

Methods for Decarbonized Aeroengine Combustor Design

A Thesis Presented for the
Master of Science
Degree

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Robert Tate Prater
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This thesis presented by Tate Prater is dedicated to his family.

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Abstract

There exists an urgent need to decarbonize the civil transportation sector in order to address global warming concerns and its effects. Alternate and sustainable fuel types for combustion systems must be pursued to mitigate the dependency on fossil fuels for power generation. Possible alternative options for aeroengine combustion and propulsion include batteries or hydrogen fuel for fuel-cells or thermal-powered gas turbines. More specifically, thermal-powered aeroengines with premixed flame regimes can promise ultra-low emissions. Future aeroengine combustors operating in highly swirled premixed mode with hydrogen/air mixtures are central to this thesis. Prior to the implementation and design of such systems, premixed combustion conditions must be correlated with gas turbine engine performance parameters. This thesis aims to address the gap by introducing a procedure to calculate relevant combustor operating conditions for the retrofit of aircraft engines in premixed mode with hydrogen/air mixture. The pursued method, a combustor performance map, leads to six key requirements: (1) the thermal power for the engine, (2) the mass conservation into the combustor, (3) the momentum conservation into the combustor, (4) the combustor energy budget, (5) the compressor-combustor-turbine power balance, and (6) a coupling relationship. The six requirements are obtained based on the derivation from fundamental conservation laws which are uniquely applied to pertain to a premixed combustor architecture. Whereas hydrogen fuel is considered, any fuel for a premixed combustion system can be determined with the present method. To obtain the performance map, thermodynamics-based turbomachinery performance and chemical kinetics data are utilized to populate the governing conservation equations. As these two data sets provide combustor boundary conditions as well as premixed hydrogen flame characteristics, the overall goal is to identify realistic operating points where all six requirements coincide.

The results from the performance map analysis are complemented with computational fluid dynamics for validation purposes. The novel combustor performance map alongside the corresponding fluid dynamics simulations provide baseline methods for the development of future hydrogen premixed combustion systems.

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Introduction

Global Warming and CO₂

One of the most critical challenges facing today's society is the threat of global warming and its consequences on Earth. The expression global warming refers to the long-term heating of the Earth's surface mostly as a result of human activities [1, 2]. Examples of human activities that contribute to global heating include deforestation, livestock farming, industrialization, and most importantly, burning of fossil fuels [3]. These processes emit enormous amounts of greenhouse gases (6 billion metric tons of CO₂ in 2021 [4]) into the atmosphere which enhances the greenhouse gas effect. Although the greenhouse gas effect is essential for sustaining life on Earth, human interference is destabilizing the planet's thermal energy balance. In this context, the most important greenhouse gases include carbon dioxide, nitrous oxide, methane, fluorinated gases, and water vapor. While these gases are naturally produced throughout the environment, human activity has greatly increased their concentrations within the atmosphere [5]. Specifically, carbon dioxide is known to be the largest contributor to global warming as civilization relies on the burning of fossil fuels for energy production [3].

In less than two-hundred years, human activities have increased carbon dioxide content in the atmosphere by fifty percent [6]. As seen in Fig. 1, global atmospheric carbon dioxide content is closely correlated to human emissions. Additionally, since 1960, the molecular presence of carbon dioxide at altitude has increased by over one hundred parts per million. Given these findings, fossil fuel burning has been identified worldwide as causing global temperatures to rise above natural levels.

