of the total fuel mass within the tanks, while the first and second terms on the right-hand side correspond to the net entering and net exiting mass flow rates, respectively. The entering mass flows are distinguished into recirculated flows returning to the tank, $\dot{m}_{rec}$, and filling flows supplied by external sources (e.g., during refuelling), $\dot{m}_{fill}$. The exiting mass flows comprise, instead, the engine feed flows, $\dot{m}_{eng}$, recirculation flows leaving the tank, $\dot{m}_{rec}$, and drain flows discharging fuel to an external sink, $\dot{m}_{drain}$. Accordingly, the fuel tank mass balance can be rewritten as Eq. (3.21):

$$\frac{dm_{FT}}{dt} = (\dot{m}_{rec} + \dot{m}_{fill}) - (\dot{m}_{eng} + \dot{m}_{rec} + \dot{m}_{drain}) \quad (3.21)$$

Provided that the reference control volume encompasses all aircraft fuel tanks, the only filling and draining flow rates are those associated with maintenance and refuelling operations, which are not considered in the present simulation. Accordingly, the fuel tank mass balance reduces to Eq. (3.22), as the recirculating mass flow terms cancel out within the global control volume:

$$\frac{dm_{FT}}{dt} = -\dot{m}_{eng} \quad (3.22)$$

Accordingly, unlike in the previous cases, the fuel tank exhibits a non-zero time derivative of its mass, which is equal and opposite to the engine fuel consumption rate. Given an initial tank fuel mass $m_{FT}(0)$, the expression is reformulated for numerical integration as the first-order ODE governing the fuel tank mass, according to Eq. (3.23):

$$\dot{m}_{FT}(t) = -\dot{m}_{eng}(t) \quad (3.23)$$

Similar to previous cases, the energy balance for the fuel tank component can be simplified to Eq. (3.24) by neglecting the internal work rate and the specific kinetic, potential, and rotational energy terms:

$$\frac{d}{dt} \int_{V_{FT}} \rho(u) dV = \dot{Q}_{FT} + \sum_{i=1}^n \left[\int_{A_{FT,i}} \rho_i(h_i) \vec{c}_i \cdot \vec{n}_i dA \right] \quad (3.24)$$

where $\dot{Q}_{FT}$ is the net heat transfer rate across the fuel tank control surface, and accounts for the tank's radiative and convective exchange with the environment, as well as any internal submerged heat sources. Assuming unidimensionality of the surface integrals, uniformity and homogeneity of the volume integral, absence of tank refilling or draining, and the approximations $u = c_v T$ and $h = c_p T$, the energy balance is reduced to Eq. (3.25), with outlet flows sharing the same temperature as the control volume $T_{FT,out} = T_{FT}$:

$$\frac{d(mc_v T)_{FT}}{dt} = \dot{Q}_{FT} + \dot{m}_{rec} c_{p,rec} T_{rec} - \dot{m}_{eng} c_{p,FT} T_{FT} - \dot{m}_{rec} c_{p,FT} T_{FT} \quad (3.25)$$

The reference fuel for this study is commercial Jet-A fuel, a kerosene-based mixture whose boiling point exceeds 150 °C. Accordingly, as for the coolant, the fuel is expected to remain liquid across the entire operational range, justifying the assumption $c_{v,fuel} = c_{p,fuel}$. As a first approximation, small variations in the fuel specific heat between entering and exiting flows can also be neglected, i.e. $c_{p,rec} \approx c_{p,FT} \approx c_{p,fuel}$. The resulting control volume time derivative $c_{p,fuel} \frac{d(mT)_{FT}}{dt}$, is solved by applying the Leibniz rule, as, unlike in the previous cases, both mass and temperature are time-dependent. Under these assumptions, Eq. (3.26) returns the fuel tank energy balance:

$$c_{p,fuel} \left(m_{FT} \frac{dT_{FT}}{dt} + T_{FT} \frac{dm_{FT}}{dt} \right) = \dot{Q}_{FT} + c_{p,fuel} (\dot{m}_{rec} T_{rec} - \dot{m}_{eng} T_{FT} - \dot{m}_{rec} T_{FT}) \quad (3.26)$$

where m_{FT} is the instantaneously stored fuel mass inside the tank, and T_{FT} its average temperature. Eq. (3.22) can then be applied to solve the time derivative of the fuel mass, yielding Eq. (3.27):

$$m_{FT} \frac{dT_{FT}}{dt} + T_{FT} \left(-\dot{m}_{eng} \right) = \frac{\dot{Q}_{FT}}{c_{p,fuel}} + \dot{m}_{rec} (T_{rec} - T_{FT}) - \dot{m}_{eng} T_{FT} \quad (3.27)$$

$$\dot{T}_{\text{FT}}(t) = \frac{1}{m_{\text{FT}}(t)} \left(\frac{\dot{Q}_{\text{FT}}(t)}{c_{\text{p,fuel}}} + \dot{m}_{\text{rec}}(t) (T_{\text{rec}}(t) - T_{\text{FT}}(t)) \right) \quad (3.28)$$

Fuel tank - boundary dynamics

The fuel tank model is completed by defining the boundary heat transfer term, $\dot{Q}_{\text{FT}}(t)$, representing the net heat flux exchanged between the tank control volume and the external environment. Following Kellermann et al. [243], the wing tanks are approximated as two identical cuboids. In the absence of submerged heat sources or internal heat exchangers, heat transfer reduces to the radiative exchange with the surroundings and the convective dissipation to the ambient air.

Radiative heat transfer comprises two distinct terms: the absorbed solar radiation and the rejected surface thermal emission expressed in Eq. (3.29):

$$\dot{Q}_{\text{rad}} = \alpha_{\text{rad}} \cdot P_{\text{rad}} - A_{\text{FT}} \epsilon_{\text{rad}} \sigma_{\text{rad}} (T_{\text{FT}}^4 + T_{\infty}^4) \quad (3.29)$$

where α_{rad} denotes the tank solar absorptance, ϵ_{rad} its emissivity, A_{FT} the exposed surface area, and P_{rad} the solar irradiation incident on the upper tank surface. T_{inf} is the undisturbed ambient temperature, while σ_{rad} corresponds to the Stefan–Boltzmann constant.

Kellermann et al. [243] propose the following reference values for commercial civil aviation: $\epsilon_{\text{rad}} = 0.5$, $\alpha_{\text{rad}} = 0.25$, and $P_{\text{rad}}/A = 1100 \text{ W m}^{-2}$. However, the authors' findings demonstrate that solar irradiation has a negligible impact in partially emptied tanks, due to the insulating effect of the ullage. Accordingly, the absorbed solar radiation is neglected herein, and radiative emission is assumed to occur primarily via the tank floor, $A_{\text{FT}} = A_{\text{FT, floor}}$, for analogous reasons. Hence, the radiative heat transfer formulation reduces to Eq. (3.30):

$$\dot{Q}_{\text{rad}} = -A_{\text{FT, floor}} \epsilon_{\text{rad}} \sigma_{\text{rad}} (T_{\text{FT}}^4 + T_{\infty}^4) \quad (3.30)$$

Following Ref. [243], analogous considerations simplify the modelling of convective heat transfer. Contributions from the vertical walls are neglected owing to their limited wetted area and low surrounding airflow velocities, while heat exchange across the upper tank surface is disregarded because of the insulating effect of the gaseous ullage–environment interface. Under these assumptions, convective heat transfer is also assumed to occur primarily through the combined wing tank floors. Accordingly, the convective heat transfer area reduces to the combined wing tank floor area, $A_{\text{FT, floor}}$, with Eq. (3.31) providing the governing expression:

$$\dot{Q}_{\text{conv}} = \frac{A_{\text{FT, floor}}}{R_{\text{eq}}} \Delta T \quad \text{with: } R_{\text{eq}} = \frac{1}{h_{\text{conv, air}}} + \frac{H_{\text{floor}}}{k_{\text{floor}}} + \frac{1}{h_{\text{conv, fuel}}} \quad (3.31)$$

where R_{eq} is the equivalent thermal resistance opposing the convective heat transfer, $\dot{Q}_{\text{conv}}$, between two fluids at a temperature difference ΔT . H_{floor} is the wall

thickness of the fuel tank floor, while the remaining terms represent: the convective heat transfer coefficient of air, $h_{\text{conv,air}}$; the tank floor thermal conductivity, k_{floor} ; and the convective heat transfer coefficient of fuel, $h_{\text{conv,fuel}}$.

In this context, the symbol h consistently denotes convective heat transfer coefficients, in accordance with heat transfer literature conventions. Due to the inherently interdisciplinary nature of the present work, the adopted formulations integrate conventions and nomenclatures from different technical domains, such as heat transfer, thermodynamics, and mechanical engineering. To preserve consistency with the established literature in each field, standard symbols have been retained even in cases of partial overlap in notation. Any resulting ambiguities are clarified in the nomenclature section of this dissertation and, where necessary, at their first occurrence in the main text.

The temperature difference between the fuel tank and the surrounding air is defined as $\Delta T = T_{\text{FT}} - T_{\text{air,recov}}$, where $T_{\text{air,recov}}$ denotes the recovery temperature of air in close proximity to the tank, calculated according to Eq. (3.32) [238]. The "air" subscript is omitted to improve readability:

$$T_{\text{recov}} = T_{\infty} \left(1 + \frac{\gamma - 1}{2} \cdot \sqrt{Pr} \cdot Ma^2 \right) \quad (3.32)$$

This equation describes the localised temperature rise resulting from the inertial compression and aerodynamic heating of air contacting the aircraft, with $\gamma = \frac{c_p}{c_v}$ being the air heat capacity ratio, Pr being the Prandtl number and Ma the aircraft Mach number.

Aluminium is assumed as the reference material for the fuel tank floor, offering a high thermal conductivity of over $100 \text{ W m}^{-1} \text{ K}^{-1}$ [258, 259]. A conservative floor thickness of 3 mm and a thermal conductivity of $150 \text{ W m}^{-1} \text{ K}^{-1}$ are adopted, consistent with typical aerospace practice.

Forced convection is expected to dominate air-side heat transfer during take-off, climbs, cruises, and descents. Roland and Rumpfkeil [241] demonstrated good agreement between classical forced-convection correlations for flat surfaces and detailed simulations of convective heat transfer over NACA airfoils. Accordingly, Eq. (3.33) is employed to estimate the air-side Nusselt number under non-zero aircraft velocities. The "air" subscript is omitted in the empirical correlation for

improved readability:

$$Nu_{\text{air}} = \frac{h_{\text{air}} W_{\text{floor}}}{k_{\text{air}}} = \begin{cases} 0.664 Re^{1/2} Pr^{1/3}, & \text{if } Pr > 0.6 \text{ and } Re < 5 \cdot 10^5 \\ 0.037 (Re^{4/5} - 871) Pr^{1/3}, & \text{if } Pr > 0.6 \text{ and } Re \geq 5 \cdot 10^5 \end{cases} \quad (3.33)$$

where W_{floor} is the characteristic length of the formulation, corresponding to the fuel tank length in the flow direction, and k_{air} is the thermal conductivity of air, assumed equal to $0.025 \text{ W m}^{-1} \text{ K}^{-1}$.

During ground operations, natural convection predominates due to near-zero external airflow velocities. Preliminary calculations using Nusselt correlations for a horizontal flat plate indicate negligible heat rejection, in agreement with observations by Kellermann et al. [243]. Accordingly, air-side heat transfer under natural convection is neglected, providing a simplified and conservative approach.

When modelling fuel-side heat transfer across the tank floor, natural convection is assumed to dominate, since no active internal mixing devices are considered at this stage. Although limited forced convection may arise from structural vibrations or sloshing, heat transfer is conservatively approximated to be governed by pure natural convection, consistent with high-level analyses in the literature [243].

Within a F-TMS configuration, fuel temperatures are generally expected to exceed those of the external air cooling the tank floor. The corresponding convective model is that of a hot fluid overlying a colder horizontal surface. However, this arrangement suppresses buoyancy-driven motion [260], leading to thermal stratification near the tank floor and to limited, conduction-dominated heat transfer. Given the modest thermal conductivity of Jet fuel (approximately $0.145 \text{ W m}^{-1} \text{ K}^{-1}$), fuel-side heat rejection remains strongly limited under these assumptions.

Convective heat transfer could instead be favoured if the heat loads were introduced near the bottom of the tank, generating localised heating that drives buoyancy-induced circulation. Such a configuration may be achieved through the incorporation of a submerged heat exchanger, as proposed in Ref. [243], or via the strategic placement of low-lying piccolo tubes near the tank floor, supplying recirculated hot fuel to promote buoyancy-driven circulation, as illustrated in Fig. 3.6.

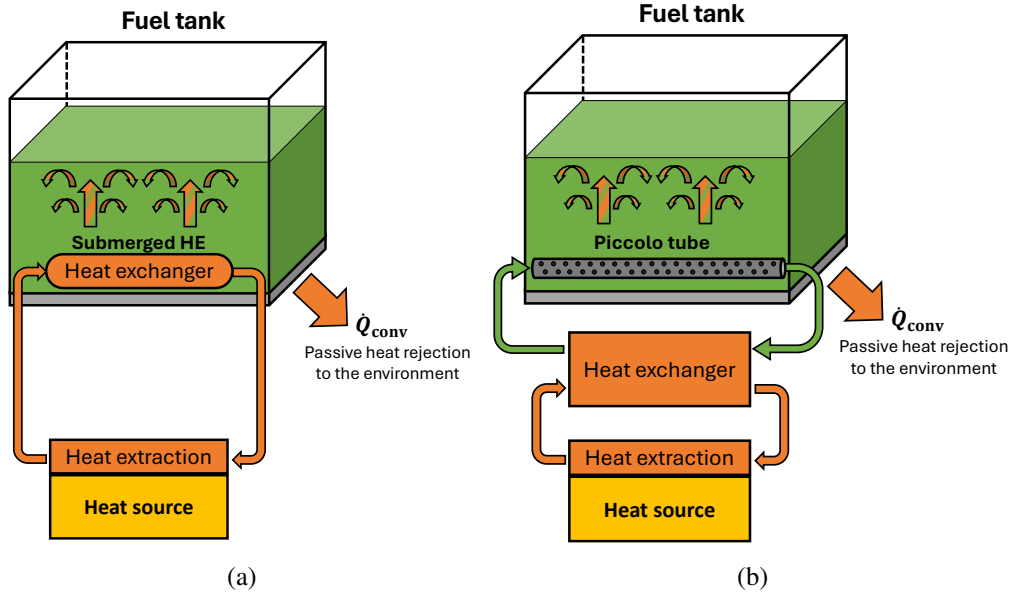


Fig. 3.6 Convection promoting configurations in heated fuel tanks. (3.6a) submerged heat exchanger-induced convection. (3.6b) low-lying piccolo tube-induced convection.

The Globe–Dropkin correlation for natural convection in rectangular cavities heated from below [261] is employed to approximate fuel-side convection for the configurations shown in Fig. 3.6. The corresponding formulation is given in Eq. (3.34), omitting the "fuel" subscript in the empirical expression for improved readability:

$$Nu_{\text{fuel}} = \frac{h_{\text{fuel}} H_{\text{fuel}}(t)}{k_{\text{fuel}}} = 0.069 Ra^{1/3} Pr^{0.074} \quad \text{for } 3 \cdot 10^5 \leq Ra \leq 7 \cdot 10^9 \quad (3.34)$$

where $H_{\text{floor}}(t)$ denotes the instantaneous depth of the fuel stock, which progressively decreases as fuel is consumed. Ra is the fuel Rayleigh number, expressed according to Eq. (3.35):

$$Ra = \frac{g \beta (T_{\text{HE},\text{fuel}} - T_{\text{FT}}) H_{\text{fuel}}(t)^3}{\nu \alpha_{\text{diff}}} \quad (3.35)$$

where g denotes the gravitational acceleration, β is the volumetric thermal expansion coefficient of the fluid, ν its kinematic viscosity, and $\alpha_{\text{diff}} = k/(\rho c_p)$ the thermal diffusivity.

However, Eq. (3.34) only provides a first-order approximation of the internal fuel tank behaviour. Several studies report that natural convection can be enhanced under vibrational excitation, with Refs. [262–264] observing 10–15% increases in heat transfer coefficients for vibrating tanks and horizontal flat plates. Dimensionless vibrational numbers, based on an induced velocity scale, are commonly used to model these effects. In particular, Shevtsova et al. [265] and Wang et al. [266] evaluate vibration-enhanced heat transfer using the Gershuni number, as defined in Eq. (3.36):

$$Gs = \frac{[a_{\text{vib}} \Omega_{\text{vib}} \beta (T_{\text{HE}} - T_{\text{FT}}) \cdot H_{\text{fuel}}(t)]^2}{2 \nu \alpha_{\text{diff}}} \quad (3.36)$$

where a_{vib} and $\Omega_{\text{vib}} = 2\pi f$ denote the amplitude and angular frequency of the excitation. Aircraft vibrations may vary widely depending on operating conditions, airframe design, and excitation sources. However, for wing structures in commercial aviation, amplitudes are typically on the order of millimetres, with dominant frequencies in the low tens of hertz, driven by global structural modes and propulsion-induced excitation [267–269]. For the present study, a representative amplitude of 10 mm and a frequency of 15 Hz are assumed.

The study by Guo et al. [270] demonstrates a strong correlation between the Nusselt and Gershuni numbers, while Huang et al. [271] introduce a $Nu \sim Gs^{1/2}$ correlation for vibration-induced heat transfer in microgravity environments. Wang et al. [266] report similar dependencies for closed systems under high oscillation frequencies ($\Omega > 1000$) and standard gravity. However, the existing heat transfer literature still does not provide a universal correlation for horizontal flat plates that concurrently accounts for the combined influence of Nu , Ra , and Gs . For engineering practice, Eq. (3.37) is proposed as a pragmatic approximation of vibration-induced heat transfer, reproducing the data from Huang et al. [271] with an average relative deviation below 10% over the range $10^5 < Gs < 10^9$:

$$Nu_{\text{fuel}} = \frac{h_{\text{fuel}} H_{\text{fuel}}(t)}{k_{\text{fuel}}} = 0.005 Gs^{1/2} \quad \text{for } 10^5 < Gs < 10^9 \quad (3.37)$$

Hence, as a first-order approximation, fuel-side convection is assumed to be governed by the dominant mechanism between buoyancy-driven and vibration-

induced convection, as expressed in Eq. (3.38):

$$Nu = \max(Nu(Ra), Nu(Gs)) \quad (3.38)$$

Sloshing effects, which may be triggered by structural vibrations, are not considered at this stage, as their intermittent nature and strong dependence on tank geometry and operating conditions make them unsuitable for inclusion in this preliminary functional analysis.

Within the model, a single fuel tank subsystem represents both the internal and boundary dynamics of the tank control volume. This element incorporates three dedicated sub-models: the internal dynamics governed by Eqs. (3.23) and (3.28), the radiative heat transfer implementing Eq. (3.30), and the convective heat exchange accounting for air, structural, and fuel convection under buoyancy-driven and vibration-induced mechanisms. Fig. 3.7 provides an overview of these three sub-models composing the global fuel tank system element.

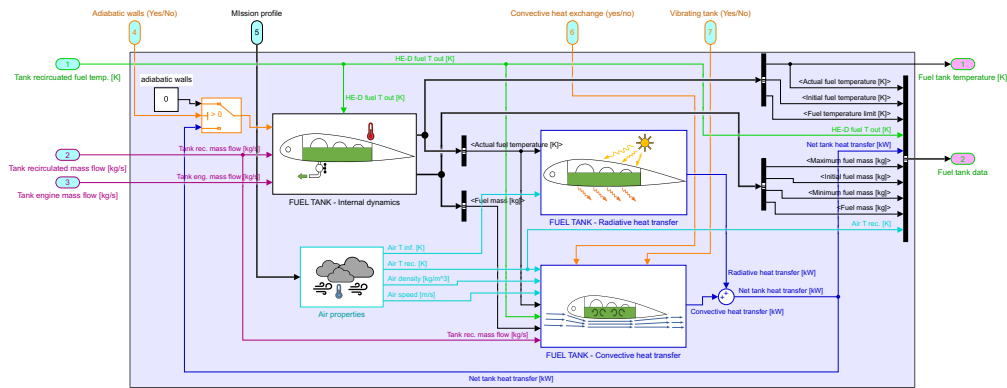


Fig. 3.7 Overview of the combined fuel tank internal and boundary numerical dynamics model.

A pre-emptive stop condition is also implemented to terminate the simulation when the stored fuel mass decreases below a minimum reserve threshold $m_{FT,min}$, thus preventing division-by-zero singularities in Eq. (3.28) as the fuel mass approaches depletion. The threshold is defined as the sum of the contingency fuel and final reserve fuel prescribed by EASA fuel planning regulations [272]. However, while this final fuel stock may theoretically be consumed under exceptional or critical

conditions, analyses of similar scenarios remain beyond the scope of the present study.

Aircraft engines

The final element of the integrated F-TMS is the turbine propulsion system. Although aircraft engines are typically associated with intense thermal phenomena, the present fuel cell-oriented F-TMS does not participate in their thermal management. Combustion processes and internal engine cooling are therefore excluded from the TMS boundaries, with the engines acting as adiabatic open systems from the TMS perspective. Under these assumptions, explicit modelling of engine mass and energy dynamics is unnecessary. Instead, the propulsion system acts solely as a mass sink to the fuel system, with the chemical energy associated with the supplied fuel mass flow being ultimately rejected to the environment via the exhaust stream.

Within the numerical model, all engines are represented by a single surrogate component converting the instantaneous turbine fuel burn rate into the corresponding mass flow rate demand, $\dot{m}_{\text{eng}}$, imposed on the tank model through Eqs. (3.23) and (3.28). The fuel burn rate history is instead provided as an external, time-dependent input, derived from the selected mission profile.

F-TMS architecture

Provided the engines fuel consumption rate, $\dot{m}_{\text{eng}}$, and the downstream coolant heat exchanger (HE_D) fuel demand, $\dot{m}_{\text{HE}_D, \text{fuel}}$, the recirculated fuel mass flow, $\dot{m}_{\text{rec}}$ remains to be defined. Assuming that the recirculated flow is supplied solely by the downstream coolant heat exchanger (HE_D), its temperature can be easily approximated as $T_{\text{rec}} = T_{\text{HE}_D, \text{fuel}}$. However, determining the actual recirculated mass flow is not as straightforward, as the model aims to compare single-tank F-TMS architectures with either parallel or series heat-exchanger recirculation.

In recirculating-parallel architectures, a separate pipeline supplies the required fuel mass flow to the HE, independently of the engines. Accordingly, the recirculated mass flow is simply equal to the HE demand, as in Eq. (3.39):

$$\dot{m}_{\text{rec}} = \dot{m}_{\text{HE}_D, \text{fuel}} \quad (3.39)$$

Conversely, in recirculating-series architectures, a single pipeline supplies fuel first to the HE and then to the engines. The exiting flow from the HE is adjusted to match the instantaneous turbine fuel demand, while the remainder is returned to the tank as recirculated flow. However, if the fuel demand from the engines exceeds that of the HE, a surplus mass flow is sent through the HE, so that $\dot{m}_{\text{HE}_D, \text{fuel}} = \dot{m}_{\text{eng}}$, without any recirculation. Accordingly, the recirculated mass flow is expressed following Eq. (3.40):

$$\dot{m}_{\text{rec}} = \max(0, \dot{m}_{\text{HE}_D, \text{fuel}} - \dot{m}_{\text{eng}}) \quad (3.40)$$

These correlations are implemented within the model through an expanded architecture-controller-pump element. This component first regulates the fuel supply to the coolant-fuel HE via the PID control scheme. Then, it combines the saturated cooling flow with the engine flow signal to determine the recirculated mass flow for the selected configuration, according to either Eq. (3.39) or Eq. (3.40). Finally, it propagates the computed $\dot{m}_{\text{HE}, \text{fuel}}$ to the HE_D fuel module, and $\dot{m}_{\text{rec}}$ to the fuel tank and fuel HE elements for subsequent calculations.

Combined with the fuel tank, fuel heat exchanger, and engine components, the architecture-controller-pump element completes the fuel system model, as illustrated in Fig. 3.8.

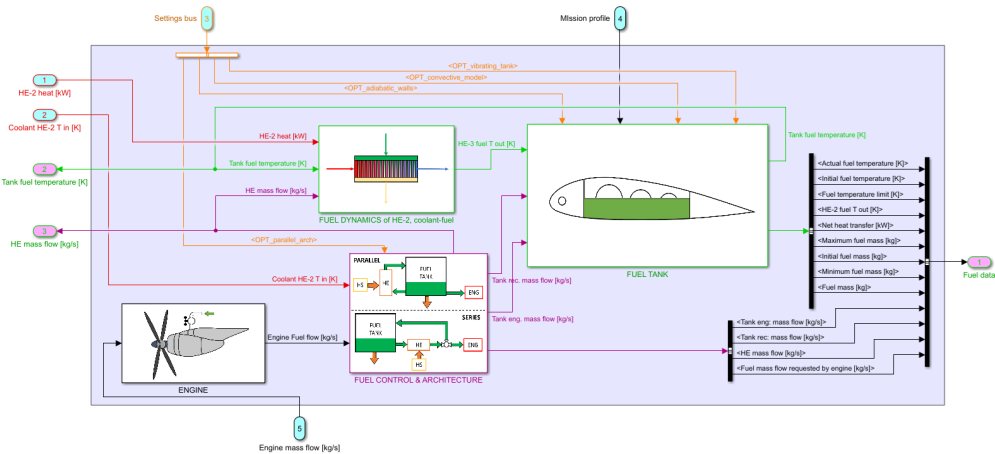


Fig. 3.8 Overview of the numerical model of the complete fuel subsystem.

3.3.4 The complete F-TMS model

Finally, a complete F-TMS model assembles the previously described heat source, coolant loop, and fuel system components in Simulink, and complements them with an option block and a mission planner module, as illustrated in Fig. 3.9.

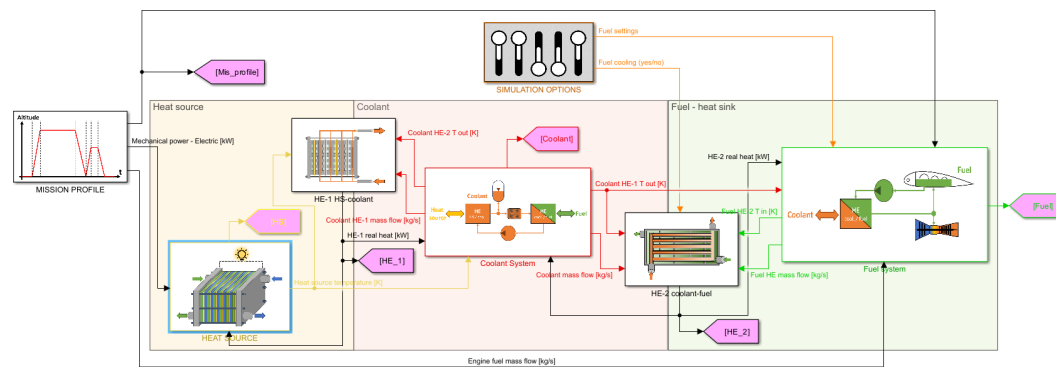


Fig. 3.9 Overview of the complete F-TMS simulation model consisting of: heat source, coolant loop, fuel system, and the intermediate HE structure components connecting these elements, together with the additional mission profile and simulation options modules.

The option block ensures high model flexibility and enables the automated configuration of simulation settings for batch execution and comparative analyses. In particular, it allows switching between adiabatic and diabatic tank assumptions, alternative convection models, and the two reference F-TMS architectures (series and parallel).

The mission planner module initializes the virtual flight scenario by importing predefined mission profiles from external .csv files, processing them, and returning the $\dot{m}_{\text{eng}}$ and $N_{\text{EM}}\dot{W}_{\text{EM,mech}}$ time-series required for the numerical simulation. The reference Pratt & Whitney PW127F engines introduced in Sect. 1.2.1 are retained, along with their certified specifications [15, 16], to construct the baseline mission profiles for a conventional turboprop aircraft benchmark.

For the present analysis, a 12-phase profile is considered, comprising: taxi-out, take-off, initial climb, mission cruise, first descent, missed approach, second climb, alternate cruise to the nearest suitable aerodrome, second descent, approach, landing, and taxi-in. A maximum operational range of 1300 km and a service ceiling of 7600 m are assumed, consistent with certified specifications for the ATR 72 regional turboprop aircraft [166]. A minimum mission range of 130 km is considered,

representative of the Dublin–Isle of Man route, one of the shortest scheduled regional services. The alternate flight segment is instead varied between 50 km to 250 km.

The climb, cruise, and descent phases collectively contribute to covering both the primary and alternate flight ranges. However, to account for manoeuvring and flight-path alignment after take-off and prior to landing, early climb and final descent occurring below 1500 m are excluded from the effective range calculation, as shown in Fig. 3.10.

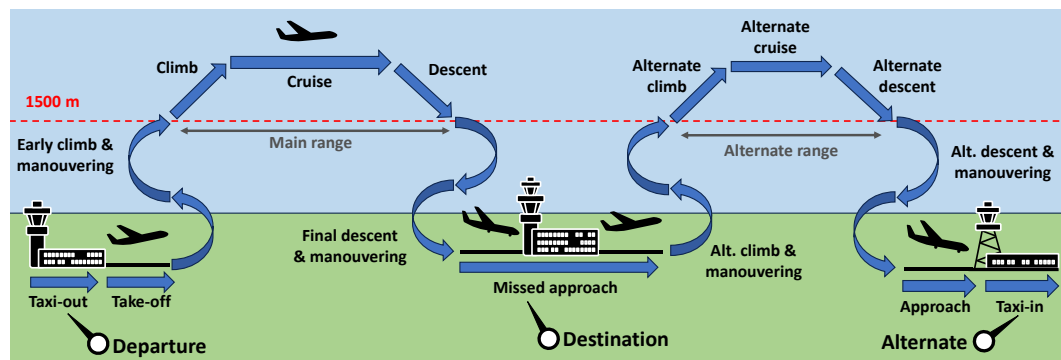


Fig. 3.10 Schematic representation of the simulated flightpath and its main phases.

Additionally, to ensure physically consistent and operationally realistic profiles, the following constraints are introduced, particularly for short-range missions:

- A **minimum cruise duration** of 10 min is enforced for both primary and alternate segments. If this requirement is not satisfied, the cruise altitude is reduced to prolong the cruise phase at the expense of climb and descent. Should the resulting cruise altitude fall below the nominal 1500 m vertical extent allocated for post-take-off and pre-landing manoeuvring, the cruise phase is further extended to account for the remainder of the manoeuvre. This extended duration is equivalent to that required to complete the missing portion of the nominal climb and descent;
- A **minimum cruise altitude** of 1000 m is imposed. In case where the previous adjustment would require a lower altitude, the 1000 m altitude is maintained, independently of the resulting cruise duration;
- A **minimum alternate range** criterion is adopted, so that the nearest suitable aerodrome is designated as the alternate destination, and may coincide with

the departure airport. For example, if the nominal alternate airport lies 150 km from the scheduled destination while the main mission range is only 130 km, the effective alternate distance is reduced to 130 km, and the departure airport becomes the alternate.

The imported datasets from these profiles include aircraft altitude, airspeed, rate of climb, engine mechanical power, and specific fuel consumption. Within the mission planner, these signals are processed to adapt the baseline configuration to the selected degree of aircraft power hybridisation H_P , according to the following hypotheses:

- Independently of the selected hybridisation level, the total mechanical power request at the propeller shafts remains the same, $N_{\text{shaft}}\dot{W}_{\text{eng}} + N_{\text{EM}}\dot{W}_{\text{EM,mech}} = N_{\text{shaft}} \cdot \dot{W}_{\text{PW127}}$, assuming $N_{\text{shaft}} = 2$ and $N_{\text{EM}} = 2$ for an hybrid twin turboprop configuration;
- The selected degree of power hybridisation and the power partition between conventional and electric propulsion remains constant throughout the entire simulation: $H_P = \frac{\dot{W}_{\text{EM,mech}}(t)}{\dot{W}_{\text{m,prop}}(t)}$;
- The conventional engines are assumed perfectly scalable, so that their specific fuel consumption rates remain unchanged: $SFC_{\text{eng}} = \frac{\dot{m}_{\text{eng}}}{\dot{W}_{\text{eng}}}$, independently of the selected degree of hybridisation: $SFC_{\text{eng}} = SFC_{\text{PW127}} = \frac{\dot{m}_{\text{PW127}}}{\dot{W}_{\text{PW127}}}$.

Accordingly, the total engine fuel consumption rate is provided through Eq. (3.41):

$$\dot{m}_{\text{eng}} = SFC_{\text{eng}} \cdot N_{\text{shaft}}\dot{W}_{\text{eng}} = \frac{\dot{m}_{\text{PW127}}}{\dot{W}_{\text{PW127}}} \cdot 2(\dot{W}_{\text{PW127}}(1 - H_P)) \quad (3.41)$$

where $N_{\text{shaft}}\dot{W}_{\text{eng}}$ is the total mechanical power delivered by the conventional powertrain, and SFC_{eng} its specific fuel consumption, while $\dot{W}_{\text{PW127}}$ and $\dot{m}_{\text{PW127}}$ denote the reference mechanical power output and fuel mass flow rates of a single PW127F engine.

The mechanical power delivered by the electric drivetrain is defined according to Eq. (3.42):

$$N_{\text{shaft}}\dot{W}_{\text{EM,mech}} = H_P \cdot 2\dot{W}_{\text{PW127}} \quad (3.42)$$

Simulation settings

A hot-day scenario with a ground temperature of 40 °C is considered for all simulations, representing typical adverse conditions for cooling system design and operation, in accordance with MIL-STD-210C [273]. The initial temperatures of the fuel tank, fuel-side HE, and coolant-side HEs are also set to 40 °C, assuming thermal equilibrium with the ambient environment. Conversely, the fuel cell stack is initialised at its minimum operating temperature of 100 °C, assuming the simulation begins once the system is activated for early ground operation.

Air properties are computed as a function of altitude, using tabulated International Standard Atmosphere (ISA) data [274] from the Simulink *ISA Atmosphere Model* block, adjusted by a 25 °C positive temperature offset to match the reference 40 °C ground scenario. Jet-A fuel behaviour is approximated using kerosene property data [275], while a 50–50 water–glycol mixture is assumed to represent the coolant.

In line with existing literature [243], two wing tanks are assumed and modelled as identical cuboids, with approximate dimensions of 745 × 132 × 31 cm, resulting in a maximum fuel capacity of approximately 5000 kg, consistent with typical regional turboprop specifications [166, 4, 164].

The maximum HE fuel mass flow rate is defined in accordance with CS-25 regulations [244], which limit local fuel velocities within channels and pipes to 6 m s⁻¹ (see Sect. 2.4). Assuming two dedicated pipelines with a 5 cm diameter delivering fuel from both wing tanks, a maximum volumetric fuel flow rate of 500 L/min is conservatively assumed per each tank. This value is consistent with typical refuelling rates for regional transport aircraft, and corresponds to an approximate maximum fuel mass flow rate of 13.5 kg s⁻¹ entering the HE. Considering that 50–50 water–glycol mixtures exhibit a specific heat capacity approximately 1.7 times higher than that of kerosene, a coolant mass flow rate of 7.8 kg s⁻¹ is sufficient to balance the fuel-side thermal capacity rate. For modelling consistency, this value is conservatively approximated to 500 L/min.

The EASA fuel planning specifications [272] also determine the initial fuel stock, $m_{\text{fuel}}(0)$, according to Eq. (3.43). For clarity, the subscript "fuel" is omitted in the equation:

$$m_0 = (1 - H_P) (m_{\text{taxi}} + m_{\text{trip}} + m_{\text{alt}} + m_{\text{cont}} + m_{\text{fr}}) \quad (3.43)$$

where $1 - H_P$ is the scaling factor accounting for the hybridisation level of the reference aircraft. The terms m_{taxi} , m_{trip} and m_{alt} respectively denote the fuel mass consumed during taxi, main flight operations, and alternate diversion to the nearest suitable aerodrome. These three contributions are collectively obtained by integrating the fuel consumption rate history along the complete mission profile. The remaining terms, m_{cont} and m_{fr} , denote the contingency fuel and final reserve fuel. According to EASA regulations, contingency fuel is equivalent to 5% of the mission fuel consumption, while the final reserve fuel corresponds to the fuel required to perform a 30 min cruise at an altitude of 450 m.

3.4 Results

The complete model is herein used to assess the feasibility of the F-TMS in supporting regional transport aircraft electrification via fuel cells. The following sections present the results, examining critical configurations and mission scenarios to identify the maximum thermally supported degree of aircraft hybridisation.

3.4.1 F-TMS mass and temperature dynamics

F-TMS performance is initially evaluated with respect to a single reference mission profile. An analysis of more than 6000 UK regional flights by Jux et al. [276] reports an average flight distance of 407 km, typically operated at altitudes around 6000 m. Based on these findings, a reference mission is defined for the present study, consisting of a 400 km primary flight conducted at an altitude of 6000 m, followed by an arbitrary 200 km alternate flight conducted at a reduced altitude of 4500 m. Fig. 3.11 illustrates the altitude profile for this reference mission and the corresponding total mechanical power demand at the propeller shafts.

The consistency of the modelled mass dynamics and their scaling with increasing levels of aircraft hybridisation is assessed first. Fig. 3.12 illustrates the total tank fuel mass and its gradual depletion over the reference mission simulation, comparing 10% and 20% hybridisation scenarios against a 0% benchmark. The results demonstrate physically consistent behaviour, with the largest differences occurring at the mission start, where cumulative variations in specific fuel consumption are most pronounced between the three configurations. These deviations gradually diminish as the mission

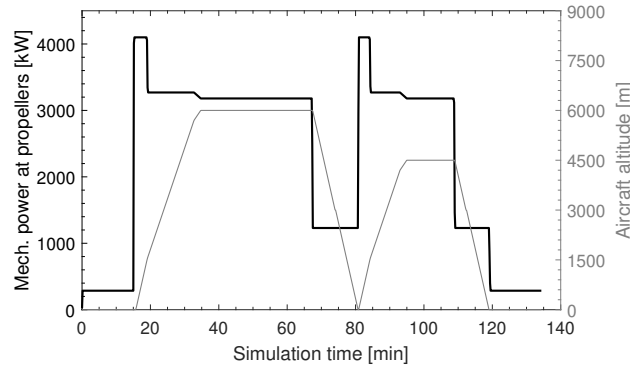


Fig. 3.11 Altitude and combined propeller shaft power trends for a 400 km, 6000 m regional flight, followed by a 100 km, 4500 m alternate.

progresses, converging towards minimal differences by the mission's end. Overall, the observed trends align with literature findings [21, 22, 25], providing additional validation of the model.

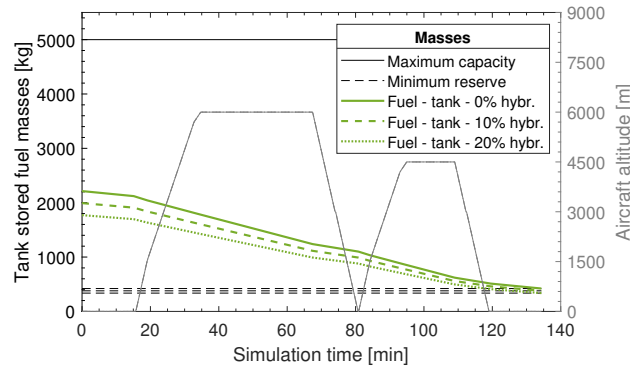


Fig. 3.12 Tank fuel mass dynamics for 0%, 10%, and 20% hybridized regional transport aircraft performing a 400 km, 6000 m main mission followed by a 200 km, 4500 m alternate flight.

The results and numerical consistency of the modelled temperature dynamics are examined next. However, since they strongly depend on the assumed boundary heat-exchange mechanisms and the selected F-TMS architecture, the comparison is restricted to a reference 5% propulsion hybridisation, while the remaining parameters are varied.

Adiabatic tank wall assumption

In line with conventional modelling approaches from the literature [246, 247, 251], adiabatic fuel tank walls are initially assumed, resulting in $\dot{Q}_{FT} = 0$. Simulations are conducted for both parallel and series F-TMS configurations, resulting in the trends illustrated in Fig. 3.13.

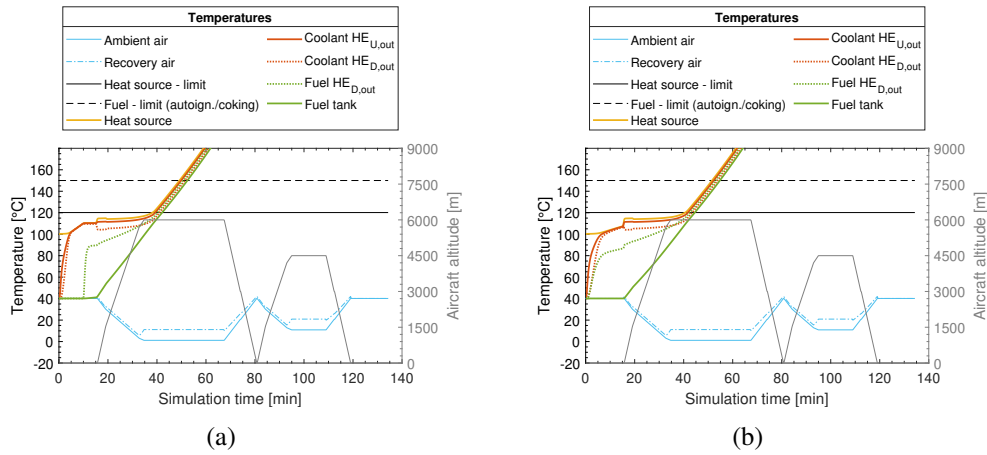


Fig. 3.13 Relevant temperature dynamics for a 5% hybrid-electric turboprop aircraft with *adiabatic* fuel tanks performing a 400 km, 6000 m main mission followed by a 200 km, 4500 m alternate flight. (3.13a) recirculating-parallel F-TMS configuration. (3.13b) recirculating-series F-TMS configuration.

Five relevant temperatures are reported: the fuel cell temperature (heat source), the HE_U and HE_D coolant outlet temperatures, the HE_D fuel outlet temperature, and the average fuel tank temperature. In the absence of numerical control strategies or safety mechanisms mitigating component overheating, the heat source temperature is allowed to exceed its maximum operating limit under the simplifying assumption that no performance degradation occurs in any component or working fluid. This modelling choice facilitates the identification of critical F-TMS operating conditions where cooling is insufficient.

Fig. 3.13 illustrates this behaviour, showing that both the parallel and series configurations exceed the maximum allowable heat source temperature after approximately 40 min of simulation, near the transition from the end-of-climb to the early-cruise phase. The two architectures show only marginally different performance, with the recirculating parallel configuration satisfying fuel cell cooling requirements for approximately 28.4% of the mission, while the recirculating series

configuration maintains acceptable thermal conditions for about 30.1% of the simulation. These results demonstrate that relying solely on the fuel's thermal inertia under the adiabatic tank assumption is insufficient to provide the fuel cell with the required cooling, even at a modest hybridisation degree of 5%.

The fuel outlet temperatures of HE_D highlight some functional differences between the parallel and series configurations. In the parallel architecture, the fuel initially remains at the tank temperature, as the cooling loop stays inactive until the coolant temperature exceeds 110°C . In contrast, the fuel temperature at the HE outlet rises from the start of the simulation in the series configuration, as fuel flows through the heat exchanger to feed the engines even at relatively low coolant temperatures. Despite these differences, both cases experience negligible changes in tank fuel temperatures before take-off, followed by a steady increase later in the mission as the heat accumulates.

Diabatic tank wall assumption with radiative and static convective heat transfer

The F-TMS performance is then examined relaxing the adiabatic wall hypothesis, and incorporating the radiative and convective terms in the fuel tank energy balance. A first batch of simulations is conducted assuming fuel as a hot fluid over a cold flat plate hypotheses. However, owing to the fluid modest thermal conductivity, results comparable to those of Fig. 3.13 are achieved, with a mere 0.6% improvement in system thermal endurance. These results confirm that layouts operating under purely conductive hypotheses effectively approach adiabatic behaviour, as shown in Fig. 3.16.

A second batch of simulations is thus performed, assuming enhanced heat transfer effects via strategic placement of the recirculation ducts (low-lying piccolo tubes), and negligible vibrations. The outcomes of this second analysis are presented in Fig. 3.14, comparing both parallel and series F-TMS configurations.

The results in Fig. 3.14 demonstrate a marked improvement in thermal endurance following the introduction of passive heat-rejection mechanisms: the parallel architecture now sustains cooling for up to 46.0% of the mission, while the series configuration extends this to approximately 62.4%. These findings confirm that heat transfer at the tank–wall interface is a relevant phenomenon that can significantly influence the feasibility of the F-TMS, consistent with the experimental observations

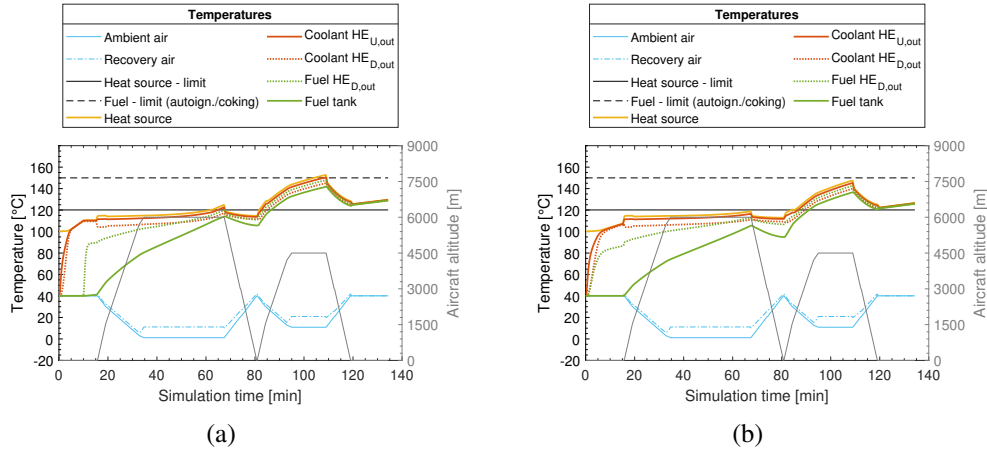


Fig. 3.14 Relevant temperature dynamics in a 5% hybrid-electric turboprop aircraft with natural convection in *adiabatic* fuel tanks performing a 400 km, 6000 m main mission followed by a 200 km, 4500 m alternate flight. (3.14a) recirculating-parallel F-TMS configuration. (3.14b) recirculating-series F-TMS configuration.

of Huang et al. [250]. The integration of passive heat transfer mechanisms also clarifies the performance gap between the parallel and series recirculation layouts. The series configuration achieves approximately 35% greater thermal endurance thanks to the fuel being supplied directly from the heat exchanger to the engines, hence being combusted at its highest available temperature. This ensures a more effective utilisation of the fuel thermal capacity compared to the parallel configuration, in which the comparatively cooler tank fuel is fed to the engines and consumed.

Despite these improvements, neither architecture ensures sufficient cooling to support the complete mission at a 5% hybridisation degree. The recirculating-series configuration, however, is now capable of covering the entire main flight segment and a small portion of the alternate climb. Given that alternate operations are infrequent and typically associated with contingency scenarios, it may be acceptable to suspend electric propulsion during these phases, when emission reduction becomes secondary to operational requirements. In this context, electric propulsion could be limited to providing a power boost during the initial phase of the alternate climb, when demand is highest, with the remainder of the mission relying solely on conventional propulsion. Under these assumptions, a 5% hybridisation level appears feasible with a dedicated series F-TMS configuration.

Ultimately, comparing the first and second climb temperature trends from Fig. 3.14 highlights the impact of progressive fuel depletion on the TMS. As the

mission advances, the reduction in tank mass decreases the characteristic thermal response time, $\tau = m_{\text{FT}}(t)/\dot{m}_{\text{rec}}(t)$, resulting in an accelerated tank temperature rise under comparable thermal loads. This reveals a non-negligible degradation in F-TMS thermal buffering at the later stages of the mission, identifying the alternate cruise and climb as the most critical segments.

The robustness of the heat-rejection model is assessed in Fig. 3.15, which compares the simulated convective heat transfer coefficients of air and fuel, as a function of their occurrence frequency over the reference mission profile for the series configuration.

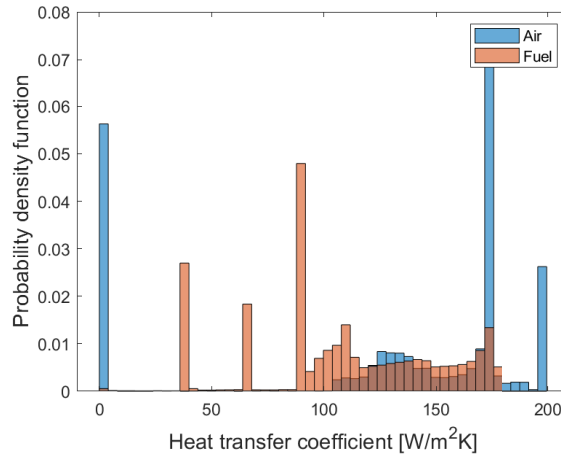


Fig. 3.15 Occurrence distribution of air and fuel convective heat-transfer coefficients at the fuel-tank interface over the reference mission profile (recirculating-series architecture for reference).