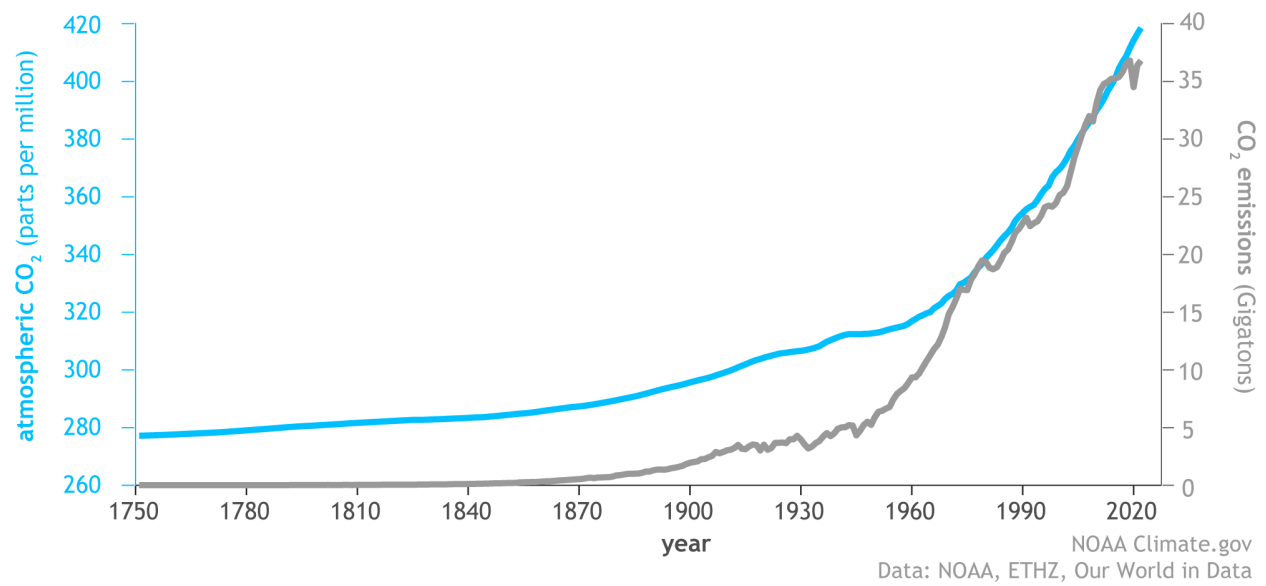


Figure 1: Global atmospheric carbon dioxide compared to annual emissions. The carbon dioxide in the atmosphere (blue line) has increased along with human emissions (gray line) since the start of the Industrial Revolution in 1750 [7].

Based on observations from the Global Monitoring Laboratory in 2021, carbon dioxide contributed to nearly two-thirds of the total heating influence of human-produced greenhouse gases [7]. In order to combat the risks of global climate change due to carbon dioxide production, government agencies across the globe are allocating time and resources towards carbon-free systems related to agriculture, transportation, and power production. For example, the Net Zero Coalition, established by the United Nations, consists of more than seventy countries that have committed to net-zero greenhouse gas emissions by the year 2050 [8]. This long-term goal requires groundbreaking advancements in technology as well as the utilization of alternate fuel sources for transportation and power production. In this thesis, the decarbonization of the civil aviation sector is the main focus as contemporary airliners and aeroengines make notable contributions to global carbon dioxide emissions [9].

To date, high-performance turbofans operate with hydrocarbon fuels that produce carbon dioxide, nitrous oxides, and soot during the combustion process. While civil aviation accounts for approximately 2.5% of global carbon dioxide emissions [10], the affects of nitrous oxide, water vapor, and soot production at high altitudes are less documented and can also play a role into the greenhouse gas effect [11]. Although aircraft manufacturers strive to reduce fuel consumption by improving engine module efficiencies, optimizing aerodynamics, and reducing overall aircraft weight, these measures cannot fully eliminate the pollutant emissions. Alternative methods of aircraft power production must then be pursued [12]. The work in this thesis focuses on retrofitting aeroengines to operate with an alternative clean fuel, hydrogen. The use of hydrogen fuel in aeronautic propulsion systems has been explored in the past for both civil and defense applications [13, 14, 15, 16], but considerable advancements in research and development are required prior to future deployment, especially for premixed hydrogen-air combustion systems. Today, engine manufacturers and organizations across the industry have recently allocated efforts towards the development of hydrogen propulsion systems in order to decarbonize the civil aviation sector [17, 18].

Hydrogen is a promising candidate for the decarbonization of the aviation industry with the potential to eliminate carbon dioxide while also lowering nitrous oxide levels when

operating in premixed mode [19]. Specifically, combustion of hydrogen fuel with air in the fully premixed regime is a long-term goal for clean propulsion systems [20]. Highly-swirled premixed flames can feature the lowest attainable nitrous oxide levels compared to other existing injection schemes such as micro-mix injectors, see Dahl and Suttrop[21], or lean direct injection (LDI) [22]. Further, fully premixed flames promote: (i) fuel-burn reduction, (ii) elimination of soot-production, and (iii) more control of engine’s exhaust composition. For implementation into an aeroengine, however, there are various components that must first be investigated such as premixing fuel-air injector systems, cryogenic hydrogen fuel storage and supply, and overall premixed combustion dynamics and flame stability [23].

Contemporary Turbofan Engines

It is worthwhile to review one of the most commonly employed gas turbine engines in the civil aeronautic transport sector, the turbofan engine as seen in Fig. 2. The characteristics and descriptions of a turbofan engine are now discussed. Turbofan aeroengines are classically described as gas turbine engines with a bypass stream that surrounds the core. These engines are known to be the optimum choice for subsonic commercial air travel given their fuel-efficient and robust operating qualities [24]. Similarly to turbojet engines, turbofans feature three main modules in the engine core: the compressor module, the combustion chamber, and the turbine module. However, the fan unit featured at the front of a turbofan is what distinguishes this engine type from the turbojet. Designed with large inlet diameters, turbofans demonstrate increased propulsive efficiencies compared to turbojets by capturing and accelerating massive amounts of air through the fan unit. A more in-depth description of turbofan gas turbines is provided next alongside their main advantages.

A nominal turbofan architecture is presented where the engine stations are numbered following the American Institute of Aeronautics and Astronautics (AIAA) standard in Figure 2. Station numbering is a standardization process for engine performance nomenclature as each number corresponds to individual turbomachinery components within the system. At these

locations, thermo-fluid properties can be calculated both experimentally and analytically to assess the specific characteristics of modules and overall performance of an aeroengine. A substantial portion of the turbofans enhanced efficiency can be attributed to the fan bypass flow rate which also acts to suppress engine noise. The engine bypass ratio, defined as $BPR = \dot{m}_{fan}/\dot{m}_{core}$, characterizes the flow path of the air where $\dot{m}_{fan} \gg \dot{m}_{core}$ in most civil transport aircrafts. On the other hand, the engine core air flow undergoes the Brayton cycle through compression, heat addition, and expansion inside the gas turbine turbomachinery modules.

Around the 1960s, Rolls-Royce and General Electric began introducing bypass turbojets with bypass ratios up to eight [25]. Currently, state of the art turbofan engines in the aviation industry, such as CFM International’s LEAP engine [26], feature bypass ratios up to twelve. While the optimized bypass ratio promises superior thrust capabilities and improved propulsive efficiency, the turbofan engines of today are constantly evolving with leading edge material and system technologies. The LEAP engine demonstrates these newly-developed technologies at nearly every engine module. Starting with the fan unit, the lightweight composite fan blades feature a geometric design that enables higher rotational speeds with minimal aerodynamic losses. Next, the compression system offers an overall engine pressure ratio of forty [27]. The combustor architecture includes swirl-stabilized flames which aim to reduce fuel consumption as well as nitrous oxide production [26]. Finally, the turbine module consists of composite materials and incorporates novel blade cooling schemes to further improve module efficiency. These advancements in design can be seen across the aeroengine manufacturing sector; however, as previously mentioned, they may not be sufficient in addressing the urgent need to decarbonize civil aeronautic transport. Thus, alternate fuels are required.