Air-side coefficients display three dominant values: $0 \text{ W K}^{-1} \text{ m}^{-2}$, $174 \text{ W K}^{-1} \text{ m}^{-2}$, and $200 \text{ W K}^{-1} \text{ m}^{-2}$ associated with the primary mission altitudes and their corresponding phases: ground operation and taxi, main cruise, and alternate cruise, respectively. Conversely, fuel-side coefficients exhibit a more uniform distribution, ranging from $38 \text{ W K}^{-1} \text{ m}^{-2}$ to $178 \text{ W K}^{-1} \text{ m}^{-2}$, consistent with typical values for natural convection in liquids reported by Mudawar [277] and Maragkos and Beji [278].

Excluding ground operations, air-side Nusselt numbers range from 5.5×10^3 to 10.5×10^3 . The maximum occurs for $Re \approx 7.7 \times 10^7$ during the alternate cruise phase, at an altitude of 4500 m and an aircraft speed of 140 m s^{-1} , in excellent agreement with the results reported by Habermann et al. [238] under identical conditions. The

corresponding maximum heat transfer coefficient of approximately $200 \text{ W K}^{-1} \text{ m}^{-2}$ remains consistent with typical ranges for gaseous forced convection reported by Mudawar [277] and Maragkos and Beji [278]. These results validate the modelling of the F-TMS passive heat exchange at the tank–environment interface, corroborating the presented methodology.

Diabatic tank wall assumption with radiative and vibration-induced convective heat transfer

The analysis is completed by incorporating the preliminary estimate of vibration-induced convection within tanks, based on the mutually exclusive formulation expressed in Eq. (3.38). An initial assessment indicates that the temperature trends predicted by the vibration-enhanced model closely match those obtained under buoyancy-driven convection alone, for both parallel and series configurations, effectively reproducing the behaviour observed in Fig. 3.14. Such results, evidenced in Fig. 3.16, indicate that, at the present level of abstraction, vibration-induced effects remain secondary to buoyancy-driven convection for the assumed 15 Hz, 10 mm harmonic excitation. This conclusion is further supported by the corresponding Nusselt numbers, with the Gershuni correlation peaking at approximately $Nu(Gs) = 49$, while Rayleigh-driven convection reaches $Nu(Ra) = 153$, thus dominating the combined correlation of Eq. (3.38).

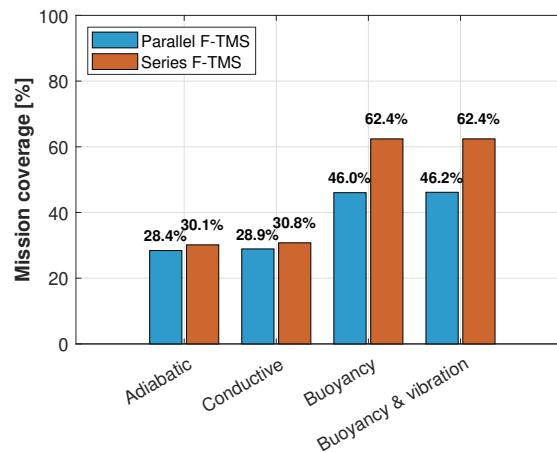


Fig. 3.16 Thermal endurance of diverse F-TMS configurations for the cooling of a MT-PEM fuel cell supporting a 5% degree of aircraft hybridisation.

It should be emphasized, however, that these findings rely solely on a first-order, mutually exclusive representation of buoyancy- and vibration-driven mechanisms, without accounting for their potential interaction. As shown by Crewdson and Lappa [279], vibration-induced convection is highly sensitive to flow configuration, thermal regime, Rayleigh number, and excitation frequency, even in simplified geometries. Properly addressing their coupled effects would require more advanced formulations and detailed system representations, which lie beyond the scope of this preliminary investigation. A more comprehensive assessment should therefore be pursued in future refinements of the tank design, potentially incorporating multiple excitation modes and free-surface sloshing effects.

Diabatic tank wall assumption with radiative and forced convective heat transfer

Finally, a preliminary assessment of an active mixing configuration inside the tank is presented for future consideration. A generic, unspecified mixing architecture is assumed, providing a constant heat transfer coefficient of $1000 \text{ W K}^{-1} \text{ m}^{-2}$, independent of instantaneous tank conditions. This value is consistent with low-to-moderate forced convection in liquids, according to Mudawar [277] and Maragkos and Beji [278]. The results of this speculative investigation, comparing the series and parallel configurations, are shown in Fig. 3.17.

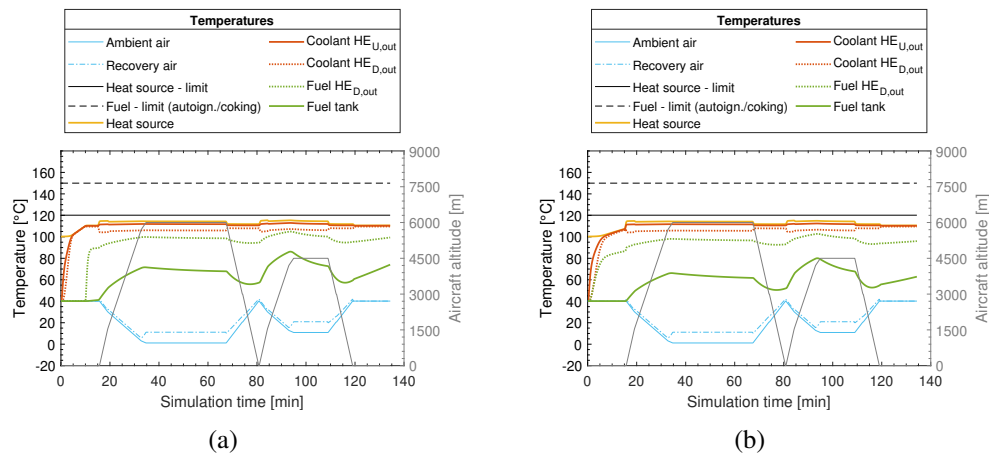


Fig. 3.17 Relevant temperature dynamics in a 5% hybrid-electric turboprop aircraft with forced convection in *diabatic* fuel tanks performing a 400 km, 6000 m main mission followed by a 200 km, 4500 m alternate flight. (3.17a) recirculating-parallel F-TMS configuration. (3.17b) recirculating-series F-TMS configuration.

The results indicate that, by increasing the convective heat transfer coefficient to $1000 \text{ W K}^{-1} \text{ m}^{-2}$, both F-TMS configurations are able to support 5% degree of aircraft hybridisation, while still retaining a substantial portion of the usable thermal inertia. Preliminary assessments further suggest that hybridisation levels up to 6.6% could indeed be sustained by a F-TMS with a series-recirculation heat exchanger.

These findings underscore the benefits of enhanced passive heat rejection at the system level. Notably, the analysis demonstrates that substantial cooling can restore the fuel's thermal buffering capacity, which can then be leveraged during high-intensity flight phases, thereby reducing the criticality of the alternate climb.

3.4.2 Influence of flight mission profile parameters

Following the previous assessment of F-TMS performance, architectural configurations, and heat-transfer assumptions, the system response is found to be governed primarily by the available fuel mass, the overall mission duration, and the external air temperature and properties. These parameters are inherently mission-dependent, making them also indirectly correlated. This interdependence complicates the identification of the most critical F-TMS operating scenario and, consequently, the determination of the maximum degree of aircraft hybridisation thermally supported by the system.

A comprehensive parametric analysis is therefore conducted to assess the influence of mission planning on F-TMS performance. The investigation focuses on three primary variables: main cruise altitude, main cruise range, and the alternate flight range. The reference 40°C hot-day scenario and a 5% propulsion hybridisation level are retained, while the design of experiments is defined around the baseline mission profile of Fig. 3.11, varying one of the three parameters at a time.

The recirculating-series configuration, enhanced through radiative and buoyancy-driven heat transfer at the tank-interface, is adopted for the remainder of this work, as it demonstrated the highest performance among the validated models. Although the forced-convection assumptions yielded promising results, these are not retained at this stage, as their implementation would require dedicated validation of the mixing architecture and its integration within the tank. Accordingly, the present analysis remains focused on passive heat-rejection mechanisms.

Influence of altitude (F-TMS)

The influence of altitude is analysed first, comparing the 6000 m average regional transport altitude reported by Jux et al. [276] with:

- 7600 m, representing the certified service ceiling of ATR-72 aircraft [166];
- 4500 m, corresponding to the test-case altitude reported in Ref. [238].

The alternate flight altitude of 4500 m is retained, along with the other reference mission flight parameters. Fig. 3.18 presents the resulting mass and temperature dynamics. For clarity, only the heat source and fuel-tank temperatures are reported, facilitating the identification of the most critical operating condition.

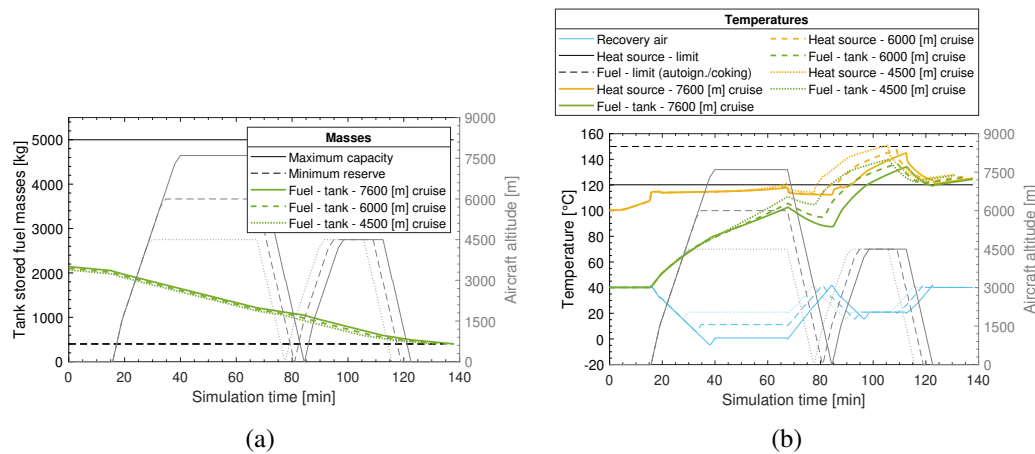


Fig. 3.18 Significant mass and temperature trends for a series F-TMS performing a reference 400 km mission at diverse altitudes, followed by a 200 km, 4500 m alternate flight (5% aircraft electrification). (3.18a) tank-fuel mass evolution. (3.18b) heat source and tank-fuel temperature evolution.

A modest increase in overall fuel consumption is observed in Fig. 3.18a, when higher cruising altitudes are considered. This trend is expected, as additional energy is required to sustain the extended climb segment. Nevertheless, since this extended climb and the subsequent descent above 1500 m contribute to the overall mission range, the additional fuel penalty remains limited. Specifically, approximately 73 kg of extra fuel are required for the 7600 m profile and 35 kg for the 6000 m mission relative to the 4500 m case, corresponding to net increases of 3.5% and 1.7%, respectively.

These penalties are compensated by improved thermal performance at higher altitudes, as illustrated in Fig. 3.18b. This behaviour is physically consistent and reflects the enhanced cooling associated with larger temperature differences, ΔT , between the fuel tanks and the environment at altitude, resulting from lower ambient temperatures. Although the reduction in air density at higher altitudes tends to weaken the convective heat rejection, the results indicate that the benefit of lower ambient temperatures remains dominant under the investigated hot-day conditions. Accordingly, the divergence among the three thermal histories becomes evident primarily during the cruise phase, with the lower-altitude case exhibiting the steepest temperature rise despite the comparable fuel masses. In practical terms, the 4500 m profile reaches the highest recorded heat-source temperature at the end of the alternate cruise, approximately 5.5 °C higher than that of the 7600 m case.

Moreover, the simulations highlight the importance of the descent phase in restoring the fuel's thermal capacity prior to the alternate climb. The reduced power demand during descent lowers waste-heat generation, establishing a negative thermal balance between the fuel tank and the external environment that results in a net cooling of the tank fuel. Operating at higher cruise altitudes inherently extends the descent phase, thus prolonging the favourable cooling period and enabling greater recovery of thermal capacity, as shown in Fig. 3.18b. These findings further reinforce the thermal advantage of high-altitude operations in F-TMS configurations relying on passive heat rejection.

Influence of main mission range (F-TMS)

The influence of the main mission range is assessed next, comparing the two following destinations against the 400 km reference flight reported by Jux et al. [276]:

- 1300 km, equivalent to the maximum certified range of an ATR-72 under maximum payload conditions [166];
- 130 km, corresponding to the Dublin-Isle of Man flight distance, representing one of the shortest scheduled commercial regional flights.

Fig. 3.18 compares the resulting mass and temperature evolutions for the heat-source and fuel tank.

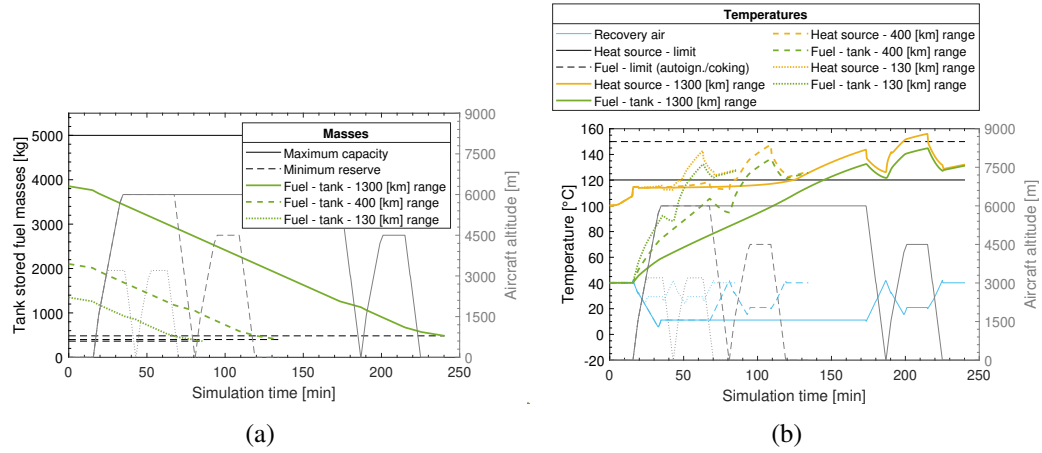


Fig. 3.19 Significant mass and temperature trends for a series F-TMS performing three different missions at a main cruise altitude of 6000 m, followed by a 200 km, 4500 m alternate flight (5% aircraft electrification). (3.19a) tank-fuel mass evolution. (3.19b) heat source and tank-fuel temperature evolution.

Comparing the altitude profiles in Fig. 3.19, the first mission-planning assumption of Sect. 3.3.4 becomes evident in the 130 km range simulation. Although the nominal cruise altitude target is 6000 m, the assumption constrains the attainable flight level to approximately 3150 m, preventing climbs that would be disproportionate to the short mission duration. The resulting lower-altitude profile aligns well with operational data for the Dublin–Isle-of-Man route. Furthermore, the third mission-planning hypothesis is also activated, with the reference alternate distance of 200 km being limited to the main mission length of 130 km, effectively designating the departure airport as the alternate. Accordingly, the alternate cruise altitude target of 4500 m is also reduced to approximately 3150 m.

In contrast to the altitude comparison, substantially different fuel-mass dynamics emerge from Fig. 3.19a, with the 1300 km mission requiring about three times the fuel mass of the 130 km flight. Although this reflects the significant difference in mission range, the scaling remains moderate relative to the tenfold distance increase. This discrepancy arises from the modelling assumption that early climbs and late descents below 1500 m do not contribute to effective range coverage. These segments, together with taxi, take-off, and missed-approach phases, constitute a baseline fuel request that is largely independent of the main mission distance. Consequently, short-range flights allocate a comparatively larger fraction of their fuel stock to low-altitude and manoeuvring phases. Moreover, fixed fuel requirements, such as

final reserve, contingency fuel, and shared alternate mission segments, further reduce the proportional difference in total fuel mass between short- and long-range missions.

The temperature evolutions shown in Fig. 3.19b exhibit progressively higher maxima as the mission range increases. The 1300 km simulation reaches a peak heat-source temperature of 156 °C, approximately 9 °C and 14 °C higher than those of the 400 km and 130 km missions, respectively. These findings highlight the critical impact of sustained waste-heat accumulation within a system of finite thermal capacity, identifying long-range flights as the most thermally demanding scenario. Conversely, although short-range missions exhibit a faster temperature rise due to reduced thermal inertia associated with smaller initial fuel masses, their shorter duration limits cumulative heat accumulation, resulting in lower maximum temperatures. This behaviour is consistent with the inverse dependence of temperature dynamics on fuel mass, $T_{FT}(t) \sim 1/m_{FT}(t)$, as described in Eq. (3.28).

However, the magnitude of waste-heat generation is shown to strongly influence the balance between these two competing effects: cumulative heat absorption during prolonged missions and reduced thermal capacity in short-range operations. This sensitivity is illustrated in Fig. 3.20, which reports the temperature trends for three analogous mission profiles at a reduced hybridisation degree of 3%.

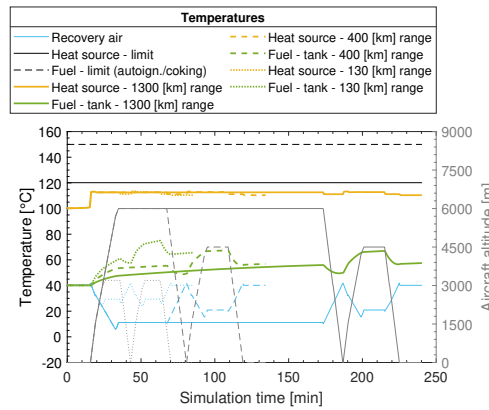


Fig. 3.20 Significant temperature trends for a series F-TMS performing three different missions at a main cruise altitude of 6000 m, followed by a 200 km, 4500 m alternate flight (3% aircraft electrification).

Under these conditions, short mission profiles become comparatively more critical with respect to tank-fuel temperatures. The reduction in heat generation weakens the dominance of cumulative heating effects, increasing the relative importance

of usable thermal inertia. Additionally, altitude saturation in particularly short flights limits the previously identified thermal benefits associated with higher cruise altitudes, further amplifying challenges for short-range missions.

The combined evidence from Figs. 3.19 and 3.20 suggests that the turning point between these competing mechanisms likely occurs near the region of most effective F-TMS utilisation. Accordingly, both minimum- and maximum-range mission configurations should be assessed when determining the maximum admissible degree of aircraft hybridisation.

Influence of alternate range (F-TMS)

Finally, the alternate flight impact is investigated, comparing the reference 200 km alternate flight against a maximum range of 350 km and a minimum of 50 km. The target alternate flight altitude of 4500 m is retained in all cases, along with the remaining parameters from the reference mission profile of Fig. 3.11.

Fig. 3.21 provides the resulting mass and temperature evolutions for the heat-source and fuel tank.

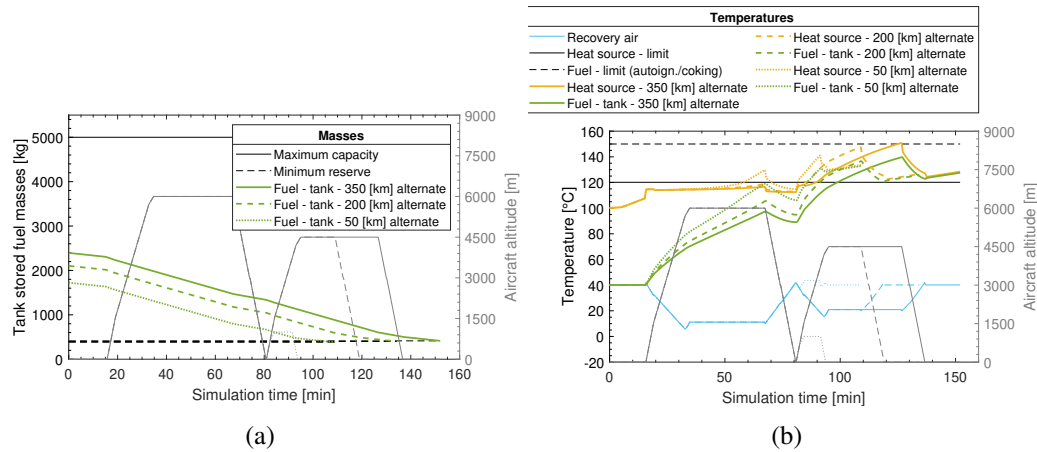


Fig. 3.21 Significant mass and temperature trends for a series F-TMS performing a 400 km, 6000 m, main cruise followed by different alternate flights. (3.21a) tank-fuel mass evolution. (3.21b) heat source and tank-fuel temperature evolution.

Similar to the previous case, the altitude-saturation hypotheses of Sect. 3.3.4 are evident for the shorter flight considered in this analysis. In particular, the first mission-planning assumption initially reduces the nominal 4500 m altitude of the

50 km alternate to prevent disproportionate climbs, with the second assumption later enforcing the minimum altitude constraint of 1000 m.

Consistent with the main mission range analysis, the fuel-mass dynamics in Fig. 3.21a show higher initial fuel requirements for longer alternate ranges. The difference in initial fuel mass among the three cases (approximately 300 kg) scales roughly with their respective distances, with the shortest alternate exhibiting an even lower initial mass due to its limited climb to 1000 m.

Temperature dynamics from Fig. 3.21b indicate higher peak temperatures for longer alternates. The 350 km alternate reaches a maximum heat-source temperature of 150 °C, approximately 3 °C and 9 °C above the 200 km and 50 km profiles, respectively. These results confirm that longer ranges are more challenging to the F-TMS, even for alternates, although previous considerations from the main mission analysis regarding the balance between heat accumulation and usable thermal capacity remain valid.

3.4.3 Maximum degree of thermally supported hybridisation

Results from the previous analyses identify the principal mission parameters influencing F-TMS performance. The most challenging operating conditions are associated with low cruise altitudes, prolonged main and alternate flight segments, and, in certain cases, minimum range missions. These findings motivate the definition of representative worst-case mission scenarios to determine the maximum degree of aircraft electrification reliably supported by the proposed solution.

Given the diminishing performance of F-TMSs at lower altitudes, a reference cruise altitude must be defined as the minimum altitude ensuring safe system operation. For the present study, the average regional transport cruise altitude of 6000 m, reported by Jux et al. [276], is adopted as a realistic reference value.

Based on these considerations, three critical mission scenarios are defined:

- **Primary Worst-Case Scenario.** This scenario combines the maximum expected main mission range with the longest alternate segment. In the present study, these correspond to 1300 km and 250 km, respectively. The main cruise altitude is set at 6000 m, while the alternate segment is assumed to be flown at 4500 m.

- **Secondary Worst-Case Scenario I.** This case combines the minimum expected operational range with an equivalent alternate segment, assuming that the departure and alternate airports coincide. A 130 km mission profile is considered. In accordance with the assumptions of Sect. 3.3.4, a saturated cruise altitude of 3150 m is adopted for both segments.
- **Secondary Worst-Case Scenario II.** This configuration considers the minimum operational main range, followed by the shortest alternate segment. A 130 km main mission, combined with a 50 km alternate, is assumed. Based on the assumptions from Sect. 3.3.4, saturated cruise altitudes of 3150 m and 1000 m are adopted for the main and alternate segments, respectively.

The maximum supported degree of aircraft electrification is determined by iteratively simulating the primary worst-case scenario and progressively refining the hybridisation level until the maximum recorded heat-source temperature approaches, without exceeding, its operating limit of 120 °C. The resulting value is then validated against the two secondary worst-case scenarios. If thermal constraints are violated in either case, the hybridisation degree is reduced accordingly to ensure compliance across all critical missions.

A preliminary assessment indicates that the maximum feasible degree of hybridisation for a recirculating-series F-TMS architecture cooling a MT-PEM fuel cell is approximately 4.0%. To corroborate this result, an extensive simulation campaign is conducted, evaluating all combinations of the following parameters:

- Three representative main cruise altitudes: 4500 m, 6000 m, and 7600 m;
- Twelve uniformly spaced main mission ranges spanning from 200 km to 1300 km, in 100 km increments, together with the minimum mission range of 130 km;
- Seven discretised alternate ranges, varying from 50 km to 350 km in 50 km increments.

The results of this campaign are presented in Figs. 3.22 and 3.23, illustrating the response surfaces of the maximum simulated heat source and tank-fuel temperatures, respectively.

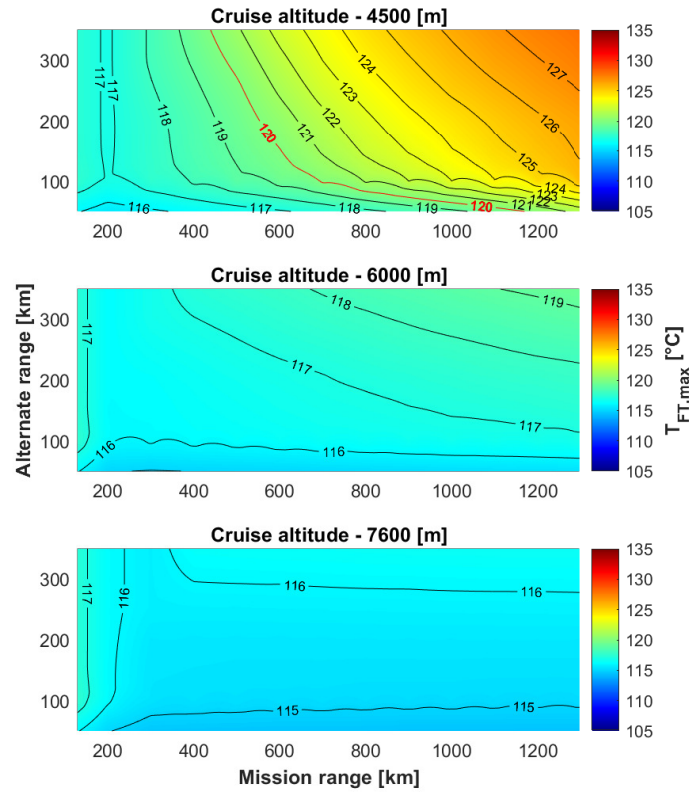


Fig. 3.22 Maximum heat source temperature for a series F-TMS as a function of main mission range, alternate range, and target cruise altitude (4% aircraft electrification). Fuel-cell operating limit of 120 °C highlighted in red. The three subfigures correspond to different cruise altitudes.

Data from Fig. 3.22 confirm that the F-TMS consistently maintains the fuel cell below its maximum temperature limit of 120 °C when the mission cruise altitude is ≥ 6000 m, validating the optimisation results. Conversely, adequate cooling cannot be guaranteed at 4500 m for mission ranges exceeding 500 km. Nonetheless, this limitation may be regarded as acceptable, since missions of comparable extent would, under realistic operational conditions, be conducted at higher cruise altitudes anyway. Hence, the results demonstrate that a F-TMS with series heat-exchanger recirculation can effectively sustain peak thermal loads of approximately 185 kW during climb, with an average heat load of about 145 kW. These findings are consistent with those of Kellermann et al. [243], who reported successful cooling of a 126 kW heat source using F-TMS. The present analysis extends their conclusions to higher

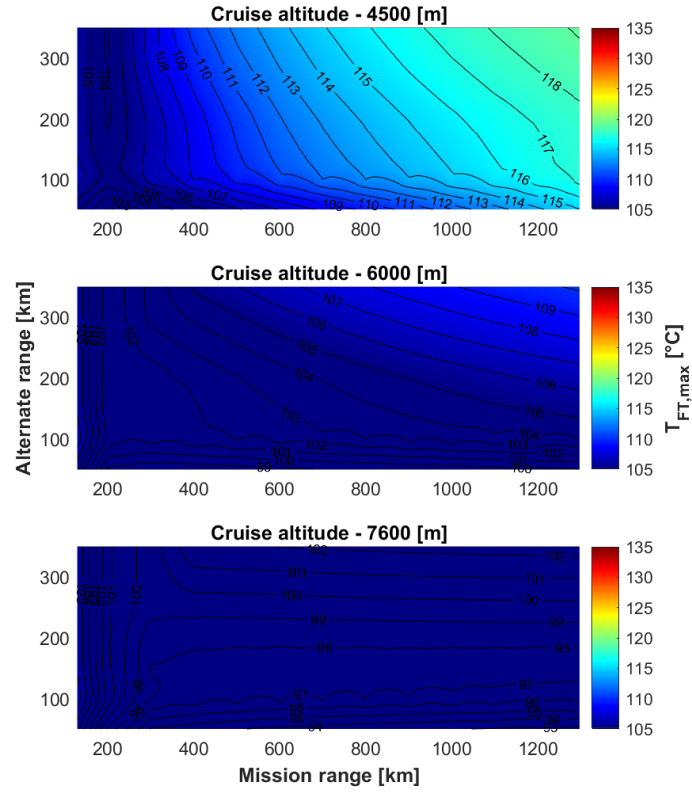


Fig. 3.23 Maximum tank-fuel temperature for a series F-TMS as a function of main mission range, alternate range, and target cruise altitude (4% aircraft electrification). The three subfigures correspond to different cruise altitudes.

overall thermal loads by explicitly accounting for the system's thermal inertia and temperature transients.

Both Fig. 3.22 and Fig. 3.23 exhibit consistent trends with earlier observations, with the highest temperatures generally occurring in the combined long-range, long-alternate scenarios. An exception arises for the 7600 m ceiling-altitude case, where the minimum main range followed by the maximum alternate segment constitutes the most critical condition, yielding the highest fuel cell and tank-fuel temperatures. This behaviour underscores the importance of validating the optimised configuration at both extremes of the mission envelope, particularly with respect to the assumptions of the Secondary Worst-Case Scenario I.

These short-range configurations also emerge as challenging scenarios as a consequence of the first mission-modelling hypothesis from Sect. 3.3.4, which truncates the attainable cruise altitude, preventing the system from benefiting from the enhanced thermal conditions associated with their nominal operating altitudes. This saturation manifests in the left-hand region of all response surfaces, where the three subfigures exhibit nearly identical behaviour due to the shared effective cruise altitude imposed by the truncation constraint. Similarly, the third mission-modelling assumption described in Sect. 3.3.4, which limits the alternate flight range to the main mission range, is also reflected in the left portion of the diagrams. In particular, for the 130 km main mission, simulation results corresponding to alternate ranges between 130 km to 350 km coincide, as the alternate segment is effectively capped to the same distance by the re-routing constraint.

Repeating the optimisation under the assumption of forced convection with a fuel-side heat transfer coefficient of $1000 \text{ W K}^{-1} \text{ m}^{-2}$ confirms the maximum hybridisation degree of 6.6%, corresponding to a peak waste-heat management capability of approximately 310 kW. In this configuration, Secondary Worst-Case Scenario I becomes the governing constraint in determining the maximum admissible electrification level, indicating that, as heat dissipation improves, the availability of usable thermal inertia becomes the discriminating factor, rendering shorter missions comparatively more critical.

Finally, assuming ideal heat transfer conditions ($h_{\text{fuel}} \rightarrow \infty$), a theoretical upper bound for the F-TMS peak heat rejection can be estimated. A conservative limit of 333 kW is obtained when the tank-floor temperature is assumed to be equal to the bulk tank-fuel temperature, corresponding to a maximum hybridisation level of 7.1%. This upper bound increases to 422 kW (9.0% hybridisation) if the tank floor is instead assumed at the recirculated fuel temperature of the low-lying piccolo tubes. This configuration is of the same order of magnitude as approximately half of the 1000 kW heat-rejection potential estimated by Roland and Rumpfkeil [241] for a comparable use case, in which the wing's top and bottom surfaces were heated to 400 K. The lower heat-rejection potential obtained in the present study is therefore consistent with expectations, as only the tank floor (i.e., the bottom wing surface) is considered, owing to the insulation provided by the ullage on the upper surface.

An intermediate estimate is thus adopted as a more plausible reference case, assuming that the tank-floor temperature equals the average between the recirculated

and bulk fuel temperatures, resulting in a peak waste-heat management of 371 kW and a maximum electrification level of 7.9%.

The resulting 4.0% optimised degree of supported propulsion electrification, together with the theoretical upper limit of 7.9%, indicates that the presently assessed F-TMS configuration is insufficient to achieve the 20% hybridisation target of this dissertation, even under ideal convection conditions inside the tanks. This underscores the inherent challenges of managing fuel cell waste-heat generation, with the proposed system remaining better suited to less demanding components, such as batteries [243], electronics, and hydraulics.

Several research directions merit further investigation to potentially enhance the feasibility of F-TMS for the cooling of diverse heat sources, including fuel cells. Advanced F-TMS architectures should be explored, including the dual-tank configurations proposed by Doman et al. [246, 247, 250], and the multi-tank layouts of Suat et al. [252], with their associated flow control strategies. Enhancements to passive heat transfer at the tank–environment interface should also be considered to approach the estimated theoretical limit of 371 kW, as transitioning from natural to moderate forced convection has been shown to nearly double the maximum sustainable peak heat load. Future analyses should incorporate high-fidelity CFD modelling of the tanks and low-lying piccolo tubes, evaluating the combined effects of buoyancy-driven, vibration-induced, and forced convection, and exploring the use of active tank-fuel mixers and forced convection promoters. To better approach the 1 MW heat-rejection potential estimated by Roland and Rumpfkeil [241], passive dissipation at the top wing surface should be promoted, for instance, via upper-wing recirculating architectures as suggested by Kellermann et al. [243]. Additionally, thermal conduction through wing longerons and ribs, along with sloshing-induced fuel mixing, merit investigation as potential enhancers of the fuel-side convective heat transfer coefficient, while integration with air–skin cooling technologies presents further opportunities for optimising air-side heat transfer beyond current theoretical thresholds.

Alternatively, F-TMS could be investigated as an auxiliary cooling system complementing conventional heat dissipation devices, such as ram-air TMS, by absorbing peak heat loads during high-intensity phases. Discontinuing F-TMS operation during non-critical phases, such as descent, could then be considered to promote fuel thermal capacity recovery, enhancing system effectiveness ahead of more critical

flight segments. Finally, the integration of emerging technologies, such as nanofluids, phase-change cooling, and heat pipes, introduces substantial opportunities to enhance both the performance and feasibility of F-TMS solutions.

The benefits of combining fuel-based thermal management with conventional ram-air cooling can be preliminarily assessed in terms of added system mass per unit of dissipated waste heat. The primary element of either system is the coolant heat exchanger, whose mass can be initially approximated using Eq. (3.44):

$$m_{\text{HE}} = \frac{\dot{Q}_{\text{HE,max}}}{(\dot{Q}/m)_{\text{HE}}} \quad (3.44)$$

where $\dot{Q}_{\text{HE,max}}$ is the nominal heat transfer rate of the HE, and $(\dot{Q}/m)_{\text{HE}}$ its gravimetric heat transfer density. In aerospace practice, typical values range from 0.2 kW kg^{-1} to 2 kW kg^{-1} for air–liquid heat exchangers [280, 281] and 1 kW kg^{-1} to 10 kW kg^{-1} for liquid–liquid devices [282]. Assuming conservative but representative values, rejecting a peak waste heat of 185 kW would require either a 185 kg air–coolant heat exchanger ($\sim 1 \text{ kW kg}^{-1}$) or a 37 kg fuel–coolant unit ($\sim 5 \text{ kW kg}^{-1}$).

Additional mass terms in the fuel-based design arise from the fuel pumps and piping. The pump mass can be estimated by substituting $\dot{Q}$ in Eq. (3.44) with the required pump power, approximated according to Eq. (3.45):

$$\dot{W}_{\text{pump}} = \frac{\dot{m} \Delta p}{\rho \eta_{\text{pump}}} \quad (3.45)$$

Assuming two high-flow boost pumps, each delivering 500 L min^{-1} of fuel, at a maximum pressure rise of 5 bar with a 0.65 efficiency, a total pumping power of 12.8 kW is obtained. Gravimetric power densities of approximately 4 kW kg^{-1} have been reported for next-generation electric aircraft pumps [283, 284]. Adopting a more conservative estimate of 1.5 kW kg^{-1} , a total installed pump mass of 8.5 kg is derived. Accounting for redundancies and backup units, this value realistically increases to approximately 17 kg , further augmented by 10 kg of piping connecting the heat exchanger, pumps, and tanks.

Summing all contributions, the total added mass of the F-TMS is approximately 64 kg, roughly one-third of the 185 kg required for an equivalent air-based solution. Fuel-based architectures also avoid parasitic aerodynamic drag, although this benefit is partially offset by the additional 12.8 kW of required pumping power. The mass of the fuel inerting system may also be considered; however, as such systems are now standard across most civil passenger aircraft [285–287], their contribution is largely independent of the selected cooling strategy.

3.5 Summary and highlights

Fuel-based thermal management practices emerge as a promising solution to the growing challenges associated with component cooling in more-electric aircraft. In this chapter, the modelling and application of fuel-based TMS are investigated as a novel approach to fuel-cell cooling within the framework of hybrid-electric regional transport aircraft propulsion.

The study builds upon existing research on the dynamic modelling of fuel temperatures in heated tanks, extending these approaches through the integration of preliminary tank heat-rejection models derived from the aircraft skin-cooling literature. Two representative single-tank recirculating architectures, the recirculating-parallel and recirculating-series configurations, are analysed and compared, with the series layout demonstrating 5–35% greater thermal endurance, depending on the assumed tank-wall heat transfer hypotheses.

The analysis also advances the existing literature on wing-tank skin cooling by explicitly incorporating fuel-side convection, system thermal inertia, and the resulting transient tank temperature evolution. Radiative and convective heat transfer mechanisms at the tank–environment interface are formulated and validated against established studies [241, 238, 243]. On the fuel side, multiple modelling assumptions are systematically assessed, including adiabatic tank walls, pure conduction, buoyancy-driven convection, vibration-induced convection, simplified forced convection, and idealised heat transfer representing the theoretical upper performance limit of the design.

Based on UK regional flight statistics [276], a representative 400 km mission with a cruise altitude of 6000 m serves as the reference case, followed by a 200 km

alternate at 4500 m. Simulation results demonstrate the benefits of integrating system-level F-TMS dynamics with tank-interface heat transfer calculations. For a 5% hybridisation level, total thermal endurance increases by 62–116% relative to adiabatic cases when buoyancy-driven passive heat rejection is included. Furthermore, accounting for the fuel's thermal inertia improves peak heat load management by up to 50% compared to earlier skin-cooling concepts [243].

Comparisons across fuel-side modelling hypotheses show that purely conductive assumptions offer only marginal benefits over adiabatic tanks (0.6%). In contrast, the strategic placement of recirculation pipes (piccolo tube layouts) substantially enhances performance, producing the previously mentioned 62–116% endurance improvements. Predicted convection levels are consistent with established correlations [277, 278] and fall within expected ranges; nevertheless, higher-fidelity simulations and experimental validation are required for the piccolo tube arrangement. Early investigations indicate that vibration-induced convection has a secondary effect compared to buoyancy at representative wing excitation frequencies and amplitudes. Finally, assuming a heat transfer coefficient of $\sim 1000 \text{ W K}^{-1} \text{ m}^{-2}$, representative of low-to-moderate forced convection in liquids, suggests up to an additional 67% increase in the maximum sustainable peak loads, motivating further research into active mixing and forced convection enhancement within fuel tanks.

The study is subsequently extended to a broader range of representative mission profiles, covering operational distances of 130 km to 1300 km, varying cruise altitudes, and diverse alternates. Higher altitude operation is shown to improve F-TMS performance under diabatic tank assumptions, whereas maximum-range missions generally pose the greatest challenge due to prolonged system usage. In some instances, however, short primary missions followed by extended alternates also emerge as critical, highlighting the risks associated with modest initial thermal capacity, particularly when tank heat rejection is more prominent. These conditions define the worst-case utilisation scenarios, which are then used to determine the maximum supported degree of propulsion electrification.

An optimised maximum electrification limit of 4.0% is obtained under worst-case constraints and verified across the full mission envelope using response surfaces of the maximum simulated system temperatures. The configuration sustains average heat loads of approximately 145 kW, with peaks up to 185 kW, but cannot achieve the 20% hybridisation target set in this dissertation. Even under idealised fuel-side

heat transfer assumptions, a theoretical upper bound of 7.9% remains insufficient. Accordingly, the present F-TMS configuration is better suited to cooling more moderate heat sources, such as hydraulic systems, power electronics, and potentially batteries [243].

Nonetheless, several research directions remain promising for enhancing the feasibility of F-TMS, even for fuel-cell cooling. These include advanced multi-tank architectures; improved modelling of fuel-side convection, sloshing, and vibration-induced mixing; deliberate promotion of forced convection within tanks; and further system integration with upper-wing skin-cooling concepts. Enhancing both passive and forced heat dissipation at the tank–environment interface could in fact bring the system closer to its estimated theoretical limit of 371 kW. Moreover, coupling advanced architectures with integrated skin-cooling strategies exploiting the upper wing surface could potentially bring heat rejection closer to the ~ 1000 kW order of magnitude reported by Roland and Rumpfkeil [241].

Alternatively, integrating F-TMS with complementary cooling technologies, both conventional (ram-air systems) and emerging (e.g., nanofluids, phase-change materials, heat pipes), represents a particularly promising pathway. In such hybrid architectures, fuel-based cooling could function as an auxiliary heat sink, absorbing partial or transient thermal loads so that the design constraints on the primary cooling system can be relaxed. This integrated approach likely constitutes the most viable strategy for enabling effective F-TMS application to fuel-cell cooling in next-generation more-electric aircraft.

Operational strategy optimisation also offers additional opportunities for F-TMS enhancement. The present analysis assumes a fixed hybridisation degree (4.0%) sustained throughout the entirety of the mission; however, adaptive or discontinuous operation of both the F-TMS and the electric propulsion may improve overall system feasibility. Temporarily reducing or suspending F-TMS usage during low-demand phases would promote thermal capacity recovery, preserving cooling potential for more critical segments. Similarly, prioritising electric propulsion during the main mission while limiting its use during alternates, except for power-boosting during peak-demand events like the second climb, could mitigate thermal constraints without materially compromising environmental benefits. Given that alternate segments are relatively infrequent and often associated with contingency conditions, a temporary

reduction in electrification during these phases may represent a pragmatic trade-off between emission performance and operational resilience.

In the next chapter, some of these research directions are further investigated, focusing on the integrated development of fuel-based and traditional thermal management systems combined with preliminary coordinated operation strategies involving discontinuous F-TMS utilisation during non-critical phases.

Chapter 4

Integrated Air–Fuel TMS

Chapter 3 explored the potential and limitations of fuel-based cooling for aerospace fuel-cell thermal management. While improvements in the modelling practices resulted in enhanced system thermal endurance and peak heat load management, the design ultimately did not achieve the cooling capacity required to support the nominal 20% fuel-cell-driven propulsion electrification, effectively managing only 185 kW of the approximately 900 kW required.

Consequently, air remains the more viable heat sink for the management of fuel-cell heat loads. Thus, in the following sections, a conventional ram-air architecture is preliminarily modelled and assessed in its capabilities and constraints.

Nonetheless, despite limitations, fuel-based thermal management demonstrated solid performance during early mission segments, offering reduced overall mass compared to air-based systems. These results motivate the development of a third integrated design, which relies primarily on ram-air cooling for fuel cell thermal management while employing fuel as a secondary heat sink.

4.1 State of the art

As anticipated in Sect. 2.3, ram-air cooling systems have been extensively investigated across the military, private, commercial, unmanned, battery-electric, and fuel cell-powered aircraft segments. Modelling efforts consistently focus on the coupled

aerodynamic, thermal, and gravimetric performance of the system, which ultimately affect the overall design feasibility.

Early design methodologies by Russ and Drela [211] employ lumped representations of the intake, heat exchanger, and nozzle–duct assembly to analyse under-wing installations operating under highly rarefied, high-altitude conditions. Their work highlights the significant aerodynamic sensitivity of aircraft to ram-air cooling component sizing, underscoring the critical importance of effective pressure recovery. Later work by Kellermann et al. [212] extends this approach by introducing a system-level analytical model that couples lumped representations of the diffuser, heat exchanger, and fan-assisted nozzle, enabling both design and off-design optimisation of air-based cooling systems. Their results are consistent with earlier aerodynamic observations, demonstrating that elevated component temperatures (<400 K) can partially offset cooling drag penalties via Meredith-effect thrust recovery [177], whereas lower-temperature operation generally leads to non-negligible induced drag and increased fuel consumption. Adler et al. [220] further extend these integrated concepts by incorporating vapour-compression cycles and thermoacoustic cooling technologies to enhance thermal management within a 180-passenger battery-hybrid airliner configuration.

Accordingly, induced drag minimisation emerges as a primary design driver for ram-air cooling architectures. Tan and Guo [213] corroborate this conclusion by specifically addressing intake drag penalties through lumped-parameter modelling and experimental validation of a scoop-type configuration for a UAV engine intake.

Higher-fidelity investigations have further refined induced drag estimation through CFD-based methodologies. Grois et al. [214] quantify aerodynamic trade-offs between scoop and flush (NACA-type) intakes for UAV propulsion, analysing pressure recovery and off-design stability. Their results confirm that aerodynamic integration frequently governs overall ram-air performance and indicate that top-mounted frontal configurations typically achieve higher pressure recovery efficiency. Flush intakes, in particular, exhibit improved flow stability at elevated angles of attack. Similarly, Eisfeld and Ruetten [210] employ CFD-based optimisation to refine fuselage-mounted, ram-air intake and outlet placement in transport aircraft, explicitly targeting system-cooling applications. Romeo et al. [215] also investigate intake design through detailed CFD analyses for fuel-cell heat removal in general aviation aircraft, while Pal and Severson [219] combine high-fidelity aerodynamic simula-

tions with advanced vapour-cycle systems to enhance sensitive component cooling under demanding operating conditions.

While CFD-based approaches provide greater fidelity and improved representation of complex geometries and flow behaviour, they remain computationally expensive and are typically confined to steady-state, design-point optimisation. Consequently, they are often decoupled from broader system-level analyses and evaluated sequentially rather than within fully integrated frameworks. For instance, subsequent work by Romeo et al. [216] expands the modelling scope to integrated representations of the fuel cell, water, and thermal management systems, relying on reduced-order intake models instead of CFD analyses. Conversely, Sain et al. [217] initially propose a lumped-parameter framework representing a complete nacelle envelope, including fuel cell, power electronics, motor, coolant loop, ram-air, and heat exchanger, and subsequently perform a detailed CFD-based parametric analysis of the intake, heat exchanger, and outlet geometries. These sequential multi-fidelity workflows underscore a persistent separation between high-resolution aerodynamic modelling and mission-level system integration.

Beyond aerodynamic considerations, Sain et al. [217] also demonstrate a non-negligible coupling between system mass and thermal effectiveness. Potamiti et al. [197] and Link et al. [218] corroborate these findings through lumped-parameter analyses of complete cooling, propulsion, and power-system envelopes, showing that coolant pumps and, in particular, coolant–air heat exchangers introduce significant mass penalties that directly affect aircraft-level feasibility.

Overall, the literature converges on two dominant design drivers: induced drag reduction and mass minimisation. Three primary modelling methodologies can then be identified:

- Lumped-parameter models of the intake, heat exchanger, and nozzle for system-level performance analyses and pressure recovery estimation. Well suited for integration with other system models, these approaches dominate early design, enabling preliminary aerodynamic and mass assessments at low computational cost.
- High-fidelity, steady-state CFD analyses and optimisation of intake and duct geometries under coupled aerodynamic constraints. These methods are typically applied during detailed component design, supporting accurate sizing

when precise boundary conditions and design parameters are available. However, the associated computational costs are often substantial.

- Sequential multi-fidelity approaches combining system-level parametric studies with detailed component aerodynamic simulations. These workflows reflect typical design evolution, progressing from early concept exploration, to system-level trade-offs and, finally, detailed device simulation and optimisation.

However, despite substantial advances in the aerodynamic, mass, and thermal modelling of ram-air cooling, analyses remain largely steady-state and design-point oriented, with limited consideration of mission-dependent dynamics and transient operability. Furthermore, aero-thermal interactions are typically coupled sequentially within broader system-level models, restricting the ability to capture feedbacks between the two domains and their sensitivity to mission and ambient variability. Hybrid integration with complementary thermal management architectures, as in Refs. [216, 217], remains largely unexplored, particularly with respect to combined air–fuel cooling. Addressing these limitations will require integrated system-level models capable of capturing aero-thermal coupling and hybrid thermal management behaviour across the mission envelope. In parallel, the development of computationally efficient multi-fidelity methodologies remains an important direction for future research.