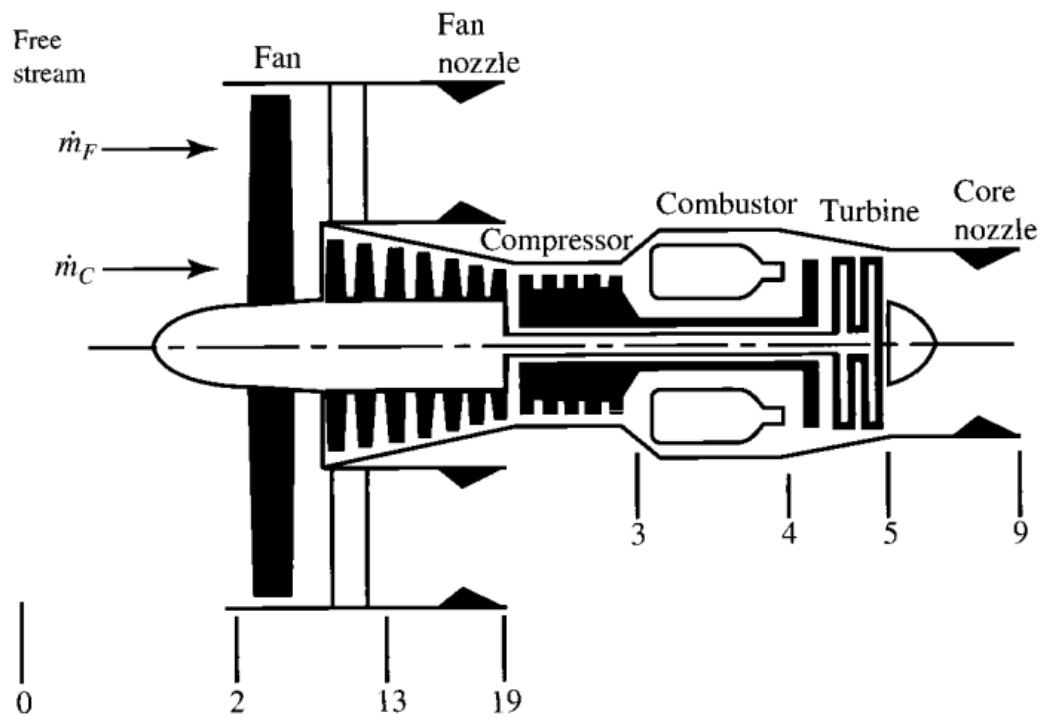


Figure 2: Turbofan engine station diagram based on AIAA standard form. Figure extracted from Ref. [24].

Aeroengine Combustor Architectures

In this section, relevant aeroengine combustors are described. Within an aeroengine core, the combustion chamber is positioned between the compressor and turbine modules with strict constraints in terms of volume and temperature limits. The high-pressure air from the compressor enters the combustor alongside fuel where chemical reactions occur and the flame sustained. This complex process requires that chemistry, aerodynamics, and mechanical design are all accounted for and linked together at the design stage. Furthermore, an aeroengine combustor must satisfy a wide range of requirements including but not limited to, weight for reducing lift and size for limiting drag impact, combustion efficiency, durability and relighting capability, uniform temperature distributions, and pollutant emissions [28]. While the implementation and design of gas turbine combustors has historically relied on experimentation, the computational breakthroughs in modeling and fluid dynamics have proven invaluable.

The most commonly utilized combustion mode in aerospace is the diffusion flame regime. A diffusion flame is characterized by the fuel and oxidizer being on opposite sides of the flame whereas a premixed flame corresponds to the case when they are mixed prior to their combustion. These flame regimes present unique attributes that must be considered when designing a burner architecture. For instance, while diffusion flames are more straightforward to implement, they are characterized by inefficient burning and high temperatures which lead to pollutant generation. However, diffusion flames are known to be less prone to combustion instability. The premixed flame regime is known to be much more susceptible to combustion instabilities but can promise a more efficient and cleaner burn. Because of its safe and stable qualities, most aeroengines combustors to date operate under diffusion flames or partially-premixed modes. The most utilized combustor architecture that relies on diffusion flames is the rich-burn quick-quench lean-burn (RQL) system depicted in Fig. 3.