4.2 Scope

This chapter addresses the identified gaps by developing an integrated, dynamic system-level model that combines ram-air, coolant loop, and fuel-based thermal management within a unified simulation framework. Unlike conventional ram-air sizing approaches, which target cooling compliance during specific operating conditions such as take-off or cruise, the present methodology evaluates thermal performance across the entire mission envelope. Within this context, the performance of a conventional air-based TMS is first assessed and subsequently compared with a combined air–fuel configuration, where ram-air serves as the primary heat sink and fuel provides supplementary thermal buffering. This enables the identification of mission-critical operating conditions in traditional air-based architectures and quantifies the extent to which integrated air–fuel cooling strategies can mitigate

design oversizing and enhance overall system effectiveness. A simplified induced-drag formulation and a first-order mass estimate are thus introduced for preliminary comparison between the two configurations.

The proposed numerical framework builds upon the modelling assumptions established in the preceding chapters of this dissertation, solving numerically the coupled temperature and mass dynamics of the heat source, coolant, air, and fuel subsystems within a unified environment. The reference hybrid-electric regional transport aircraft, MT-PEM fuel cell stack, and 20% nominal hybridisation target are retained. For the fuel-side modelling of the TMS, the recirculating-series architecture from Chap. 3 is adopted as the baseline configuration. All prior assumptions regarding fuel degradation, coking, flammability, and flow management are retained. In addition, a refined fuel control strategy is introduced to assess the feasibility of discontinuous fuel-based cooling and the deliberate recovery of fuel thermal capacity.

Consistent with previous analyses, system performance is evaluated across a set of representative mission profiles to examine the influence of range, altitude, and alternate conditions on cooling. This approach enables the identification of the most critical operating scenarios and the assessment of effective cooling strategies for the reference configurations.

As the study focuses on preliminary system-level performance and architectural trade-offs, detailed component design and failure-mode analyses remain outside the scope of this work.

4.3 Methodology

Following the modelling rationale established in Chap. 3, a system-level model of an air-based thermal management system (A-TMS) for fuel-cell cooling is developed in MATLAB/Simulink. The reference architecture, shown in Fig. 4.1, comprises three primary subsystems: the fuel cell, the coolant loop, and the air heat sink, interconnected via two dedicated heat exchanger elements (HE_U and HE_M).

The Eulerian formulation for open systems introduced in Fig. 3.2 is recalled once again to derive the mass and energy balances of each component, under the lumped-parameter assumptions of homogeneous, isotropic, and spatially uniform properties. The LMC (Eq. (3.1)) and FLT (Eq. (3.2)) are specialised to define the governing

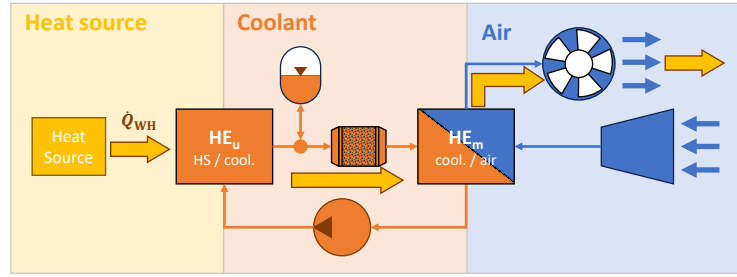


Fig. 4.1 Functional scheme of a ram-air-based A-TMS for the cooling of a fuel cell.

balance equations for the air-side components of the TMS, while previously assessed elements, such as the fuel cell, coolant loop, and heat exchangers, are inherited from Chap. 3 without modification.

The air-based model is subsequently extended to a combined air–fuel TMS (AF-TMS) by integrating the fuel-side architecture analysed in Chap. 3 with this new air-driven baseline. In this configuration, the fuel system operates as an auxiliary heat sink and is connected downstream of the air cooling branch within the coolant loop, as illustrated in Fig. 4.2.

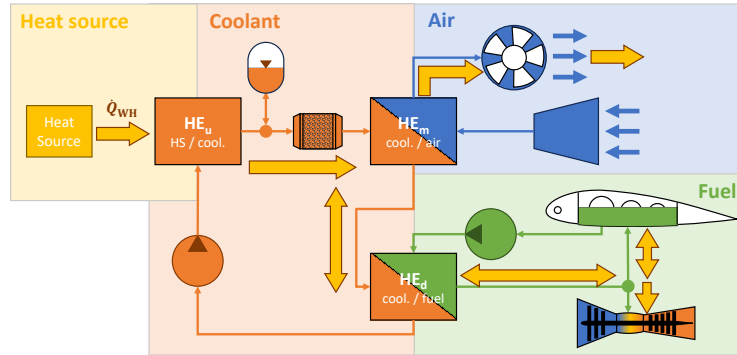


Fig. 4.2 Functional scheme of a ram-air, recirculating-series single tank, AF-TMS for the cooling of a fuel cell.

This arrangement is selected over alternative layouts based on the following considerations:

- Locating the coolant–air heat exchanger (HE_M) immediately downstream of the heat source–coolant exchanger (HE_U) ensures that the highest-temperature coolant is first directed to the primary cooling architecture. This maximises

the temperature gradient between the working fluids, enhancing air-side heat dissipation and promoting favourable pressure recovery due to the increased outlet air temperature (moderate Meredith effect).

- Positioning the coolant–fuel heat exchanger (HE_D) downstream of HE_M ensures that the fuel is exposed to comparatively lower coolant temperatures. During high-intensity operation, this mitigates peak fuel temperatures, particularly within the tanks, thus reducing flammability risks and evaporation losses. However, this arrangement inherently limits the maximum exploitable thermal capacity of the fuel, effectively constraining the overall cooling potential of the fuel-based subsystem.
- Additionally, placing HE_D downstream of HE_M may create operating conditions in which the incoming coolant is at a temperature lower than that of the bulk fuel stored in tanks, particularly during low-intensity phases. A portion of the stored fuel thermal energy may thus be transferred to the coolant and subsequently dissipated through the air-side heat exchanger, enabling partial recovery of the fuel’s thermal capacity without relying exclusively on passive heat dissipation at the tank wall interface. Under such circumstances, the fuel subsystem can deliberately operate as an active thermal buffer: absorbing excess heat during high-demand phases and releasing it to the coolant for dissipation during low-demand operation, as illustrated in Fig. 4.3.

The integrated architecture of Fig. 4.2 can thus serve as the reference framework for a unified air/fuel TMS model capable of representing sole-fuel, sole-air, and combined air–fuel configurations within a single simulation environment. This versatility is achieved by selectively activating or nullifying the heat transfer terms associated with the air- and fuel-side heat exchanger submodels, effectively neglecting either heat sink depending on the desired configuration.

Consistent with previous observations, the balance equations governing the fuel-side components, including the fuel branch of HE_D , the control loop, the engines, and the storage tanks, are all retained from Chap. 3 and embedded within the integrated framework. However, some elements of the F-TMS model require updating to ensure consistent thermodynamic and numerical integration within the coupled architecture, while air-side components must still be formulated in full.

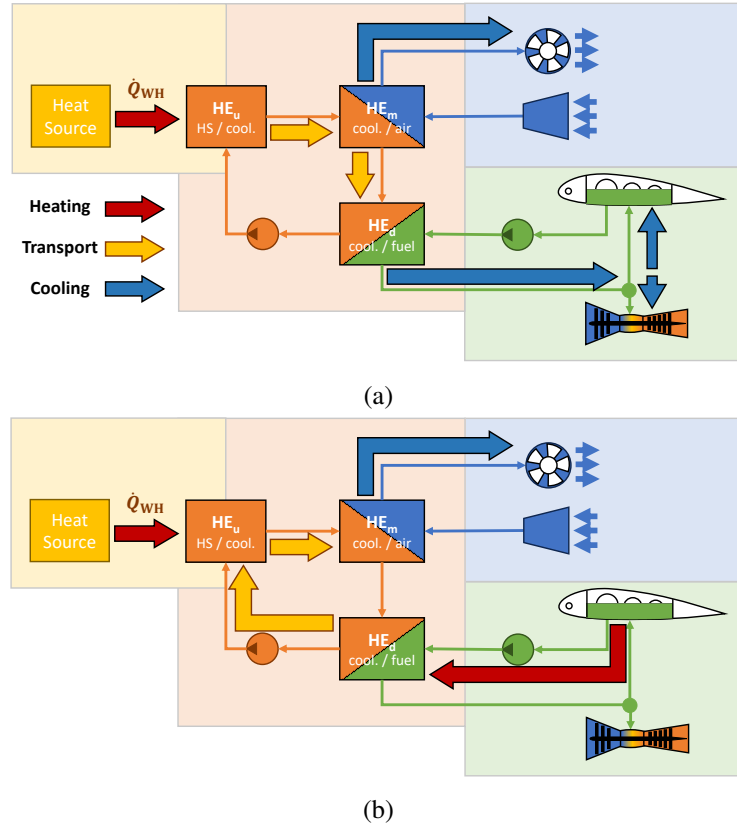


Fig. 4.3 Schematic representations of fuel-based thermal buffering for an AF-TMS: (4.3a) F-TMS cooling (absorbing heat from the coolant) and (4.3b) F-TMS recharging (releasing heat to the coolant).

The following sections introduce the additional modelling assumptions, governing equations, and subsequent numerical implementation specific to the AF-TMS, derived through appropriate specialisation of the LMC (Eq. (3.1)) and FLT (Eq. (3.2)) formulations.

4.3.1 The air system

The air system extracts heat from the coolant loop through the intermediate coolant–air heat exchanger (HE_M), located between the heat source–coolant heat exchanger (HE_U) and the coolant–fuel heat exchanger (HE_D). A ram-air intake supplies the cooling airflow to the air side of HE_M, where it governs the heat rejection rate and the resulting air outlet temperature.

Hence, analogously to the fuel system described in Sect. 3.3.3, the air system of the integrated AF-TMS is herein represented by a set of dedicated submodels governing its numerical simulation, as described in the following sections.

Air heat exchanger

The air-side heat exchanger submodel constitutes the first of these elements. It receives the heat transfer rate signal from the intermediate structural model of HE_M , which couples the coolant and the air subsystems, and applies it to the airstream.

Assuming that the airflow velocity downstream of the intake remains below 100 ms^{-1} , compressibility effects may be neglected and incompressible flow formulations adopted. Under these assumptions, the modelling approach employed for the fuel side of HE_M in Sect. 3.3.3 can be consistently extended to the air side, following appropriate adjustment of the thermophysical properties.

Accordingly, Eq. (4.1) expresses the integral mass balance over the air-side control volume of HE_M :

$$\int_{A_{HE_M, \text{air}, \text{in}}} \rho_{\text{in}} \vec{c}_{\text{in}} \cdot \vec{n}_{\text{in}} dA = \int_{A_{HE_M, \text{air}, \text{out}}} \rho_{\text{out}} \vec{c}_{\text{out}} \cdot \vec{n}_{\text{out}} dA \quad (4.1)$$

Consistently with the coolant loop, the air system is modelled as a single stream without branching. Assuming a single inlet and a single outlet for the heat exchanger, a unique air mass flow rate can be defined: $(\dot{m}_{HE_M, \text{in}})_{\text{air}} = (\dot{m}_{HE_M, \text{out}})_{\text{air}} = \dot{m}_{\text{air}}$.

The corresponding energy balance for the air-side control volume is thus given by Eq. (4.2):

$$\frac{d}{dt} \int_{V_{HE_M, \text{air}}} \rho u dV = \dot{Q}_{HE_M} + \int_{A_{HE_M, \text{air}, \text{in}}} \rho_{\text{in}} h_{\text{in}} \vec{c}_{\text{in}} \cdot \vec{n}_{\text{in}} dA - \int_{A_{HE_M, \text{air}, \text{out}}} \rho_{\text{out}} h_{\text{out}} \vec{c}_{\text{out}} \cdot \vec{n}_{\text{out}} dA \quad (4.2)$$

Under the assumptions of spatially uniform, homogeneous, and isotropic properties within the control volume, together with the constant specific heat approximation, Eq. (4.2) reduces to the lumped-parameter ordinary differential equation reported in Eq. (4.3). For improved readability, the "air" subscript is omitted in the resulting

expression:

$$\dot{T}_{\text{HEM}}(t) = \frac{1}{m_{\text{HEM}}} \left(\frac{\dot{Q}_{\text{HEM}}(t)}{c_p} + \dot{m}(t) (T_{\text{HEM},\text{in}}(t) - T_{\text{HEM}}(t)) \right) \quad (4.3)$$

The equation is solved numerically within the model, given an initial air-side HE temperature $T_{\text{HEM},\text{air}}(0)$.

Air intake

Unlike the coolant and fuel subsystems, where active control strategies can be readily implemented, airflow regulation is less straightforward, as it is primarily governed by the aircraft's relative velocity with respect to the external environment. A dedicated digital intake component is therefore introduced to model the dependency between aircraft airspeed and the supplied air mass flow rate.

Two identical scoop-type intakes are assumed to provide the $\dot{m}_{\text{air}}$ airflow, as these configurations are generally preferred over NACA flush inlets when higher mass flow rates and elevated heat rejection requirements are expected. Strictly speaking, ram-air intake flows should be treated using compressible flow formulations. However, given the low-to-moderate subsonic operating conditions typical of regional transport aircraft ($Ma_{\text{air}} < 0.4\text{--}0.5$), density variations remain moderate, such that simplified modelling approaches remain acceptable as first-order estimates. Accordingly, Eq. (4.4) provides a preliminary estimate of the total mass flow rate entering the scoop intakes as a function of the local air density and velocity:

$$\dot{m}_{\text{air}}(t) = \rho_{\text{air}}(t) u_{\text{air}}(t) (N_{\text{intake}} A_{\text{intake}}) \eta_{\text{intake}} \quad (4.4)$$

where the $\rho_{\text{air}}(t)$ term indicates the instantaneous air density at the intake entrance, which is assumed, as a first approximation, to be equal to the local ambient air density. $u_{\text{air}}(t)$ is the air velocity at the intake inlet, which can be preliminarily assumed to coincide with the instantaneous aircraft speed provided by the mission planner module described in Sect. 3.3.4. This assumption represents a first-order approximation of the actual intake behaviour, neglecting external aerodynamic effects such as local flow acceleration, incidence angles, and flow distortions. Neverthe-

less, it remains consistent with the fidelity levels typically adopted in preliminary system-level analyses of aircraft thermal management systems.

The $(N_{\text{intake}} A_{\text{intake}})$ product denotes the total intake capture area, defined as the product of the number of intakes and their individual inlet area (assuming all intakes are identical). The effectively captured air mass flow rate is finally reduced by an η_{intake} factor, representing a lumped intake efficiency parameter accounting for spillage and other non-ideal intake phenomena. For the preliminary sizing of scoop-type ram-air intakes, a constant value of $\eta_{\text{intake}} = 0.75$ is typically adopted. This conservative assumption is consistent with first-order mass capture modelling practices commonly employed in early-stage aircraft systems analyses.

Alternative approaches for regulating intake airflow are also possible. Variable-geometry intakes, for instance, can provide airflow modulation at moderate to high speeds by modifying the inlet section and internal pressure evolution. However, these require detailed knowledge of aircraft operating conditions and duct geometries for proper design, which are not available at the current stage of the system-level analysis. Accordingly, these solutions are not considered further.

Downstream puller-fan architectures, on the other hand, represent the primary candidates for airflow modulation during low-speed conditions, when the cooling capability provided by ram-air alone becomes insufficient. Although detailed modelling of the puller fan lies beyond the scope and fidelity level of the present study, a simplified representation of its function is included to account for cooling during ground operations. Typical airflow velocities in radiator applications range between 3 ms^{-1} to 10 ms^{-1} . Consistent with the current level of abstraction, a single puller-fan configuration is assumed, providing a minimum airflow velocity of 5 ms^{-1} when the ram-air contribution from aircraft motion is insufficient. Within the numerical model, this behaviour is represented by imposing a lower saturation limit of 5 ms^{-1} on the velocity signal used in Eq. (4.4).

Air properties

A dedicated submodel computes the air properties required for both the intake and air-side HE elements. Air properties are computed from the Simulink *ISA Atmosphere Model* block as a function of the current aircraft altitude, based on tabulated data from the International Standard Atmosphere (ISA) [274].

To represent a hot day scenario, the computed ISA temperature is increased by a positive offset of 25 °C, $T_{\text{air}} = T_{\text{ISA}} + 25^\circ\text{C}$, resulting in the reference ground temperature of 40 °C. This linear correlation provides a reasonable approximation of hot-day atmospheric temperatures up to 10.000 m and remains valid within the typical service ceiling of turboprop regional transport aircraft (~ 7.600 m). Air density is then calculated accordingly, assuming perfect gas behaviour through Eq. (4.5):

$$\rho_{\text{air}} = \frac{p_{\text{ISA}}}{R_{\text{air}}(T_{\text{ISA}} + 25^\circ\text{C})} \quad (4.5)$$

where $R_{\text{air}} = 287 \text{ J kg}^{-1} \text{ K}^{-1}$ is the ideal gas constant of air, in accordance with standard thermodynamic literature [260]. The resulting density is used to compute the intake mass flow rate in Eq. (4.4).

As the air approaches the HE, it decelerates and compresses, resulting in a heated flow. Accurate modelling of this effect requires detailed knowledge of the diffuser, heat exchanger, and nozzle geometries, which are not available at this preliminary design stage. Thus, the stagnation (total) temperature of the airflow entering the intake is assumed to represent the maximum temperature achievable under compression heating. It combines both the thermal and kinetic energy terms of the stream as given by Eq. (4.6):

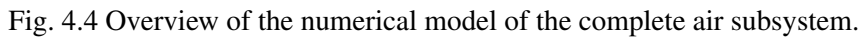
$$T_{\text{air,stag}} = T_{\text{air}} + \frac{u_{\text{air}}^2}{2c_{p,\text{air}}} \quad (4.6)$$

As a first-order approximation, this stagnation temperature is applied immediately upstream of the air-side inlet of HE_M , providing a conservative thermal boundary condition to Eq. (4.3).

The complete air-system model, comprising the air properties submodel, the air-intake component, and the air-side heat exchanger, is thus illustrated in Fig. 4.4.

4.3.2 The updated coolant system

Compared to the model in Sect. 3.3.2, the new coolant loop of the AF-TMS framework is expanded to include the intermediate heat exchanger HE_M , located between


$$\int_{A_{\text{HE}_M, \text{cool}, \text{in}}} \rho_{\text{in}} \vec{c}_{\text{in}} \cdot \vec{n}_{\text{in}} dA = \int_{A_{\text{HE}_M, \text{cool}, \text{out}}} \rho_{\text{out}} \vec{c}_{\text{out}} \cdot \vec{n}_{\text{out}} dA \quad (4.7)$$

The energy balance equation for the coolant side of HE_M also retains the same structure as Eq. (3.15), here written as Eq. (4.8), with the "cool" subscript omitted to improve readability:

The resulting ODE is solved numerically within the model, given an initial coolant temperature $T_{\text{HEM,cool}}(0)$. The computed temperature change rate is then integrated to determine the coolant temperature history $T_{\text{HEM,out}}(t)$.

During most operating scenarios, the heat transfer rate $\dot{Q}_{\text{HEM}}(t)$ is assumed negative for the coolant side in Eq. (4.8) and positive for the air side in Eq. (4.3). Its value is computed according to Eq. (4.9), which governs heat transfer across the intermediate structure element of HE_M , following the same modelling assumptions

introduced in Sect. 3.3.2:

$$\dot{Q}_{\text{HE}_M} = \varepsilon_{\text{HE}_M} C_{\text{HE}_M, \min} (T_{\text{HE}_M, \text{cool}, \text{in}} - T_{\text{HE}_M, \text{air}, \text{in}}) \quad (4.9)$$

The reference heat exchanger thermal effectiveness of $\varepsilon_{\text{HE}_M} = 0.7$ is retained, while the minimum heat capacity rate, $C_{\text{HE}_M, \min}$, is defined according to Eq. (4.10):

$$C_{\text{HE}_M, \min} = \min(c_{p, \text{cool}} \dot{m}_{\text{cool}}, c_{p, \text{air}} \dot{m}_{\text{HE}, \text{air}}) \quad (4.10)$$

Within the numerical model, a dedicated HE_M structure element computes both Eqs. (4.9) and (4.10), connecting the coolant loop to the air system. This element also serves as a model truncation point, allowing air-side heat transfer to be neglected when modelling purely fuel-based TMSs. Analogous considerations also apply to the intermediate element of HE_D , which is updated to allow the full disregard of fuel-side heat transfer when considering solely air-based cooling.

Fig. 4.5 illustrates the updated three-HE coolant loop and the outer, intermediate structure elements connecting it to the heat source, air system, and fuel system.

The coolant controller remains unchanged, supporting a maximum mass flow rate of 500 L min^{-1} , which increases linearly across a $T_{\text{HS}} = 100^\circ\text{C}$ to 115°C activation-to-saturation temperature range.

4.3.3 The updated fuel system

The fuel system model of Sect. 3.3.3 is retained, together with the governing equations for the fuel side of HE_D , the engines, and the fuel tank submodels. The fuel controller element, however, requires modification to account for the introduction of the intermediate heat exchanger and the auxiliary role of the fuel in the new integrated AF-TMS architecture.

Eq. (3.19) provides the original controller logic regulating the fuel flow rate through HE_D . The reference temperature $T_{\text{HE}_U, \text{cool}}(t)$ is retained as the controller feedback in the updated model, as it represents the highest coolant temperature within the system and thus a suitable indicator of the need for supplementary cooling. The regulation bandwidth is shifted to lower values to favour earlier system response while maintaining a width of 8°C . Specifically, the controller now activates once the

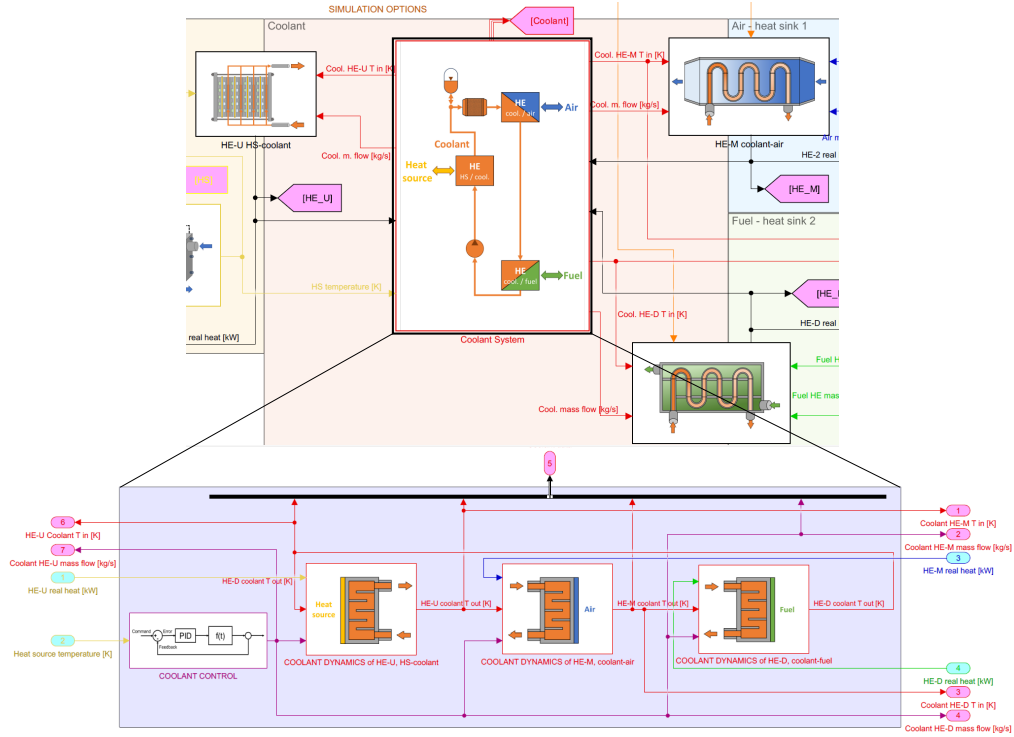


Fig. 4.5 Overview of the numerical model of the coolant loop subsystem and outer HE structure elements connecting to the fuel cell, air system, and fuel system.

coolant temperature exceeds 100°C and saturates at 108°C . These settings define the baseline controller logic adopted in the AF-TMS design.

However, this baseline formulation does not fully exploit the additional capabilities introduced by the integrated air–fuel architecture. As discussed in Sects. 4.2 and 4.3, the inclusion of ram-air cooling and the selected HE arrangement enable the recovery of fuel thermal capacity by transferring heat from the fuel back to the coolant and subsequently rejecting it through the air-side heat exchanger. For this process to occur, two conditions must be satisfied: the fuel must circulate through the heat exchanger, and the coolant entering HE_D must be cooler than said fuel. The baseline controller cannot support this behaviour in its current formulation, as the fuel is supplied to the heat exchanger only when the coolant temperature exceeds the activation threshold of 100°C . Consequently, heat return from the fuel would only occur under rare circumstances in which $T_{\text{HE}_{U,\text{cool}}}(t)$ exceeds the F-TMS activation temperature while $T_{\text{HE}_{M,\text{cool}}}(t)$ remains below the tank fuel temperature. Moreover,

such conditions are generally undesirable, as introducing additional heat into the coolant loop would likely aggravate an already demanding thermal scenario.

To address this limitation, two mutually exclusive operating modes are defined at the functional level: a *fuel-based cooling mode* and a *tank thermal-capacity recharge mode*. The fuel-based cooling mode is activated once the maximum coolant temperature exceeds the activation threshold of 100 °C and the coolant entering HE_D is hotter than the tank fuel. Under these conditions, the fuel acts as a heat sink, absorbing heat from the coolant. Conversely, the tank thermal-capacity recharge mode is activated when the maximum coolant temperature remains below 100 °C and the coolant entering HE_D is also colder than the tank fuel. In this regime, the fuel behaves as a heat source, transferring its thermal energy to the coolant and restoring the thermal endurance of the F-TMS. In all other cases, no fuel flow is supplied to the coolant-fuel heat exchanger (HE_D). Tab. 4.1 summarises the logical conditions governing the various operating modes.

Table 4.1 Operating modes of the updated air–fuel TMS control logic governing the fuel heat-exchanger mass flow rate.

	$T_{\text{HEU,cool}}(t) < T_{\text{fuel,sat}} - \Delta T_{\text{fuel,rbw}}$	$T_{\text{HEU,cool}}(t) > T_{\text{fuel,sat}} - \Delta T_{\text{fuel,rbw}}$
$T_{\text{HEM,cool}}(t) < T_{\text{FT}}(t)$	<i>tank thermal capacity recharge mode</i>	zero fuel mass flow rate
$T_{\text{HEM,cool}}(t) > T_{\text{FT}}(t)$	zero fuel mass flow rate	<i>fuel-based cooling mode</i>

with $T_{\text{fuel,sat}} - \Delta T_{\text{fuel,rbw}} = 100\text{ °C}$.

Within the model, this logic is implemented by retaining the saturated PID control of the fuel mass flow and subsequently gating the signal according to the conditional activation of the operating modes. For simplicity, the PID controller is mirrored around the threshold temperature, so that the commanded mass flow rate becomes zero when $T_{\text{HEU,cool}}(t)$ approaches 100 °C, and saturates once the temperature exceeds 108 °C or falls below 92 °C. The resulting PID control signal is given by Eq. (4.11):

$$\dot{m}_{\text{HED,fuel,PID}}(t) = \text{sat}_{[-\dot{m}_{\text{fuel,max}}, \dot{m}_{\text{fuel,max}}]} \left(\frac{\dot{m}_{\text{fuel,max}}}{\Delta T_{\text{fuel,rbw}}} [T_{\text{HEU,cool}}(t) - (T_{\text{fuel,sat}} - \Delta T_{\text{fuel,rbw}})] \right) \quad (4.11)$$

where the baseline parameters are retained as $T_{\text{fuel,sat}} = 108^\circ\text{C}$ and $\Delta T_{\text{fuel,rbw}} = 8^\circ\text{C}$. Under this formulation, the PID controller returns a positive output when the coolant temperature exceeds the activation threshold, indicating the need for fuel-based cooling, and a negative output when the temperatures are lower, indicating the potential for fuel thermal capacity recovery.

Conditional activation of the operating modes can then be determined using binary signals, expressed according to Eqs. (4.12) and (4.13):

$$S_1 = \frac{1}{2} \left[1 + \text{sign} \left(T_{\text{HEU,cool}}(t) - (T_{\text{fuel,sat}} - \Delta T_{\text{fuel,rbw}}) \right) \right] \quad (4.12)$$

$$S_2 = \frac{1}{2} \left[1 + \text{sign} \left(T_{\text{HEM,cool}}(t) - T_{\text{FT}}(t) \right) \right] \quad (4.13)$$

Eq. (4.12) returns $S_1 = 1$ when the maximum coolant temperature exceeds the 100°C threshold, and 0 otherwise. Conversely, Eq. (4.13) yields $S_2 = 1$ when the coolant entering HE_D is hotter than the tank fuel, and 0 otherwise. Under these assumptions, the fuel-based cooling mode activates only when $S_1 = S_2 = 1$, whereas the tank thermal-capacity recharge mode activates only when the opposite is true ($S_1 = S_2 = 0$). The corresponding activation conditions can thus be written according to Eqs. (4.14) and (4.15), respectively.

$$S_{\text{cool}} = S_1 S_2 \quad (4.14)$$

$$S_{\text{rech}} = (1 - S_1)(1 - S_2) \quad (4.15)$$

This formulation ensures that the two modes remain mutually exclusive and allows the controller logic to conditionally gate the absolute value of the regulated PID signal $\dot{m}_{\text{HE}_\text{D},\text{fuel,PID}}(t)$, according to Eq. (4.16).

$$\dot{m}_{\text{HE}_\text{D},\text{fuel}}(t) = \text{sat}_{[0,\dot{m}_{\text{fuel,max}}]} \left(\left| \dot{m}_{\text{HE}_\text{D},\text{fuel,PID}}(t) \right| \cdot (S_{\text{cool}} + S_{\text{rech}}) \right) \quad (4.16)$$

A saturation operator $\text{sat}_{[0,\dot{m}_{\text{fuel,max}}]}$ finally ensures that the commanded cooling mass flow remains non-negative and bounded, returning zero when neither of the two operating modes is active.

However, switching between modes through discrete logical operators ($\text{sign}()$), introduces discontinuities in the control signal, which can potentially lead to numerical instabilities and premature termination of the simulation. To mitigate this issue, switching functions are replaced with smooth sigmoid approximations based on hyperbolic tangent functions, reformulated according to Eqs. (4.17) and (4.18).

$$S_1 = \frac{1}{2} \left[1 + \tanh \left(\frac{T_{\text{HEU,cool}}(t) - (T_{\text{fuel,sat}} - \Delta T_{\text{fuel,rbw}})}{\Delta T_1} \right) \right] \quad (4.17)$$

$$S_2 = \frac{1}{2} \left[1 + \tanh \left(\frac{T_{\text{HEM,cool}}(t) - T_{\text{FT}}(t)}{\Delta T_2} \right) \right] \quad (4.18)$$

where ΔT_1 and ΔT_2 are the smoothing parameters controlling the sharpness of the transition, here assumed to be equal, $\Delta T_1 = \Delta T_2 = 1^\circ\text{C}$.

The numerical implementation of the baseline and upgraded control logics is presented in Fig. 4.6.

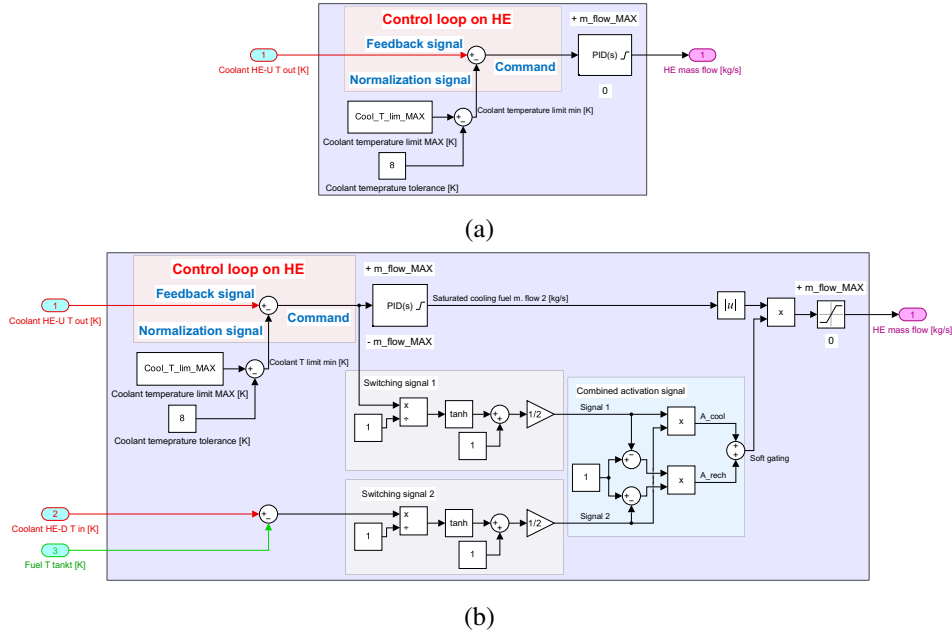


Fig. 4.6 Overview of the baseline (4.6a) and updated (4.6b) fuel-flow controllers for the AF-TMS.

4.3.4 The integrated AF-TMS model

The complete Simulink model of the integrated air–fuel thermal management system is ultimately shown in Fig. 4.7, connecting all relevant subsystems of the proposed cooling architecture.

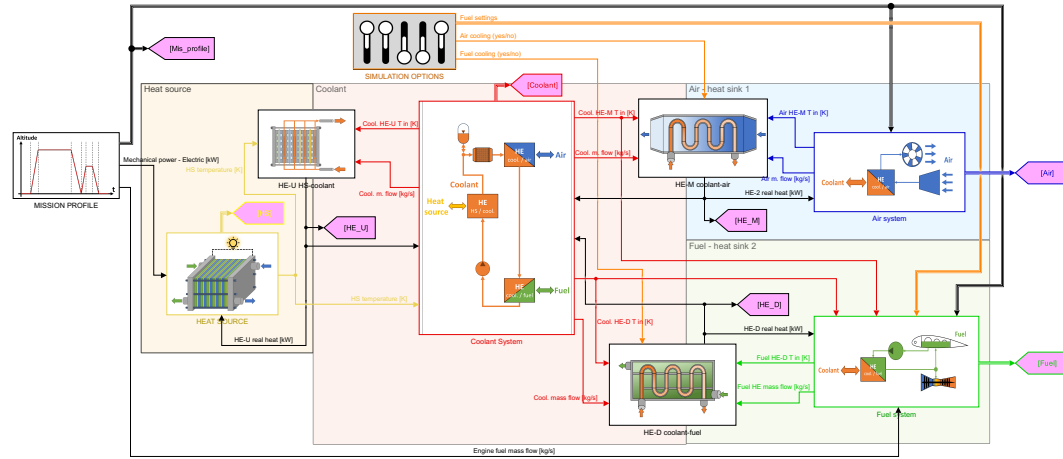


Fig. 4.7 Overview of the numerical model of the complete air–fuel thermal management system in Simulink.

Beyond the previously described subsystems, the model retains the Mission Planner element and the option block from the previous framework. However, the option block is also expanded to command the selective deactivation of either the air-driven or the fuel-based TMS, enabling flexible simulation of individual or combined cooling strategies.

4.4 Results

The presented AF-TMS model is herein used to assess the feasibility of both purely air-based and air–fuel thermal management strategies for supporting fuel cell cooling in hybrid-electric regional aircraft. The following sections present simulation results for both configurations across representative mission scenarios, consistent with the previously adopted rationale of Sect. 3.4. The analysis aims to compare their respective thermal performances and evaluate the potential benefits of integrating fuel-based thermal buffering within conventional air-cooling architectures.

4.4.1 Air-based TMS dynamics

The system-level performance of a baseline A-TMS is assessed first. The reference flight scenario of Fig. 3.11 is retained, consisting of a 6000 m, 400 km main mission [276], followed by a 4500 m, 200 km alternate flight. A hybridisation degree of 20% is assumed, in line with the design targets of this dissertation. Finally, a baseline intake configuration consisting of two air intakes is adopted, each with an inlet area of $A_{\text{intake}} = 0.15 \text{ m}^2$, corresponding to a total capture area of 0.30 m^2 .

Assuming a single air stream and neglecting internal air spillage or storage within the aircraft, mass dynamics can be disregarded for the purely air-based configuration. Unlike in the fuel-based systems, neither the cooling air nor the coolant circuit contains significant mass reserves subject to substantial thermal storage or progressive depletion. Consequently, temperature dynamics represent the primary factor governing system performance. The resulting temperature histories are shown in Fig. 4.8, illustrating the system effectiveness in maintaining the fuel cell temperature within its operating limits.

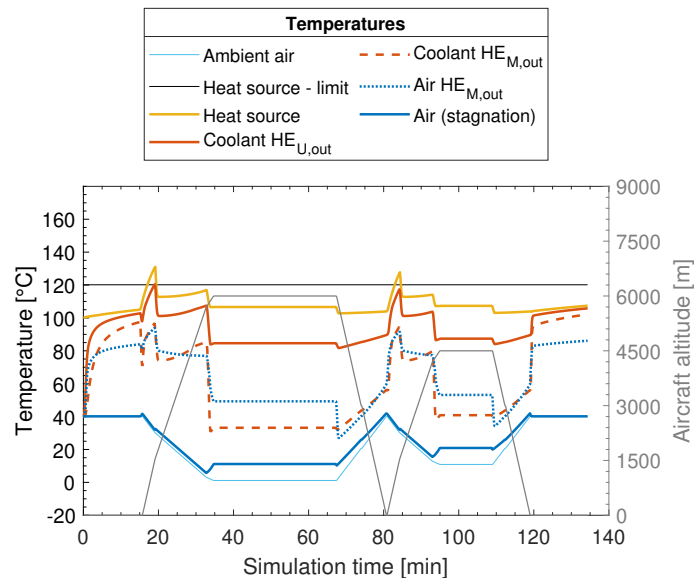


Fig. 4.8 Relevant temperature trends of an A-TMS with two 0.15 m^2 intakes. Cooling results for a 20% hybrid-electric turboprop aircraft performing a 400 km, 6000 m main mission followed by a 200 km, 4500 m alternate flight.

The results indicate insufficient cooling under these initial assumptions, with the heat source exceeding its maximum operating temperature in two separate instances corresponding to the early climb segments of the main and alternate missions. These phases clearly emerge as the most critical operating conditions, resulting from a combination of maximum power demand, which drives peak heat generation, and relatively moderate longitudinal aircraft velocities, leading to reduced incoming air mass flows.

For the remainder of the mission, temperature profiles remain relatively stable. Unlike in the previously analysed F-TMS case, the external environment effectively acts as an infinite heat sink, with sufficient thermal inertia to absorb the rejected heat without appreciable changes in ambient temperature. This behaviour is particularly evident during the cruise segments, where the system rapidly reaches thermal equilibrium and all temperature trends remain nearly constant.

Apart from the early climb phases, the heat source temperature remains consistently below its operational limit. This result indicates that while the system is under-dimensioned for the most demanding conditions, it remains underutilised during the rest of the mission. These observations motivate the integration of fuel-based thermal buffering, which is investigated in the following section. However, before introducing this additional capability, a purely air-based benchmark must be defined in order to quantify the potential benefits of the combined air–fuel architecture. In particular, intake sizing plays a dominant role in the design and performance of air-based thermal management solutions, as larger inlet areas generally enable greater mass flow rates and increased heat rejection. The influence of intake sizing on airflow and cooling effectiveness is illustrated in Fig. 4.9, where the baseline configuration of 0.15 m^2 per intake is compared with 0.10 m^2 and 0.20 m^2 alternatives. For clarity, only the heat source and outlet air temperature trends are reported.

The simulations confirm the expected trends, with larger intakes improving cooling and maintaining lower heat source temperatures. In particular, the configuration with two 0.20 m^2 intakes consistently provides sufficient cooling throughout the entire simulation scenario, whereas smaller 0.10 m^2 intakes yield significantly higher peak temperatures, making the end-of-climb segment the most critical condition. Nonetheless, outside the climb segments, all configurations keep their heat source temperatures within the upper operating limit, confirming that the system is in-

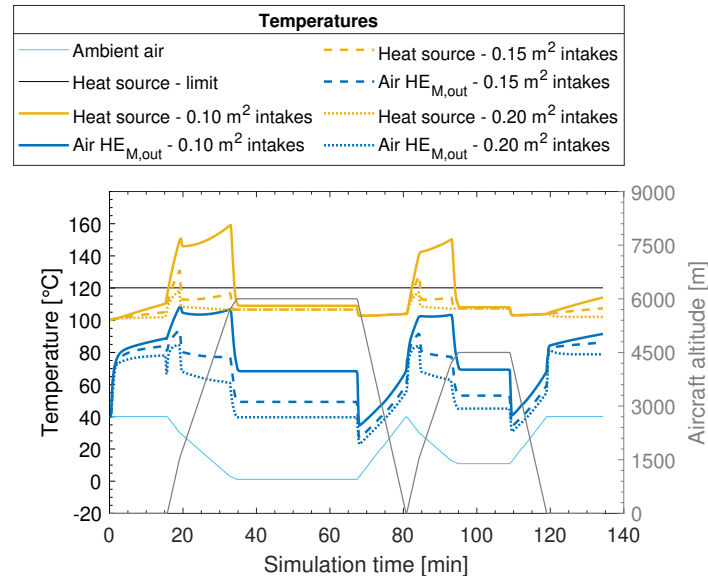


Fig. 4.9 Relevant temperature trends of an A-TMS with two 0.10 m², 0.15 m², and 0.20 m² intakes. Cooling results for a 20% hybrid-electric turboprop aircraft performing a 400 km, 6000 m main mission followed by a 200 km, 4500 m alternate flight.

deed oversized for less demanding phases, strengthening the appeal of integrating fuel-based thermal buffering.

Fig. 4.9 also illustrates the effects of different puller-fan architectures. Maintaining a minimum air velocity of 5 ms⁻¹ in a 0.10 m² duct results in a lower cooling mass flow rate compared to a 0.20 m² duct. Consequently, the smaller intake configuration experiences progressively higher heat source temperatures during taxi-out and taxi-in phases, whereas the larger layout maintains relatively stable temperature levels. However, sustaining the same airspeed through a larger intake inevitably requires a more powerful fan, leading to increased parasitic power consumption.

These observations highlight some of the trade-offs associated with increasing intake area, as larger intakes also introduce a series of additional system penalties. Beyond the increased electrical power demand of the puller fan, wider intakes may significantly aggravate parasitic aerodynamic drag, which should instead be minimised whenever possible. Moreover, larger intake areas typically require expanded ducts, heat exchangers, and associated components, potentially increasing system mass. Finally, higher cooling mass flow rates generally produce lower outlet air

temperatures, reducing the potential thrust recovery associated with the Meredith effect and further aggravating the drag penalty.

Accordingly, the intake capture area must be carefully sized to ensure sufficient cooling across the mission envelope while minimising these penalties. To this end, a brief comparative analysis is conducted across different altitudes, main mission ranges, and alternate flight distances to identify the most challenging scenario for appropriately sizing the aircraft intakes, following the rationale of Sect. 3.4.2.

Influence of altitude (A-TMS)

The influence of the main cruise altitude is assessed first, by comparing cases at 4500 m, 6000 m, and 7600 m for the reference configuration with two 0.15 m^2 intakes. Fig. 4.10 reports the resulting fuel cell and HE_M outlet air temperature dynamics.

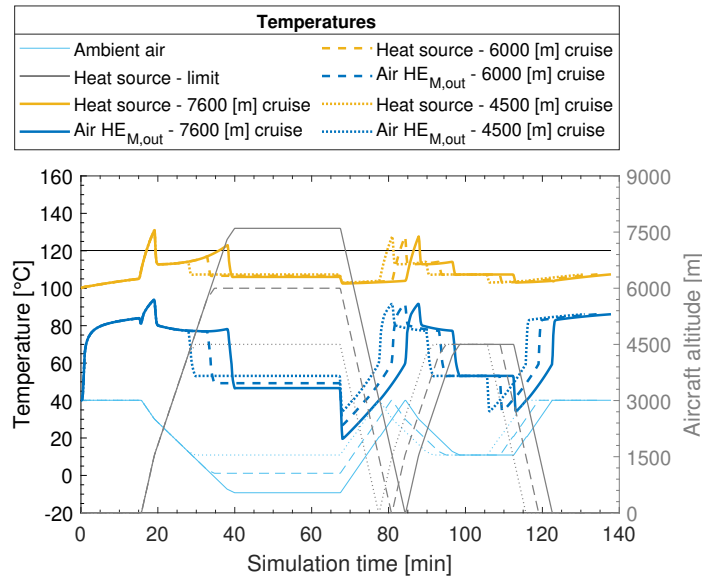


Fig. 4.10 Relevant temperature trends of an A-TMS with two 0.15 m^2 intakes. Cooling results for a 20% hybrid-electric turboprop aircraft performing a 400 km main mission at different altitudes, followed by a 200 km, 4500 m alternate flight.

The results consistently identify early climb segments, occurring at low altitudes of 0 m to 1500 m, as the most thermally demanding phases across all simulations. For the remainder of the climb, however, higher cruise altitudes lead to generally

higher heat source temperatures due to cumulative heating of the fuel cells. This behaviour suggests that the higher-altitude missions are the more demanding scenario, reflected in the non-linear increase of the end-of-climb temperatures with altitude. These results are consistent with expectations as, although the air-to-heat-source temperature gradient increases with altitude, it is typically offset by the reduction in air density, which significantly reduces the captured cooling airflow.

During the cruise and descent phases, temperature differences among the three altitude cases remain moderate, with slightly more effective cooling observed at higher altitudes instead. This behaviour results from the modelling assumption of adopting a minimum constant heat exchanger effectiveness of 0.7 across all simulations for early comparative assessment. While reasonably applicable to the previous fuel and coolant heat exchange, this assumption represents a coarser approximation of air-side heat transfer at varying altitudes. In practice, a fixed heat exchanger designed to maintain an effectiveness of 0.7 at 7600 m would likely achieve higher effectiveness at lower altitudes, making low-altitude flights generally more favourable, as observed during climb. This factor should be considered when extrapolating these early design results to the performance of a fully defined system.

These observations confirm the early climb phases as the primary thermal constraint for the system, while higher-altitude missions define the generally more challenging scenario, particularly near the end-of-climb segment.

Influence of main and alternate ranges (A-TMS)

The influence of the main and alternate mission ranges is assessed next by comparing main mission distances of 130 km, 400 km, and 1300 km, and alternate segments of 50 km, 200 km, and 350 km. Figs. 4.11a and 4.11b present the resulting fuel cell and outlet air temperature dynamics for the main and alternate mission analyses, respectively.

Consistent with observations from Sect. 3.4.2, altitude and range saturation effects appear both in Figs. 4.11a and 4.11b as a consequence of the adopted mission-planning assumptions. Specifically, the 130 km mission range shown in Fig. 4.11a is subject to an altitude limitation of 3150 m, while the alternate distance is capped at 130 km, since the alternate airfield located at 200 km exceeds the departure airport

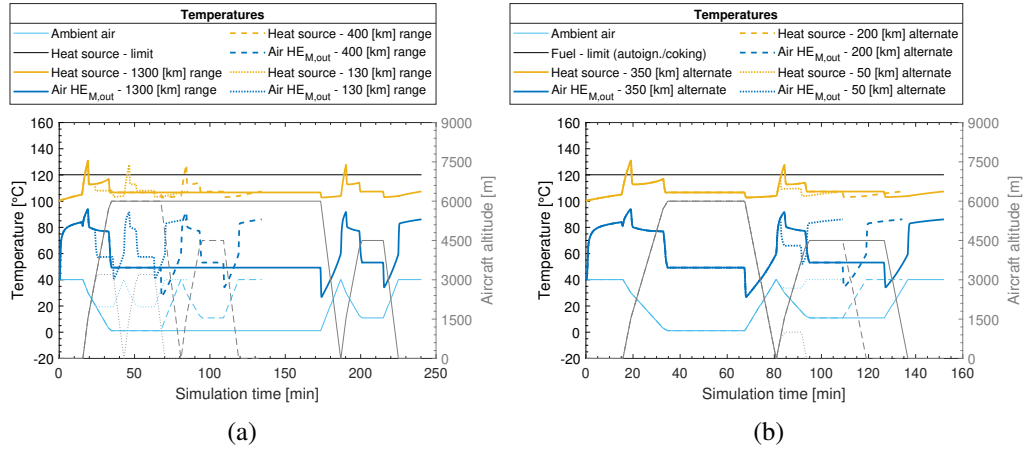


Fig. 4.11 Relevant temperature trends of an A-TMS with two 0.15 m^2 intakes. Cooling results for a 20% hybrid-electric turboprop aircraft performing different mission profiles at a 6000 m main cruise altitude followed by a 4500 m alternate segment. (4.11a) comparison of results for 130 km, 400 km, and 1300 km main flight ranges. (4.11b) comparison of results for 50 km, 200 km, and 350 km alternate flight ranges.

distance. Conversely, the 50 km alternate case in Fig. 4.11b requires a reduction of the cruise altitude to the minimum allowed value of 1000 m.

However, when comparing the temperature profiles of the six cases, variations in the main and alternate mission ranges have negligible effects on thermal performance, except when specific ranges impose altitude limitations. In such cases, the trends observed in the previous altitude comparison apply. These results are consistent with expectations and align with the earlier observation that air behaves as an effectively infinite thermal sink, unlike fuel. Consequently, sustained operation does not significantly alter system temperatures once thermal equilibrium is established, meaning that mission range, unlike altitude, has a limited influence on A-TMS sizing.

Air intake design optimisation (A-TMS)

Based on the preceding analyses, altitude emerges as the dominant factor affecting A-TMS performance, while mission ranges only play a marginal role. These observations enable the definition of a representative worst-case scenario for intake size optimisation in purely air-based cooling.

Given the reduced performance of A-TMS at higher altitudes, the aircraft service ceiling is assumed to be the maximum altitude at which safe system operation must

be ensured. Accordingly, a single worst-case scenario can be adopted for system optimisation:

- **Single Worst-Case Scenario.** This scenario assumes a flight performed at the aircraft service ceiling, with main and alternate segments sufficiently long to avoid altitude capping from mission-planning constraints. For the present study, a 400 km main mission and a 200 km alternate segment are considered, performed at cruising altitudes of 7600 m and 4500 m, respectively.

The minimum intake area required to consistently satisfy the heat-source cooling demand is determined by iteratively simulating the reference scenario and adjusting the inlet size until the maximum heat-source temperature approaches, without exceeding, its operating limit of 120 °C. For the reference use case of a 20% hybrid-electric regional transport aircraft powered by MT-PEM fuel cells, a first intake area estimate of 0.190 m² is obtained for two intakes (total 0.380 m²).

To verify the robustness of this result, an extensive simulation campaign is conducted across the full mission envelope, considering all relevant combinations of the following mission parameters:

- 4500 m, 6000 m, and 7600 m altitudes;
- 130 km and 200 km to 1300 km main mission ranges in 100 km increments;
- 50 km to 350 km alternate flight ranges in 50 km increments.

The simulation campaign confirms that the optimised A-TMS maintains the fuel cell temperature below its upper operating limit of 120 °C across all evaluated scenarios, with the maximum heat source temperature being consistently reached by the end of the first climb. Since this segment shares analogous dynamics across all profiles, three identical and flat response surfaces are obtained across all reference altitudes, which are not reported here due to their limited additional insight.

To provide numerical evidence of these statements, an updated diagram of the three reference altitudes is provided in Fig. 4.12, illustrating the temperature responses at the three reference altitudes, when the intake area is increased from 0.15 m² to 0.190 m².

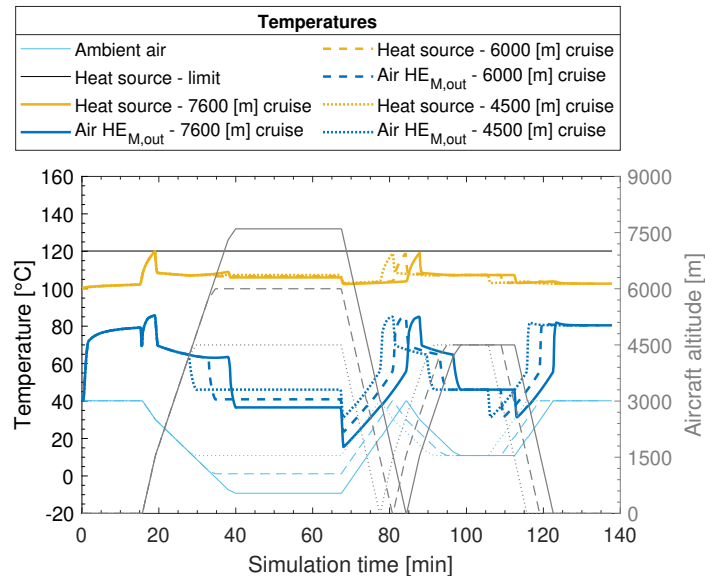


Fig. 4.12 Relevant temperature trends of an A-TMS with two 0.190 m^2 intakes. Cooling results for a 20% hybrid-electric turboprop aircraft performing a 400 km main mission at different altitudes, followed by a 200 km, 4500 m alternate flight.

4.4.2 Integrated air–fuel TMS dynamics

The results from Figs. 4.8 and 4.9 indicate that the A-TMS could be substantially improved by mitigating peak heat loads during early climb phases, motivating the introduction of fuel-based thermal buffering during high-intensity segments. Building on the benchmark system-level performance of the A-TMS, the integrated AF-TMS architecture is now evaluated, comparing the baseline and updated fuel control strategies described in Sect. 4.3.3.

The reference flight scenario from Fig. 3.11 is retained, consisting of a 400 km main mission at 6000 m, followed by a 200 km alternate segment at 4500 m. The 20% target hybridisation level is recalled with the initial intake area assumption of $A_{\text{intake}} = 0.15 \text{ m}^2$ for both intakes. The resulting temperature trends are presented in Figs. 4.13a and 4.13b for the baseline and updated controllers, respectively, while the fuel mass dynamics are omitted, since they remain consistent with the 20% hybridisation case shown in Fig. 3.12.

Compared to the A-TMS benchmark, the integrated AF-TMS maintains adequate cooling throughout the entire mission, even with the original 0.15 m^2 intake, thus

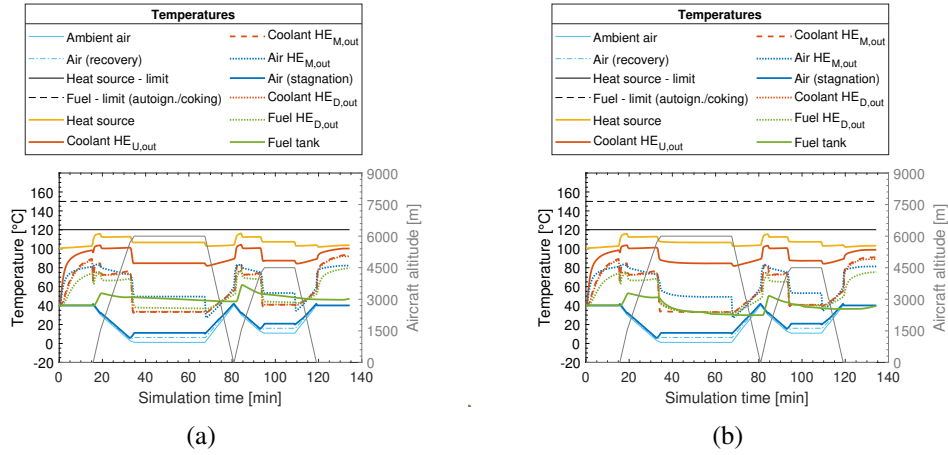


Fig. 4.13 Relevant temperature trends of an AF-TMS with two 0.15 m^2 intakes and a single tank recirculating-series arrangement. Cooling results for a 20% hybrid-electric turboprop aircraft performing a 400 km, 6000 m main mission followed by a 200 km, 4500 m alternate flight. (4.13a) baseline fuel-flow controller. (4.13b) updated fuel-flow controller.

already outperforming the air-only design. The early climbs remain the most thermally demanding phases, corresponding to the two observed temperature peaks in the main and alternate segments.

Both control strategies exhibit similar HS temperature profiles, with a first-peak maximum of 115.8°C . Differences emerge at the second peak, with Figs. 4.13a and 4.13b showing maxima of 116.2°C and 115.8°C , respectively. The improved performance of the second configuration arises from the updated control logic, which enables active recovery of the fuel's thermal capacity during cruise and descent, establishing a larger thermal gradient between the fuel and the HS and ultimately reducing the maximum HS temperature by approximately 0.4°C .

Tank fuel temperatures reflect and amplify this behaviour, remaining consistently cooler under the updated controller, as shown in Fig. 4.13b. This effectively reduces flammability risks and improves safety. In practice, the maximum tank fuel temperature reaches 61.8°C with the baseline controller, reducing to 53.0°C when the updated logic is introduced. Moreover, by increasing the temperature gradient between the coolant and the tank fuel, the system achieves more efficient cooling at lower mass flows, with the baseline controller requiring a maximum of 7.35 kg s^{-1} during peak cooling, against the 6.21 kg s^{-1} of the improved logic.

These results clearly demonstrate the advantages of integrating fuel as a thermal buffer in the A-TMS and actively managing its thermal capacity. Accordingly, the AF-TMS configuration with the updated fuel flow control logic is adopted herein as the reference configuration for the remainder of this dissertation.

A subsequent mission analysis is conducted once again, comparing altitudes, main mission ranges, and alternate flight distances to capture the full dependencies of AF-TMS performance across the expected mission envelope. For these simulations, the reference settings assume a 20% hybridisation degree, a maximum fuel flow rate of 13.5 kg s^{-1} , and two smaller 0.10 m^2 air intakes. The reduced intake areas are intentionally selected to stress the system and clearly highlight critical use-case dependencies, as the AF-TMS has already demonstrated adequate cooling with 0.15 m^2 intakes.

Influence of altitude (AF-TMS)

A first simulation campaign is conducted to compare heat source, tank fuel, and outlet air temperatures for a 400 km mission at 4500 m, 6000 m, and 7600 m, followed by a 200 km, 4500 m alternate. Fig. 4.14 reports the relevant simulation dynamics.

The results highlight higher-altitude flights as the more demanding scenario, as evidenced by the peak heat source temperatures of the first climb, which progressively increase with the target cruising altitude, peaking at 120.6°C , 127.3°C , and 134.8°C respectively for the 4500 m, 6000 m, and 7600 m cases. These findings contrast with earlier analyses on pure F-TMS from Sect. 3.4.2, but remain consistent with previous observations from the A-TMS. This result can be reasonably expected, since air-cooling remains the primary cooling method, while actively cooling the fuel renders passive heat rejection through prolonged descent a less influential factor in determining the system's thermal endurance. This is also reflected in the cruise phase, where higher-altitude flights generally result in increased HS temperatures.

The temperature profiles of the tank fuel follow similar trends, with the high-altitude flight actually struggling to sustain active fuel thermal capacity recovery, as indicated by the almost stable tank-fuel temperature compared with lower-altitude cases. Nonetheless, the 7600 m flight still manages a substantial decrease in the stored-fuel temperature during the descent phase, reaching temperatures comparable

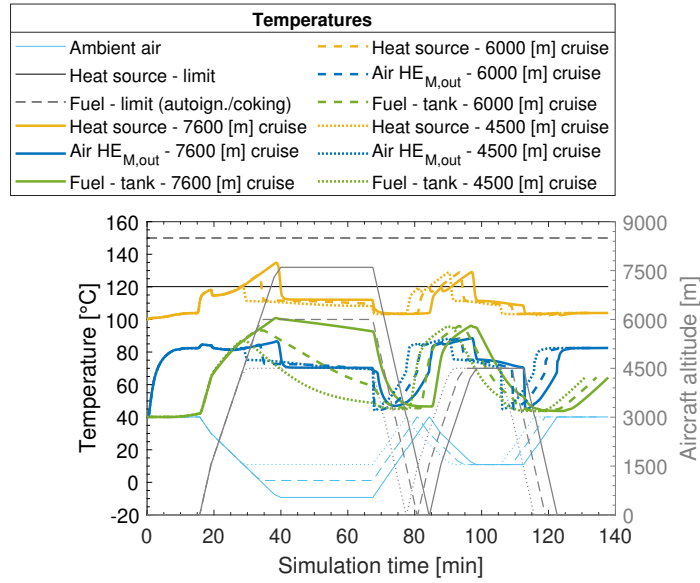


Fig. 4.14 Relevant temperature trends of an AF-TMS with two 0.10 m^2 intakes and a single tank recirculating-series arrangement. Cooling results for a 20% hybrid-electric turboprop aircraft performing a 400 km main mission at different altitudes, followed by a 200 km, 4500 m alternate flight.

to those of the other cases by the missed-approach phase, effectively recovering most of the fuel thermal capacity before the alternate climb segment.

As a result, comparable trends can be observed in the alternate segment for all temperature profiles, demonstrating that previous thermal histories play a negligible role, consequent to the air's ideal thermostat behaviour and the recovery of the fuel's thermal capacity during the preceding cruise and descent phases.

Influence of main mission range (AF-TMS)

The influence of the main mission range is addressed next by comparing heat source, tank fuel, and outlet air temperatures for 130 km, 400 km, and 1300 km missions, conducted at 6000 m, and followed by a 200 km, 4500 m alternate. The results of this analysis are reported in Fig. 4.15.

The simulation results reveal distinct temperature trends among the different mission ranges, although the peak temperatures are consistently reached at the end of the second climb. However, identifying the absolute maximum temperature amongst

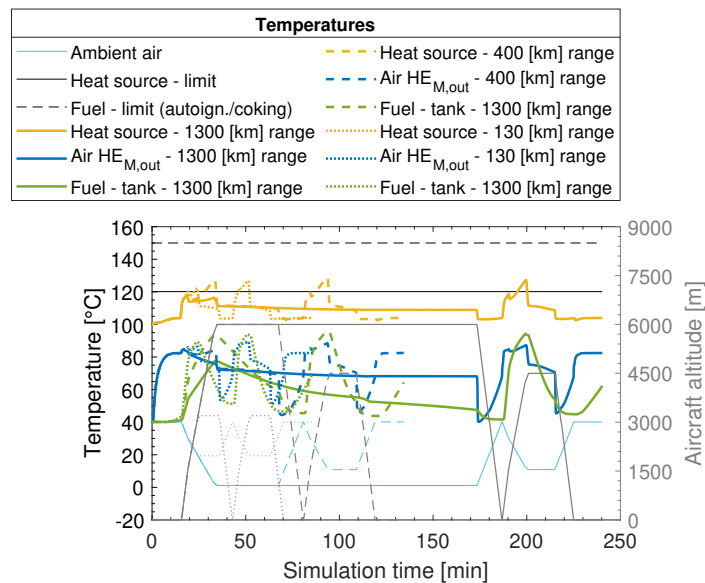


Fig. 4.15 Relevant temperature trends of an AF-TMS with two 0.10 m^2 intakes and a single tank recirculating-series arrangement. Cooling results for a 20% hybrid-electric turboprop aircraft performing different missions at a main cruise altitude of 6000 m, followed by a 200 km, 4500 m alternate flight.

these three cases is not as straightforward and requires a more in-depth comparison. Nonetheless, some insights can instead be deduced from the first climb peaks, where the intermediate 400 km profile clearly emerges as the more demanding use case, peaking at 127.3°C , against the 122.2°C of the 130 km case and the 117.3°C of the 1300 km mission. Indeed, comparable dependencies can be observed from a detailed analysis of the second climb peaks, reported in Tab. 4.2.

Table 4.2 Maximum heat source temperatures simulated during main flights of 130 km, 400 km, or 1300 km at an altitude of 6000 m, followed by a 200 km, 4500 m alternate flight.

	Mission range		
	130 km	400 km	1300 km
First peak	122.2°C	127.3°C	117.3°C
Second peak	126.8°C	128.9°C	127.2°C

Comparable trends are also observed in the tank fuel temperatures, which, excluding the alternate segments of the mission, remain consistently higher for the 400 km profile.

These findings are verified in Fig. 4.16, which compares the maximum simulated HS temperatures of the AF-TMS for 13 different ranges, for a reference cruising altitude of 6000 m and a subsequent 200 km, 4500 m alternate.

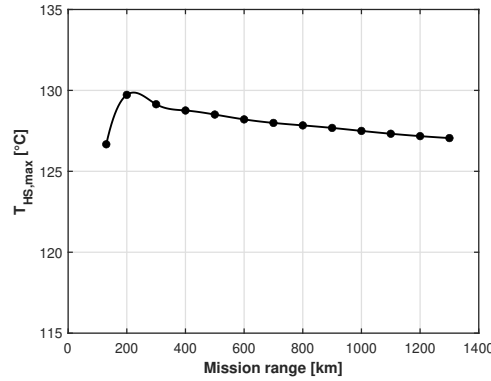


Fig. 4.16 Heat source temperature trends of an AF-TMS with two 0.10 m^2 intakes and a single tank recirculating-series arrangement. Cooling results for a 20% hybrid-electric turboprop aircraft performing diverse missions at a main cruise altitude of 6000 m, followed by a 200 km, 4500 m alternate flight.

The results of this more in-depth analysis identify the most challenging scenarios in the 200 km to 300 km range. This outcome is attributed to two main factors: a sustained high load application rate, combined with a reduced initial thermal inertia. Longer flights are less prone to early overheating owing to a higher initial fuel mass, which slows fuel heating and maintains larger thermal gradients with the heat source, resulting in more effective cooling. The effects of this higher thermal inertia are also evident when comparing cruise fuel cooling for the 400 km and 1300 km cases, with the latter experiencing a much slower cooling owing to the higher stored mass.

Short-range flights, on the other hand, are hindered by a limited fuel thermal inertia. However, as the mission profile becomes shorter, the effective cruise altitude is capped at lower values, according to the mission-planning hypotheses of Sect. 3.3.4. This capping shortens the overall climb phase duration, effectively reducing the net thermal stress applied to the systems during the peak heat generation segment of the mission. Moreover, this shorter heat generation burst is more effectively diluted by the train of thermal inertias connecting the heat source to the sink, effectively dampening the critical effects of the early climb segment.

Accordingly, the most challenging profile can be identified in the shortest mission profile that is not subject to altitude capping, as it combines maximum sustained heat generation with minimum thermal inertia for fuel thermal buffering. Given the assumptions of this work, this critical main flight range lies within 200 km to 300 km, depending on the reference altitude of choice (4500 m to 7600 m), and ensures a minimum cruise duration of 10 min.

Influence of alternate range (AF-TMS)

The alternate flight range influence is assessed last, comparing heat source, tank fuel, and outlet air temperatures for a 400 km, 6000 m mission, followed by 50 km, 200 km, and 350 km alternate flights at a nominal altitude of 4500 m. The results of this analysis are reported in Fig. 4.17.

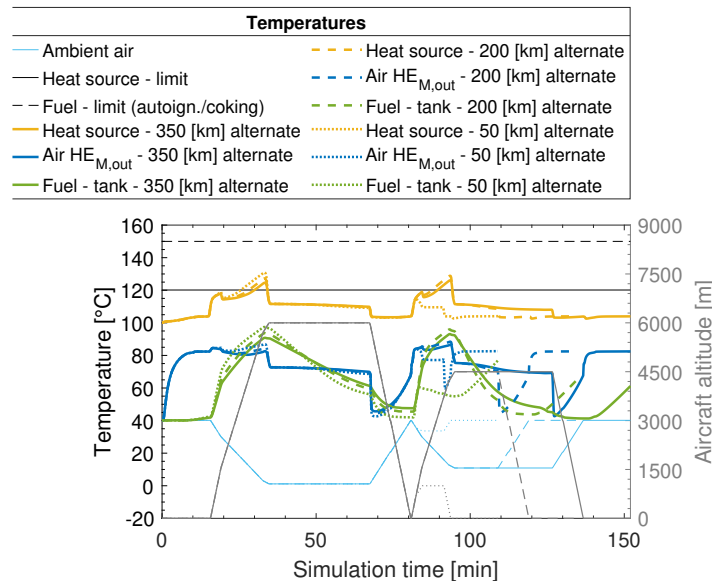


Fig. 4.17 Relevant temperature trends of an AF-TMS with two 0.10 m^2 intakes and a single tank recirculating-series arrangement. Cooling results for a 20% hybrid-electric turboprop aircraft performing a 400 km, 6000 m main cruise followed by different alternate flights.

The results delineate two opposite behaviours when comparing the first and second climb segments. Compared to the previous case, the short alternate profile clearly emerges as the most challenging scenario during the first climb segment. This is expected, as the reduced fuel mass for the brief alternate limits fuel thermal inertia,

resulting in less effective cooling during early mission segments. The beneficial reduction of altitude does not occur for the first climb, since all cases share the same 400 km main flight, so that only thermal inertia affects this phase. The associated temperature peaks are 130.9 °C, 127.3 °C, and 124.7 °C for the 50 km, 200 km, and 350 km cases, respectively.

However, when addressing the second climb, previous observations concerning altitude saturation apply once again, with the extremely short 50 km alternate being capped at the minimum altitude of 1000 m, prescribed by the third mission-planning hypothesis of Sect. 3.3.4. This results in a very brief climb phase, with moderate total heat generation compared to the longer 200 km and 350 km alternate segments, which instead rise up to 4500 m. Between these two, the 200 km alternate is the more challenging one, owing to the reduced fuel mass and thermal inertia, in line with previous considerations on the main range influence.

Ultimately, the most challenging alternate scenario varies depending on whether the first or second climb is considered. Accordingly, a single worst-case scenario cannot be defined a priori. Depending on the considered case and flight envelope, it may correspond to the shortest alternate range, or to the minimum alternate profile that does not enforce altitude capping, as in the primary-mission-range case.

Integrated air–fuel TMS design optimisation

Having identified the key correlations between mission parameters and AF-TMS performance, a selection of worst-case scenarios can be defined and considered for the optimisation of this new, integrated air–fuel framework. The analyses identified more challenging operating conditions associated with high altitude, low-to-medium range flights, and short-to-moderate alternate segments. Given these observations, the following worst-case scenarios can be defined:

- **Primary Worst-Case Scenario (AF-TMS).** This scenario considers the shortest main flight that does not enforce altitude capping, conducted at the service ceiling altitude, and followed by the shortest alternate flight within the considered envelope. Given the assumptions of the present analysis, this corresponds to a 300 km mission conducted at 7600 m, followed by a 50 km alternate segment at 1000 m.

- **Secondary Worst-Case Scenario (AF-TMS).** This case combines the shortest main and alternate flights that do not enforce altitude capping, each at its respective maximum altitude. For the present study, these correspond to a 300 km mission at 7600 m, followed by a 200 km alternate segment at 4500 m.

Based on these reference scenarios, the air–fuel TMS is optimised by combining the methods described in Sects. 3.4.3 and 4.4.1. The baseline 20% aircraft hybridisation is retained for simulating the primary worst-case scenario and refining the air-side intake area until the maximum recorded heat source temperature approaches, without exceeding, its upper operating limit of 120 °C. Subsequently, the optimised intakes are verified against the secondary worst-case scenario, with the inlet area increased if the thermal constraints are violated.

A brief optimisation campaign confirms the Primary Worst-Case Scenario as the more demanding use case for the proposed AF-TMS configuration and selected application. Iterations identify a minimum intake area of $A_{\text{intake}} = 0.128 \text{ m}^2$ as the smallest viable area for supporting system cooling via two intakes (total area 0.256 m^2). To corroborate these results, one final simulation campaign is conducted, evaluating the optimised AF-TMS across the entire mission envelope for:

- 4500 m, 6000 m, and 7600 m altitudes;
- 130 km and 200 km to 1300 km main mission ranges in 100 km increments;
- 50 km to 350 km alternate flight ranges in 50 km increments.

The response surfaces of the maximum simulated heat source temperatures are presented in Fig. 4.18.

The results of Fig. 4.18 corroborate the optimisation outcomes, verifying that the proposed AF-TMS framework consistently supports fuel cell cooling across the entire mission envelope for each of the three flight altitudes. The response surface confirms previous observations regarding altitude and range dependencies, with the more demanding cases concentrating around the 300 km mission mark, particularly when performed at the aircraft service ceiling.

Between the two more demanding alternate cases, the shortest 50 km segment is confirmed as the most challenging. However, when lower main cruising altitudes are considered, 100 km alternate segments are generally met with greater challenges.

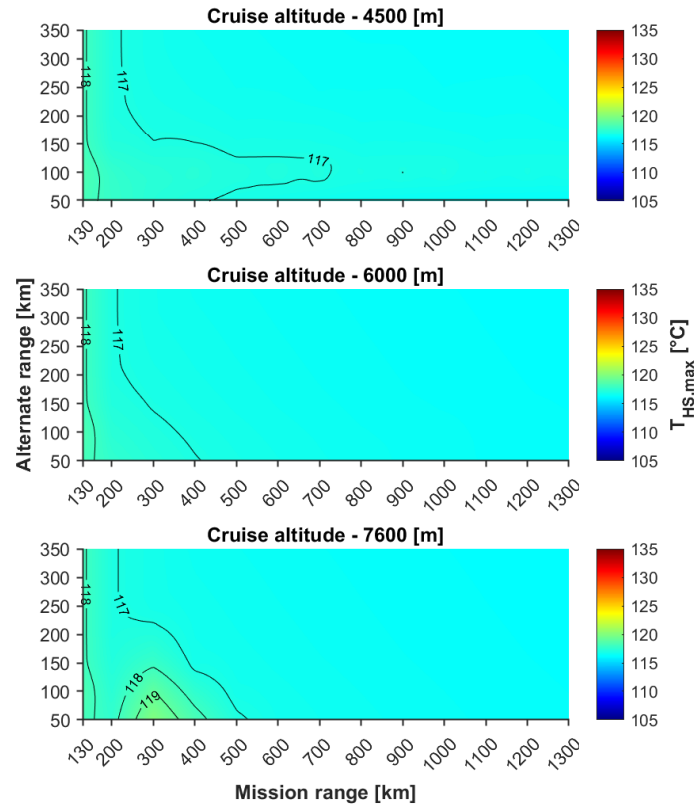


Fig. 4.18 Maximum heat source temperatures for an integrated AF-TMS with two 0.128 m^2 intakes and a single tank recirculating-series arrangement in a 20% hybrid-electric aircraft configuration. Response surfaces of main and alternate mission ranges. The three subfigures correspond to the different main cruise altitudes.

Finally, observations from Sect. 3.4.3 concerning altitude and alternate range saturation in the left-hand region of the diagrams are retained. Specifically, the values for the 130 km main mission remain constant across all three reference altitudes, as the mission altitude is capped at the same effective value to compensate for the brief cruise duration associated with the short range. Similarly, the temperatures associated with alternate flights exceeding 130 km are all identical when the assumed main flight range is 130 km, as the effective alternate range is capped to the minimum value between the prescribed alternate range and the actual main flight range. These same considerations apply, less strictly, to the 200 km and 300 km cases.

Ultimately, this analysis demonstrates that the proposed integrated AF-TMS is capable of providing adequate cooling for a fuel cell supporting a 20% propulsion

electrification in a regional transport aircraft. Compared to a baseline A-TMS, this new framework demands smaller airflows, effectively reducing the required intake area by approximately 32%. Furthermore, unlike in the F-TMS, active thermal capacity recovery ensures that worst-case scenarios are observed at relatively low ranges and exceptionally high altitudes. These conditions are unlikely to be encountered during typical flight operations, providing a safety margin to the optimised design. Moreover, the response surfaces of Fig. 4.18 suggest that extended alternate and main flights could realistically be considered without affecting AF-TMS performance, provided that sufficient energy reserves are made available. These promising results motivate further research into the development and integration of thermal buffering and fuel-based cooling strategies for high-heat-load applications.

4.4.3 Optimised A-TMS and AF-TMS comparison

Given the optimised configurations for the sole-air and integrated air–fuel TMSs, it is now possible to compare and assess the mass and power benefits associated with integrating fuel-based thermal management with conventional ram-air cooling, recalling the preliminary mass assessments of Sect. 3.4.3.

Eq. (3.44) is recalled for computing the air and fuel HE masses, retaining the previously adopted specific heat transfer rates of $\sim 1 \text{ kW kg}^{-1}$ [280, 281] for the air–liquid heat exchanger and $\sim 5 \text{ kW kg}^{-1}$ [282] for the fuel–coolant unit. For the optimised A-TMS configuration, a maximum heat transfer rate of $\dot{Q}_{\text{HEM},\text{max}} = 938 \text{ kW}$ is obtained at the air heat exchanger. Contrarily, peak cooling rates of $\dot{Q}_{\text{HEM},\text{max}} = 675 \text{ kW}$ and $\dot{Q}_{\text{HEU},\text{max}} = 365 \text{ kW}$ are obtained at the air and fuel heat exchangers of the AF-TMS. Given these values, all HEs for both configurations can be preliminarily sized as reported in Tab. 4.3.

Eq. (3.45) is employed once again to estimate fuel pumping power and the associated pump masses, given a gravimetric power density of approximately 1.5 kW kg^{-1} [283, 284]. These values remain consistent with those of the original F-TMS, as the two-pump arrangement delivering 500 L min^{-1} of fuel at a maximum pressure rise of 5 bar and an efficiency of 0.65 is retained, along with the redundancy assumptions. Accordingly, a combined fuel pipe mass of 10 kg is also retained.

Additionally, new parasitic power and mass terms need to be added to account for the effects of the puller fan in the air-cooling design. Eq. (3.45) can be adapted

into Eq. (4.19) to derive the fan pumping power:

$$\dot{W}_{\text{fan}} = \frac{u_{\text{fan}} (A_{\text{intake}} \eta_{\text{intake}}) \Delta p_{\text{fan}}}{\eta_{\text{fan}}} \quad (4.19)$$

where $u_{\text{fan}} = 5 \text{ m s}^{-1}$ is the air velocity provided by the fan and A_{intake} is the optimised intake area, assuming $\eta_{\text{intake}} \simeq 1$ for low speeds. A fan efficiency of $\eta_{\text{fan}} = 0.65$ is assumed, while a pressure drop of $\Delta p_{\text{fan}} = 500 \text{ Pa}$ is considered, since fan systems rarely exceed these values.

The fan mass can be approximated following the empirical correlation of Ref. [24], here adjusted into Eq. (4.20) for use with the International System of Units (SI):

$$m_{\text{fan}} = 0.4386 (\rho_{\text{air}} A_{\text{intake}}) + \frac{0.1104}{2.205} \quad (4.20)$$

where $\rho_{\text{air}} = 1.2 \text{ kg m}^{-3}$, assuming the fan is operating on ground at sea level.

The combined results of these observations on the cooling systems' power consumption and masses are collected in Tab. 4.3, for both the A-TMS and the AF-TMS.

The results of this preliminary analysis demonstrate that, by incorporating F-TMS practices with conventional ram-air cooling, system-level masses can be reduced by up to 17.4%, bringing the total system mass down from approximately 940 kg to 776 kg.

The reduced dependency on air cooling also enables the downsizing of the puller fan, reducing the associated mass and power requirements by approximately 31%, although the savings are marginal in absolute terms. Conversely, this integration is offset by the introduction of additional parasitic power consumption by the fuel pumps, amounting to a maximum of 12.8 kW. However, while these terms are significantly larger than those for the fan, the total power consumption does not exceed 1.3% of the total fuel cell power output. Moreover, similar power consumptions are only achieved when the pumps operate at full regime, delivering the maximum fuel mass flow rate, which rarely occurs during a mission and is generally limited to periods of peak cooling demand, such as climbs.

Table 4.3 System mass and maximum power comparison between A-TMS and AF-TMS architectures.

Component	A-TMS		AF-TMS	
	Power/Heat rate (max) kW	Mass kg	Power/Heat rate (max) kW	Mass kg
HE _M (1 kW kg ⁻¹)	938	938	675	675
Air fan	1.46	1.05	0.98	0.72
Air fan (redundancy)		2.10		1.45
HE _D (5 kW kg ⁻¹)	0	0	365	73
Fuel pump (1.5 kW kg ⁻¹)	0	0	12.8	8.5
Fuel pump (redundancy)		0		17.1
Fuel pipes		0		10.0
Total	1.46	940.1	13.8	776.5
Net change			12.3	-163.5
Relative change			+845%	-17.4%

Finally, integrating air and fuel cooling into a combined AF-TMS framework also enables the reduction of the total intake area by 32.6%, moving from a two-intake solution with 0.190 m² sections to a smaller 0.128 m² per inlet. Alternatively, a single 0.256 m² intake could also be considered for the optimised AF-TMS. Accordingly, a cooling system drag reduction of approximately 30% can realistically be anticipated with the presented integrated cooling framework.

Eq. (4.21) allows a preliminary quantification of the induced drag of an intake:

$$F_{\text{drag}} = C_D \frac{1}{2} \rho_{\text{air}} (u_{\text{air}})^2 A_{\text{intake}} \quad (4.21)$$

where ρ_{air} is the free-stream air density and C_D is the intake drag coefficient, consistent with standard aerospace and aerodynamics literature. The ESDU 86002 standard indicates $0.6 < C_D < 2+$ for scoop intakes [288]. Considering a typical regional transport aircraft cruising at 140 m s⁻¹ at an altitude of 6000 m, and an air

density of 0.66 kg m^{-3} under ISA conditions, a preliminary estimate of the drag savings can be drafted. Assuming well-designed scoops, with $C_D \simeq 1$, a total drag force of 2.46 kN can be estimated for the A-TMS, reducing to 1.66 kN when the integrated AF-TMS framework is employed.

Eq. (4.22) returns the approximate power penalty associated with the intake-induced drag:

$$\dot{W}_{\text{drag}} = F_{\text{drag}} u_{\text{air}} \quad (4.22)$$

resulting in approximately 344.1 kW for the A-TMS solution and 231.8 kW for the AF-TMS, corresponding to a net propulsive power saving of over 100 kW by employing this new integrated design. However, it should be noted that this preliminary estimate does not consider pressure recovery effects or drag recovery via the Meredith effect, which are expected to reduce these benefits in terms of net propulsive power savings.

4.5 Summary and highlights

Integrating fuel-based thermal management with conventional ram-air cooling emerges as a promising approach to combine the benefits of both architectures while mitigating their respective limitations. In this chapter, an integrated modelling framework is proposed by extending the baseline fuel-based TMS model of Sect. 3.3 with a preliminary ram-air cooling representation. A simplified scoop-intake model governs the incoming airflow and air-system performance, given an inlet area and estimated capture ratio. The framework is generalised to represent three reference configurations (A-TMS, F-TMS, and integrated AF-TMS) by selectively activating the two heat sinks at their interfaces with the coolant loop: HE_M and HE_D .

The study builds upon existing work on dynamic modelling of F-TMS by integrating fuel- and air-based cooling strategies in a single architecture and investigating the use of fuel as a secondary heat sink and thermal buffer. In particular, the model explores the active recovery of fuel thermal capacity during low-demand phases through a preliminary control logic featuring two operating modes: a fuel-based cooling mode and a fuel thermal capacity recovery mode. In the first mode, fuel acts as a heat sink absorbing heat during high-load segments, whereas in the second, it

behaves as a heat source, returning stored thermal energy to the coolant to reduce the tank fuel temperature.

Contemporarily, the analysis also extends existing literature on ram-air cooling by introducing auxiliary cooling through fuel thermal inertia, thus reducing the thermal burden on the air system. In this context, fuel buffering is investigated as a means of smoothing the most critical mission segments, particularly during early climb and take-off, allowing the downsizing of the A-TMS while maintaining equivalent cruise-phase cooling capability.

The A-TMS demonstrates substantially higher cooling capability than the purely fuel-based concept of Chap. 3, supporting the 20% hybridisation target of this dissertation, provided that the air intakes are adequately sized. Unlike in F-TMS, the air system is not constrained by finite thermal capacity and therefore remains largely insensitive to variations in main mission or alternate range. Instead, performance is primarily affected by altitude, as reduced air density decreases captured mass flow rates, offsetting the benefits of the larger thermal gradient between the heat source and ambient air. Early climb and take-off consistently emerge as the most demanding phases across all simulated missions, owing to a combination of high heat generation and modest airflows at reduced flight velocities, ultimately driving intake sizing in purely air-based architectures. Preliminary optimisations of a two-intake configuration indicate a minimum inlet area of 0.190 m^2 for the reference case of this dissertation.

When air-based and fuel-based thermal management are combined, overall cooling capability is improved, allowing for smaller air intakes. Moreover, introducing a two-mode fuel control strategy, in which the fuel alternately absorbs heat from the coolant and releases it back during low-demand phases, further enhances system safety and performance. Compared with a baseline strategy in which the fuel only absorbs heat, the updated controller achieves 0.4°C lower heat-source temperatures, an 8.8°C reduction in maximum tank fuel temperature, and a 1.14 kg s^{-1} decrease in required fuel flows.

The interaction between mission parameters and AF-TMS performance reflects the combined characteristics of the air and fuel subsystems. Since ram air remains the primary heat sink during cruise and extended flight phases, high-altitude missions continue to represent the most demanding operating conditions. Range dependencies, however, differ from those observed in the original F-TMS analysis due to the

presence of active cooling. The most critical main mission corresponds to the shortest flight likely to be conducted at the aircraft service ceiling, while the most demanding alternate scenario occurs for the shortest alternate segment within the expected mission envelope. Accordingly, a worst-case scenario is defined as a 300 km mission conducted at a service ceiling of 7600 m, followed by a short 50 km alternate segment at 1000 m. Optimisation of a two-intake configuration with a series F-TMS architecture and two-mode control logic yields a minimum inlet area of 0.128 m^2 for the reference 20% hybrid-electric regional aircraft.

A preliminary comparison between A-TMS and AF-TMS architectures highlights several trade-offs associated with the integration of fuel thermal buffering within ram-air cooling systems:

- A 17.4% reduction in total TMS mass, decreasing from 940 kg to 776 kg, including heat exchangers, pumps, fans, and preliminary pipe estimates;
- A net increase of 12.8 kW in maximum fuel-pumping power;
- A reduction of over 30% in the required intake area, from 0.190 m^2 to 0.128 m^2 for two intakes. Early drag estimates, neglecting pressure recovery and Meredith effects, indicate a potential reduction in the propulsive power associated with intake drag of more than 100 kW, decreasing from 344.1 kW for the A-TMS to 231.8 kW for the AF-TMS.

The results of this preliminary analysis highlight the potential benefits of integrating fuel thermal buffering into air-based cooling architectures for aircraft fuel-cell thermal management. Nevertheless, several limitations remain. The current air-side modelling approach is highly simplified and does not yet represent intake aerodynamics or heat-exchanger behaviour in detail. Future work should focus on improving intake modelling fidelity, including pressure losses, drag, and spillage effects. Additional developments should also address more detailed heat-exchanger modelling, enabling explicit dependencies between mass flow rates and thermal effectiveness, which may significantly affect system performance and component masses.

Furthermore, the present controller logic is implemented only at a functional level and approximates pump dynamics and circuit inertias. Future studies should incorporate detailed pump correlations, circuit pressure modelling, and more rigorous control-law formulations, validated against real-world pump dynamics. Improvements to passive heat-rejection modelling at the fuel-tank boundary, as discussed

in Chap. 3, may also be considered, although active cooling reduces their relative importance. Ultimately, comprehensive experimental validation will be required before the AF-TMS concept can be fully assessed.

Beyond methodological refinements, several system-level extensions may also be explored. In particular, air-skin cooling and wing-fuel recirculation concepts [243] could further reduce ram-air cooling requirements, although their mass and energy balances must be carefully assessed. Fuel cooling could also be considered to replace puller-fan systems during ground operations. Finally, the introduction of hydrogen-based cooling concepts, exploiting the large temperature gradients between the fuel-cell stacks and cryogenic hydrogen, may offer additional benefits while simultaneously preheating the fuel prior to use.

In the following chapters, some of these research directions are further developed, focusing on the refinement of heat-exchanger modelling for early component characterisation and design. While the previous chapters addressed system-level thermal management, the subsequent analyses shift the focus towards the detailed modelling of the components themselves.

Chapter 5

Heat exchanger modelling

Results from Chaps. 3 and 4 demonstrate the significant influence of heat exchanger components on the performance and feasibility of thermal management systems, particularly in terms of added system mass. Current models of air-, fuel-, and integrated air–fuel TMS, however, rely on the simplifying assumption of constant heat exchanger thermal effectiveness. While this provides a conservative basis for preliminary sizing of the systems, it fails to capture their full dynamic behaviour.

Accordingly, accurate modelling, design, and optimisation of heat exchangers become strategic enablers for meeting the increasing cooling demands of aerospace systems, particularly in the context of transport electrification and decarbonisation, while significantly enhancing the fidelity of the simulation of the proposed architecture.

Concurrently, recent advances in Additive Manufacturing (AM) introduce significant opportunities for enhancing heat exchanger performance by enabling the realisation of unprecedented, highly optimised geometries that were previously unattainable through conventional manufacturing techniques. Numerous studies [289–291] highlight the potential of AM to address emerging thermal challenges in next-generation transport applications. However, substantial gaps remain in the literature about modelling of AM-based heat exchangers. In particular, discrepancies between the available experimental results and current modelling approaches remain insufficiently resolved, while robust, computationally efficient optimisation frameworks are also required to fully exploit the design freedom offered by additive technologies.

5.1 State of the art

Several modelling approaches are proposed in the literature for heat exchanger design, characterisation, and optimisation. Khaled et al. [292] employ a two-dimensional component discretisation combined with empirical regression of the global heat transfer coefficient U and pressure drops to characterise the thermohydraulic behaviour of an innovative multi-passage HE. Torregrosa et al. [293] similarly determine the UA value of a compact heat exchanger for an electric vehicle dual-purpose air heater-cooler through experimental data fitting. However, Ref. [293] adopts the lumped $\varepsilon - NTU$ formulation for unmixed cross-flow HEs (Incropera [260]), trading detailed modelling of local temperatures for a simplified heat transfer solution. Ruivo et al. [294] follow a similar approach to characterise the thermal performance of a finned-tube HE using calibrated Nusselt number correlations for both fluids. However, unlike Khaled et al. [292], the studies of Torregrosa [293] and Ruivo [294] neglect pressure drop modelling entirely. Conversely, Kim et al. [295] apply the $\varepsilon - NTU$ correlation to characterise a louvered fin compact heat exchanger, addressing the component's heat transfer coefficient and pressure drops via the Colburn j -factor and Fanning f -factor, respectively. Experimental data fitting is then used to determine the dependence of both factors on the Reynolds number, improving accuracy under dry/wet operating conditions.

Khaled et al. [292] report a maximum relative error of 3.76% in the simulated heat transfer, while Ruivo et al. [294] achieve a maximum relative deviation of 0.9% for their best calibrated model. Torregrosa et al. [293] observe a higher maximum relative error of 14%, attributed to limited experimental accuracy at low mass flows, although the average relative error remains low at 2.4%. Kim et al. [295] report larger deviations due to the simplified formulation of the Colburn and Fanning factors, leading to a rms error of $\pm 16.9\%$ and $\pm 13.6\%$ when determining the j -factor and f -factor, respectively. These results indicate that data regression is a valuable tool for accurate HE performance characterisation, consistent with the recommendation of Scott and Joshi [296] to use calibrated models instead of complex Computational Fluid Dynamic (CFD) simulations. However, the strong reliance on regression ties the formulations to the reference HE, limiting their applicability to the investigation of new geometries.

Alternative modelling approaches characterise HE performance through physical laws, allowing the formulations to be decoupled from a specific component and linked

to its design parameters. For instance, the studies by Pujol et al. [297] and Kleiner et al. [298] provide insights into the development of thermal resistance models for cooling channels in electronic heat sinks. Rasouli et al. [299] employ the reference $\varepsilon - NTU$ equation [260] to quantify the heat transfer of an innovative micro-fin compact HE produced via . The global heat transfer coefficient U is determined through a thermal resistance model based on physical equations and the channels' shapes and dimensions. Pressure drops are still addressed through experimental data regression due to the geometrical complexity of the device, which prevents complete decoupling of the modelling approach from the reference configuration. Ref. [299] also includes a FEM structural analysis of the HE and considerations related to AM manufacturing constraints. Da Silva et al. [11] model the thermal performance of an additively manufactured heat exchanger using a combined NTU method and channel thermal resistance framework, extending the physical modelling to the characterisation of HE pressure drops [300].

Rasouli et al. [299] report an average absolute deviation of 3.3% in the simulated component efficiency, with a maximum error of 9.2%. Similarly, da Silva et al. [11] achieve an average heat transfer error of 3.3% with a maximum error close to 10%. These results suggest that the greater flexibility of physics-driven modelling is often accompanied by reduced correlation accuracy. In particular, Ref. [11] reports significant deviations in the modelling of HE pressure drops, with an average relative error of 17.9% and a maximum recorded error of 27%. These findings highlight the challenges of accurately simulating hydraulic losses in AM heat exchangers and are consistent with the trends discussed in the reviews by Kaur and Singh [290] and Careri et al. [291].

Lim et al. [301] demonstrate the use of $\varepsilon - NTU$ formulations and thermal resistance frameworks to support the design optimisation of bare-tube heat exchangers. Arie et al. [302] apply the same concept to the refinement of a microchannel-manifold heat exchanger fabricated via , achieving appreciable performance improvements. However, similarly to the findings of da Silva et al. [11], the study also highlights challenges in accurately modelling pressure drops in AM components, leading to hydraulic loss underestimations of up to 31%.

Alternatively, Nacke, Northcutt, and Mudawar propose an innovative modelling approach for the characterisation of a finned micro-channel test-plate for AM [303], later extending the methodology to the simulation of a complete HE composed of

several stacked modules [304]. The study computes plate heat transfer by applying differential equations to an infinitesimal two-dimensional discretisation of the element, subsequently integrating the differential contributions into a unified sum equation to simplify the numerical solution. The global heat transfer coefficient U is determined through a thermal resistance model of the plate, whereas pressure drops are neglected. An average error of approximately 3% is achieved for the calculated heat transfer, with a maximum absolute deviation of 15% at lower mass flow rates, similarly to the findings of Torregrosa et al. [293]. The work of Zhuang et al. [305] demonstrates the use of globalised plate models and thermal resistance frameworks for the design optimisation of heat exchangers, specifically for an innovative rhombus-fractal heat sink design. The textbook by Nellis and Klein [306] provides detailed descriptions of more traditional discretised plate methods based on surface meshing to capture local HE properties and performance. Although these approaches offer greater modelling flexibility and spatial resolution, they significantly increase the computational cost of simulations, as the behaviour of each mesh node must be evaluated.

Building on these discretised approaches, Petrovic et al. [307] employ full Computational (Thermo-)Fluid Dynamic analyses (CFD) for the simulation of an additively manufactured heat exchanger with complex geometries. The study addresses both heat transfer and pressure drops, achieving satisfactory agreement between predicted and experimental Colburn j -factors, with deviations between 5% and 15%. However, pressure drop predictions are less consistent, underestimating the numerical Fanning f -factors by 11–32%. Hathaway et al. [308] adopt a similar approach, focusing on the integration of self-sustaining geometries for fabrication via AM. However, consistency with experimental data proves challenging, resulting in a 27% maximum heat transfer overestimation and a 300% pressure drop underestimation. The study attributes these discrepancies to powder residues inside the pipes, which significantly affect the experimental measurements. These results further corroborate the challenges highlighted by Kaur [290] and Careri et al. [291] regarding the effective modelling of AM heat exchanger performance, particularly for complex geometries.

CFD analyses offer unmatched modelling versatility but are hindered by significant computational costs, limiting their viability for iterative design optimisation. Consequently, while CFD is frequently used to support HE design refinement, studies often focus on optimising only a limited number of key geometric param-

eters [309, 310], rather than the full component configuration. More extensive strategies embed CFD simulations within topology optimisation loops, but these approaches are typically restricted to small control domains due to prohibitive computational requirements. Feppon et al. [311] provide relevant insights into advances and challenges in HE topology optimisation, while recent studies explore the use of periodic or cellular geometries to reduce the computational cost of these methodologies [312, 313].

Alternatively, Beltrame et al. [284] and Mohapatra et al. [314] propose the use of surrogate models to enable fast design optimisation of heat exchangers. These reduced-order models significantly improve computational efficiency but require extensive training on databases of pre-optimised designs. As a result, their applicability remains limited to the geometries included in the training set. Recent efforts therefore explore the training of Artificial Neural Networks on CFD simulations as a promising strategy for developing more generalisable surrogate models [315, 316].

Ultimately, a variety of modelling strategies are presented for assessing the thermal and hydraulic performance of heat exchangers. The selection of a given method comports a trade-off between simulation accuracy, application flexibility, and computational cost, as summarised in Tab. 5.1 [10]. Nonetheless, despite the already extensive body of work, evolving thermal management targets and emerging opportunities from processes still drive the need for increasingly advanced methodologies for HE simulation, design, and optimisation. In particular, the growing demand for efficient system cooling and the optimisation of existing configurations motivates the development of flexible and robust numerical tools capable of retaining strong coupling between component geometry and performance at a modest computational cost. Moreover, accurately capturing the behaviour of AM-fabricated components remains a significant research challenge, particularly with respect to pressure drop modelling.

5.2 Scope

Within this chapter, a renewed and integrated modelling approach is presented to address identified gaps in the literature and enable the robust and flexible simulation of compact, cross-flow heat exchangers manufactured through AM, featuring either rectangular or circular channels. This class of HE and the associated channel

Table 5.1 Significant modelling approaches for HE thermal characterisation, simulation, and design. Adapted from the author's previous work [10], licensed under CC BY 4.0.

Methodology	Accuracy	Flexibility	Computational efficiency
ϵ -NTU & empirical regression of: • UA [293] • Nusselt num. [294] • Colburn j-factor [295]	• High – avg. deviation: 2.5% • Very high – max. deviation: 0.9% • Low – RMS error: $\pm 16.9\%$	Very low – empirical model calibration	Very high – no discretisation nor iteration (once fitted)
Bidimensional discretisation & empirical regression of U [292]	High – max. deviation: $\simeq 3.7\%$	Low – empirical model calibration	Medium-low – reduced by discretisation
ϵ -NTU formulation & thermal resistance model [299, 11]	Medium – avg. deviation: 3.3%, max. $\simeq 10\%$	Medium – limited to the adopted ϵ -NTU correlation	Medium-high – no discretisation, can involve iterative solving
Global heat transfer & physical thermal resistance models [303, 304]	Medium-low – avg. deviation: 3%, max. $\simeq 15\%$	High – limited to the adopted thermal resistance model	Medium – no discretisation, but includes iterative solving
CFD analyses [307, 308]	Medium-low – 5–15% j -factor deviation [307]	Very high – supports arbitrary geometries	Very low – requires fine discretisation and extensive iteration

geometries are selected due to their widespread use in aviation and transport cooling applications, as well as their high flexibility, allowing implementation in both liquid-liquid and liquid-gas configurations. These characteristics make the cross-flow HE well suited for early HE design on both sides of the AF-TMS architecture presented in this dissertation.

The proposed methodology integrates established thermal and hydraulic correlations to enable robust and computationally efficient simulations of cross-flow heat exchangers. It extends the flexible framework developed by Nacke et al. [303, 304] and incorporates its with the hydraulic modelling approach adopted by da Silva et al. [11]. The resulting integrated framework is then validated against relevant experimental data from the literature. Additionally, the study investigates the dependencies between AM manufacturing processes, fabrication defects, and the resulting pressure losses, proposing two corrective formulations that reduce numerical deviations from published experimental data.

Key characteristics of the proposed method include: a strong alignment between the governing equations and the underlying HE physics and geometry, a high degree of model adaptability to different component shapes and dimensions, and a modest computational cost for the simulations. Particular attention is also devoted to the numerical implementation, with the aim of enhancing numerical robustness, enabling straightforward upgrades of the computational tool, and facilitating its future integration into a design optimisation framework, which will be addressed in the following chapter of this dissertation (Chap. 6).

Ultimately, the simulation framework presented in this analysis aims to provide a comprehensive characterisation of heat exchanger performance based on its primary geometrical features, serving as an early component design tool.

5.3 Methodology

An integrated simulation methodology is developed based on established correlations for the thermal, hydraulic, and gravimetric performance of heat exchangers. Each of these three physical domains is first independently formulated and then implemented within a unified Python-based numerical framework, as detailed in the following of this section.

5.3.1 Thermal domain

The thermal domain of the HE is evaluated first to determine the heat transfer rate between the two working fluids and predict their outlet temperatures as a function of the inlet operating conditions. For the reference configuration of compact heat exchangers operating in a cross-flow arrangement, the device can be virtually decomposed into multiple layers of orthogonal channels perpendicularly stacked to the two fluid streams.

Assuming:

- **Steady-state operation.**
- **Uniform flow distribution** across all plates and channels within each plate.
- **Uniform global heat transfer coefficient (U)**, representing the average value of the local heat transfer coefficients across the plate.
- **Negligible heat losses** to the external environment.
- **Negligible axial conduction**, implying that heat transfer occurs only perpendicularly to both fluid flows.

Each pair of mutually orthogonal channel layers can be treated as a single thermal unit, periodically repeated along the stacking direction of the heat exchanger [303, 304, 10]. Each of these layer pairs is hereafter referred to as a *plate* for the remainder of this dissertation, representing the smallest thermally self-contained modelling unit.

Assuming negligible temperature gradients along the stacking direction, the description of each plate can be then simplified from a three-dimensional representation to a bi-dimensional one, as illustrated in Fig. 5.1.

A two-dimensional Cartesian system of coordinates is applied to the reference plate element and aligned with the flow directions of the two streams. Heat transfer occurs perpendicularly to this plane and can be evaluated by differential energy balances to a generic infinitesimal element of the plate. Specifically, Equation (5.1) models the heat transfer across the differential element, Eq. (5.2) details the corresponding heat balance from the perspective of the hot fluid, and Eq. (5.3)

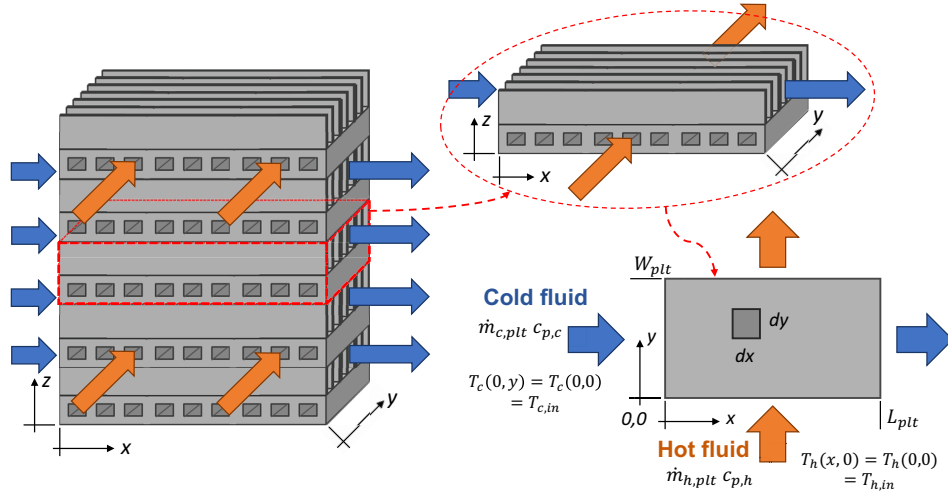


Fig. 5.1 Representation of the whole heat exchanger component and detail of a single plate module with the associated two-dimensional system of coordinates. Reproduced from the author's previous work [10], licensed under CC BY 4.0.

provides the analogous formulation for the cold fluid, governing their respective temperature variations.

$$d\dot{Q} = U (T_h(x, y) - T_c(x, y)) dx dy \quad (5.1)$$

$$d\dot{Q} = -\frac{\dot{m}_{h,plt}}{L_{plt}} c_{p,h} \frac{\partial (T_h(x, y))}{\partial y} dx dy \quad (5.2)$$

$$d\dot{Q} = +\frac{\dot{m}_{c,plt}}{W_{plt}} c_{p,c} \frac{\partial (T_c(x, y))}{\partial x} dx dy \quad (5.3)$$

By combining, differentiating, and further manipulating these three governing equations, a dimensionless temperature difference ratio θ can be defined, as originally proposed by Mason [317]. This ratio, reported in Eq. (5.4), represents the local temperature difference between the hot and cold fluids normalized by the inlet temperature difference.

$$\theta(x', y') = \frac{T_h(x, y) - T_c(x, y)}{T_h(x, 0) - T_c(0, y)} = e^{-(ay' + bx')} \cdot \sum_{n=0}^{\infty} \frac{(abx'y')^n}{(n!)^2} \quad (5.4)$$

where $T_h(x, 0) = T_h(0, 0)$ and $T_c(0, y) = T_c(0, 0)$ are the inlet temperatures of the hot and cold streams, and:

$$a = \frac{U W_{\text{plt}} L_{\text{plt}}}{\dot{m}_{h,\text{plt}} c_{p,h}}, \quad b = \frac{U W_{\text{plt}} L_{\text{plt}}}{\dot{m}_{c,\text{plt}} c_{p,c}} \quad (5.5)$$

with:

$$x' = \frac{x}{L_{\text{plt}}}, \quad y' = \frac{y}{W_{\text{plt}}} \quad (5.6)$$

representing the normalized axes of the plate element.

The heat transfer rate of a single plate is obtained by integrating Eq. (5.1) over the entire plate surface, leading to Eq. (5.7):

$$\dot{Q}_{\text{plt}} = U \int_{y=0}^{W_{\text{plt}}} \int_{x=0}^{L_{\text{plt}}} (T_h(x, y) - T_c(x, y)) \, dx dy \quad (5.7)$$

This formulation rewritten into Eq. (5.8), by introducing the dimensionless spatial coordinates of Eq. (5.6) and normalizing the local temperature difference by the inlet value:

$$\dot{Q}_{\text{plt}} = U W_{\text{plt}} L_{\text{plt}} [T_h(0, 0) - T_c(0, 0)] \int_0^1 \int_0^1 \frac{T_h - T_c}{T_h(0, 0) - T_c(0, 0)} \, dx' dy' \quad (5.8)$$

Combining Eqs. (5.4) and (5.8), the dimensionless parameter ϕ , representing the ratio between the mean temperature difference across the plate and the inlet temperature difference, can be defined according to Eq. (5.9), as proposed by Nacke et al. [303]:

$$\phi = \frac{\dot{Q}_{\text{plt}}}{U W_{\text{plt}} L_{\text{plt}} [T_h(0, 0) - T_c(0, 0)]} = \int_0^1 \int_0^1 \theta(x', y') \, dx' dy' \quad (5.9)$$

By substituting the analytical expression of θ from Eq. (5.4) into Eq. (5.9), the dimensionless parameter ϕ can be expressed in the series form of Eq. (5.10), following the developments reported in Ref. [303]. This formulation reduces the discretised plate model to a single analytical expression, eliminating the need for a spatial mesh while retaining the dependence of the thermal response on the governing

parameters.

$$\phi = \frac{1}{ab} \sum_{n=0}^{\infty} \left[1 - e^{-a} \sum_{k=0}^n \frac{a^k}{k!} \right] \left[1 - e^{-b} \sum_{k=0}^n \frac{b^k}{k!} \right] \quad (5.10)$$

As a result, the computational cost of the model is significantly reduced, enabling efficient simulations while maintaining a consistent representation of the coupling between plate geometry, flow conditions, and heat transfer performance.

The resulting area-averaged dimensionless temperature difference ϕ can then be used to compute the total heat transfer rate of the plate by rearranging Eq. (5.9) into Eq. (5.11):

$$\dot{Q}_{\text{plt}} = U W_{\text{plt}} L_{\text{plt}} [T_h(0,0) - T_c(0,0)] \phi \quad (5.11)$$

Similarly, by combining Eq. (5.9) with the differential energy balances of the hot and cold streams (Eqs. (5.2) and (5.3)), the average outlet temperatures of the hot and cold fluids can be expressed as Eqs. (5.12) and (5.13):

$$\bar{T}_{\text{plt,h,out}} = T_h(0,0) - a [T_h(0,0) - T_c(0,0)] \phi \quad (5.12)$$

$$\bar{T}_{\text{plt,c,out}} = T_c(0,0) + b [T_h(0,0) - T_c(0,0)] \phi \quad (5.13)$$

Building on the previously discussed symmetries in compact cross-flow heat exchangers, the results obtained for a single plate module can be extended to represent the entire device. The methodology developed in Refs. [303, 304] specifically addresses configurations in which both fluids have an equal number of channel layers. In such cases, each layer of hot fluid channels can be paired with an adjacent layer of cold fluid channels to form a plate, leaving no unpaired layers.

In practical applications, however, the number of layers for one fluid may differ from that of the other. This is exemplified by the compact heat exchanger studied in Ref. [11], which comprises 10 layers of cold fluid channels alternating with 9 layers of orthogonal hot ducts. To account for these scenarios, a corrective approach is introduced, in which any unpaired layers are assumed to behave adiabatically relative to the surrounding plates. Accordingly, the total number of plates contributing to the heat transfer is taken as the number of layers of the fluid with fewer rows, as expressed in Eq. (5.14):

$$N_{\text{plt}} = \min(N_{\text{h, layers}}, N_{\text{c, layers}}) \quad (5.14)$$

The total heat transfer rate of the exchanger is obtained by summing the contributions of all active plates, according to Eq. (5.15):

$$\dot{Q}_{\text{HE}} = \dot{Q}_{\text{plt}} N_{\text{plt}} \quad (5.15)$$

Any unpaired layers are assumed to remain at the inlet temperature of the corresponding fluid, in accordance with the adiabatic layer assumption. Consequently, the average outlet temperature of the fluid is computed as a weighted mean of the plate outlet temperatures and the unpaired layers, as expressed in Eq. (5.16):

$$\bar{T}_{\text{HE,out}} = \frac{\bar{T}_{\text{plt,out}} \cdot N_{\text{plt}} + \bar{T}_{\text{plt,in}} (N_{\text{layers}} - N_{\text{plt}})}{N_{\text{layers}}} \quad \text{Where: } \bar{T}_{\text{in}} = T_{0,0} \quad (5.16)$$

Thermal resistance model of plate

Having established the plate heat transfer formulation, its global heat transfer coefficient, U , remains to be determined. To this end, a thermal resistance network is introduced to evaluate the overall heat transfer conductance UA_{plt} , where A_{plt} denotes the reference plate area defined as the product of the two main plate dimensions, W_{plt} and L_{plt} .

Additionally, by decoupling the ϕ formulation from the thermal resistance network, the two components can be treated independently and selectively adapted to different heat exchanger configurations. For instance, Eq. (5.10) is extended in Ref. [304] to describe both rectangular and annular heat exchanger geometries, whereas the thermal resistance formulation is modified in Ref. [303] to represent either mixed- or separate-flow cross-flow heat exchangers.

The geometric symmetry of the heat exchanger is thus exploited once more to redefine the plate control volume around the channels of a single fluid stream, while preserving the dimensions and thermal balance of the domain. In this representation, the channel walls behave as fins within the thermal resistance framework, with the channels centreline delineating the fin tips, under the following assumptions:

- **Constant and uniform fluid properties**, evaluated as the mean value between the average inlet and outlet conditions.

- **Adiabatic fin tips**, as these coincide with the symmetry planes separating the redefined control volume from the adjacent plate modules, resulting in zero net heat transfer.

Accordingly, three symmetry planes associated with the fin tips can be identified within a single plate module: two corresponding to the boundary planes with the adjacent modules and one located at the plate centreline. This symmetry allows to virtually divide the plate module into two identical elements, so that the thermal network can be represented as a parallel of two analogous thermal resistance branches, from which the global UA_{plt} value is obtained according to Eq. (5.17):

$$UA_{\text{plt}} = \frac{1}{R_{\text{plt}}} = \frac{1}{R_{\text{branch}}} + \frac{1}{R_{\text{branch}}} \quad (5.17)$$

Each branch of the resistance network can be further decomposed into three resistances in series, corresponding to convection from the hot fluid, conduction through the separating wall, and convection to the cold fluid, as expressed in Eq. (5.18):

$$R_{\text{branch}} = R_{\text{conv,h}} + R_{\text{cond,w}} + R_{\text{conv,c}} \quad (5.18)$$

The complete process of redefining the plate control volume and deriving the equivalent thermal resistance network is illustrated in Fig. (5.2).

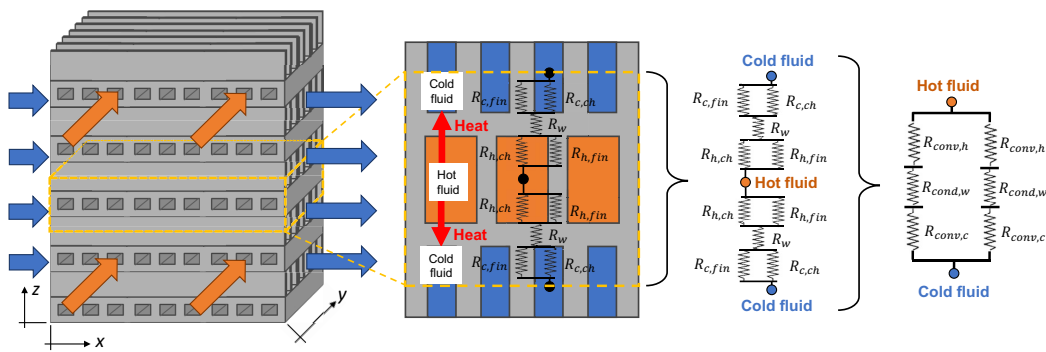


Fig. 5.2 Representation of the plate control volume centred around the channels of one fluid, and extrapolation of the equivalent thermal resistance model. To ease visualization, the channels of both fluids are shown with cross-sections perpendicular to the reader, although the heat exchanger operates in a cross-flow configuration. Reproduced from the author's previous work [10], licensed under CC BY 4.0.

Eq. (5.19) returns the conductive thermal resistance:

$$R_{\text{cond,w}} = \frac{H_w}{k_w W_{\text{plt}} L_{\text{plt}}} \quad (5.19)$$

where H_w represents the wall thickness separating the two fluids and k_w the thermal conductivity of the material composing the heat exchanger.

Following Ref. [303], the convective resistance terms of the fluids can be further decomposed into two separate contributions for a square channel configuration: a base-channel resistance and a sidewall-fin resistance, as illustrated in Fig. 5.2. These two components effectively behave as thermal resistance parallel, which can be rewritten into a combined equivalent resistance term through Eq. (5.20), under the adiabatic fin-tip assumption:

$$R_{\text{conv}} = \frac{1}{N_{\text{fin}} (\eta_a \bar{h}_{\text{fin}} A_{\text{fin}} + \bar{h}_{\text{base}} A_{\text{base}})} \quad (5.20)$$

where N_{fin} denotes the number of fins associated with a given fluid within a plate module, corresponding to the fluid's number of channels per plate. The parameter η_a represents the fin efficiency of the rectangular channel sidewalls, while the $\bar{h}_{\text{fin}}$ and A_{fin} terms denote the average heat transfer coefficient and wetted area of a single fin, respectively. Similarly, $\bar{h}_{\text{base}}$ and A_{base} represent the average heat transfer coefficient and wetted area associated with the base surface of the channel.

Ref. [11] provides a more generalised formulation of the convective thermal resistance. For consistency with the adopted modelling approach, this expression is reformulated to account for half of the channel domain, yielding Eq. (5.21):

$$R_{\text{conv}} = \frac{2}{N_{\text{fin}} \eta_0 \bar{h}_{\text{chan}} A_{\text{chan}}} \quad (5.21)$$

In this formulation, $\bar{h}_{\text{chan}}$ represents the average convective heat transfer coefficient within the channels and A_{chan} the total wetted area of a single duct. The term η_0 expresses a generalised fin efficiency factor accounting for the contribution of the channel sidewalls.

Ref. [11] derives the generalised fin efficiency η_0 according to Eq. (5.22), which has been validated for the modelling of circular channel geometries.

$$\eta_0 = 1 - \frac{A_{\text{fin}}}{A_{\text{fin}} + A_{\text{base}}} (1 - \eta_a) \quad (5.22)$$

In this approach, the fin efficiency η_a is evaluated with respect to an equivalent rectangular geometry, corresponding to the smallest square channel circumscribing the reference circular duct, such that $B_{\text{chan}} = H_{\text{chan}} = D_{\text{chan}}$. The underlying geometrical assumption is illustrated in Fig. 5.3.

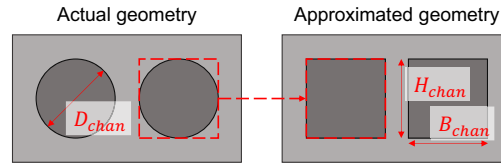


Fig. 5.3 Circular ducts approximation assumed to simplify fin efficiency calculation. Reproduced from the author's previous work [10], licensed under CC BY 4.0.

This approximation simplifies the fin efficiency calculation and enables both rectangular and circular channel geometries to be solved based on the same definition of η_a . Accordingly, a unified fin-efficiency equation is proposed for this analysis, by combining Eqs. (5.20), (5.21), and (5.22) into Eq. (5.23):

$$R_{\text{conv}} = \frac{2}{N_{\text{fin}} \left(\eta_a \bar{h}_{\text{fin}} \frac{A_{\text{fin}}}{A_{\text{fin}} + A_{\text{base}}} + \bar{h}_{\text{base}} \frac{A_{\text{base}}}{A_{\text{fin}} + A_{\text{base}}} \right) A_{\text{chan}}} \quad (5.23)$$

where the wetted area of a single channel, A_{chan} , is computed as $A_{\text{chan}} = B_{\text{chan}} H_{\text{chan}} L_{\text{chan}}$ for rectangular geometries and $A_{\text{chan}} = \frac{\pi}{4} D_{\text{chan}}^2 L_{\text{chan}}$ for circular ducts. The fin and base areas, A_{fin} and A_{base} , instead refer exclusively to the equivalent rectangular geometry of Fig. 5.3 to evaluate a rectangular fin efficiency according to Eq. (5.24).

$$\eta_a = \frac{\tanh \left(m_{\text{fin}} \frac{H_{\text{chan}}}{2} \right)}{m_{\text{fin}} \frac{H_{\text{chan}}}{2}} \quad (5.24)$$

where $\frac{H_{\text{chan}}}{2}$ represents a single fin height, corresponding to half of a channel height under the previously introduced symmetry-based separation of the plate module into two identical elements.

Eq. (5.25) provides the fin efficiency parameter m_{fin} :

$$m_{\text{fin}} = \sqrt{\frac{\bar{h}_{\text{fin}} P_{\text{fin,cs}}}{k_w A_{\text{fin,cs}}}} \quad (5.25)$$

where $P_{\text{fin,cs}}$ and $A_{\text{fin,cs}}$ denote the cross-sectional perimeter and area of a rectangular fin, respectively defined as $P_{\text{fin,cs}} = 2(W_{\text{fin}} + L_{\text{fin}})$ and $A_{\text{fin,cs}} = W_{\text{fin}} L_{\text{fin}}$.

Ultimately, the heat transfer coefficients for the channel base and fins, $\bar{h}_{\text{base}}$ and $\bar{h}_{\text{fin}}$, are determined using well-established empirical correlations from the literature, summarised in Tab. 5.2. These correlations are formulated in terms of the standard dimensionless numbers defined in Eq. (5.26):

Table 5.2 Convective heat transfer coefficients of fluids. Adapted and refined from the author's previous work [10], licensed under CC BY 4.0.

Hypothesis	Regime	Equation
Finned-plate hypothesis For: $\delta_{\text{bl}} < H_{\text{chan}}/2$, $\delta_{\text{bl}} < B_{\text{chan}}/2$ Assumes: $L_{\text{char}} = L_{\text{chan}}$,	Laminar, $Re < Re_{\text{crit}}$	$Nu = 0.664 Re^{1/2} Pr^{1/3}$ [260]
	Turbulent, $Re \geq Re_{\text{crit}}$	$Nu = \left[0.664 Re_{\text{crit}}^{1/2} + 0.037 \left(Re^{4/5} - Re_{\text{crit}}^{4/5} \right) \right] Pr^{1/3}$ [260]
Internal-channel hypothesis For: $\delta_{\text{bl}} \geq H_{\text{chan}}/2$, $\delta_{\text{bl}} \geq B_{\text{chan}}/2$ Assumes: $L_{\text{char}} = D_{\text{hyd}}$, $D_{\text{hyd}} = \frac{4A_{\text{chan}}}{P_{\text{chan}}}$	Laminar, $Re < 1800$	$Nu_{\text{lam}} = [4.364^3 + 0.6^3 + a^3 + b^3]^{1/3}$, $a = 1.953 \sqrt[3]{Pr Re \left(\frac{D_{\text{hyd}}}{L_{\text{chan}}} \right)}$, $b = 0.924 \sqrt[3]{Pr} \sqrt{Re \left(\frac{D_{\text{hyd}}}{L_{\text{chan}}} \right)}$ [318, 319]
	Transitional, $1800 \leq Re < 4000$	$Nu = (Nu_{\text{lam}}^6 + Nu_{\text{turb}}^6)^{1/6}$ [320]
	Turbulent, $Re \geq 4000$	$Nu_{\text{turb}} = \frac{(f/8)(Re - 1000)Pr}{1 + 12.7(f/8)^{1/2}(Pr^{2/3} - 1)} \left(1 + \left(\frac{L_{\text{chan}}}{D_{\text{hyd}}} \right)^{-2/3} \right)$, with $f = \frac{1}{4} (1.8 \log Re - 1.5)^{-2}$ [321, 319]

$$Re = \frac{\rho u L_{\text{char}}}{\mu}, \quad Pr = \frac{c_p \mu}{k}, \quad Nu = \frac{\bar{h} L_{\text{char}}}{k} \quad (5.26)$$

where L_{char} denotes the characteristic length associated with the selected heat transfer correlation, based on the specific flow configuration being considered.

A fundamental distinction exists between correlations developed for open and enclosed flow regimes. Within each regime, the flow may be further classified as either laminar, transitional, or turbulent, based on the current Reynolds number. Entrance-region effects are accounted for in the internal flow correlations. The Gnielinski correlations for developing internal flow are employed for both laminar and turbulent regimes, providing reliable predictions across a wide range of thermal and hydrodynamic entrance conditions.

In most compact heat exchanger configurations, internal flow regimes are expected to dominate, whereas external flow conditions may arise only in specific cases where the channel cross-sections are comparatively large relative to the fin length. Following the methodology adopted by Mudawar et al. [303, 304], internal flow correlations are applied when boundary layer merging occurs, namely, when the maximum boundary layer thickness exceeds half the distance between two opposing channel walls. Conversely, flat-plate correlations are employed when short fins or an interrupted fin pattern prevent boundary layer merging by disrupting its development at an early stage.

Accordingly, boundary layer development plays a key role in distinguishing between open and enclosed convection regimes. The boundary layer thickness can be estimated using the interpretation of the Blasius boundary layer definition proposed by Schlichting [322], expressed in Eq. (5.27):

$$\begin{cases} \delta_{\text{bl}} \simeq 5x Re_x^{-1/2} & \text{if } Re_x < 5 \cdot 10^5 \text{ (laminar)} \\ \delta_{\text{bl}} \simeq 0.37x Re_x^{-1/5} & \text{if } Re_x > 5 \cdot 10^5 \text{ (turbulent)} \end{cases} \quad (5.27)$$

where Re_x denotes the Reynolds number evaluated at a distance x from the leading edge. For the reference case, assuming unmixed flow and continuous channel-fins, the maximum layer thickness occurs at $x_h = L_{\text{fin,h}} = W_{\text{plt}}$ for the hot channels and $x_c = L_{\text{fin,c}} = L_{\text{plt}}$ for the cold ones.

Numerical implementation and computational cost analysis

A numerical model developed in Python implements the presented formulations for the thermal domain of the heat exchangers. It computes the heat transfer rate and outlet fluid temperatures for a given geometry, provided the upstream temperatures and pressures and the downstream pressures of the fluids.

The infinite series of Eq. (5.10) is solved numerically through an iterative approach that leverages the diminishing contribution of higher-order terms in the ϕ summation. The series is initially approximated as a finite sum with an upper summation limit of 1 and progressively refined by increasing this limit and including additional terms. The refinement continues until the contribution of the newly added terms falls below 0.01% of the current sum value, or until a maximum of 50 iterations is reached. In practice, convergence was achieved after only two iterations for all simulations in this study (corresponding to the first three terms of the ϕ series), well before approaching the maximum iteration limit.

The model derives the relevant fluid properties (ρ , μ , and c_p) using the Python CoolProp library by Bell et al. [322]. These properties are evaluated at the inlet and outlet sections of the heat exchanger and averaged to solve the plate heat transfer equations. However, since the downstream temperatures are unknown a priori, an iterative procedure is introduced to estimate and refine the mean fluid properties. The procedure begins by assuming a negligible heat transfer rate, corresponding to an initial guess of $T_{\text{out}} = T_{\text{in}}$. These initial temperature estimates are then used to evaluate the outlet thermophysical properties, compute the average plate properties, and update the predicted heat transfer rate and outlet temperatures. The procedure is repeated iteratively until the newly computed outlet temperatures converge with the previously assumed values, given a convergence tolerance of 0.01 °C.

Ultimately, the proposed thermal modelling approach retains the robustness of physics-based models while providing the framework flexibility required to simulate diverse geometries and potentially accommodate the introduction of new configurations. Moreover, while the lumped representation of the heat transfer coefficients reduces the overall simulation fidelity, it enables rapid solving of the governing equations at a fraction of the computational cost associated with comparable discretised methods.

This reduction in computational cost is quantified by comparing the proposed ϕ -based model with the finite-difference routine presented by Nellis and Klein [323] for unmixed cross-flow plate heat transfer. The numerical tool, originally implemented in MATLAB by the authors, is reproduced in Python to enable a direct comparison with the present methodology. The main steps of the discretised model are summarised in the following:

1. The algorithm defines a virtual mesh of $M \times N$ grid points. For simplicity, $M = N$ is assumed;
2. A uniform temperature distribution is initially assumed across all mesh points for both fluids;
3. The CoolProp library [322] is used to evaluate the local fluid properties at each mesh point;
4. With the inlet boundary conditions fixed, the local energy balance between adjacent nodes is computed across the mesh according to Eq. (5.28);
5. The resulting temperature fields are updated and compared with the previous temperature distribution;
6. The process is repeated from Step 3 onward until the root-mean-square error between the new and previous temperature fields falls below a predefined tolerance of 0.01°C , ensuring consistency with the ϕ -model.

$$\begin{cases} \frac{\dot{m}_h c_{p,h}}{M} (T_{h,i,j} - T_{h,i,j+1}) = \frac{UA}{MN} \left[\frac{T_{h,i,j} + T_{h,i,j+1}}{2} - \frac{T_{c,i,j} + T_{c,i+1,j}}{2} \right] \\ \frac{\dot{m}_c c_{p,c}}{N} (T_{c,i+1,j} - T_{c,i,j}) = \frac{UA}{MN} \left[\frac{T_{h,i,j} + T_{h,i,j+1}}{2} - \frac{T_{c,i,j} + T_{c,i+1,j}}{2} \right] \end{cases} \quad (5.28)$$

for $i = 1, \dots, M, j = 1, \dots, N$

An automatic mesh refinement routine is implemented to adjust the grid discretisation according to the desired solution accuracy, similarly to how the finite approximation of Eq. (5.10) is refined in the ϕ -model. The procedure starts with a coarse 2×2 mesh and progressively increases the resolution by doubling the number of grid points at each step, as defined in Eq. (5.29). After each refinement, the temperature distribution is recomputed, and the process continues until the change in

average outlet temperatures between successive meshes falls below a user-defined tolerance of 0.01 °C.

$$M_n = N_n = \text{round}(M_0 r^n) \quad \text{With: } M_0 = 2, \quad r = \sqrt{2}, \quad n = 0, 1, 2, 3, \dots \quad (5.29)$$

Accordingly, the discretised solution is governed by two convergence criteria: one for the iterative convergence of the temperature field within a given mesh, and one for convergence between consecutive mesh refinements. For simplicity, the same temperature tolerance is applied to both criteria. Consequently, a relaxed tolerance of 0.1 °C will result in a coarser mesh and faster, less strict temperature-field convergence, whereas a tighter tolerance of 0.001 °C will require finer meshes and yield more precise temperature distributions.

The computational times of the discretised approach are compared with those of the proposed ϕ -model across a range of temperature refinement tolerances from 1 °C to 0.00001 °C, and for three iteration refinement thresholds in the numerical solution of Eq. (5.10): 1%, 0.01%, and 0.0001%. For consistency, each model is run 100 times per tolerance target, assuming a 0.0295 kg s⁻¹ water stream at 60 °C and a 0.0028 kg s⁻¹ airflow at 25 °C, with the inlet temperatures randomized by ± 10 °C for each run.

The results, presented in Fig. 5.4, compare the average computational costs of the discretised and ϕ -based models over 100 simulations, as well as the cold-fluid outlet temperature for the reference, non-randomized case.

Fig. 5.4a confirms the substantially lower computational cost of the lumped ϕ -model, which remains nearly constant across all tested summation accuracies (1%, 0.01%, 0.0001%). These results highlight the negligible impact of iterating the numerical solution of Eq. (5.10), which converged using only 2, 3, and 4 terms for the 1%, 0.01%, and 0.0001% thresholds, respectively. Compared to the discretised approach, the ϕ -model requires only 25% of the finite-difference method runtime when a temperature refinement tolerance of 0.1 °C or higher is assumed. For the more stringent tolerance of 0.01 °C, however, the ϕ -model requires merely 2% of the computational time of the discretised model, as the cost of solving the ϕ equation increases marginally, while the runtime of the discretised approach grows exponentially due to finer mesh requirements and additional iterations.

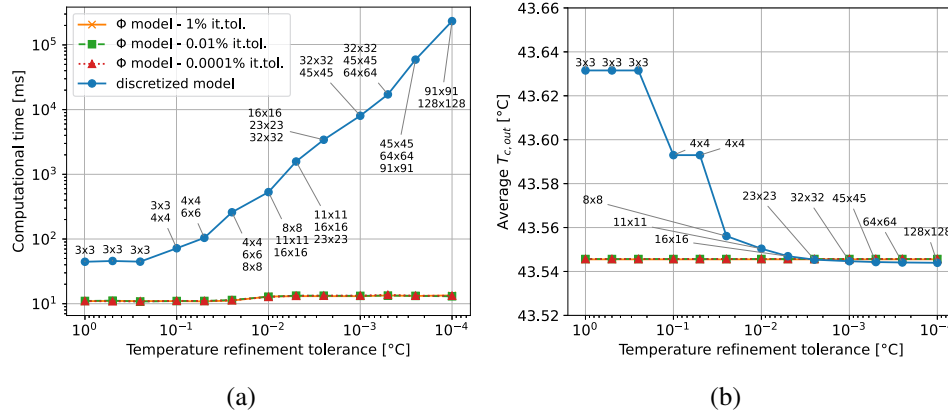


Fig. 5.4 Comparison of the average computational costs (5.4a) and reference simulation outcomes (5.4b) for the proposed global ϕ -model and a finite difference approach, as a function of the temperature refinement tolerance. Global model results are provided for three minimum summation refinement thresholds in solving the ϕ equation: 1%, 0.01%, 0.0001%. The converged mesh sizes associated with each tolerance are also reported. Reproduced from the author's previous work [10], licensed under CC BY 4.0.

Fig. 5.4b illustrates the rapid numerical convergence of the ϕ -model, showing reduced sensitivity to refinement tolerances on iterations and stable predictions across the full range. This behaviour arises because the temperature refinement tolerance affects only the determination of global fluid properties, rather than the spatial resolution of the solution. By contrast, in the discretised model, this tolerance also governs the mesh resolution, with appreciable convergence being observed only when the mesh exceeds a 10×10 grid, consistent with Nellis and Klein [323]. In the present implementation, this level of convergence is reached for a temperature refinement tolerance of 0.01 °C, or smaller. Conversely, the ϕ -model produces consistent numerical results across all tested summation thresholds (1%, 0.01%, and 0.0001%), indicating that the first two terms of Eq. (5.10) already capture most of the heat transfer behaviour, with no need for substantial refinement of ϕ .

5.3.2 Hydraulic domain

The hydraulic domain of the HE is evaluated to determine the characteristic pressures within the component and the associated pressure losses on both fluid streams. According to Shah and Sekulić [300], three primary mechanisms contribute to pressure variation within compact heat exchangers: flow contraction at the inlet,

frictional and thermal effects along the channels, and flow expansion at the outlet, as illustrated in Fig. 5.5. Each contribution can be decomposed into ideal and irreversible components, which can be combined into a unified formulation under the following assumptions:

- **Uniform velocity profile.**
- **Constant fluid properties.**
- **Incompressible fluid flow**, valid for Mach numbers below 0.3 [324]. Since air is the only compressible fluid considered in this study, the assumption remains acceptable provided that the flow velocity does not exceed 100 m s^{-1} .

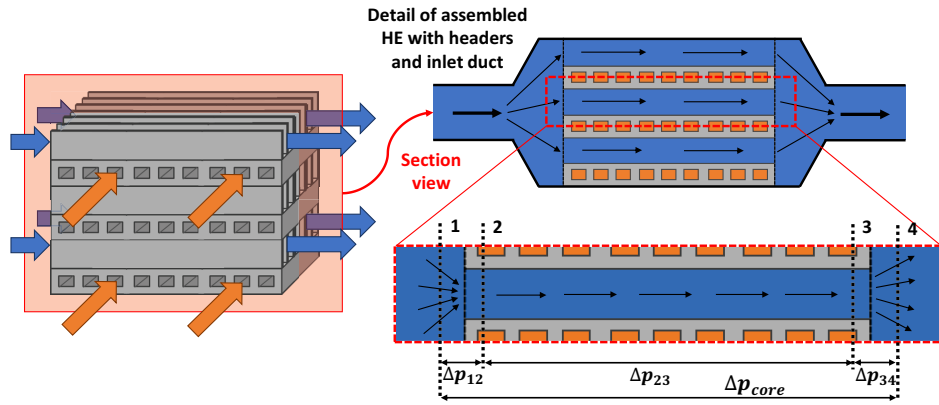


Fig. 5.5 HE representation and subdivision according to the inlet, mid, and outlet pressure drop contributions adopted in the modelling approach. Reproduced from the author's previous work [10], licensed under CC BY 4.0.

Under these assumptions, the comprehensive pressure-drop formulation for compact heat exchangers is expressed in Eq. (5.30), following Shah and Sekulić [300] and implemented according to Ref. [11].

$$\Delta p_{14} = \frac{G^2}{2\rho_2} \left[(1 - \sigma_{\text{HE}}^2) + K_{\text{contr}} + \right. \\ \left. + 2 \left(\frac{\rho_2}{\rho_3} - 1 \right) + \left(f_D \frac{L_{\text{hyd}}}{D_{\text{hyd}}} \rho_2 \overline{\left(\frac{1}{\rho} \right)} \right) - \frac{\rho_2}{\rho_3} (1 - \sigma_{\text{HE}}^2) + \frac{\rho_2}{\rho_3} K_{\text{exp}} \right] \quad (5.30)$$

In this formulation, the first two terms represent the inlet pressure losses, the third and fourth terms describe the pressure variation within the channels, and the final

two terms correspond to the outlet expansion effects. The parameter σ_{HE} denotes the heat exchanger porosity, defined as the ratio between the total cross-sectional area of the channels and the upstream frontal area of the HE, consistent with established thermal–fluid literature conventions. It is assumed identical at the inlet and outlet sections and is defined as

$$\sigma_{\text{HE}} = \frac{A_{2,\text{cs}}}{A_{1,\text{cs}}} = \frac{A_{3,\text{cs}}}{A_{4,\text{cs}}} \quad (5.31)$$

The mass flux G is defined as the mass flow rate divided by the total cross-sectional area of the channels, which are assumed equal at sections 2 and 3:

$$G = \frac{\dot{m}_{\text{HE}}}{A_{2,\text{cs}}} = \frac{\dot{m}_{\text{HE}}}{A_{3,\text{cs}}} \quad (5.32)$$

The coefficients K_{contr} and K_{exp} represent the irreversible losses associated with flow contraction and expansion, respectively. These coefficients are derived from the empirical diagrams of Kays and London [325] and are implemented here through tabulated data to allow automatic adjustment based on the simulated geometry. The fluid densities at the channel inlet and outlet are denoted by ρ_2 and ρ_3 . Under the incompressible flow assumption, they can also represent the upstream and downstream densities, such that $\rho_1 = \rho_2$ and $\rho_4 = \rho_3$.

The parameters L_{hyd} and $D_{\text{hyd}} = 4A_{\text{chan,cs}}/P_{\text{chan,cs}}$ define the channel length and hydraulic diameter, which are assumed constant across all ducts. The term $\overline{\left(\frac{1}{\rho}\right)}$ denotes the mean reciprocal density within the channel and may be approximated as $(\bar{\rho})^{-1}$ when density variations remain small. The Darcy friction factor f_{D} is evaluated using the Churchill correlation of Eq. (5.33) [326]. This formulation is preferred over the classical Darcy–Colebrook equation [260], as it provides an explicit expression valid across all flow regimes without discontinuities.

$$\begin{cases} f_{\text{D}} = 8 \left[\left(\frac{8}{Re} \right)^{12} + (\Phi_1 + \Phi_2)^{-1.5} \right]^{\frac{1}{12}} \\ \Phi_1 = \left[-2.457 \ln \left(\left(\frac{7}{Re} \right)^{0.9} + 0.27 \frac{\epsilon}{D_{\text{hyd}}} \right) \right]^{16} \\ \Phi_2 = \left[\frac{37530}{Re} \right]^{16} \end{cases} \quad \text{Where } Re = \frac{\rho u D_{\text{hyd}}}{\mu} \quad (5.33)$$

This formulation also incorporates the effect of surface roughness in transitional and turbulent regimes. To account for variations in Reynolds number along the

channels, the Darcy friction factor is evaluated using the mean flow properties, namely the average velocity $\bar{u}$, density $\bar{\rho}$, and dynamic viscosity $\bar{\mu}$, defined between sections 2 and 3 in Fig. 5.5. However, potential reductions in the critical Reynolds number caused by highly textured surfaces are neglected at the present stage of the study.

The parameter ε denotes the equivalent Colebrook sand–grain roughness, which does not correspond to a directly measurable physical quantity. As a first-order approximation, the ratio $\varepsilon/D_{\text{hyd}}$ is estimated using Ra/D_{hyd} , following Refs. [11, 327, 328]. This approximation relies on the widespread use of the arithmetic mean roughness Ra in experimental characterisation [327], although it provides only a partial description of the surface texture.

Numerical implementation

The hydraulic model computes the pressure losses for both fluids and determines the corresponding inlet pressures, using the outlet pressures and fluid temperatures as inputs.

In contrast to the thermal model, the hydraulic calculation proceeds from the HE outlet towards the inlet. Absolute pressures are determined sequentially by adding the loss contributions obtained from Eq. (5.30), as illustrated in Fig. 5.6. This formulation reduces the likelihood of obtaining negative pressures within the component, which could arise if the losses were subtracted from upstream values. Such conditions would not only produce non-physical results but could also introduce numerical instabilities when querying fluid property databases, ultimately limiting the robustness of the model in optimisation procedures.

Two iterative loops are implemented to refine the local fluid properties at sections 2 and 3 and to accurately estimate the expansion coefficient K_{exp} and the mean channel properties. Initially, a negligible pressure drop is assumed, leading to $p_3 = p_4$ and $p_2 = p_3$. Fluid properties are then evaluated at these conditions and used to compute the HE pressure losses and the corresponding upstream pressures. The updated pressures are subsequently employed to recompute the fluid properties, and the process is repeated until convergence between assumed and calculated pressures is achieved. The procedure is first applied to section 3 and subsequently to

section 2. A convergence tolerance of 100 Pa is adopted as the termination criterion for the iterative process.

5.3.3 Volumetric and gravimetric domains

The volumetric and gravimetric domains are finally assessed to determine the component encumber, both in terms of the device volume and its wet and dry masses. The overall device volume is first estimated using Eq. (5.34), approximating the component as a rectangular cuboid. The volumes occupied by the hot and cold fluids are then calculated using Eq. (5.35) and Eq. (5.36), respectively, assuming straight prismatic channels. The channel cross-sectional area is defined as $A_{\text{chan,cs}} = \pi D_{\text{chan}}^2/4$ for circular ducts, and $A_{\text{chan,cs}} = B_{\text{chan}}H_{\text{chan}}$ for rectangular geometries. The structural volume of the heat exchanger is finally obtained by subtracting the fluid volumes from the total device volume, as expressed in Eq. (5.37).

$$V_{\text{HE}} = L_{\text{HE}} \cdot W_{\text{HE}} \cdot H_{\text{HE}} \quad (5.34)$$

$$V_h = (N_{\text{h,layers}} \cdot N_{\text{h,fin}}) \cdot A_{\text{h,chan,cs}} \cdot W_{\text{HE}} \quad (5.35)$$

$$V_c = (N_{\text{c,layers}} \cdot N_{\text{c,fin}}) \cdot A_{\text{c,chan,cs}} \cdot L_{\text{HE}} \quad (5.36)$$

$$V_s = V_{\text{HE}} - V_h - V_c \quad (5.37)$$

Here, L_{HE} , W_{HE} , and H_{HE} denote the heat exchanger length, depth, and height, respectively, while the products $N_{\text{h,layers}} \cdot N_{\text{h,fin}}$ and $N_{\text{c,layers}} \cdot N_{\text{c,fin}}$ represent the total number of channels for the hot and cold fluids.

The masses of the fluids and structural material are then obtained by multiplying the corresponding volumes by their densities $m = V\rho$. Mean densities $\bar{\rho} = (\rho_2 + \rho_3)/2$ are adopted for the working fluids, while tabulated material properties are used for the structural elements. The total heat exchanger mass is therefore expressed as

$$m_{\text{HE}} = V_h \cdot \rho_h + V_c \cdot \rho_c + V_s \cdot \rho_s \quad (5.38)$$

which accounts for the contributions of both fluids and the structural components.

5.3.4 Integrated HE numerical model

A numerical simulation model combines the implemented thermal and hydraulic models, and integrates them with the last gravimetric equations previously described. High tool modularity is ensured by having each of these three aspects be independently managed through a dedicated, fully functional, Python script, allowing for the proposed framework to be easily updated and expanded to the analysis of alternative design and configurations. A master script, finally orchestrate the complete HE model, activating the execution of the three base codes based on the rationale illustrated in Fig. (5.6).

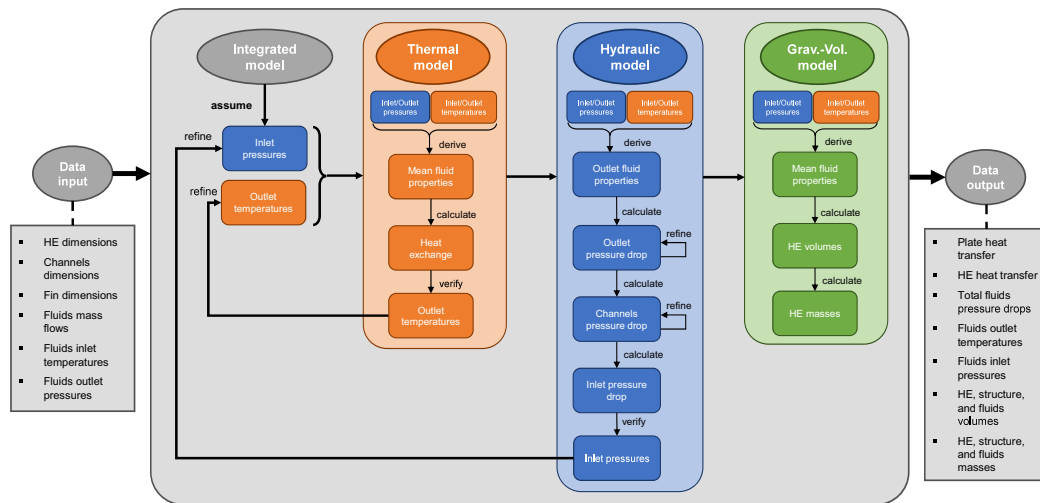


Fig. 5.6 Functional scheme of the integrated HE characterisation model, including detail of the thermal, hydraulic, and gravimetric/volumetric sub-models. Reproduced from the author's previous work [10], licensed under CC BY 4.0.

The model supports the rapid and automated assessment of compact, cross-flow heat exchangers performance, provided the geometric inputs illustrated in Fig. 5.7. The remaining geometrical parameters of interest, such as channels' wet surfaces and hydraulic diameters, are instead extracted directly by the tool. The fluid's mass flow rates, inlet temperatures, and outlet pressures define the key operating conditions for the simulation, while all relevant fluid properties are determined within the model via the Python Cool-prop library [322] and the accuracy refinement strategies presented for the thermal and hydraulic domains.

The master script, which integrates and orchestrates the previously the individual thermal, hydraulic and gravimetric models, combines the base script results to expand

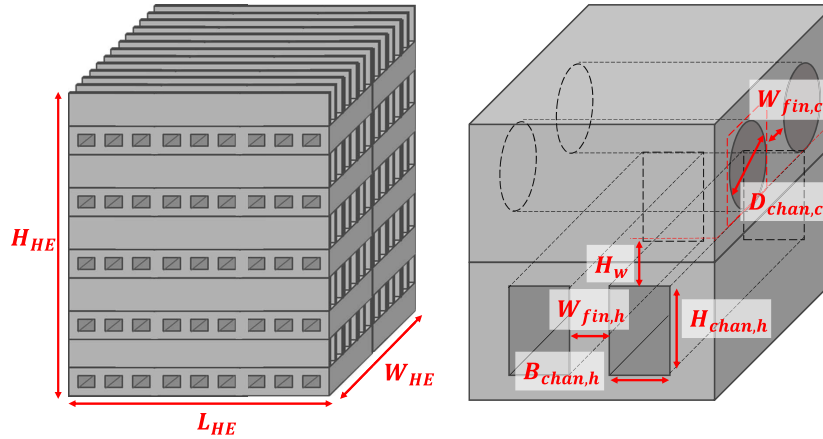


Fig. 5.7 Representation of all geometrical inputs required by the model. A combined circular-rectangular channel design is provided, on the left, to evidence differences between parameters of the two shapes. Reproduced from the author's previous work [10], licensed under CC BY 4.0.

the methodology, so that only the fluids' inlet temperatures and outlet pressures are required as inputs, returning the simulated outlet temperatures and inlet pressures instead. To do so, the integrated model must account for the fact that both the inlet pressures and outlet temperatures are unknown a priori, despite being required as inputs to the hydraulic, and thermal models respectively. An external loop is thus introduced, iterating between the two models to progressively refine the determination of both parameters. The initial assumptions of negligible heat transfer and pressure drops are retained, combining the initial hypotheses of $T_4 = T_1$, and $p_1 = p_4$. The thermal model is run first, calculating a first estimate heat transfer rate and iteratively refining the estimated outlet temperatures T_4 . Subsequently, the hydraulic model is executed, adopting the refined outlet temperatures to compute the pressure drops and update the inlet pressures p_1 . These updated pressures are then provided as new inputs for a second run of thermal model, and the process is repeated until the assumed values for the thermal and hydraulic domains converge with the newly computed results. Once this double iteration is concluded, the refined mean fluid properties are saved and the volumetric-gravimetric performance is last assessed.

The simulation outputs for this entire framework are: the heat transfer rate of a single plate and the entire heat exchanger, the total pressure drop experienced by

both fluids, their respective temperature and pressures at the relevant device sections (1, 2, 3, and 4), the dry mass, wet mass, and volume of the component.

5.4 Model validation

The proposed methodology and its numerical implementation are validated against experimental data from the literature in order to consolidate the model predictions before extending the framework to design optimisation of HE, which is addressed in the final chapter of this dissertation (Chap. 6).

5.4.1 Reference validation case

Validation is first conducted using the experimental data reported by Ref. [11] for a compact heat exchanger manufactured via (AM). AISI 316 stainless steel is adopted as the reference structural material, with an average density of $\rho_s \simeq 8000 \text{ kg m}^{-3}$ and thermal conductivity $k_s \simeq 15 \text{ W m}^{-1} \text{ K}^{-1}$. The channel surface roughness is assumed equal to the experimentally measured value $Ra = 12.21 \text{ }\mu\text{m}$.

For completeness, a schematic representation of the experimental test rig used by da Silva et al. [11] is shown in Fig. 5.8, while the geometry and dimensions of the reference heat exchanger are reported in Fig. 5.9.

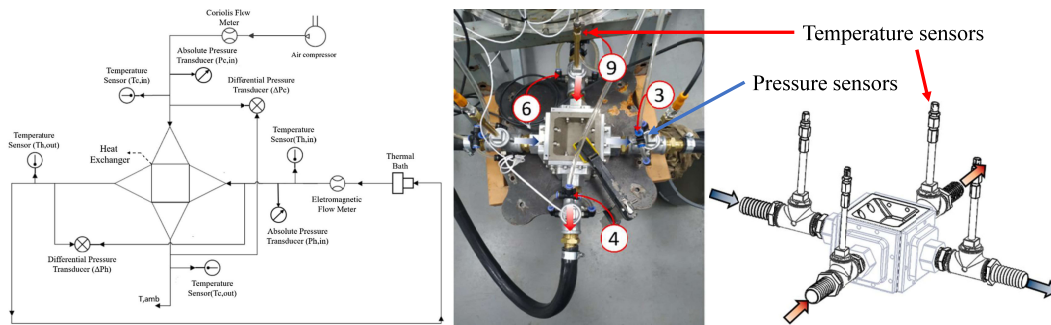


Fig. 5.8 Literature-based test rig and sensor placement for the experimental measurement of pressure drops and temperatures. Adapted from da Silva et al. [11]. ©2021 Elsevier.

The reference heat exchanger has a porosity $\sigma_{HE} = 0.1$ and consists of nine layers of hot-fluid channels alternating with ten layers of cold-fluid channels. Each layer comprises a single row of 19 straight circular ducts with an average manufactured

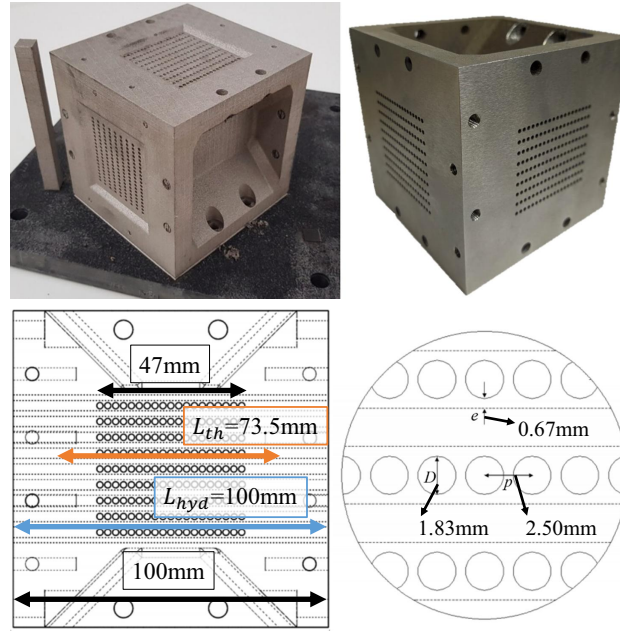


Fig. 5.9 Photographs, geometries, and dimensions of the reference literature heat exchanger. Adapted from da Silva et al. [11]. ©2021 Elsevier.

diameter of 1.83 mm. The spacing between channel axes within the same layer is 2.5 mm, while the distance between the median planes of adjacent layers is also 2.5 mm.

For the pressure drop calculations of Eq. (5.30), a hydraulic length L_{hyd} of 100 mm is adopted, corresponding to the total flow path length within the component and equal, in this case, to the main HE dimensions L_{HE} and W_{HE} .

It should be noted that the experimental pressure measurements in Ref. [11] were performed slightly upstream and downstream of the heat exchanger due to practical limitations in sensor placement, as illustrated in Fig. 5.8. Consequently, the measured pressure drops include additional losses associated with the surrounding test rig. To enable a consistent comparison between simulations and experiments, these additional losses are explicitly modelled following the formulations proposed by the authors [11], allowing the experimental measurements to be converted into equivalent heat exchanger core conditions.

The experimental facility includes several components, although only a subset contributes to the measured pressure losses. These are the upstream and downstream manifolds, which connect the assembly to the rest of the rig and house the sensors,

and two headers linking the manifolds to the heat exchanger core. Each of these elements introduces additional pressure losses that are captured in the experimental measurements.

To account for these effects, the pressure-drop model is extended by introducing additional sections, as illustrated in Fig. 5.10. Sections 0 and 5 represent the sensor locations and the boundaries of the rig assembly, while sections 1' and 4' correspond to the interfaces between the manifolds and the headers.

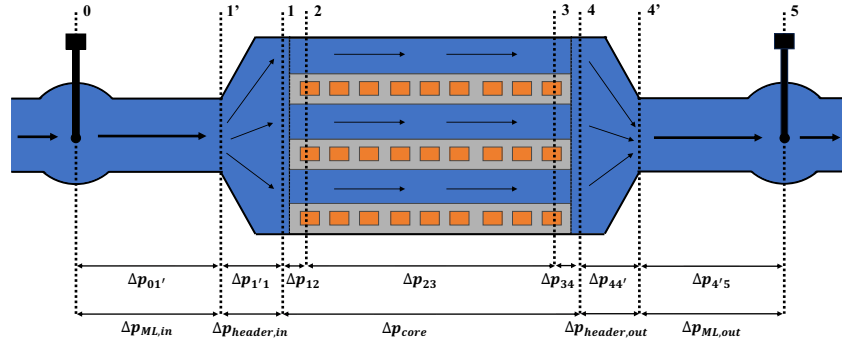


Fig. 5.10 Representation of the heat exchanger core, headers, and manifolds with the corresponding sections used for pressure-drop evaluation according to the test rig in Ref. [11]. Reproduced from the author's previous work [10], licensed under CC BY 4.0.

Pressure losses in the headers are evaluated using Eq. (5.39) for the inlet and Eq. (5.40) for the outlet, following Idel'Chik [329]. The irreversibility coefficients are set to $\zeta_{\text{exp}} = 0.5$ at the inlet and $\zeta_{\text{con}} = 0.25$ to 0.097 at the outlet, consistent with Ref. [11]. The value $\zeta_{\text{con}} = 0.25$ corresponds to the lowest Reynolds numbers, while $\zeta_{\text{con}} = 0.097$ applies to the highest.

$$\Delta p_{1'1} = \Delta p_{\text{header}_{\text{in}}} = \frac{\rho_1 u_1^2}{2} \left[1 - \left(\frac{A_1}{A_{1'}} \right)^2 + \zeta_{\text{exp}} \right] \quad (5.39)$$

$$\Delta p_{44'} = \Delta p_{\text{header}_{\text{out}}} = \frac{\rho_4 u_4^2}{2} \left[\left(\frac{A_4}{A_{4'}} \right)^2 - 1 + \zeta_{\text{con}} \right] \quad (5.40)$$

Pressure losses in the manifolds are instead evaluated using Eq. (5.41) and Eq. (5.42) for the inlet and outlet, respectively, following Çengel and Cimbala [330]. A loss coefficient $K_L = 0.53$ is adopted in accordance with the value reported in

Ref. [11].

$$\Delta p_{01'} = \Delta p_{\text{MLin}} = \frac{\rho_0 K_L u_0^2}{2} \quad (5.41)$$

$$\Delta p_{4'5} = \Delta p_{\text{MLout}} = \frac{\rho_5 K_L u_5^2}{2} \quad (5.42)$$

The total pressure drop measured by the test rig is therefore obtained by combining the rig-related losses with those occurring in the heat exchanger core:

$$\Delta p_{\text{rig}} = \Delta p_{\text{MLin}} + \Delta p_{\text{headerin}} + \Delta p_{14} + \Delta p_{\text{headerout}} + \Delta p_{\text{MLout}} \quad (5.43)$$

By rearranging Eq. (5.43), the experimental value of Δp_{14} can be extracted and directly compared with the predictions of the baseline numerical model.

5.4.2 Thermal length validation

Following Ref. [11], a thermal length L_{th} of 73.5 mm is adopted in the thermal calculations. This value is defined as the average between the full channel length (100 mm) and the 47 mm section operating under true cross-flow conditions. The definition accounts for heat transfer occurring in the inlet and outlet regions surrounding the central cross-flow core, which would otherwise be neglected. Conversely, assuming the entire 100 mm length to operate under ideal cross-flow conditions would lead to an overestimation of thermal performance, as demonstrated by da Silva et al. [11].

Given the geometric symmetry of the component, identical thermal lengths are assumed for both fluids, $L_{th,c}$ and $L_{th,h}$, and are used to define L_{plt} and W_{plt} in the numerical thermal model. Notably, the formulations of Eqs. (5.10)–(5.13) and the associated parameters a and b remain unaffected by the thermal length definition. This occurs because these relations depend on the product $U L_{\text{plt}} W_{\text{plt}}$, rather than on the plate dimensions individually. This product is determined through the thermal resistance model, which instead depends strongly on the assumed thermal length, particularly in Eqs. (5.19), (5.20), and (5.25), as well as in the selection of the appropriate Nusselt correlation from Tab. (5.2).

For consistency, the adopted thermal length definition is further validated using experimental data from Morteau et al. [12], who investigated an AISI-316 compact heat exchanger manufactured via diffusion bonding. The reference geometry is a $167\text{ mm} \times 167\text{ mm}$ water–water heat exchanger with 450 square channels of 3 mm side arranged in 15 alternating layers. The proposed thermal model is used to simulate the heat exchange using both the reference surface area reported by the authors (0.9418 m^2) and the area derived from the thermal length definition of da Silva et al. [11]. The corresponding results are presented in Fig. 5.11a.

The same validation approach is applied to the experimental geometries reported by Luo et al. [13] and Hein and Morteau [14]. Luo et al. [13] investigated the influence of flow maldistribution in an aluminium cross-flow heat exchanger consisting of 128 round channels distributed across eight layers with various inlet and outlet distributor configurations. The present model and thermal length definition are applied to this geometry, and the results are shown in Fig. 5.11b.

Hein and Morteau [14] studied heat transfer in an AM compact heat exchanger with 84 round channels arranged in seven layers and manufactured via Selective Laser Sintering of Polyamide 2200. Simulations are performed using both the thermal conductivity reported in the reference study ($0.23\text{ W m}^{-1}\text{ K}^{-1}$) and the value from the EOS technical data sheet for sintered Polyamide 2200 ($0.144\text{ W m}^{-1}\text{ K}^{-1}$) [331]. The corresponding results are presented in Fig. 5.11c.

Results from Fig. 5.11a indicate that the numerical predictions for the geometry studied by Morteau et al. [12] are consistent with other models reported in the literature, showing comparable magnitudes and deviations. In particular, adopting the thermal length definition proposed by da Silva et al. [11] provides the most accurate predictions and aligns closely with the advanced numerical approaches reported in the reference study, including finite-element simulations.

For the heat exchanger investigated by Luo et al. [13], the results shown in Fig. 5.11b demonstrate satisfactory agreement with the theoretical predictions presented in the reference study. The model reproduces the experimental trends particularly well under conditions of uniform flow distribution (configuration 1).

Finally, for the polymer heat exchanger investigated by Hein and Morteau [14], simulations assuming a thermal conductivity of $0.23\text{ W m}^{-1}\text{ K}^{-1}$ do not closely match the experimental results. However, adopting the conductivity value reported

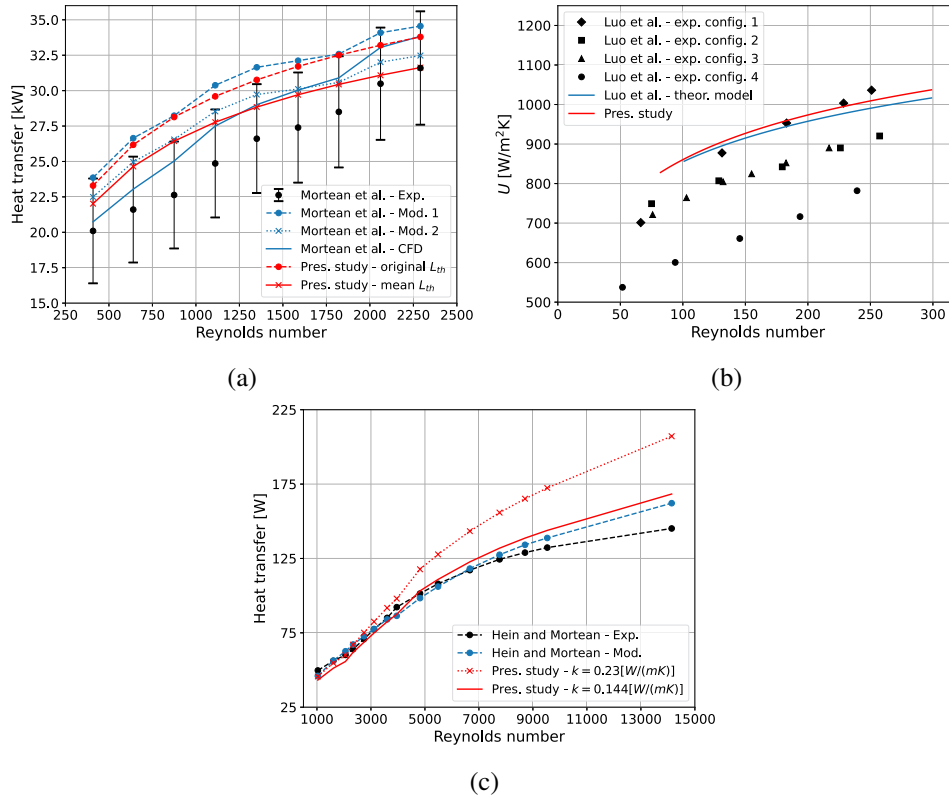


Fig. 5.11 Validation of the proposed thermal length definition against experimental data (black) and numerical predictions (blue) from Mortean et al. [12] (5.11a), Luo et al. [13] (5.11b), and Hein and Mortean [14] (5.11c). Present-model predictions (red) use either the reference heat transfer area (dashed) or the thermal length proposed by da Silva et al. [11] (solid). In (5.11c), the dotted and solid lines assume the thermal conductivities reported by Hein and Mortean [14] and in the EOS data sheet for sintered Polyamide 2200, respectively. Reproduced from the author's previous work [10] under CC BY 4.0.

in the manufacturer's data sheet ($0.144 \text{ W m}^{-1} \text{ K}^{-1}$) yields predictions that are in acceptable agreement with the experimental data.

These comparisons support the validity of both the proposed thermal model and the adopted thermal length definition, contributing to the overall validation of the modelling framework.

5.4.3 Complete model validation

After incorporating the additional losses associated with the experimental test rig and importing the reference HE geometries shown in Fig. 5.9, the thermal-hydraulic

performance of the component is simulated. The results are presented in Fig. 5.12 for both the HE core and the complete test rig assembly, and compared with the experimental measurements and numerical predictions reported by da Silva et al. [11].

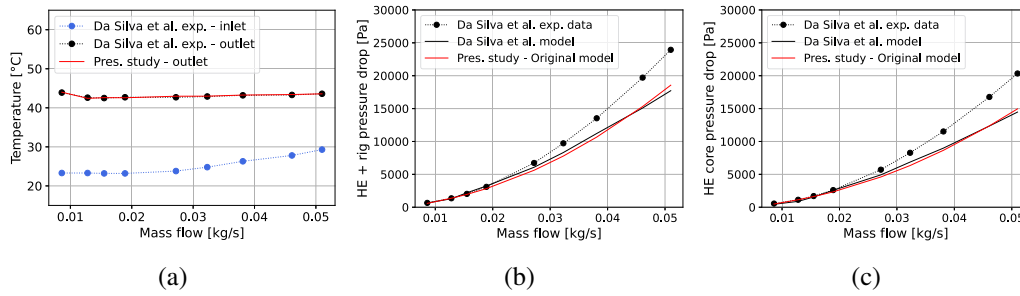


Fig. 5.12 Thermal-hydraulic results of the proposed integrated model (red line) compared to experimental data (black dotted line) and numerical predictions (black line) from Ref. [11]. (5.12a) Inlet and outlet air temperature measurements and numerical results. (5.12b) air pressure drop measurements and numerical results for the test rig. (5.12c) air pressure drop measurements and numerical results for the sole heat exchanger (core). Reproduced from the author's previous work [10], licensed under CC BY 4.0.

Fig. 5.12a shows strong agreement between the present numerical results and the experimental data of Ref. [11] in terms of thermal performance, with a maximum relative deviation of 1.4% in the temperature difference. These results validate the assumptions adopted to extend the modelling approach to heat exchangers featuring circular ducts and an uneven number of channel layers, highlighting the flexibility of the proposed thermal formulation. Furthermore, they confirm that the thermal length definition introduced by da Silva et al. [11] is suitable for modelling heat exchanger plates through Eqs. (5.9)–(5.13).

By contrast, Fig. 5.12b and Fig. 5.12c show satisfactory agreement between the predicted pressure drops and the numerical results reported in Ref. [11]. Minor discrepancies are observed and can be attributed to small differences in the implementation of empirical loss coefficients. Consequently, the present model also inherits the deviations of the original methodology relative to the experimental data, resulting in pressure drop underestimations of up to -22.4% for the test rig and -26.4% for the HE core, particularly at higher Reynolds numbers.

Such discrepancies exceed the typical expectations for models based on well-established correlations for compact heat exchanger characterisation [300]. Instead,

the deviations are most likely associated with the (AM) process used to fabricate the component. Accurate performance prediction of AM heat exchangers remains an active research topic, as discussed by Careri et al. [291] and Kaur and Singh [290]. In particular, limitations in dimensional accuracy and repeatability can significantly affect component geometry, including variations in channel dimensions, shape, and surface roughness.

Da Silva et al. [11] accounted for these effects by considering a measured surface roughness of $12.21\text{ }\mu\text{m}$ and a noticeable reduction in the manufactured channel diameter, which decreased from the design value $D_{\text{chan,design}} = 2.00\text{ mm}$ to an average fabricated value of $\bar{D}_{\text{chan,AM}} = 1.83\text{ mm}$. These factors contributed to the trends observed in Fig. 5.12, with maximum experimental deviations reaching 27%. The study further concluded that surface roughness has a limited influence on pressure drop predictions, whereas channel diameter variations exert a significantly stronger effect, as later confirmed by Ref. [332].

5.4.4 Corrected formulations for AM

A significant relative roundness error of 0.08 was also reported for the channels in Ref. [11]. Similar dimensional contractions and roundness deviations were observed by Kempen et al. [333], who attributed these deformations to the characteristics of the laser melting process. In particular, the formation of the melt pool alters the component geometry by shifting the designed profile by approximately the melt pool radius. For internal features such as channels, this effect reduces the effective dimensions by approximately the melt pool diameter. Based on the measured shrinkage from 2 mm to 1.83 mm, the reference heat exchanger was therefore likely produced with a melt pool diameter of approximately $170\text{ }\mu\text{m}$.

Additionally, Kempen et al. [333] observed the formation of dross on channel ceilings when the channel axes are not perpendicular to the build platform. This phenomenon originates from uneven cooling of the melt pool into underlying layers of unsintered powder, generating a protrusion with a characteristic size comparable to the melt pool diameter and contributing to the observed roundness errors.

For a melt pool diameter of $170\text{ }\mu\text{m}$ and an average channel diameter of 1.83 mm, the formation of such dross results in an estimated relative roundness error of

approximately 0.09, which is consistent with the 0.08 value reported by da Silva et al. [11].

To account for this effect, a corrective formulation based on the presence of AM-induced dross is introduced. Although the roundness error is acknowledged in Ref. [11], it does not appear to be explicitly incorporated into the hydraulic model. The present approach therefore assumes the presence of a triangular dross protrusion at the top of the channel. Two geometrical approximations are considered: an isosceles right triangle and an equilateral triangle. The dross area is subtracted from the nominal circular cross section to obtain a corrected channel area according to Eq. (5.44), as illustrated in Fig. 5.13.

$$\begin{cases} A_{\text{corr}} = \frac{\pi}{4} D_{\text{chan}}^2 - H_{\text{dross}}^2 & \text{for isosceles right triangle} \\ A_{\text{corr}} = \frac{\pi}{4} D_{\text{chan}}^2 - \frac{\sqrt{3}}{3} H_{\text{dross}}^2 & \text{for equilateral triangle} \end{cases} \quad (5.44)$$

Where H_{dross} represents the dross height, assumed equal to the melt pool diameter ($170 \mu\text{m}$). A corrected cross-sectional perimeter is similarly defined according to Eq. (5.45).

$$\begin{cases} P_{\text{corr}} \simeq \pi D_{\text{chan}} + 2H_{\text{dross}} (\sqrt{2} - 1) & \text{for isosceles right triangle} \\ P_{\text{corr}} \simeq \pi D_{\text{chan}} + \frac{2}{3} \sqrt{3} H_{\text{dross}} & \text{for equilateral triangle} \end{cases} \quad (5.45)$$

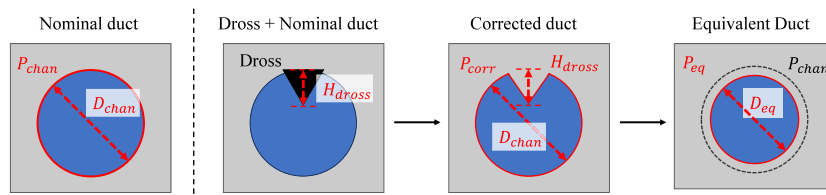


Fig. 5.13 Functional scheme of the corrected channel cross-sections, accounting for the presence of a triangular-shaped laser melting dross at the top of the duct. Reproduced from the author's previous work [10], licensed under CC BY 4.0.

A corrected hydraulic diameter is then obtained from the modified area and perimeter as $D_{\text{hyd,corr}} = 4A_{\text{corr}}/P_{\text{corr}}$. This value is used to define an equivalent

circular geometry for the hydraulic calculations according to Eq. (5.46).

$$\begin{cases} D_{eq} = D_{hyd,corr} \\ P_{eq} = \pi D_{eq} \\ A_{eq} = \pi/4(D_{eq})^2 \end{cases} \quad (5.46)$$

For the reference geometry ($D_{chan} = 1.83$ mm), both assumed cross shapes yield a similar reduction in hydraulic diameter of approximately $3.7\% \pm 0.2\%$, and are therefore treated as equivalent in the following analysis. Pressure drops are recalculated using this corrected diameter, with the results presented in Fig. 5.14. Overall agreement improves, particularly at higher Reynolds numbers, where the maximum pressure drop underestimation decreases from 26.4% to 11.4%.

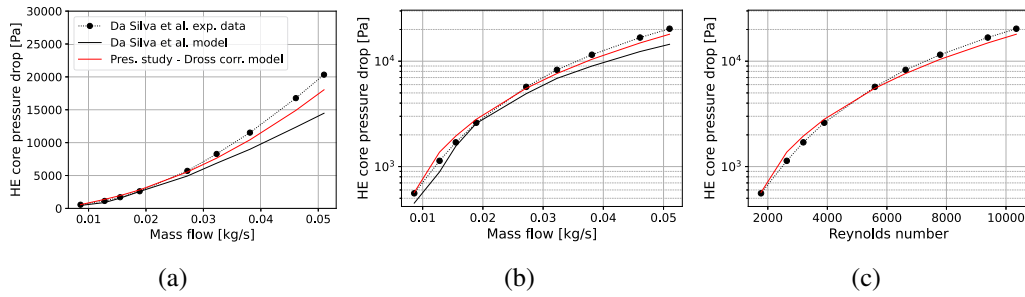


Fig. 5.14 Hydraulic results of the dross-corrected model (red line) compared to experimental data (black dotted line) and simulation outcomes (black line) by da Silva et al. [11]. Detail of the air pressure drops of the heat exchanger core with respect to the fluid mass flow rate on a linear (5.14a) and logarithmic (5.14b) scales, and for the Reynolds numbers on a logarithmic scale (5.14c). Reproduced from the author's previous work [10], licensed under CC BY 4.0.

An alternative formulation is also proposed by introducing an uncertainty factor K_p into the HE core pressure drop calculation, as expressed in Eq. (5.47).

$$\Delta p_{14,corr} = K_p \Delta p_{14} \quad (5.47)$$

This formulation provides a conservative estimate of hydraulic losses while preserving the geometric correlation of the original model and retaining the nominal hydraulic diameter. Calibration against the experimental data of Ref. [11] yields a preliminary correction factor of $K_p = 1.3$. The resulting predictions, shown in

Fig. 5.15, demonstrate improved agreement at higher Reynolds numbers, reducing the maximum pressure drop underestimation to 4.3%.

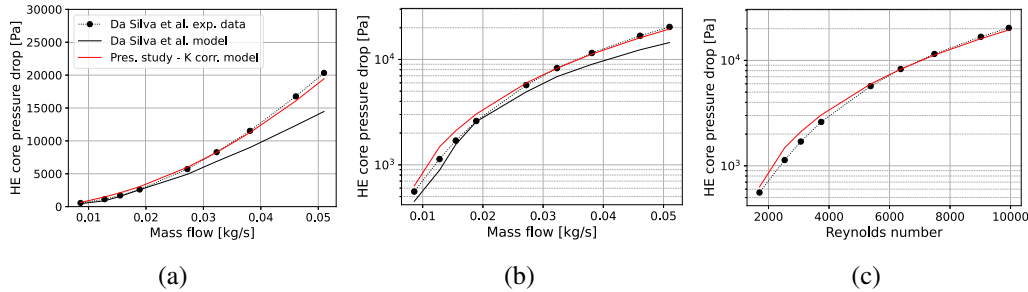


Fig. 5.15 Hydraulic results of the K_p -corrected model (red line) compared to experimental data (black dotted line) and simulation outcomes (black line) from Ref. [11]. Detail of the air pressure drops of the heat exchanger core with respect to the fluid mass flow rate on a linear (5.15a) and logarithmic (5.15b) scales, and for the Reynolds numbers on a logarithmic scale (5.15c). Reproduced from the author's previous work [10], licensed under CC BY 4.0.

Both corrective approaches increase the predicted pressure drops at low Reynolds numbers, leading to some overestimation of hydraulic losses near the transitional flow regime. This effect is more pronounced for the K_p formulation (Eq. (5.47)), which exhibits a maximum relative deviation of 30.8% around $Re \approx 2500$. However, the associated absolute errors remain small and have negligible practical impact. In contrast, the substantial reduction in underestimation at higher Reynolds numbers improves the overall predictive accuracy in operating conditions that are most relevant to heat exchanger performance.

A comparison of pressure drop predictions obtained from the original, dross-corrected, and penalty-factor-corrected models is provided in Fig. 5.16.

The impact of both corrections on the thermal predictions is illustrated in Fig. 5.17. Since the outlet temperatures reported in Fig. 5.12a are very similar, Eq. (5.48) is applied to the experimental measurements from Ref. [11] to estimate the actual heat transfer rate and highlight deviations among the models.

$$\dot{Q} = \dot{m} \cdot c_p (T_{\text{out}} - T_{\text{in}}) \quad (5.48)$$

Results in Fig. 5.17 indicate a slight increase in thermal deviations for the dross-corrected formulation. Conversely, the K -factor approach has negligible influence

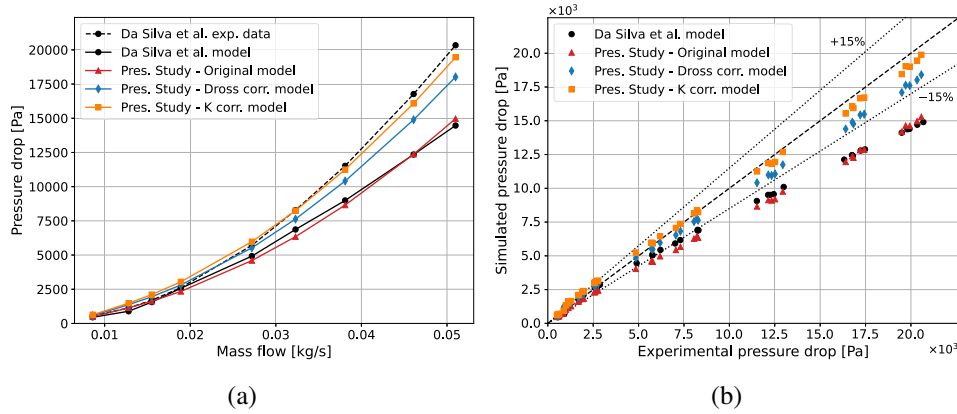


Fig. 5.16 Comparison of results from the various models simulating the air pressure drops in the core HE from Ref. [11]. Detail of trends with respect to the air mass flow rates (5.16a) and experimental deviations (5.16b). Reproduced from the author's previous work [10], licensed under CC BY 4.0.

on the predicted outlet temperatures, closely reproducing the results of the original model. This behaviour arises because the reduced channel cross section in the dross-corrected geometry slightly increases the Reynolds number, leading to a small overestimation of heat transfer. Nevertheless, the maximum relative deviation in the cold fluid temperature change increases only from 1.4% to 3.7%, remaining well below the commonly accepted 5% accuracy threshold.

5.5 Summary and highlights

The robust and flexible modelling of heat exchangers emerges as a key research direction for addressing the growing need for rapid geometry optimisation and early system-level thermal management design. In this chapter, a renewed modelling methodology is proposed, integrating established literature correlations for the characterisation of high-performance cross-flow heat exchangers featuring rectangular and circular channels compatible with additive manufacturing.

The proposed framework captures the thermal, hydraulic, and gravimetric behaviour of the components, prioritizing physics-based formulations over empirical correlations to preserve modelling flexibility. Extensive parameterization ensures sensitivity to geometric variations, while a modular structure of the numerical tool facilitates future extensions and upgrades to alternative geometries and operating

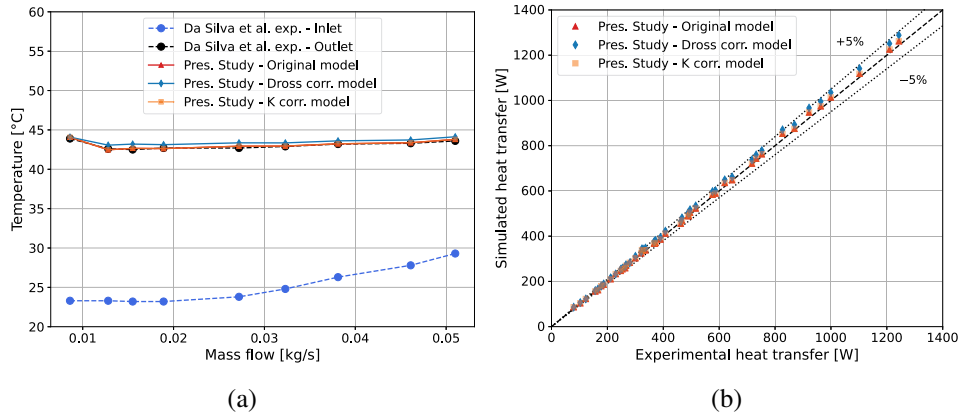


Fig. 5.17 Thermal model results following the application of corrective formulations: detail of the outlet temperature trends (5.17a) and experimental deviations of the associated heat transfer rates (5.17b). Reproduced from the author's previous work [10], licensed under CC BY 4.0.

configurations. Particular attention is devoted to the prediction of hydraulic losses in additively manufactured components, for which two preliminary corrective formulations are introduced, significantly reducing deviations at higher Reynolds numbers.

To ensure computational efficiency while retaining modelling flexibility, the thermal formulation employs lumped representations of the heat exchanger properties. In particular, the averaged dimensionless temperature difference ϕ , originally introduced by Mudawar et al. [303, 304], is adopted to condense discretised thermal contributions into a single analytical formulation. A preliminary comparison with a finite discretisation framework shows that the lumped formulation requires only 2% of the discretised model runtime for a temperature convergence tolerance of 0.01 °C.

The analysis contributes to the literature on thermal modelling based on the averaged dimensionless temperature difference ϕ , extending the case studies of Refs. [303, 304] to the geometries investigated in Refs. [13, 12, 11, 14], which differ in channel geometry, number of layers, or both. To accommodate configurations with an uneven number of channel rows, a simplifying hypothesis is introduced by which excess channel layers are treated as adiabatic. This assumption demonstrates strong agreement with experimental observations, broadening the applicability of the original methodology. Furthermore, the present study incorporates and validates the thermal length definition proposed by da Silva et al. [11] within the plate model formulation of Ref. [303], demonstrating consistency across alternative modelling approaches.

Contemporarily, the modelling approach expands the methodology of da Silva et al. [11] by integrating the hydraulic and thermal formulations within a unified simulation environment. Both domains are coupled through an iterative integration routine that updates fluid properties and boundary conditions until convergence is achieved. As a result, the complete heat exchanger performance can be evaluated using only three thermo-hydraulic inputs: inlet temperatures, outlet pressures, and mass flow rates, making the tool suitable for automated simulation, design, and optimisation workflows.

To improve numerical stability:

- The hydraulic losses are computed by propagating pressure calculations upstream from a prescribed outlet pressure, preventing the non-physical negative pressures that may arise from downstream propagation from the inlet, particularly when simulating previously unexplored designs;
- The Churchill correlation is adopted for the Darcy friction factor f_D , providing a continuous and explicit formulation across all flow regimes;
- The series expression in the ϕ formulation is numerically solved through an iterative summation procedure, progressively introducing additional terms until convergence is reached.

The complete methodology is validated against experimental data from the literature, showing good agreement between numerical predictions and empirical results. When applied to the additively manufactured heat exchanger reported by da Silva et al. [11], the thermal model achieves excellent accuracy, with a maximum deviation of 1.4% in the predicted heat transfer rates and temperature changes. Conversely, the hydraulic model initially underestimates pressure losses, with deviations reaching 26.4% at higher Reynolds numbers.

Comparisons with existing studies on additive manufacturing suggest that these discrepancies are primarily associated with fabrication-induced geometric deviations, particularly, the significant roundness errors caused by dross formation on channel ceilings. Two corrective strategies are presented. The first introduces a geometry-based correction through a modified hydraulic diameter accounting for the dross formation, while the second applies a conservative empirical uncertainty factor calibrated against experimental data. The geometry-based correction reduces the

maximum pressure-drop underestimation to 11.4%, whereas the uncertainty-factor approach further lowers it to 4.3%. Both corrections remain consistent with the thermal model, having only a limited impact on thermal predictions: the dross-corrected formulation increases the maximum heat-transfer deviation to 3.7%, while the K -factor approach preserves the original 1.4% deviation.

Despite the improvements achieved at higher Reynolds numbers, some discrepancies remain near the transitional flow regime, where both corrections tend to overestimate hydraulic losses. Nonetheless, these deviations remain small in absolute terms and do not significantly affect the validity of the overall results. Future work should focus on improving the fidelity of the corrective formulations at lower Reynolds numbers, extending their experimental validation range, and refining the K -term regression.

Several research directions emerge from this initial analysis to further improve model fidelity and extend its applicability. These include:

- Exploring alternative thermal fluids, such as water–glycol mixtures or fuels, and different structural materials such as aluminium alloys;
- Extending the methodology to the modelling of additional heat exchanger geometries, including triangular channels, wavy plates, or the rhomboid–fractal configuration proposed by Zhuang et al. [305];
- Expanding the explored physical domains by introducing simplified structural assessments of the fins and walls or incorporating automated manufacturability checks, providing a more comprehensive design evaluation.

Beyond improving the tool fidelity, application range, and overall completeness, significant opportunities also stem from its early integration with potential applications. In particular, the key features of this tool make it well suited for rapid performance evaluation of compact heat exchangers, preliminary design exploration, and numerical integration within optimisation workflows for component design. Additionally, the methodology shows strong potential for integration with additive manufacturing design pipelines, enabling accelerated transitions from numerical concept development to rapid prototyping.

In the next chapter, some of these research directions are further investigated, focusing on the integration of comprehensive and rapid heat-exchanger simulation for early design optimisation.

Chapter 6

Heat exchanger optimisation

The results presented in Chaps. 3 and 4 demonstrated the significant influence of heat exchangers on the performance, feasibility, and mass characteristics of aerospace TMS. Within this context, the development of rapid, system-oriented heat exchanger optimisation strategies remains an important research direction in the thermal management literature.

In Chap. 5, a robust and computationally efficient methodology for the numerical simulation of heat exchangers was therefore proposed, developed, and validated. This unified and flexible simulation framework constitutes the core element linking a specified heat exchanger geometry to the corresponding simulation of its performance.

In this sixth and final chapter, this baseline framework is employed to drive the design optimisation of the reference air–coolant and fuel–coolant heat exchangers considered in this dissertation, with the objective of maximizing their thermal performance per unit of added system mass.

6.1 State of the art

Design optimisation methods can be broadly classified into three main categories [334]: size optimisation, shape optimisation, and topology optimisation (TO). Size optimisation represents the most straightforward approach and focuses solely on adjusting predefined feature dimensions. Shape optimisation extends this concept by allowing

the selection and refinement of geometries from a predefined set. Finally, topology optimisation employs more advanced techniques capable of generating entirely new and unconstrained layouts, offering potentially higher performance improvements. However, these methods generally require substantially higher computational resources and still encounter difficulties when addressing interface flow behaviour, multi-flow arrangements, and multi-objective problems. Nonetheless, integrating these approaches with Additive Manufacturing (AM) is essential to fully exploit the potential of HE design optimisation, as discussed by Fawaz et al. [334]. The extensive review by Kaur et al. [290] further supports this perspective, summarizing recent advancements in AM-supported HE design and development. Likewise, the work of Careri et al. [291] emphasises the relevance of combining AM and design optimisation in aerospace applications, where compactness and weight reduction are critical requirements.

Sensitivity analyses and parametric studies represent the most straightforward techniques for preliminary size and shape optimisation. These approaches provide early insight into the influence of individual design parameters on component performance but typically require manual interpretation and user intervention. Moreover, they rarely identify the true design optimum when multiple parameters or objectives are involved. For instance, Northcutt and Mudawar [304] perform an extensive parametric analysis of a finned microchannel module by independently varying channel and fin widths and heights to identify performance trends maximizing heat transfer. Their results show a significant increase in heat transfer when the air-side channel width is reduced to 0.1 mm and the height increased to approximately 2.5 mm, with similar but less pronounced trends observed for the fuel channels. Similarly, Rasouli et al. [299] conduct a preliminary parametric study of a novel 3D-printed polymeric HE to evaluate the trade-off between thermal effectiveness and pressure drops resulting from variations in channel length and width. The study identifies several feasible configurations without converging to a single optimum and demonstrates that thermal effectiveness above 80% can be achieved with air pressure drops below 100 Pa for channels 1.6 mm to 1.8 mm wide and 190 mm to 250 mm tall.

Several studies combine parametric analyses with CFD simulations to investigate complex geometries while mitigating the high computational cost of fully automated optimisation. Modi et al. [309] illustrate this approach by comparing four winglet configurations in a tube-fin HE through two-dimensional CFD simulations to maximise the heat transfer-to-pressure-drop ratio. The study evaluates performance using

the j/f ratio, where j represents the thermal Colburn j -factor and f the hydraulic Fanning f -factor. Lee et al. [310] extend this approach by performing a campaign of three-dimensional simulations for twenty-one wavy fin-tube HEs, optimising a $j/f^{1/3}$ metric against three geometric parameters: wave-fin height, wavelength, and fin pitch. By contrast, Arjmandi et al. [335] adopt a structured Design of Experiments (DoE) to guide twenty three-dimensional CFD simulations of vortex generator designs in double-pipe HEs. The resulting dataset is used to perform a parametric analysis and construct response surfaces for two key parameters, namely the vortex generator angle and pitch-to-length ratio. These maps are then combined with a Response Surface Methodology (RSM) to simplify design selection and identify near-optimal configurations numerically.

Recent research suggests integrating structured DoE with advanced techniques such as Artificial Neural Networks (ANNs), enabling higher-resolution response mapping and more effective optimisation with multiple design parameters. Abbasi et al. [315] demonstrate this approach through a campaign of thirty-six CFD simulations applied to a porous baffle concept for shell-and-tube HEs. Several configurations are evaluated by varying the number, thickness, and angle of the baffles, and the resulting dataset is used to train and validate a five-layer ANN for performance prediction. The trained network is subsequently coupled with a Genetic Algorithm (GA) to optimise the baffle geometry and trace the Pareto front between heat transfer and pressure drop. The study reports a strong linear correlation between total heat transfer and the number of baffles, with reduced free-flow areas caused by smaller baffle angles further enhancing heat transfer. However, similar trends are observed for pressure drops, requiring trade-off analysis through Pareto fronts. Sarangi et al. [316] extend this methodology to the optimisation of vortex generators in fin-tube HEs, simulating 125 winglet configurations to train an ANN and subsequently performing a GA-based optimisation.

In practice, ANNs provide powerful tools for high-resolution performance mapping and optimisation, often outperforming traditional RSM techniques. However, their development requires significant computational effort, including simulation campaigns, network training, and final optimisation. Moreover, ANN applications must be carefully managed to avoid issues such as data overfitting, anomaly sensitivity, limited generalisation capability, and low interpretability, as discussed in the review by Zou et al. [336]. The scalability of these approaches also remains

uncertain and continues to be investigated [337], as most studies in the literature restrict optimisation to only two or three design parameters [315, 316, 338].

The fields of electronics cooling and thermoelectric generation provide notable examples of response surface mapping applied to thermal design optimisation. Kleiner et al. [298] present an early study on the enhancement of conventional finned-plate heat sinks. Using computationally inexpensive physics-based models, the authors systematically vary channel and fin widths to generate a matrix of geometric configurations. The resulting simulations produce performance maps describing the influence of these parameters on thermal resistance, pressure drop, and pumping power, ultimately enabling identification of an optimal configuration. Pujol et al. [297] extend this approach to the optimisation of a thermoelectric generator, balancing heat transfer enhancement and pressure drop reduction to maximise net power output. However, both analyses consider only two design parameters, since performance map readability decreases with dimensionality while computational cost grows exponentially. Zhuang et al. [305] address this limitation by adopting an enumeration method to optimise a rhombus-fractal heat sink defined by three parameters. The algorithm progressively explores the design space by incrementally adjusting each parameter until performance criteria are satisfied. Although effective, the method remains computationally demanding and does not guarantee identification of the global optimum. Culham and Muzychka [339] propose an alternative entropy minimisation strategy, essentially a gradient-descent method, to optimise fin number, fin height, fin width, and airflow velocity in conventional heat sinks. The algorithm converges rapidly to the nearest numerical optimum, but similar gradient-based approaches typically struggle to locate the global optimum in complex response surfaces with multiple local minima and may suffer from numerical instability depending on initialisation conditions [340]. Consequently, gradient-based methods are highly effective for low-dimensional optimisation problems but become increasingly prone to suboptimal convergence as problem complexity grows, such as when transitioning from single-fluid electronic cooling to multi-fluid heat exchanger systems.

When optimising thermal components involving multiple working fluids, such as heat exchangers, genetic algorithms often represent a preferred tool, particularly when handling numerous design parameters, navigating complex design spaces, or pursuing multiple objectives simultaneously. Lim et al. [301] illustrate this concept by implementing a Multi-Objective Genetic Algorithm (MOGA) optimisation to

refine up to five design parameters of bare-tube HEs while maximizing heat transfer. The study traces Pareto-optimal frontiers for heat transfer and air-side pressure drops and compares the optimised results with those of a reference louvered fin HE. Similarly, Li et al. [341] apply a MOGA to optimise the thermo-hydraulic performance of a serrated-fin compact HE by refining four fin size parameters and the HE length. The study introduces entransy theory to quantify the heat transfer potential of the system through $E_{hv} = \frac{1}{2}mc_vT^2$, defining an entransy-dissipation-based thermal resistance $R_{hv} = E_{hv}/Q^2$. This formulation yields improved non-dominated solutions compared with those obtained using traditional Colburn j and Fanning f factors. Garcia et al. [342] instead address multi-objective optimisation of a microchannel HE through linear scalarization of the optimisation problem, as described in Eq. (6.1):

$$f(x) = \sum_{i=1}^n q_i f'_i \quad \text{with} \quad \sum q_i = 1 \quad (6.1)$$

where f'_i is the normalised objective function of a generic optimisation function f_i . This formulation reduces the multi-objective problem to a single-objective optimisation, enabling Ref. [342] to maximise performance relative to occupied volume. Multiple scalarised optimisations are performed by progressively varying the error function weights q_i from 0 to 1 with increments of 0.05, thereby tracing the Pareto frontier and linking each non-dominated solution to a corresponding scalarised objective function. Arie et al. [302] avoid repeated optimisation runs by adopting a hybrid-MOGA approach to optimise up to 12 design parameters in a manifold-microchannel HE manufactured via AM. The optimisation simultaneously maximises thermo-hydraulic performance and minimises component mass by reshaping the entire architecture rather than simply refining channel geometry. The optimised HE achieves up to a 60% increase in gravimetric heat transfer density compared with conventional designs, although further improvements are expected as AM manufacturing limits for microchannels, fins, and manifolds are reduced. However, consistent with other studies in the literature [308, 290, 291], Ref. [302] confirms that existing numerical models still tend to underestimate pressure drops in AM heat exchangers.

The work of Mekky et al. [343] extends GA-driven optimisation to the refinement of 2 500 parameters, enabling preliminary topological optimisation of serrated-fin

profiles in compact HEs. Each parameter defines a mesh node classified as solid or fluid and is iteratively adjusted to maximise a $Q/dp^{1/3}$ ratio estimated through CFD simulations, resulting in performance improvements of 45% for single-fin domains and 89% for two-fin domains. Petrovic et al. [307] also employ the Finite Element Method (FEM) to progressively refine serrated-fin profiles, introducing mesh adaptation after each iteration to improve resolution at the fluid–solid interface. The analysis exploits geometric periodicity to restrict optimisation to a two-dimensional refinement of a narrow core section, after which the resulting topology is extended to the full-scale component and validated numerically through three-dimensional CFD simulations and experimentally after fabrication via AM. Feppon et al. [311] further integrate evolutionary algorithms with advanced topology optimisation tools by combining meshed and level-set domain representations solved through FEM. The methodology is first applied to the optimisation of a single-flow finned-tube HE using two-dimensional CFD simulations and later extended to more complex two- and three-dimensional multi-flow configurations. The study demonstrates improved robustness and accuracy of the topology optimisation, particularly at fluid–solid interfaces, but remains computationally demanding, with three-dimensional optimisations requiring up to eight days of continuous computation for relatively small design domains.

Ultimately, numerous optimisation strategies emerge from the literature, with many active areas of research reflecting the growing interest in HE optimisation. A summary of the principal methods, their characteristics, advantages, and limitations, is presented in Tab. 6.1.

AM is consistently identified as a key enabling technology in this context, allowing optimisation algorithms to overcome many of the geometric constraints traditionally associated with heat sinks and heat exchangers. Most studies focus on optimising internal HE features to maximise variations of the heat transfer to pressure drop ratio, typically expressed as either j/f , $j/f^{1/3}$, or $Q/dp^{1/3}$. In contrast, relatively few works investigate reshaping the overall HE envelope or minimizing component mass and volume, despite these being key parameters in applied thermal engineering design, particularly for aerospace applications. Moreover, limited research addresses optimisation under application-oriented constraints, such as achieving specific heat transfer targets, respecting predefined design-space boundaries, or maintaining pressure drops within allowable limits.

Table 6.1 Principal methodologies applied to HE design optimisation.

Methodology	Features	Parameters	Objectives	Level	Refs.
Parametric analyses	Simple, intuitive, fast; require active analysis of results, struggle to find true optima	1–2 per opt. instance	Single (primarily): heat transfer, Δp	Size	[304, 299]
Parametric analyses + CFD	Suited for complex shapes (CFD), moderate computational costs (par. analyses); dimensionality remains limited, require active analysis of results, struggle to find true optima	1–2	Single: performance ratios (j/f , $j/f^{1/3}$)	Size / shape	[309, 310]
Response Surface Methodologies	Simple, intuitive, relatively fast, accurate response surfaces, numerically identified optima; dimensionality remains limited	2	Single: heat transfer, Δp , net power	Size	[298, 297]
Response Surface Methodologies (with CFD + DoE)	Manage complex shapes (CFD), structured comparisons (DoE), numerically identified optima (RSM); dimensionality remains limited, surrogate response surfaces	2	Single: Nu , f	Size / shape	[335]
ANN + GA (with CFD + DoE)	Structured analysis of complex shapes (CFD + DoE), accurate response surface mapping (ANN), fast simulation (ANN), flexible optimisation (GA); high upfront CFD cost; ANN risks (overfitting, generalisation)	2–3 (4+ future)	Multiple: heat transfer vs Δp	Shape	[315, 316, 338, 337]
Enumeration / progressive search	Automated, simple; computationally expensive, no guarantee of global optimum	3	Meet performance targets	Size / shape	[305]
Gradient-based methods	Automated, rapid convergence to local optimum; sensitive to initial conditions, no guarantee of global optimum	3–4	Single: entropy	Size / Shape	[339]
Genetic Algorithms (Scalarised)	Automated, robust, flexible, simplifies multi-objective optimisation to a single objective; relatively slow, requires multiple runs to trace a Pareto front.	2–4+	Single: scalarised function	Size / Shape	[342]
Genetic Algorithms (NSGA-II)	Automated, robust, flexible, tailored for multi-objective problems, returns the Pareto fronts, allows full envelope reshaping; relatively slow, does not provide a single optimum	10+	Multiple: heat transfer, Δp , entransy, density, mass	Size / Shape	[302, 341, 302]
Topology optimisation	Automated, maximum flexibility, strong performance improvements; extremely computationally expensive, challenges with accurate interface handling, challenges with multi-flow design	10^2+	Single (primarily): $Q/\Delta p^{1/3}$	Topology	[343, 307, 311]

6.2 Scope

This chapter addresses several of the aforementioned gaps, particularly the limited availability of optimisation methodologies explicitly driven by end-user performance requirements or involving the reshaping of the complete HE envelope. In contrast to most approaches that focus on isolated component metrics, like maximizing the overall heat transfer coefficient U or the j/f ratio, the present methodology targets application-level objectives such as meeting a required heat transfer rate, allowable pressure drops, or a component mass constraint. It introduces a flexible numerical framework, supporting the simultaneous and rapid optimisation of multiple heat exchanger design parameters, including the shape and primary dimensions of channels and fins and the overall heat-exchanger envelope. This capability enables geometry-adaptive optimisations in which both internal flow features and the overall component dimensions are iteratively reshaped to meet system-level targets within a predefined design space.

The proposed approach is specifically developed to facilitate the multi-objective optimisation of additively manufactured heat exchangers, here applied to support hydrogen-powered propulsion electrification of regional transport aircraft through MT-PEM fuel cells. The analysis builds upon the existing simulation frameworks and results of Chap. 5 and Ref. [10], integrating the validated heat-exchanger model within a dedicated optimisation rationale, as outlined in Fig. 6.1.

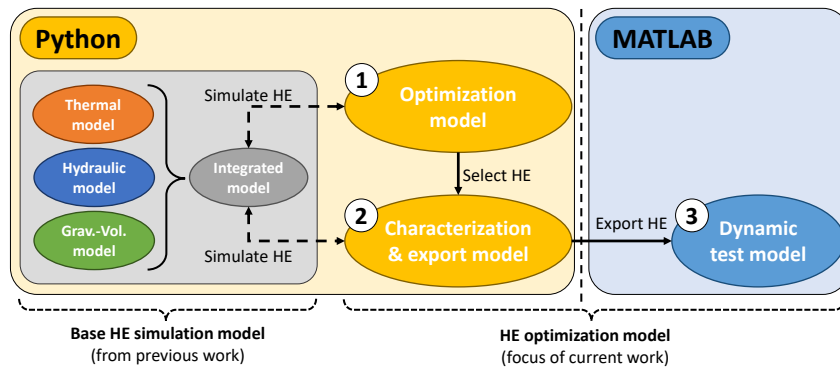


Fig. 6.1 Integration of the thermo-hydraulic–gravimetric model from Ref. [10] into the proposed genetic optimisation framework, including comprehensive component characterisation and export to alternative simulation environments for dynamic system-level analysis (i.e. MATLAB/Simulink).

The heat exchanger specifications derived for the optimised A-TMS and AF-TMS configurations in Chap. 4 define the baseline settings for the present analysis:

- A 938 kW air-coolant heat exchanger for the A-TMS;
- A 675 kW air-coolant heat exchanger for the A-TMS;
- A 365 kW fuel-coolant heat exchanger for the A-TMS;

As a first-order estimate, a cuboidal space compatible with the standard aisle dimensions of typical regional transport aircraft is adopted as the available design space for the coolant–air heat exchanger, assuming installation between the hydrogen tank and the passenger cabin [146]. This corresponds to an approximate design volume of $226 \times 226 \times 135$ cm [3, 166].

For the coolant–fuel heat exchanger, a rectangular envelope approximately matching the height and width of the fuel tank is assumed, enabling integration within the wing structural frame, as illustrated in Fig. 6.2. Accordingly, a design space of approximately $250 \times 130 \times 30$ cm is considered, although the system could alternatively be implemented as two separate modules each occupying half of this volume.

6.3 Methodology

A dedicated design optimisation framework is developed to rapidly extract refined heat exchanger geometries for a given optimisation target. The validated numerical model from Chap. 5 serves as the core simulation element of this framework, assessing the thermal, hydraulic, volumetric, and gravimetric behaviour of candidate geometries generated by the optimisation routine. For the hydraulic characterisation, the corrected form of Eqs. (5.30) and (5.47) is adopted, implementing the corrective factor $K_p = 1.3$ [10] to mitigate pressure-drop underestimation at higher Reynolds numbers. Owing to its lumped formulation, the core simulation model retains rapid steady-state evaluation of heat exchanger performance, supporting broad design exploration, batch simulations, and stochastic optimisation strategies such as GA and MOGA. Accordingly, a GA-driven framework is adopted because of its robustness and suitability for the multi-objective optimisation of numerous design parameters, as

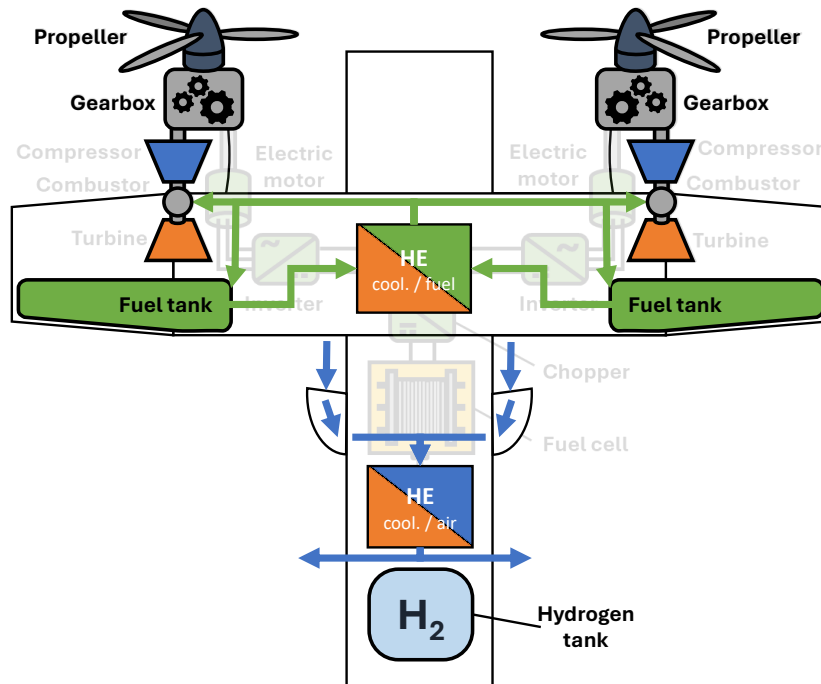


Fig. 6.2 Early HE placement concept for a hybrid electric regional transport aircraft. Components are not to scale.

demonstrated by Arie et al. [302]. The typically slower convergence of GA methods is here offset by the low computational cost of the core heat-exchanger model.

To ensure the optimisation results are accessible beyond the optimisation environment, a complete characterisation and export routine is also implemented. The performance of the non-dominated solution(s) is evaluated across multiple flow regimes and compiled into performance maps suitable for export to other simulation platforms. Fig. 6.3 illustrates the full simulation framework, from design optimisation to geometry characterisation, result export, and final integration into the dynamic simulation model.

The key steps of this workflow are described in detail in the following sections.

6.3.1 The genetic optimisation routine

A genetic optimisation routine is first implemented using the PyGaD Python library [344], which configures and orchestrates the evolutionary algorithm according to the functional scheme shown in Fig. 6.4.

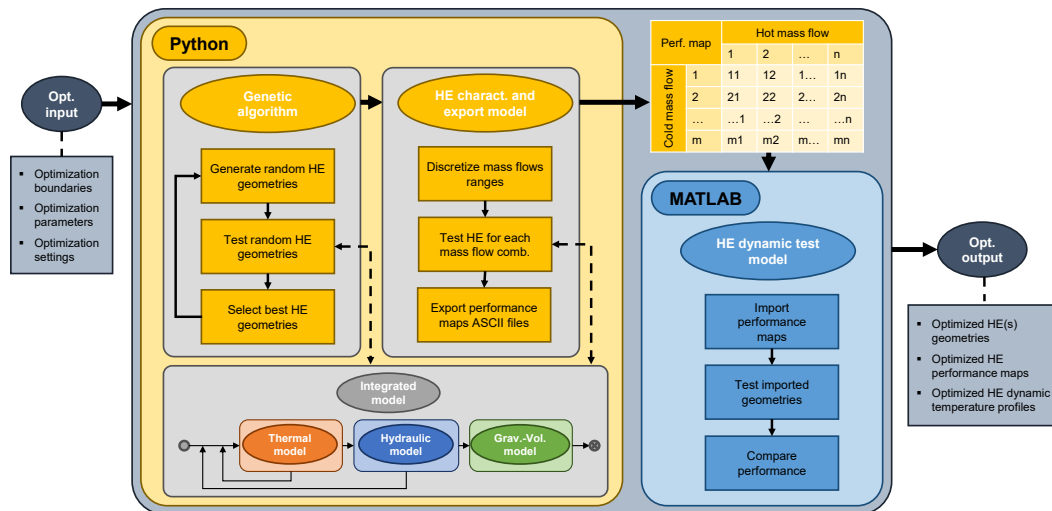


Fig. 6.3 Functional scheme of the proposed HE design optimisation loop. The representation details the main steps of the process, highlighting the interfaces between the various sub-models and simulation environments.

The routine begins by generating an initial population of randomly created solutions (chromosomes). The expected performance of each chromosome is subsequently evaluated by invoking the HE modelling script described in Chap. 5. The resulting solutions are ranked according to the defined optimisation targets. The highest-performing chromosomes are retained as reference candidates, while lower-performing designs are replaced by new solutions that combine features of the best-performing chromosomes (crossover). Subsequently, a few randomly generated variations are applied to these chromosomes to explore additional regions of the design space (mutation). Once the new population is established, a new generation is defined and the HE characterisation model is invoked again to evaluate it. This evolutionary loop is then repeated until a preset number of generations has been investigated.

Within this framework, four categories of parameters constitute the essential inputs to the optimisation: the optimisation genes, the optimisation boundaries, the optimisation targets, and the optimisation settings.

Optimisation genes

The optimisation genes constitute the collection of randomized variables linked to each chromosome. Within a design optimisation framework, every gene corresponds

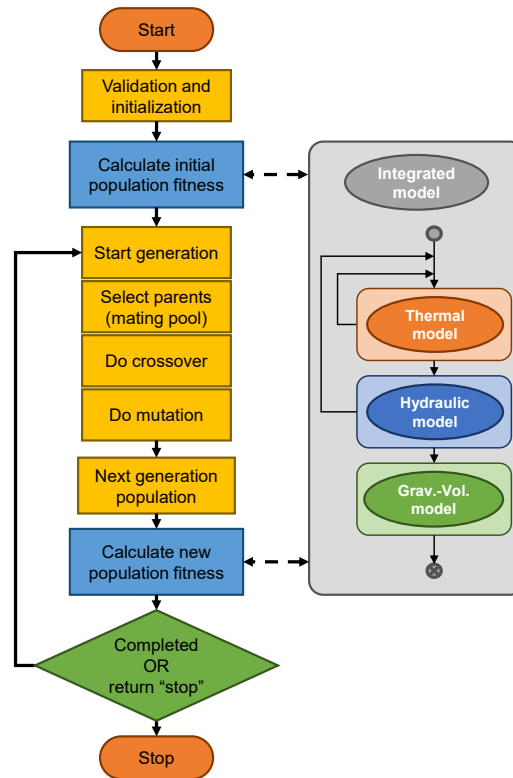


Fig. 6.4 Functional scheme of the genetic-algorithm optimisation loop.

to a particular geometric parameter that can be randomized and refined during the optimisation process. In the present study, which focuses on optimising an additively manufactured compact HE with either circular or rectangular ducts in cross-flow arrangement, a total of 10 genes is sufficient to fully characterise the component geometry, as shown in Fig. 6.5.

Accordingly, a fixed number of genes, randomly generated within predefined bounds, is assigned to each chromosome at every generation.

The genes controlling the main heat exchanger dimensions are constrained by the maximum design envelope of the application, corresponding to $226 \times 226 \times 135$ cm for the two air-HE cases and $250 \times 130 \times 30$ cm for the fuel-HE case. To prevent non-physical configurations resulting from randomly generated dimensions approaching zero, a non-null lower boundary must also be imposed. For the present study, a minimum overall HE dimension of 5 cm is assumed. Beyond ensuring numerical robustness, this constraint also reduces the effective optimisation space, accelerating numerical convergence to non-dominated solutions.

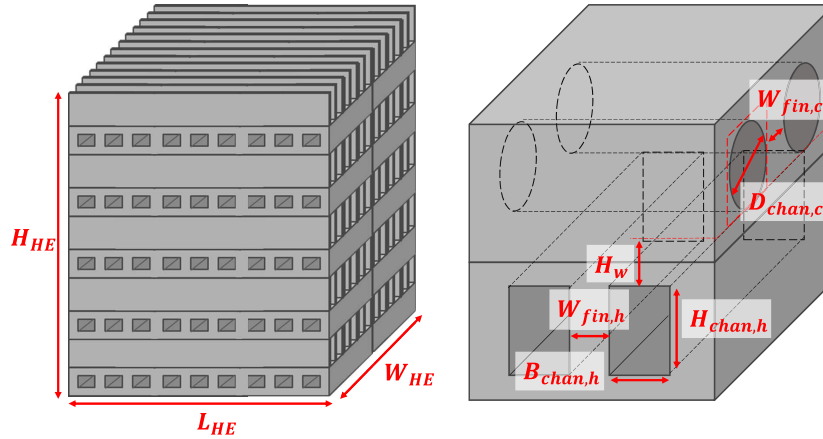


Fig. 6.5 Representation and nomenclature of every chromosome gene employed for the evolutionary design optimisation of the HE. A combined circular-rectangular channel configuration is provided, on the left, to evidence differences between optimisation parameters of the two shapes.

Assuming fabrication via AM techniques, the laser melting pool size constrains the minimum achievable thicknesses of internal fins and walls. Considering a representative melt pool diameter of $160\text{ }\mu\text{m}$, comparable with the values reported by Kempen [333], Minetola et al. [345], and da Silva et al. [11], a minimum fin and wall thickness of 0.17 mm is adopted. An upper boundary of 10 mm is instead imposed for the fin thickness.

Accordingly, a minimum rectangular channel width and height of 0.84 mm are assumed, corresponding to a minimum design feature size of $1 \times 1\text{ mm}$ reduced by the melt pool diameter. The maximum channel width is set equal to the corresponding design space width, while the maximum channel height is limited to approximately $1/3$ of the design space height for configurations featuring an odd number of channel layers (3, 5, 7, ...), and to $1/2$ of the design space height for even-layer layouts (2, 4, 6, ...).

Tab. 6.2 summarises all genes considered in this study and their respective value ranges.

The combination of all genes in a chromosome defines the randomly generated geometry. However, logical validation checks need to be applied when translating the chromosome genes into the actual geometrical parameters used for performance evaluation within the integrated HE model. These checks prevent randomly generated geometries from becoming unfeasible, when a channel base length extends beyond

Table 6.2 Genetic algorithm chromosome genes.

Gene name	Symbol	Lower bound [mm]	Upper bound [mm]
Main dimensions			
HE height	H_{HE}	50	1350 (air-HE) / 300 (fuel-HE)
HE length	L_{HE}	50	2260 (air-HE) / 1300 (fuel-HE)
HE depth	W_{HE}	50	2260 (air-HE) / 2500 (fuel-HE)
Internal dimensions			
Separating-wall thickness	H_{w}	0.17	10
Hot-side channel height (even layers)	$H_{\text{chan,h}}$	0.84	675 (air-HE) / 150 (fuel-HE)
Hot-side channel height (odd layers)	$H_{\text{chan,h}}$	0.84	450 (air-HE) / 100 (fuel-HE)
Hot-side channel width	$B_{\text{chan,h}}$	0.84	2260 (air-HE) / 1300 (fuel-HE)
Hot-side fin width	$B_{\text{fin,h}}$	0.17	10
Cold-side channel height (even layers)	$H_{\text{chan,c}}$	0.84	675 (air-HE) / 150 (fuel-HE)
Cold-side channel height (odd layers)	$H_{\text{chan,c}}$	0.84	450 (air-HE) / 100 (fuel-HE)
Cold-side channel width	$B_{\text{chan,c}}$	0.84	2260 (air-HE) / 2500 (fuel-HE)
Cold-side fin width	$B_{\text{fin,c}}$	0.17	10

the corresponding HE dimension, or inconsistent, such as having more rows of channels than the heat exchanger dimensions could fit. Analogous validations also ensure that the height of a single plate does not exceed the overall HE height gene, and that the assembled plate stack remains within the total exchanger height.

Eq. (6.2) prevents incompatible gene combinations from producing unfeasible channels whose base length exceeds the available heat exchanger width:

$$\begin{cases} B_{\text{chan,h}} = \min \left(B_{\text{chan,h}}^{\text{gene}}, L_{\text{HE,h}}^{\text{gene}} - 2B_{\text{fin,h}} \right) \\ B_{\text{chan,c}} = \min \left(B_{\text{chan,c}}^{\text{gene}}, L_{\text{HE,c}}^{\text{gene}} - 2B_{\text{fin,c}} \right) \end{cases} \quad (6.2)$$

where $B_{\text{chan,h}}^{\text{gene}}$ and $B_{\text{chan,c}}^{\text{gene}}$ are the randomized base channel genes, and $B_{\text{chan,h}}$ and $B_{\text{chan,c}}$ the corresponding actual channel dimensions supporting a realistic geometry. The correction effectively constrains the base channel base length to the minimum value between the gene-defined channel base and the available HE base length reduced by two fin-wall thicknesses.

Eq. (6.3), instead, prevents incompatible gene combinations from producing unfeasible single plate modules whose total height exceeds the available HE height, whenever $H_{\text{chan,h}}^{\text{gene}} + H_{\text{chan,c}}^{\text{gene}} + nH_{\text{w}} > H_{\text{HE}}^{\text{gene}}$:

$$\begin{cases} H_{\text{chan,h}} = \min \left(H_{\text{chan,h}}^{\text{gene}}, \frac{H_{\text{HE}}^{\text{gene}}}{n-1} - nH_{\text{w}} \right) \\ H_{\text{chan,c}} = \min \left(H_{\text{chan,c}}^{\text{gene}}, \frac{H_{\text{HE}}^{\text{gene}}}{n-1} - nH_{\text{w}} \right) \end{cases} \quad (6.3)$$

with $n = 3$ for even number of layers.

Where $H_{\text{chan,h}}^{\text{gene}}$ and $H_{\text{chan,c}}^{\text{gene}}$ are the randomized channel height genes, and $H_{\text{chan,h}}$ and $H_{\text{chan,c}}$ the corresponding actual channel dimensions supporting a realistic geometry. The correction effectively reduces the hot and cold channel height to ensure that a single plate module remains compatible with the overall HE height constraint. When uneven numbers of layers are considered, the n term of Eq. (6.3) is increased to 4 and the threshold changed to $2H_{\text{chan,h}}^{\text{gene}} + H_{\text{chan,c}}^{\text{gene}} + nH_{\text{w}} > H_{\text{HE}}^{\text{gene}}$ when the hot layers exceed the cold ones or $H_{\text{chan,h}}^{\text{gene}} + 2H_{\text{chan,c}}^{\text{gene}} + nH_{\text{w}} > H_{\text{HE}}^{\text{gene}}$, when the opposite is true.

Eq. (6.4) ensures geometric consistency, preventing situations in which the number of channels would exceed the available base plate width, and determines the maximum number of channels that can be accommodated within a plate module given the available base length:

$$\begin{cases} N_{h,plt,chan} = \frac{L_{HE,h}^{gene} - B_{fin,h}}{B_{chan,h} + B_{fin,h}} \\ N_{h,plt,chan} = \frac{L_{HE,c}^{gene} - B_{fin,c}}{B_{chan,h} + B_{fin,c}} \end{cases} \quad (6.4)$$

where $N_{h,plt,chan}$ and $N_{h,plt,chan}$ are the numbers of hot and cold channels per plate. Geometrically consistent HE base lengths ($L_{HE,h}$ and $L_{HE,c}$) are subsequently obtained through the inverse formulation of Eq. (6.4), reducing $L_{HE,h}$ and $L_{HE,c}$ to a value equal to or smaller than the original gene-defined dimensions $L_{HE,h}^{gene}$ and $L_{HE,h}^{gene}$.

Finally, Eq. (6.5) ensures geometric consistency, preventing situations in which the plates stacks would exceed the available HE height, and determines the maximum number stacks that can be accommodated within the HE given the available height:

$$N_{plates} = \begin{cases} \frac{H_{HE}^{gene} - H_w - (H_{chan,h} + H_w)}{H_{chan,h} + 2H_w + H_{chan,c}} & \text{if } N_{h,layers} > N_{c,layers} \\ \frac{H_{HE}^{gene} - H_w}{H_{chan,h} + 2H_w + H_{chan,c}} & \text{if } N_{h,layers} = N_{c,layers} \\ \frac{H_{HE}^{gene} - H_w - (H_{chan,c} + H_w)}{H_{chan,h} + 2H_w + H_{chan,c}} & \text{if } N_{h,layers} < N_{c,layers} \end{cases} \quad (6.5)$$

where $(H_{chan,h} + H_w)$ and $(H_{chan,c} + H_w)$ respectively represent a extra hot and cold layer in uneven-layer configurations, whereas $H_{chan,h} + 2H_w + H_{chan,c}$ denotes a single plate height. As in the previous case, the inverse formulation of Eq. (6.5) is subsequently employed to obtain a geometrically consistent HE heights, H_{HE} from the original height gene H_{HE}^{gene} .

Optimisation boundaries

Unlike the optimisation genes, the optimisation boundaries constitute a set of pre-defined physical parameters, whose values remain constant throughout the entire optimisation process. These define the case scenario upon which the component is

optimised and should be representative of recurrent and critical conditions of the heat exchanger usage. For the present analysis, the following boundary conditions are considered for the hot and cold fluids: inlet temperatures, fluid outlet pressures, and fluid mass flow rates, consistent with the simulation inputs for the reference model of Chap. 5.

Recalling the A-TMS and AF-TMS results of Chap. 4, two separate worst-case scenario should be considered, one per each configuration, based on a hot-day mission profile for a 300 km flight conducted at a ceiling altitude of 7600 m and followed by a 50 km alternate flight at 1000 m. For consistence, the relevant temperature trends for the two simulations are reported in Fig. 6.6, based on TMS models and assumptions from Chap. 4.

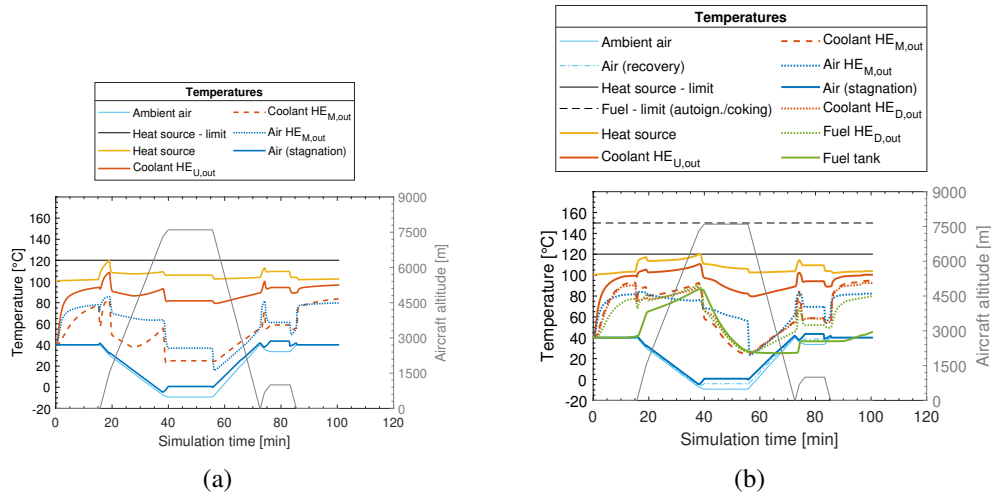


Fig. 6.6 Relevant temperature trends of optimised A-TMS and AF-TMS for a 20% hybrid-electric turboprop aircraft performing a 300 km, 7600 m main mission followed by a 50 km, 1000 m alternate flight. (6.6a) A-TMS configuration with two 0.190 m² intakes. (6.6b) AF-TMS configuration with two 0.128 m² intakes.

For the A-TMS, the early climb and take-off represent the more challenging mission segments. Accordingly, a 40 °C, 18.13 kg s⁻¹ air mass flow is assumed for the air-HE, corresponding to the captured ground airflow from two 0.190 m² intakes with a capture ratio of 0.75 at an airspeed of approximately 56.5 m s⁻¹. Consistently, a coolant mass-flow of 8.75 kg s⁻¹ at 108.7 °C is assumed for the hot side of the HE.

Conversely, for the AF-TMS, the end-of-climb is the more demanding phase of the scenario. Accordingly, a -4.5 °C, 7.15 kg s⁻¹ air mass flow is assumed for the air-

HE in this configuration, corresponding to the captured airflow from two 0.1128 m^2 , with a capture ratio of 0.75 at an altitude of 7600 m and a longitudinal airspeed of approximately 72.2 m s^{-1} . A coolant mass-flow of 8.75 kg s^{-1} at 110.4°C is assumed in this case.

For the fuel-side of the AF-TMS, a coolant mass-flow of 8.75 kg s^{-1} is retained at a reduced inlet temperature 92.7°C , consequent to the pre-cooling from the air-side of the TMS. A maximum fuel flow of 13.67 kg s^{-1} is assumed, at a 87.7°C temperature, corresponding to the maximum registered tank fuel temperature in the simulations.

Since the current system level models of Chap. 4 neglect precise modelling of pressure drops, a heat exchanger outlet pressure of 1.01325 bar is initially assumed across all simulations, both air and fuel.

Tab. 6.2 summarises all optimisation boundaries considered in this study and their respective values.

Objective functions

The objective functions define the goals of the optimisation. They process the outputs of the numerical simulation model to rank each chromosome based on its placement with respect to the objective, enabling the identification of the best-performing solutions. In the PyGAD framework [344], such objective functions are referred to as fitness functions, which the genetic algorithm aims to maximise.

For the present analysis, the minimisation of the heat exchanger mass is adopted as the primary optimisation objective. Eq. (6.6) defines a fitness formulation that promotes the reduction of the component mass within the optimisation framework:

$$f'_m = K_m / m_{\text{HE}} \quad (6.6)$$

where f'_m represents the normalised figure of merit defined with respect to a corrective factor k_m corresponding to the optimisation benchmark for the analysis. Accordingly, in this case, k does not represent a thermal conductivity. Assuming a single objective optimisation, this value can be set equal to the nominal HE masses

Table 6.3 GA optimisation boundary conditions and design targets.

Boundary name	Symbol	A-TMS	F-TMS
Integrated HE model inputs			
Air-HE coolant T in	$T_{\text{HE}_M, \text{h}}(0, 0)$	108.7 °C	110.4 °C
Air-HE coolant mass flow	$\dot{m}_{\text{HE}_M, \text{h}}$	8.75 kg s ⁻¹	8.75 kg s ⁻¹
Air-HE air T inlet	$T_{\text{HE}_M, \text{c}}(0, 0)$	40 °C	-4.5 °C
Air-HE air mass flow	$\dot{m}_{\text{HE}_M, \text{c}}$	18.13 kg s ⁻¹	7.15 kg s ⁻¹
Fuel-HE coolant T in	$T_{\text{HE}_D, \text{h}}(0, 0)$	/	92.7 °C
Fuel-HE coolant mass flow	$\dot{m}_{\text{HE}_D, \text{h}}$	/	8.75 kg s ⁻¹
Fuel-HE fuel T in	$T_{\text{HE}_D, \text{c}}(0, 0)$	/	87.7 °C
Fuel-HE fuel mass flow	$\dot{m}_{\text{HE}_D, \text{c}}$	/	13.67 kg s ⁻¹
Optimisation targets			
Air-HE heat transfer rate	$\dot{Q}_{\text{HE}_M}^{\text{target}}$	~938 kW	~675 kW
Air-HE coolant p drop	$\Delta p_{\text{HE}_M, \text{h}}^{\text{target}}$	< 50 kPa	< 50 kPa
Air-HE air p drop	$\Delta p_{\text{HE}_M, \text{c}}^{\text{target}}$	< 5 kPa	< 5 kPa
Fuel-HE heat transfer rate	$\dot{Q}_{\text{HE}_D}^{\text{target}}$	/	~365 kW
Fuel-HE coolant p drop	$\Delta p_{\text{HE}_D, \text{h}}^{\text{target}}$	/	< 50 kPa
Fuel-HE cold p drop	$\Delta p_{\text{HE}_D, \text{c}}^{\text{target}}$	/	< 50 kPa

anticipated during early system characterisation, namely 938 kg for air-HE in the A-TMS, and 675 kg and 73 kg for the air- and fuel-HE in the AF-TMS, respectively.

To this end, a heat transfer rate maximisation function is also defined, according to Eq. (6.7):

$$f'_Q = k_Q \dot{Q}_{HE} \quad (6.7)$$

where f'_Q is the normalised fitness function associated with heat transfer, defined to be directly proportional to the simulated heat exchanger performance $\dot{Q}_{HE}$ to ensure its maximisation. This contrasts with the m_{HE} term of Eq. (6.6), which appears in the denominator to enforce the minimisation of the heat exchanger mass.

To frame the optimisation within an application-oriented set-up, corrective factors are applied to the normalised fitness functions, to promote solutions that meet the design objectives, while penalizing those that deviate from the requirements of the intended use case.

Eq. (6.8) is introduced to reduce the fitness values of designs whose heat dissipation differs by more than 1 kW from the targets identified in Sect. 4.4.3. This 1 kW tolerance ensures that the denominator of the correction term never decreases below unity, which would otherwise produced increased fitness values that could compromise the genetic algorithm's ability to converge towards the true optimum. Dividing the fitness value by the square of the error ensures a strictly positive denominator while also aggravating the penalization for solutions that deviate further from the optimisation target.

$$f'_{\text{corr}_1} = \frac{f'}{(\dot{Q}_{HE} - \dot{Q}_{HE}^{\text{target}})^2} \quad \text{for: } |\dot{Q}_{HE} - \dot{Q}_{HE}^{\text{target}}| > 1\text{ kW} \quad (6.8)$$

Naturally, in case Eq. (6.7) is applied to maximise the HE heat transfer rate, the corrective formulation of Eq. (6.8) is dropped to open the exploration of diverse heat transfer rates.

Although pressure drop modelling is currently neglected in the system-level representations of the A-TMS and AF-TMS, a corrective function penalizing excessive heat exchanger pressure losses is introduced nonetheless. Evidence from the literature indicates that optimisation strategies focused solely on mass minimisation, while

neglecting pressure-drop constraints, often lead to infeasible designs or non-real optima.

Accordingly, two additional corrective factors are implemented to discourage solutions exceeding prescribed pressure-drop threshold per the hot and cold fluids, according to Eqs. (6.9) and (6.10):

$$f'_{\text{corr}_2} = f'_{\text{corr}_1} \left(\frac{\Delta p_{\text{HE,h}}^{\text{target}}}{\Delta p_{\text{HE,h}}} \right)^2 \quad \text{for: } \Delta p_{\text{HE,h}} > \Delta p_{\text{HE,h}}^{\text{target}} \quad (6.9)$$

$$f'_{\text{corr}_3} = f'_{\text{corr}_2} \left(\frac{\Delta p_{\text{HE,c}}^{\text{target}}}{\Delta p_{\text{HE,c}}} \right)^2 \quad \text{for: } \Delta p_{\text{HE,c}} > \Delta p_{\text{HE,c}}^{\text{target}} \quad (6.10)$$

For the purpose of this study, a maximum tolerable hydraulic loss of 0.5 bar is considered for both the coolant and the fuel, reducing to 0.05 bar for the air-HE.

Tab. 6.3 collects the reference optimisation targets for the corrective formulations of Eqs. (6.8), (6.9), and (6.10).

Optimisation settings

The optimisation settings constitute the final group of parameters governing the optimisation procedure. These parameters define the genetic algorithm hyper-parameters, including the total number of generations, the number of chromosomes per generation, and the number of genes within each chromosome. While the number of genes is fixed at 10, the number of generations, population size, and other hyper-parameters are progressively tuned to achieve the best compromise between convergence effectiveness and computational efficiency. In particular, the elitism rate, parent selection rate, mutation rate, and mutation strategy can significantly influence both the convergence speed of the algorithm and the ability to discover new non-dominated solutions.

For parent selection, a tournament selection strategy with a tournament size of three is adopted for single-objective optimisations. In contrast, multi-objective optimisations involving two or more fitness functions rely on a Non-dominated Sorting Genetic Algorithm II (NSGA-II). A small elitism rate of 5% is introduced to preserve the best-performing solutions while still allowing broad exploration of the

design space. Parents are selected from the top 60% of the population, ensuring that higher-performing chromosomes participate to the crossover. A mutation rate of 10% is assumed, proportional to $1/N_{\text{genes}}$ applied to each gene of every chromosome.

Tab. 6.4 summarises the key optimisation settings and hyper-parameters adopted for this study.

Table 6.4 GA optimisation hyper-parameters.

Setting	Value
Number of generations	900
Number of chromosomes	300
Number of genes	10
Elitism rate	5%
Selection rate (mating chromosomes)	60%
Selection method (single-objective)	Tournament (size = 3)
Selection method (multi-objective)	NSGA-II
Crossover type	Uniform
Mutation rate	10%
Mutation type	Random resetting

6.3.2 The characterisation and export routine

To ensure that the optimisation results can be reused and integrated into other simulation environments, a dedicated export routine computes and stores the relevant performance data of the non-dominated solutions identified by the GA. This routine performs a comprehensive characterisation of each optimised design by simulating its performance beyond the boundary conditions used during the optimisation process. The resulting data are then organized into structured performance maps suitable for export and subsequent use in system-level simulations. The rationale behind the characterisation and export routine is described as follows.

Two discretized mass-flow-rate vectors are defined, one per each fluid, spanning from a minimum non-zero value up to the maximum expected mass flow rate for the target application, increased by 10% to provide some operational margin.

For the coolant-air heat exchangers (HE_M), a maximum air mass flow rate of 40.18 kg s^{-1} is considered for the A-TMS configuration. This corresponds to the airflow captured by two 0.190 m^2 intakes during a 141 ms^{-1} cruise at the minimum altitude of 1000 m. Including the 10% margin this value increases to 44.20 kg s^{-1} . For the AF-TMS configuration, the maximum air mass flow rate becomes 27.74 kg s^{-1} , and increases to 30.51 kg s^{-1} , corresponding to the captured airflow from two 0.128 m^2 intakes operating under analogous conditions.

For the coolant-fuel heat exchanger (HE_D), a maximum fuel mass flow rate of 13.67 kg s^{-1} is assumed instead and increased to 15.04 kg s^{-1} .

Finally, a maximum coolant mass flow rate of 8.75 kg s^{-1} is considered for all cases and similarly increased by 10% to 9.62 kg s^{-1} .

The two discretized vectors are subsequently combined into a matrix representing all possible combinations of hot and cold fluid mass flow rates. The integrated heat exchanger model described in Chap. 5 is then recalled once again to simulate the performance of the optimised design for each mass-flow-rate pair within the matrix. Finally, the corresponding heat exchanger thermal effectiveness is evaluated for every simulated case according to Eq. (6.11):

$$\epsilon_{HE}(\dot{m}_h, \dot{m}_c) = \frac{|\dot{Q}_{HE}|}{|C_{\min}(T_h(0,0) - T_c(0,0))|} \quad (6.11)$$

with:

$$C_{\min} = \min(\bar{c}_{p,h}\dot{m}_h, \bar{c}_{p,c}\dot{m}_c) \quad (6.12)$$

The resulting effectiveness surface $\epsilon_{HE} = f(\dot{m}_h, \dot{m}_c)$ is then structured according to the Matlab 2-D lookup table format and exported into a dedicated .mat file. In addition, three complementary parameters are exported to complete the heat exchanger characterisation within the MATLAB–Simulink dynamic simulation model: the dry heat exchanger mass, the hot-fluid mass, and the cold-fluid mass. These parameters are key to define the thermal inertias associated with the corresponding components in the dynamic system model.

6.3.3 Integration with AF-TMS dynamic simulation model

The results obtained from the heat exchanger simulation model presented in Chap. 5, combined with the optimisation and characterisation procedures described herein, are integrated into the AF-TMS framework of Chap. 4 to enable a refined evaluation of system-level performance.

The optimised hot-fluid, cold-fluid, and dry heat exchanger masses are imported and parametrized within the dynamic system model for all heat exchangers belonging to a given configuration (A-TMS or AF-TMS). In parallel, the corresponding thermal effectiveness maps are implemented via two-dimensional lookup tables within the intermediate coolant–air (HE_M) and coolant–fuel (HE_D) heat exchanger modules. This approach ensures a higher-fidelity representation of the system behaviour, while retaining the possibility to switch back to the original approximated formulation, as illustrated in Fig. 6.7.

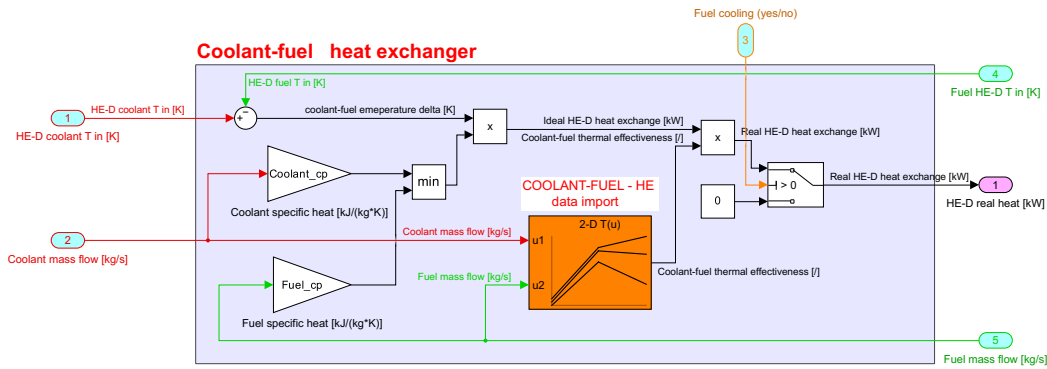


Fig. 6.7 Updated intermediate heat exchangers modules in the AF-TMS model and detail of the look-up table implementing the thermal effectiveness response surface in HE_D (coolant-fuel).

6.4 Results

The presented optimisation and export routine is herein applied to the use cases of this dissertation.

6.4.1 Methodology validation

Before proceeding with the refinement of the three reference heat exchangers of this analysis, the presented optimisation routine is tested and verified on the baseline geometry from Ref. [11]. Accordingly, a first round of optimisations is performed assuming the following optimisation parameters.

The 10 optimisation genes of Tab. 6.2 are retained, allowing to optimise both the reference design internal features and its main dimensions. To remain consistent with the orders of magnitude of the original geometry, the design space is confined to the reference dimensions of the baseline component $100 \times 100 \times 100\text{mm}$. The upper bounds of the H_{HE} , L_{HE} , W_{HE} and channel genes are also actualized accordingly, accounting for the 25 mm upper and lower HE flanges adopted in the original configuration for attaching the component to the test-rig headers. Similarly, the maximum genes values for the channel fins and intermediate walls are also reduced to 2 mm, more in line with the overall dimensions of the component.

The specifics of the T60C5 experimental test from Ref. [11] are assumed as the optimisation boundaries for this analysis and consider: a 60.2°C , 0.2665 kg s^{-1} hot water mass flow rate, and a 23.8°C , 0.0272 kg s^{-1} cold airflow.

To assess the capability of the tool of supporting component mass minimisation while retaining high thermal performance, a multi-objective optimisation is performed recalling both fitness functions of Eqs. (6.6) and (6.7). The corrected expression of Eq. (6.8) is dropped instead to allow for unconstrained heat transfer rate optimization. Moreover, to remain consistent with measured pressure ranges from Ref. [11], the minimum pressure drop is increased to 25 kPa.

Accordingly, a NSGA-II parent-selection method is adopted to support the multi-objective optimisation, while the remaining hyperparameters are retained from Tab. 6.4.

Congruent with the baseline configuration, AISI 316 stainless steel is adopted for the heat exchanger structure. The material is compatible both with additive and conventional fabrication methods, presenting an average density of approximately 8000 kg m^{-3} , and a thermal conductivity of $15\text{ W m}^{-1}\text{ K}^{-1}$. A channel surface roughness of $12.21\text{ }\mu\text{m}$ is also assumed, in line with the experimental measurements reported by da Silva et al. [11] for compact heat exchangers manufactured via additive processes.

Circular geometries are assumed for the optimisation, consistent with the features from the baseline component. However, the optimisation is also extended to encompass: square channels, rectangular channels with a maximum aspect ratio of 1 : 5, and rectangular channels of unbounded aspect ratio.

The results of the four optimisations are collected in Fig. 6.8, showing the performance distribution of all chromosomes, for each of the four channel shape optimisations.

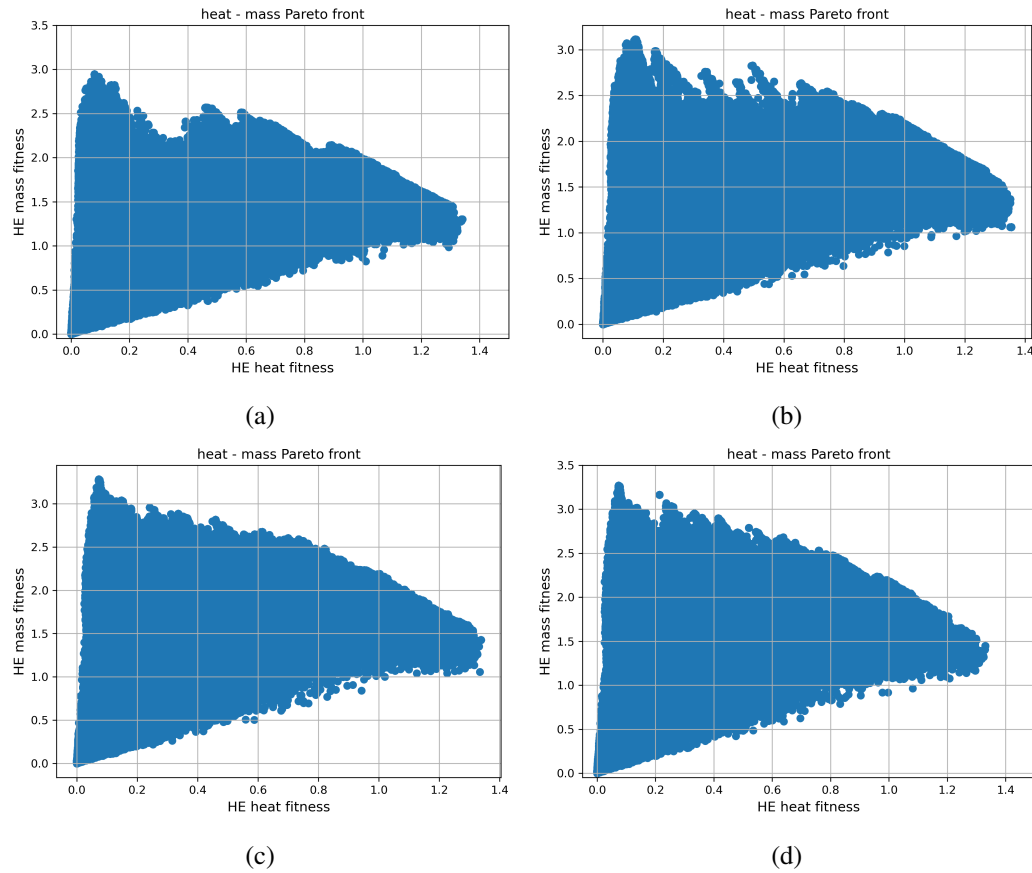


Fig. 6.8 Performance distribution of all chromosomes from the mass and heat transfer optimisation of the baseline HE geometry of Ref. [11] for: (6.8a) circular channel shape; (6.8b) square channel shape; (6.8c) rectangular channel shape, with a maximum aspect ratio of 1 : 5; (6.8d) rectangular channel shape, with an unbounded aspect ratio. The (1, 1) point indicates the baseline geometry performance.

Each distribution delineates a clear landscape, with the upper boundary of this landscape collecting the non-dominated solution of the Pareto optimal frontier.

Fig. 6.9, collects and highlights these frontiers, comparing them with the performance of the baseline geometry.

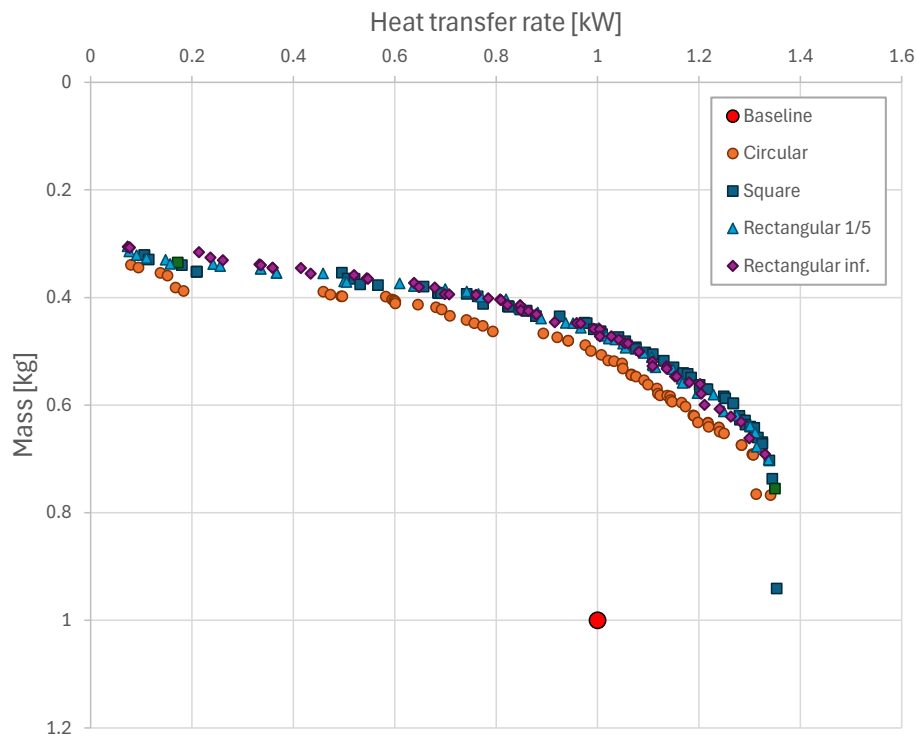


Fig. 6.9 Baseline HE [11] mass and heat transfer rate comparison with the Pareto frontiers of non-dominated solutions featuring: circular, square, rectangular (maximum aspect ratio of 1 : 5), or unbounded rectangular channels, optimised within the same design space.

The results demonstrate that the optimisation routine effectively reduces the overall component mass while maintaining or enhancing the heat transfer rate, which corroborates the proposed rationale. When the baseline circular geometries are optimised, the heat exchanger mass is reduced by more than 3 kg, corresponding to nearly 50% weight savings, while preserving the baseline heat transfer performance. Alternatively, when heat transfer is prioritized, performance increases up to 31% while still achieving a mass reduction of approximately 24%.

Even greater improvements are obtained when switching to rectangular channels. In this case, mass reductions of up to 55% relative to the baseline geometry are achieved, or, alternatively, heat transfer enhancements of up to 35% combined with mass savings of approximately 25%. However, only minor differences can be

observed between the three optimised rectangular configurations, indicating limited sensitivity to constraints on the maximum allowable aspect ratio.

Overall, these results indicate that rectangular channels represent the preferred configuration for optimising the baseline HE, particularly when minimizing component mass, yielding, on average 11% lower masses compared to circular geometries. This behaviour is attributed to a more efficient utilisation of the available volume, as rectangular cavities require less surrounding material than circular ducts of comparable dimensions. In contrast, circular geometries necessitate thicker fin roots, increasing the material volume per channel and, consequently, the overall weight.

These results are made more evident in Fig. 6.10, which derives the heat transfer rate per unit mass of the baseline and optimised solutions, expressed as $\frac{\dot{Q}_{HE}}{m_{HE}}$.

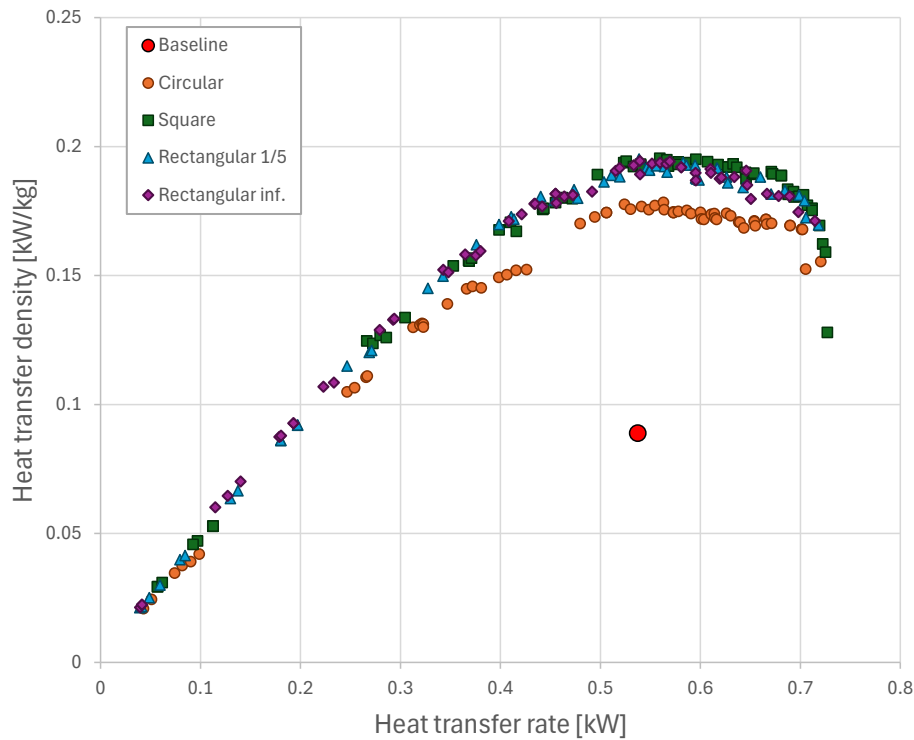


Fig. 6.10 Heat transfer rate per unit mass for the baseline HE [11] and the optimised non-dominated solutions featuring circular, square, rectangular (maximum aspect ratio of 1 : 5), and unconstrained rectangular channels.

The results indicate relatively modest values overall, with peak values on the order of 0.2 kW/kg). However, this behaviour is consistent considered the substantial mass

contribution of the component flanges which, in most cases, exceed the dimensions of the core plate stack.

A selection of representative cases is extracted from the Pareto frontiers and exported to deduce some key dependencies between the geometries, heat transfer rates, and masses. A scalarization function is applied for this purpose, as in Garcia et al. [342], to reduce the multi-objective optimisation into a set of single-objective problems. A Chebyshev scalarization is adopted over linear scalarization, since it provides greater control over the selection of non-dominated solutions, regardless of the Pareto front shape [346]. The formulation is reported in Eq. (6.13), here expressed as a maximin problem to comply with the PyGaD optimisation logic, which aims to maximise a fitness function rather than minimizing an error function.

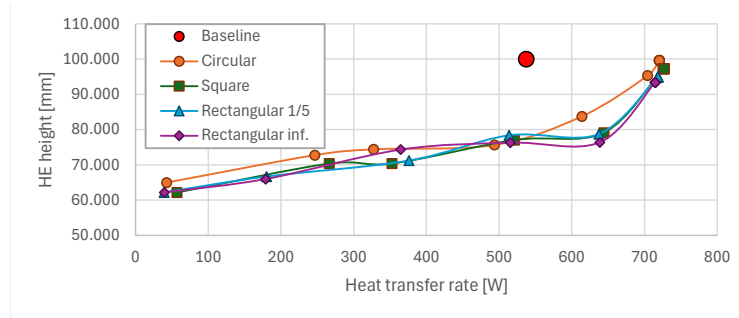
$$f_{\text{Ch}} = \max_{x \in X} \min_{i=1, \dots, n} \left\{ \frac{f'_i(x)}{q_i} \right\} \quad \text{with} \quad \sum_1^n q_i = 1, \quad n = 2 \quad (6.13)$$

Representative solutions are extracted by defining a set of 11 Chebyshev scalarization functions based on Eqs. (6.6) and (6.7). This is achieved by progressively varying the weights q_m and $q_Q = 1 - q_m$ in increments of 0.1, yielding 11 representative geometries sampled from the Pareto front of each optimisation.

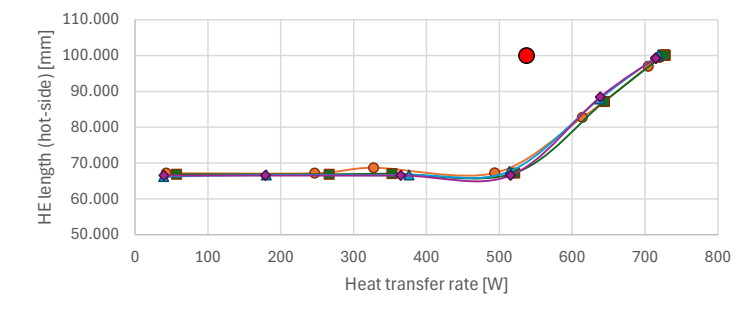
Fig. 6.11 illustrates the key dependencies between HE height, width and length and the associated component heat transfer rates for each of the four channel shapes.

The results exhibit mostly consistent geometric trends across all of the optimised main HE dimensions, with values generally ranging between 60 mm to 100 mm. These outcomes are consistent with expectations, as the HE flange inherited from the baseline geometry already occupies approximately 50 mm in each direction.

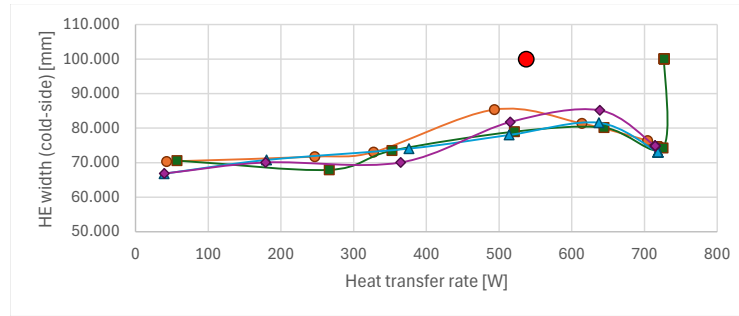
Notably, a reduced number of points is observed compared to the 11 nominal design identified via the scalarization. This occurs because the Chebyshev scalarization partitions the fitness space of Fig. 6.8 into 11 equiangular directions and selects the closest non-dominated solution to each divider. Since the lower-right region of the fitness map doesn't contain any Pareto-optimal solutions, scalarizations corresponding to $q_Q = 0.6 - 1.0$ converge to the same point, resulting in overlapping selections.



(a)



(b)



(c)

Fig. 6.11 Key relationships between the heat transfer performance of the optimised heat exchangers and their main dimensions for circular, square, rectangular (maximum aspect ratio of 1 : 5), and unconstrained rectangular channels. (6.11a) HE height. (6.11b) HE length (hot-side). (6.11c) HE width (cold-side).

Overall, the results from Fig. 6.11a indicate that the optimiser reduces the HE height when minimising component mass, while progressively increasing it as heat transfer becomes the dominant objective to accommodate a larger number of plates. Fig. 6.11c shows analogous, although less pronounced, trends for the HE width, corresponding to the frontal section exposed to the cold fluid. A single outlier is observed from this trend in the square channel geometry, optimised to achieve the absolute maximum heat transfer rate. This behaviour suggests that the optimiser limits increases in the air-side inlet area and retains relatively few channels per layer, likely to maintain high internal flow velocities and promote higher Reynolds numbers and turbulence.

A distinct trend is observed in Fig. 6.11b, characterised by relatively stable behaviour up to approximately 500 W, followed by a marked increase as higher heat transfer rates are targeted. These results indicate that the optimiser persistently maintains minimal hot-side inlet lengths when prioritising mass reduction, likely to limit the added coolant mass, resorting to larger inlet sections and increased numbers of channels only after other design adjustments have been exhausted.

Notably, these trends are consistent across all optimisation cases, regardless of the selected internal features, with the only minor difference being slightly higher HE heights for circular geometries.

Fig. 6.12 illustrates the key dependencies between the optimised internal features of the HE and the resulting heat transfer rates instead.

The results reveal representative optimisation trends for the key internal features of the heat exchanger geometry. Notably, trends from Figs. 6.12a—6.12b and Figs. 6.12c—6.12d overlap for the circular and square cases due to the enforced unitary aspect ratio of 1 : 1.

A consistent pattern of channel height minimisation can be observed across all non-dominated solutions in Fig. 6.12a for the hot channels and Fig. 6.12c for the cold ones. Lightweight designs feature taller channels, in the range of 3.5 mm to 5 mm, decreasing to less than 1 mm as the heat transfer rates exceed 500 W, with the hot channels generally converging faster. This behaviour reflects the optimiser's preference for larger channels and minimal material usage when mass reduction is prioritized. Conversely, as thermal effectiveness becomes more critical, channel heights are progressively reduced to accommodate additional plates and enhance heat transfer.

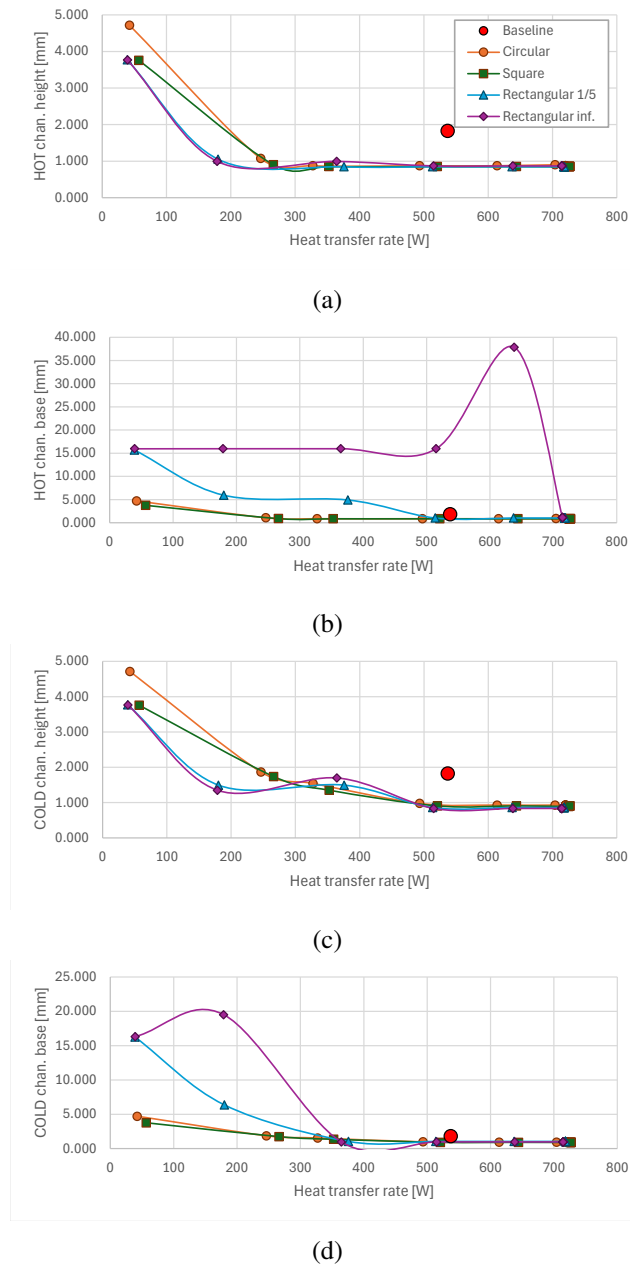


Fig. 6.12 Key relationships between the heat transfer performance of the optimised heat exchangers and their main dimensions for circular, square, rectangular (maximum aspect ratio of 1 : 5), and unconstrained rectangular channels. (6.12a) water channel height/diameter (hot-side). (6.12b) water channel base/diameter (hot-side). (6.12c) air channel height/diameter (cold-side). (6.12d) air channel base/diameter (cold-side)

Similar trends are observed for the cold channel base widths in Fig. 6.12d, with rectangular channels, unconstrained by a unitary aspect ratio, exhibiting more pronounced differences between low mass and high heat transfer designs. Specifically, for rectangular channels bounded to a maximum aspect ratio of 1 : 5, the maximum channel width reaches 16.2 mm before rapidly converging to the minimum threshold of 0.83 mm. The unbounded geometries show a peak width at 179 W instead, before converging to the same results.

Fig. 6.12b reveals a distinct trend for the hot-side channel base widths. While most cases follow the general behaviour of progressive size reduction with increasing thermal effectiveness, unbounded rectangular channels exhibit a unique pattern: an initially stable base width of 16 mm, followed by a 38 mm peak and a minimum of 1.1 mm. These results indicate that the optimiser aims to maximise the hot channel volume, likely to reduce material usage for fins and minimise structural mass. This pattern also suggests that heat transfer on the hot side is already highly effective and does not require additional fins, a consideration consistent with the hot fluid being a liquid and the cold fluid being air. This behaviour is corroborated by the 1 : 5 rectangular case, in which the channel width decreases following the hot channel height trend of Fig. 6.12a due to the imposed aspect ratio constraint, rather than following the true optimum.

Finally, correlations between heat transfer rates, wall thicknesses, and fin widths are omitted, as the results consistently converge to the lower gene boundary of 0.170 mm, corresponding to the selected AM melting pool diameter. These findings indicate that the optimiser consistently seeks to minimise these dimensions and would converge to even smaller values if allowed. Accordingly, it is shown that the melting pool diameter can effectively limit the achievable design optimum. Reducing the overall size of the laser melting pools is therefore critical, particularly considering that conventional manufacturing techniques can achieve fin thicknesses as low as 0.05 mm to 0.1 mm. However, it should be noted that introducing structural considerations to the model might offset these trends, favouring thicker walls and fins to sustain higher pressures.

6.4.2 A-TMS and AF-TMS HE optimisation

Having validated the optimisation methodology and identified the key dependencies among geometrical features, the framework is now applied to the refinement of the air and fuel heat exchangers listed in Tab. 4.3.

The reference genes from Tab. 6.2 are adopted for this analysis, along with the optimisation boundaries and objective functions reported in Tab. 6.3, and the hyper-parameter settings in Tab. 6.4. Since the primary objective of this study is the minimisation of the heat exchanger mass, a single-objective optimisation is adopted, based on the tournament parent selection method. The objective function of Eq. (6.6) governs the optimisation, corrected according to the formulations of Eqs. (6.8), (6.9), and (6.10) to ensure compliance with system-level heat transfer requirements and acceptable pressure drop constraints.

Rectangular channels are selected as the reference geometry, having previously demonstrated superior gravimetric performance. In particular, rectangular channels with a maximum aspect ratio of 1 : 5 are chosen over the unitary 1 : 1 case to allow greater design freedom. Unbounded designs are avoided instead to prevent excessively elongated channels that may prove infeasible in higher-fidelity modelling and analysis.

For a more representative analysis, a flange design inspired by Ref. [11] is incorporated. The flange extends 15 mm perpendicular to the plate stacking direction and 20 mm above and below the HE envelope, connecting to the case via a 45° ramp.

To improve the overall thermo-gravimetric performance of the components, AlSi12Mg is assumed as the base material instead of AISI 316. This lightweight aluminium alloy is suitable for both additive and conventional manufacturing, with an average density of 2650 kg m^{-3} and a thermal conductivity of approximately $120 \text{ W m}^{-1} \text{ K}^{-1}$ [347]. A channel surface roughness of $12.21 \text{ } \mu\text{m}$ is retained. Finally, since kerosene properties are not available in the CoolProp library [348], a brief property characterisation script is implemented to capture the thermal behaviour of Jet-A1 fuel using data tables from Simcenter Amesim [349].

To handle wide numerical ranges of the design-space genes, which span several orders of magnitude, an iterative approach is employed. The procedure begins with a faster, coarse optimisation using a reduced population and fewer generations applied to a subspace of the full design space. The subspace boundaries are then

progressively refined whenever optimised solution genes approach their limits. After three to five iterations, the most promising subspace is identified, and the final optimisation is conducted over this refined region, requiring approximately 20 min.

A first batch of optimisations yields promising results for the air heat exchangers in both the A-TMS and AF-TMS configurations, with each optimised design achieving less than half the mass of the baseline. However, preliminary results for the fuel heat exchanger indicate that the assumed design space is insufficient to attain the nominal thermal effectiveness of 0.7, given the high mass flow rates and the very small temperature difference, approximately 5 °C, specified for the two fluids in Tab. 6.3.

These findings highlight the limitations of the constant effectiveness assumption adopted in Chaps. 3 and 4, particularly for the fuel side, as it undergoes a progressive temperature rise. This results in reduced temperature gradients with the coolant, which not only necessitate higher mass flow rates to meet the cooling targets but also demand progressively larger heat exchangers to maintain the nominal thermal effectiveness at peak operating conditions. More broadly, these results emphasise the strong interdependence between component-level design and system-level analysis, underscoring the need for tighter integration between the two modelling approaches. In this context, an iterative coupling of system and component models emerges as a promising strategy to achieve an optimal balance between overall system performance and component feasibility.

To address these limitations, the optimisation of the fuel heat exchanger is reformulated. Rather than seeking to outperform the baseline design directly, an inverse optimisation approach is adopted, in which the objective is to match the baseline component mass while maximizing the heat transfer rate. To avoid constraining the optimisation within unfeasible performance targets, the maximum registered temperature difference between the coolant and fuel in HE_D is derived from Fig. 6.6, corresponding to approximately 53 °C.

The optimisation is carried out for each of the three reference heat exchangers, following the respective strategies outlined above. The resulting optimised designs for HE_M and HE_D in the A-TMS and AF-TMS configurations are reported in Tab. 6.5.

The corresponding response surfaces of the computed heat exchanger thermal effectiveness are presented in Fig. 6.13, together with the associated heat transfer rates at fixed temperature difference.

Table 6.5 Optimised designs for HE_M and HE_D in the A-TMS and AF-TMS configurations.

Case	Performance and external dimensions						
	Q_{HE} [kW]	V_{HE} [m ³]	Dry mass [kg]	Wet mass [kg]	H_{HE} [m]	L_{HE} [m]	W_{HE} [m]
A-TMS air-HE	938.1	0.58	207.1	285.7	701.7	819.5	1066.1
AF-TMS air-HE	675.5	0.31	116.4	145.4	461.5	1226.4	596.1
AF-TMS fuel-HE	746.4	0.035	59.4	70.3	79.6	893.0	921.1

Case	Internal dimensions						
	H_w [mm]	$H_{ch,h}$ [mm]	$W_{ch,h}$ [mm]	$W_{fin,h}$ [mm]	$H_{ch,c}$ [mm]	$W_{ch,c}$ [mm]	$W_{fin,c}$ [mm]
A-TMS air-HE	0.17	1.00	5.64	0.18	5.51	4.07	0.17
AF-TMS air-HE	0.17	0.99	5.60	0.17	8.44	5.60	0.17
AF-TMS fuel-HE	0.18	1.12	1.16	1.04	1.15	1.37	0.39

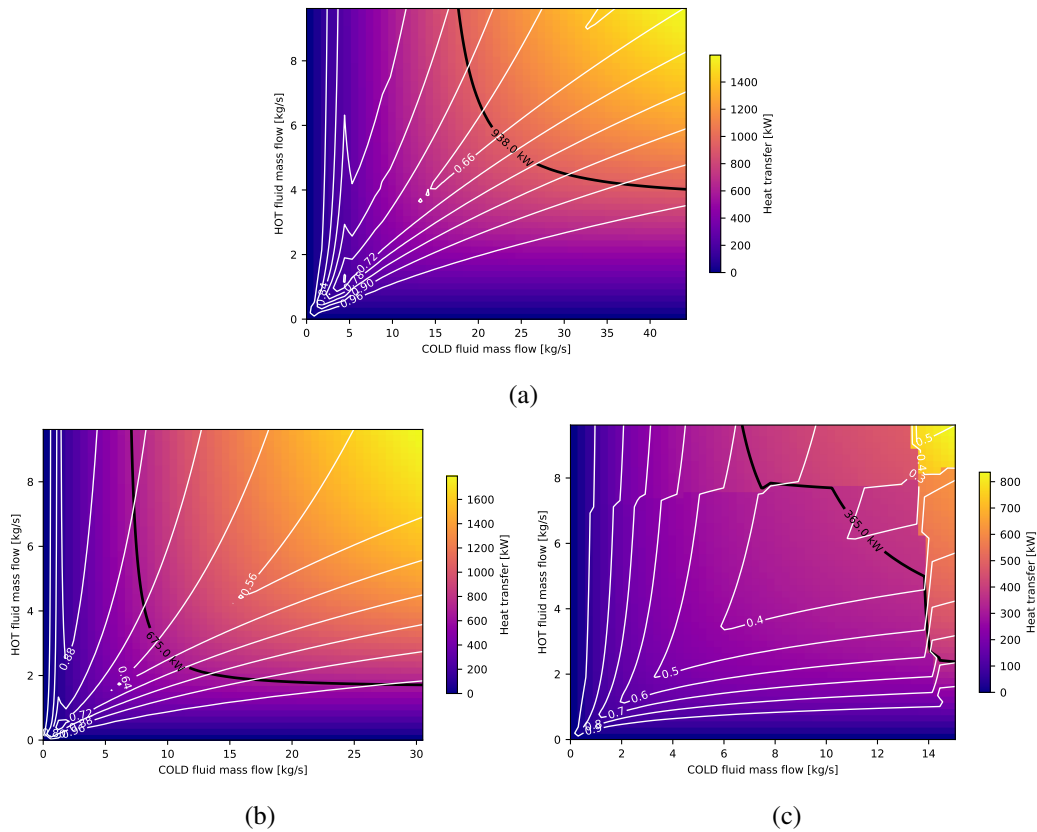


Fig. 6.13 Response surfaces of thermal effectiveness and heat transfer rates for the exported heat-exchanger optimised designs. (6.13a) coolant-air HE of the A-TMS. (6.13b) coolant-air HE of the AF-TMS. (6.13c) coolant-fuel HE of the AF-TMS.

The optimised geometries demonstrate clear performance improvements relative to the baseline assumptions presented in Chap. 3. Several factors contribute to this outcome. The high thermal conductivity and low density of Al-based alloys provide competitive thermal performance at a modest mass. The small features typical of compact heat exchanger designs ensure a high heat transfer area and dense channel packing. Finally, the relatively unconstrained design space allows for favourable component proportions from a thermo-gravimetric perspective.

In practice, several factors may moderate these first-order gains and are here accounted for to produce a more conservative estimate of component performance. First, the available design space may be more limited at higher-fidelity levels of analysis, potentially forcing less favourable geometries; this is addressed by restricting the maximum heat exchanger height to 500 mm for the air-TMS and 250 mm for the fuel-TMS. Second, the small fin and channel sections of the optimised geometries may be prone to fouling, which can significantly degrade performance. Northcutt et al. [304] report that a minimum channel dimension of 0.762 mm is required to prevent substantial clogging from coking, debris, or sand; accordingly, a conservative minimum channel height and base length of 1.5 mm are assumed, approximately doubling the recommended value. Third, practical challenges may emerge with the additive manufacturing of thin fins; these are accounted for by assuming a more conservative minimum fin thickness of 0.35 mm, roughly equivalent to double the melting pool diameter.

Additional considerations, such as the alignment of rectangular channel geometries for effective manufacturing, may further influence the final design. However, these factors lie beyond the current level of abstraction of this analysis and are not explicitly incorporated within the presented the optimisation framework.

The updated results for this new set of simulations are reported in Tab. 6.6.

6.4.3 Refined A-TMS and AF-TMS simulation

Finally, the presented thermal effectiveness maps of Fig. 6.13 are integrated into the dynamic simulation framework of Chap. 4, updated according to the scheme of Fig. 6.7.

The reference worst-case scenario defined in Sect. 4.4.2 is here revisited to assess the updated simulation model. The mission consists of a 300 km flight at an altitude

Table 6.6 Conservative optimised designs for HE_M and HE_D in the A-TMS and AF-TMS configurations.

Case	Performance and external dimensions						
	Q_{HE} [kW]	V_{HE} [m ³]	Dry mass [kg]	Wet mass [kg]	H_{HE} [m]	L_{HE} [m]	W_{HE} [m]
A-TMS air-HE	938.0	1.00	426.7	544.1	451.3	1345.2	1792.3
AF-TMS air-HE	674.4	0.43	199.3	266.6	294.3	1487.7	1119.2
AF-TMS fuel-HE	641.5	0.03	60.5	71.0	84.4	922.3	786.3

Case	Internal dimensions						
	H_w [mm]	$H_{ch,h}$ [mm]	$W_{ch,h}$ [mm]	$W_{fin,h}$ [mm]	$H_{ch,c}$ [mm]	$W_{ch,c}$ [mm]	$W_{fin,c}$ [mm]
A-TMS air-HE	0.33	1.60	3.48	0.33	9.55	5.00	0.33
AF-TMS air-HE	0.33	1.55	8.39	0.33	6.56	9.84	0.33
AF-TMS fuel-HE	0.33	1.73	1.55	1.85	2.01	1.56	0.53

of 7600 m, followed by a 50 km alternate segment at a minimum altitude of 1000 m. The results of this final iteration are presented in Fig. 6.14 for both the A-TMS and AF-TMS configurations.

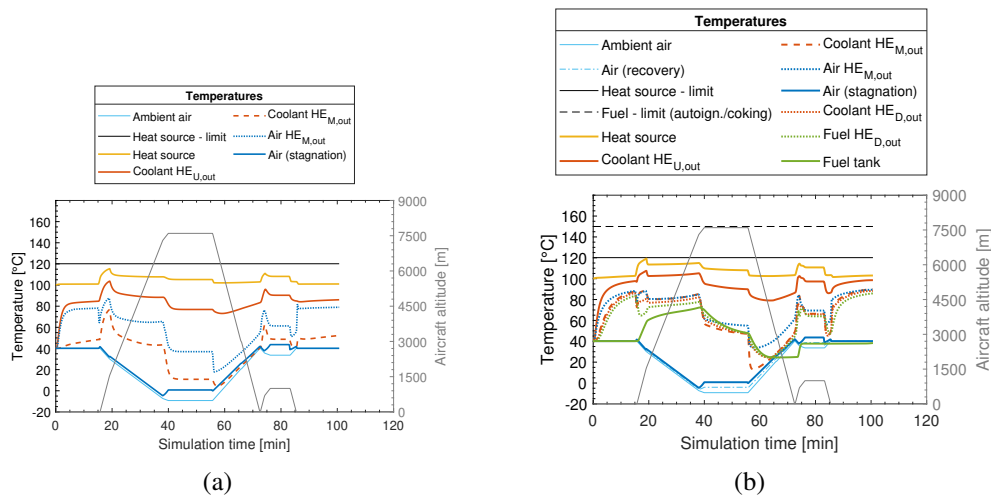


Fig. 6.14 Temperature evolution of the optimised A-TMS and AF-TMS configurations for a 20% hybrid-electric turboprop aircraft performing a 300 km, 7600 m main mission followed by a 50 km, 1000 m alternate flight. (6.14a) A-TMS configuration with two 0.190 m² intakes. (6.14b) AF-TMS configuration with two 0.128 m² intakes.

Despite the reduced peak thermal performance of the fuel heat exchanger, the integration of detailed performance maps results in improved system-level behaviour overall. Two main factors are responsible for this outcome: the refined estimation of fluid masses and the enhanced effectiveness of the heat exchangers in off-design operating conditions. The updated component and fluid masses improve the representation of the thermal inertias, resulting in the system being able to better smooth transient loads. Moreover, although the fuel heat exchanger exhibits reduced effectiveness at maximum operating regimes, its performance improves at lower flow rates, promoting heat transfer before critical cooling demand is reached. Similar considerations also apply to the air heat exchangers, which already benefit from higher nominal effectiveness.

Indeed, these improvements enable a further optimisation of the A-TMS and AF-TMS intake systems, which can now be reduced to 0.157 m^2 and 0.116 m^2 , respectively.

These final considerations consolidate the methodology developed in this dissertation and enable a refined mass and drag trade-off between the A-TMS and AF-TMS configurations. An updated set of optimised trade-offs is thus derived, following the rationale of Sect. 4.4.3, and summarised in Tab. 6.7.

In the end, the adoption of a heat-exchanger-optimised AF-TMS yields a net mass reduction of 43.4 kg (-15.1%) relative to comparable pure A-TMS configurations. Furthermore, preliminary intake drag analyses indicate a corresponding reduction in drag-related power of approximately 74.25 kW (-26.1%).

6.5 Summary and highlights

The design optimisation of heat exchangers emerges as a key research direction in addressing the increasing need for effective thermal management at the early stages of system-level design. In this chapter, a dedicated optimisation framework is developed, leveraging validated heat exchanger simulation models to refine and characterise preliminary HE concepts derived from system-level analyses.

The proposed approach contributes to addressing the limited body of literature on optimisation methodologies tailored to applied engineering design. In contrast to conventional strategies focusing primarily on maximizing intrinsic performance

Table 6.7 System masses and maximum power penalties for A-TMS and AF-TMS architectures following the integration of refined HE characterisation.

	A-TMS	AF-TMS
	Parasitic power [kW]	
Air fan	1.21	0.89
Fuel pump (1.5 kW kg^{-1})	0	12.82
Total	1.21	13.71
Net change		+12.50
Relative change		+1033%
	Added mass [kg]	
HE _M	285.7	145.4
Air fan (redundant)	1.75	1.32
HE _D	0	70.3
Fuel pump (redundant)		17.1
Fuel piping		10.0
Total	287.5	244.1
Net change		-43.4
Relative change		-15.1%
	Induced-drag power [kW]	
Air intake	284.3	210.1
Net change		-74.2
Relative change		-26.1%

metrics such as the $j/f^{1/3}$ ratio, the present work adopts a system-oriented perspective, by which the optimisation objectives are directly linked to the functional requirements of the application. To this end, corrective terms are incorporated into the objective functions to penalise solutions that fail to meet heat transfer and mass targets or exceed prescribed pressure drop limits.

When combined with the capability to continuously adjust internal geometrical features and global component dimensions within a predefined design space, the framework facilitates the extraction of optimised heat exchanger configurations that achieve high thermo-hydraulic performance while simultaneously satisfying system-level constraints. Accordingly, the proposed methodology constitutes a practical and computationally efficient tool to support early-stage design, evaluation, and integration of thermal management components in real-world engineering applications.

The analysis also details the export of optimised designs to external simulation environments through the full characterisation and storage of design-specific thermal performance maps and simulation data. These maps are subsequently integrated into the dynamic system-level model of Chap. 4, enabling a final simulation campaign in which the proposed AF-TMS configuration is assessed against baseline A-TMS layouts.

The methodology is validated against a benchmark geometry from the literature [11], demonstrating significant improvements in both thermal and gravimetric performance following optimisation. Specifically, weight reductions of up to 55% are achieved while preserving the original heat transfer rate. Heat transfer enhancements of up to 35% are also obtained alongside a mass reduction of approximately 25%. The analysis further indicates that rectangular channel geometries are particularly effective for minimizing component mass, yielding, on average, 11% lower masses compared with optimised circular configurations of equivalent thermal performance.

The study also identifies consistent geometric optimisation trends based on non-dominated solutions extracted from the Pareto frontiers. The overall HE height and cold-side length increase almost linearly with the target heat transfer rate. In contrast, the hot-side length of the HE remains minimised across most solutions and increases only when approaching maximum thermal performance. The channel heights share similar behaviours for the hot and cold sides, with larger values being favoured when minimizing mass, then rapidly converging to the minimum size limits to enable

increased plate stacking as higher heat transfer rates are required. The cold-side channel widths follow analogous, though more pronounced, trends, whereas the hot-side channels tend to favour wider geometries, reflecting the optimiser's tendency to replace fin material with fluid volume due to its reduced density.

Finally, wall thicknesses and fin widths consistently converge to the lower bound of 0.170 mm, corresponding to the assumed additive manufacturing melting pool diameter. This indicates that the optimiser systematically drives these parameters towards minimal values and would select even smaller dimensions if permitted. Consequently, reducing the characteristic size of the laser melting pool emerges as a key enabler for further performance improvements, particularly when considering that conventional manufacturing techniques can achieve fin thicknesses as low as 0.05 mm to 0.1 mm.

The methodology is subsequently applied to the reference cases of Chap. 4 to refine system-level characterisation. For the air heat exchangers, a clear performance improvement is achieved relative to the baseline configurations initially assumed. Specifically, the A-TMS air-HE mass is reduced by 69.5% compared to the original configuration, while the AF-TMS counterpart achieves a reduction of 78.5%. These significant gains result from the combination of optimised geometries, the adoption of aluminium-based materials, and the relatively relaxed design space constraints. When more conservative design assumptions are introduced, these reductions decrease to 42.0% and 60.5%, respectively.

In contrast, the fuel heat exchanger presents a more challenging design problem. Early analyses indicate that the prescribed design space is insufficient to meet the required thermal performance. Accordingly, the optimisation strategy is reformulated using an inverse approach, which maximises the heat transfer rate while constraining the solution to match the baseline component mass.

The design loop is then completed by integrating the optimised heat exchanger geometries into the dynamic simulation models of the A-TMS and AF-TMS configurations. The integration results in an overall improvement of the system-level performance, driven by the enhanced off-design heat exchanger effectiveness and the more accurate representation of fluid masses and associated thermal inertias. In particular, the higher coolant masses obtained from the optimisation increase the system's thermal inertia relative to the initial estimates, allowing more effective smoothing of the peak heat loads and facilitating thermal management. The result-

ing improvement in system-level performance enables a second iteration of system optimisations for both the A-TMS and AF-TMS, respectively leading to reduced intake-areas of 0.157 m^2 and 0.116 m^2 per intake. Compared to the initial optimisation, the overall system mass is decreased by 70–78%, while the total induced drag drops by 9–17%.

Building on the outcomes of this early design optimisation loop and its integration with system-level analyses, several directions for future work can be identified. First, the completion of the design loop through experimental validation of the optimised geometries is required to fully corroborate the proposed methodology. In particular, further integration of the framework within an additive-manufacturing-oriented design workflow represents a promising direction for future investigation.

Moreover, extending the analysis to include multi-pass heat exchanger configurations may unlock additional opportunities for enhancing thermal effectiveness while reducing component mass and volume. In this context, the incorporation of more advanced geometrical features, together with the development of a dedicated low-cost structural model, represent promising topics for further refinement of the optimisation framework, as discussed in Chap. 5.

The results obtained from the integration of the optimised HE designs with the dynamic system model also suggest that a more tightly coupled approach is essential to fully harness the interdependencies between component and system-level simulations. Accordingly, the development of an automated, integrated framework where the system-level outputs inform the optimisation process, and the refined geometries are iteratively fed back into the system model, emerges as a key step towards maximising the effectiveness of the proposed methodology.

Finally, future work should address the refinement of intake modelling within the AF-TMS dynamic framework and extend the current design rationale. With a more accurate representation of both intake and heat exchanger behaviour, pressure drop modelling can be consistently incorporated into the overall system framework presented in Chap. 4.

Chapter 7

Conclusions

The Thermal Management System is a critical aircraft subsystem that ensures safe and reliable operation by maintaining optimal temperature ranges for the crew, passengers, and equipment. Recent advances in electrification, propulsion, and onboard systems have substantially increased waste heat generation, challenging conventional cooling solutions and highlighting the need for new and enhanced thermal management strategies. This dissertation investigates the emerging thermal challenges associated with the electrification of a hybrid-electric regional transport aircraft with a 20% fuel-cell power contribution. The work analyses a selection of integrated TMS architectures that combine air- and fuel-based cooling with active thermal buffering practices to manage peak heat loads. It assesses their system-level performance, mass, and operational feasibility through the modelling and optimisation of key components, particularly heat exchangers, while delivering validated tools and methodologies for the design and refinement of next-generation thermal management systems for aviation.

7.1 Outcomes

This research investigates the thermal challenges arising from the hybridisation of regional transport aircraft, with a particular focus on fuel-cell cooling through the integration of air- and fuel-based heat sinks. The study progresses from a comprehensive review of established and emerging thermal management strategies in aviation to the development of dynamic system-level TMS models, component-

level heat exchanger design methodologies, and application-driven optimisation approaches, establishing a holistic framework for the analysis and design of advanced aircraft thermal management systems.

Initial analyses of the relevant scientific literature indicate that air-based thermal management systems remain the most widely adopted solution in aerospace applications, relying primarily on ram-air concepts to provide effective cooling with relative simplicity and high operational safety. However, these systems present notable limitations, including aerodynamic drag penalties, additional system mass, and sensitivity to operating conditions such as altitude and impinging airspeeds. Although less prevalent, fuel-based thermal management systems represent a promising alternative that utilises onboard fuel as a primary heat sink for low-to-moderate heat loads or as a secondary heat sink within more demanding thermal architectures. Their implementation requires careful consideration of key constraints, including limited thermal capacity, susceptibility to chemical degradation, and inherent flammability concerns, necessitating appropriate recirculation strategies and stringent safety measures. Despite these challenges, the application of fuel-based thermal management to hybrid-electric fuel-cell cooling remains insufficiently explored, motivating further investigation within this emerging aircraft domain.

Accordingly, a dynamic modelling framework for fuel-based TMS is developed to capture temperature and mass transients in a system comprising a heat source (the fuel cell), a coolant loop, two intermediate heat exchangers, and a fuel subsystem including the fuel-coolant heat exchanger, fuel-flow control, engines, and wing fuel tanks. The approach combines dynamic system-level modelling practices with steady-state passive heat rejection models from the skin-cooling literature to capture both fuel-tank thermal inertia and passive heat transfer effects at the tank-wall interface. The resulting framework enables the assessment of system thermal endurance, defined as the maximum duration over which a TMS with limited thermal capacity can sustain the cooling requirements of a heat source before reaching thermal constraints. A comparative evaluation of recirculating architectures demonstrates that series arrangements provide superior thermal endurance, extending the achievable operating range by 5–35% compared to parallel layouts. The study then assesses different fuel-side convection models and identifies the key dependencies between mission parameters and system thermal endurance. Simulations across representative mission profiles indicate that, under worst-case conditions, the maximum feasible degree of propulsion electrification is approximately 4.0%, with a theoretical upper

limit of 7.9% assuming ideal heat transfer at the tank floor. These results demonstrate that fuel-based cooling alone is insufficient to manage high-power fuel-cell systems but remains appealing for more moderate heat sources, such as power electronics, hydraulics, and batteries.

Building on these insights, an integrated thermal management strategy combining fuel- and air-based heat sinks is developed to exploit the complementary strengths of both approaches. The baseline F-TMS model is extended with preliminary representations of air intakes and heat exchangers, creating a unified simulation framework capable of assessing three reference architectures: fuel-based TMS, air-based TMS, and integrated air–fuel TMS. Within the AF-TMS architecture, fuel thermal inertia is leveraged to smooth thermal loads imposed by the heat source and enable significant air-intake downsizing. A two-mode fuel control strategy is also introduced for this layout, enabling fuel to absorb heat during peak-demand phases while restoring its thermal capacity during low-intensity operation. The analysis systematically evaluates the influence of key mission parameters, including altitude, range, and alternate flight segments, identifying the most demanding operating conditions for air intake minimisation across each architecture. For the AF-TMS layout, the most critical mission corresponds to a short-range, service ceiling flight followed by a low-altitude alternate segment, here represented by a 300 km mission at 7600 m followed by a 50 km alternate at 1000 m. For the reference 20% hybridisation case, the AF-TMS is shown to provide significant advantages over single-heat-sink architectures, reducing total TMS mass by 17.4% and intake area by 32% relative to the A-TMS, with only a 12.8 kW increase in parasitic fuel pumping power. Preliminary estimates further indicate potential propulsive power savings exceeding 100 kW through the associated reduction in air-intake drag. These results demonstrate that fuel-based thermal buffering can effectively complement ram-air cooling by mitigating mission-dependent thermal peaks and overcoming the limitations inherent to single-heat-sink architectures.

Having established the system-level performance of F-TMS, A-TMS, and AF-TMS architectures, the focus shifts to the component-level characterisation of the principal heat exchangers within the A-TMS and integrated AF-TMS concepts. To this end, a physics-based modelling methodology for compact, additively manufactured heat exchangers is developed to accurately predict their thermal, hydraulic, volumetric, and gravimetric characteristics. The resulting framework provides a robust, flexible, and computationally efficient tool suitable for integration into early-

stage design workflows and optimisation routines. To balance modelling fidelity with computational efficiency, the thermal and hydraulic domains are represented using lumped formulations, in which the heat transfer rate is evaluated from the average dimensionless temperature difference between the hot and cold fluids (ϕ). This approach enables the efficient simulation of the heat exchanger performance while maintaining modelling flexibility and satisfactory accuracy relative to alternative discretised methods. Moreover, the framework extends existing approaches by introducing corrective strategies that account for manufacturing-induced geometric deviations, particularly dross formation on channel ceilings. These corrections significantly improve the prediction accuracy of hydraulic losses at higher Reynolds numbers, reducing pressure-drop deviations from 26.4% to below 11.4%. The resulting model is finally validated against experimental data from the literature, demonstrating strong agreement for both thermal and hydraulic predictions and establishing a consistent numerical relationship between heat-exchanger geometry and overall performance.

Subsequently, a dedicated heat exchanger design optimisation framework is developed by coupling the validated modelling approach with a genetic algorithm to iteratively refine candidate geometries according to application-specific requirements. The methodology addresses the relatively limited body of literature on optimisation strategies for applied engineering design by adopting a system-oriented perspective. Rather than optimising isolated metrics, such as the j/f ratio of individual geometric features, the objective functions are directly linked to system-level requirements through corrective terms that penalise solutions failing to satisfy the prescribed heat transfer, mass, and pressure-drop targets. Additionally, the framework simultaneously optimises both the overall heat exchanger dimensions and the internal geometric features within the prescribed design space, enabling complete component reshaping rather than solely refining of the flow passages. The methodology is first validated against a reference case from the literature, demonstrating mass reductions of up to 55% and increased maximum heat transfer rates of up to 35%. The optimisation results reveal consistent design trends, with rectangular channels being generally favoured over circular ducts when minimising component mass. At the component scale, the overall heat exchanger dimensions exhibit a clear design trade-off, decreasing as lower mass is prioritised and expanding as higher heat transfer rates are required. Conversely, the internal geometric features follow the opposite trend, favouring larger flow passages to minimise mass and smaller passages to

maximise heat transfer, with only minor exceptions. Wall thicknesses and fin widths systematically converge towards the minimum values permitted by additive manufacturing constraints, highlighting the critical influence of manufacturing limitations on achievable performance. Finally, the optimisation framework is applied to the reference air and fuel heat exchangers of the A-TMS and AF-TMS architectures considered in this dissertation, achieving substantial improvements in both thermal and gravimetric performance, with mass reductions of up to 78% relative to early design estimates.

The integration of the optimised heat exchanger designs into the AF-TMS dynamic model completes the component-to-system design loop, improving the fidelity of system performance predictions. A resulting increase in the coolant thermal inertia improves peak heat-load smoothing and the overall thermal management performance, while detailed thermal effectiveness mapping enhances the prediction of off-design behaviour. These improvements enable a further iteration of intake optimisations for both the A-TMS and AF-TMS configurations, achieving additional intake size reductions compared with the initial designs. This newly refined AF-TMS configuration reduces the total system mass by 63.9% and the intake area by an additional 9.4% relative to first design iterations. Compared with the baseline A-TMS, this optimised AF-TMS maintains a 15.1% lower total mass and a 26.1% reduction in intake-induced drag. These results highlight the strong interdependence between component-level optimisation and system-level performance, demonstrating that tightly integrated design approaches are essential to fully exploit the potential of advanced thermal management architectures.

Collectively, the results presented in this dissertation advance the state of the art in hybrid-electric aircraft thermal management. They demonstrate the feasibility and limitations of fuel-based TMS, quantify the benefits of integrated air–fuel architectures with active thermal buffering, and provide validated tools for heat exchanger modelling and optimisation. The work establishes a practical and scalable methodology for early-stage system design, providing a clear pathway towards the development of more efficient, lightweight, and mission-adapted thermal management solutions for next-generation aircraft.

7.2 Future work

Despite the comprehensive modelling, optimisation, and system-level analyses presented in this dissertation, several research directions remain open for further investigation to improve the accuracy, versatility, and technological maturity of the proposed thermal management systems. In particular, the dynamic fuel-system model still relies on simplified representations of convective heat transfer within the fuel tank. Higher-fidelity models accounting for internal fuel motion, interactions with structural elements such as ribs and longerons, and the combined effects of buoyancy, vibration, and sloshing should be developed to improve the prediction accuracy of tank thermal behaviour. Moreover, experimental testing and validation of both fuel-side convection mechanisms and the proposed low-lying piccolo-tube configuration are essential to corroborate the predicted performance of purely fuel-based TMSs. Indeed, while some studies have experimentally investigated F-TMS, available data on heat transfer induced by internal fuel motion remain limited, with existing research being predominantly numerical. At the system level, further developments of advanced fuel-based TMS architectures should be pursued, including multi-tank arrangements, forced-convection strategies, and active mixing approaches, to enhance their feasibility and thermal endurance.

Future studies should also address the substantial simplifications introduced in the modelling of the air intakes for the A-TMS and AF-TMS architectures through the incorporation of variable capture ratios, dynamic drag terms, and comprehensive pressure-loss models, following approaches such as those described in the ESDU 86002 report [288]. Combined with the detailed thermal and hydraulic characterisation of heat exchangers developed in this dissertation, these improvements would further enhance predictive accuracy and confidence in system-level assessments. Similar considerations also apply to the coolant loop, where a more accurate representation would benefit from improved modelling of pump characteristics, deionisation filter losses, expansion tank behaviour, overall system thermal inertia, and pressure distribution. This aspect is particularly relevant for the AF-TMS, where multiple components connected in series contribute to cumulative pressure losses and may ultimately require a reassessment of the heat exchanger optimisation boundaries.

Concurrently, adaptive and mission-aware control strategies could be incorporated into the dynamic model to regulate the instantaneous degree of electrification and thermal resource utilisation throughout the mission. In particular, the integration

of fully electric ground operation with complete fuel-based cooling represents a promising opportunity to improve overall thermal management efficiency during low-power phases, provided that the associated thermal constraints are appropriately addressed.

Numerous research directions also remain open at the component level, particularly when integrated with the proposed design optimisation framework. These include extending the modelling approach to multi-pass heat exchanger configurations and exploring alternative geometries, such as triangular channels, louvred fins, serrated fins, geometric patterns, and other advanced architectures compatible with the reference plate layout. The incorporation of preliminary structural analyses, together with automated verification of manufacturing constraints, represents a promising pathway to improve the robustness and effectiveness of the optimisation process, particularly for additively manufactured components. Furthermore, experimental validation of the optimised designs will be essential to further validate the modelling framework, corroborate numerical predictions, and refine the proposed corrective strategies for manufacturing-induced geometric deviations.

Additionally, exploring complementary and emerging cooling technologies, such as phase-change materials, nanofluids, air-skin cooling concepts, and hydrogen-based approaches, may provide further enhancements in overall system effectiveness while reducing mass and aerodynamic penalties.

Ultimately, the tighter integration of component- and system-level simulations represents the next critical step in the proposed design methodology. In this framework, system-level requirements guide component optimisation, while refined component characteristics are iteratively fed back into the system model to improve performance predictions and enable more informed design decisions. Pursuing these research directions would not only expand the feasibility of fuel-cell cooling solutions for hybrid-electric aircraft but also establish a generalisable framework for the integrated design and optimisation of advanced thermal management systems, supporting the development of next-generation aviation technologies.

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