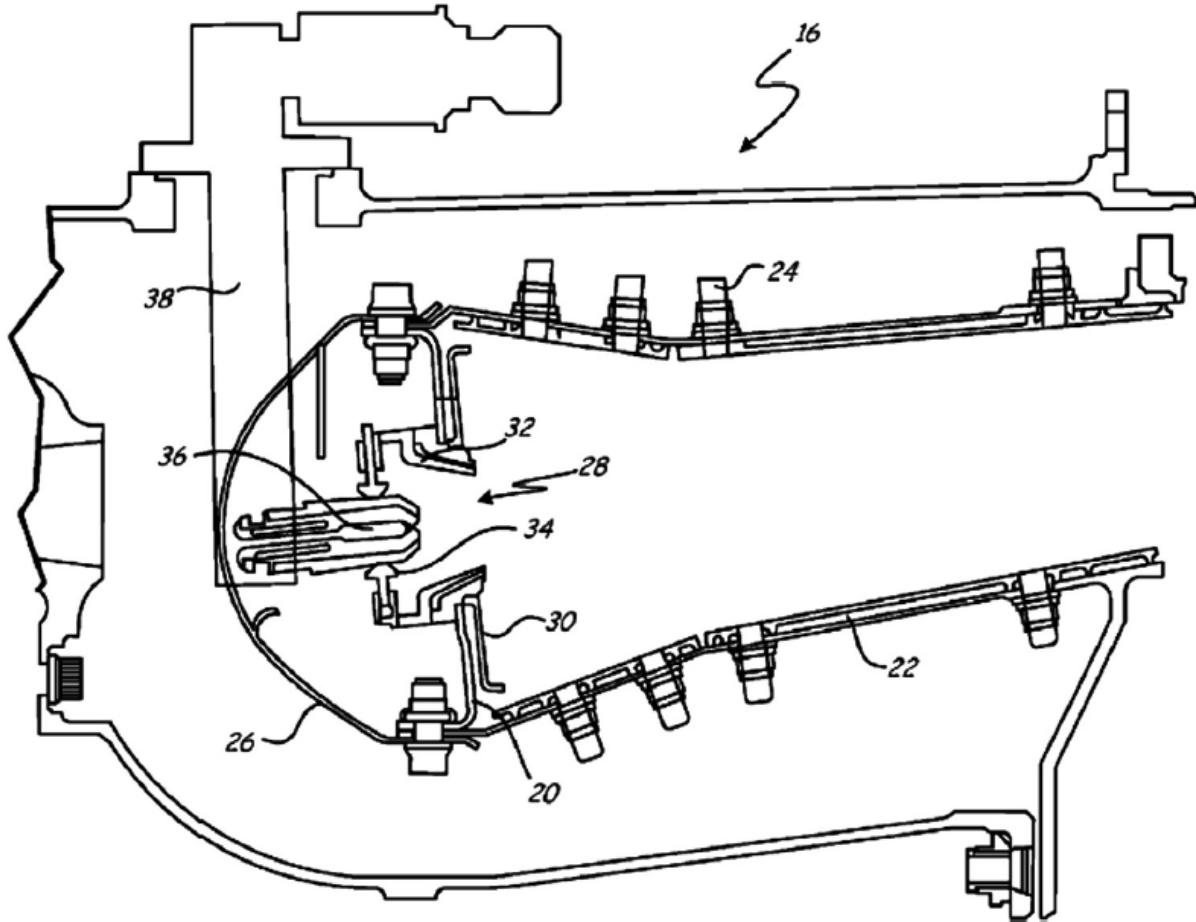


Figure 3: Jet engine RQL combustor geometry taken from Ref. [20, 29]. The dilution holes are not featured in the combustor longitudinal slice.

Figure 3 illustrates a nominal jet engine RQL combustor geometry. In this system, the combustion between fuel and air occurs in two separate locations. The most upstream location of the combustor features an overall rich combustion zone where an excess amount of fuel is provided. This excess fuel is then burned in the downstream combustor region alongside air flow from the dilution jet streams. By operating at rich conditions in the primary zone, the combustion scheme aims to reduce nitrous oxide formation by limiting the adiabatic flame temperature. However, as this zone operates in the diffusion flame regime where the reaction process occurs at the fuel/air stoichiometric ratio of one, substantial nitrous oxide production levels are reached [30]. Although this is a major design issue in terms of emissions, the dynamic stability of this combustor architecture compared to other systems leads to its reliability in contemporary aeroengines.

Moving from the diffusion flame regime towards premixed combustor systems, well-known methods include lean premixed/prevaporized (LPP), low swirl injector (LSI), and lean fully premixed (LFP) injector concepts. These combustor architectures exhibit unique designs and flow schemes that target the minimization of emissions while accounting for overall combustion stability. The LPP methodology aims for the complete vaporization of the fuel and complete mixing between the fuel and air prior to combustion. This design however raises several concerns regarding dynamic instabilities. The LSI system is another premixed concept that applies to land-based gas turbine engines only. This ultra-low emissions technology demonstrates a swirl-stabilized premixed flame that greatly reduces nitrous oxide formation. Another notable feature of the LSI's swirl-stabilized flame is the absence of an inner recirculation zone. Finally, the LFP injector concept has features of the previously defined combustor designs to target a lean fully premixed swirl-stabilized flame. This architecture consists of a sprayless system where all fuel is fully vaporized before mixing with air, and the design features dilution holes around the combustion chamber similar to the RQL geometry in contrast to the initial LPP operating without dilution holes [22]. The LFP combustor concept, presented in Fig. 4, is central to this thesis according to the design from Palies [20].

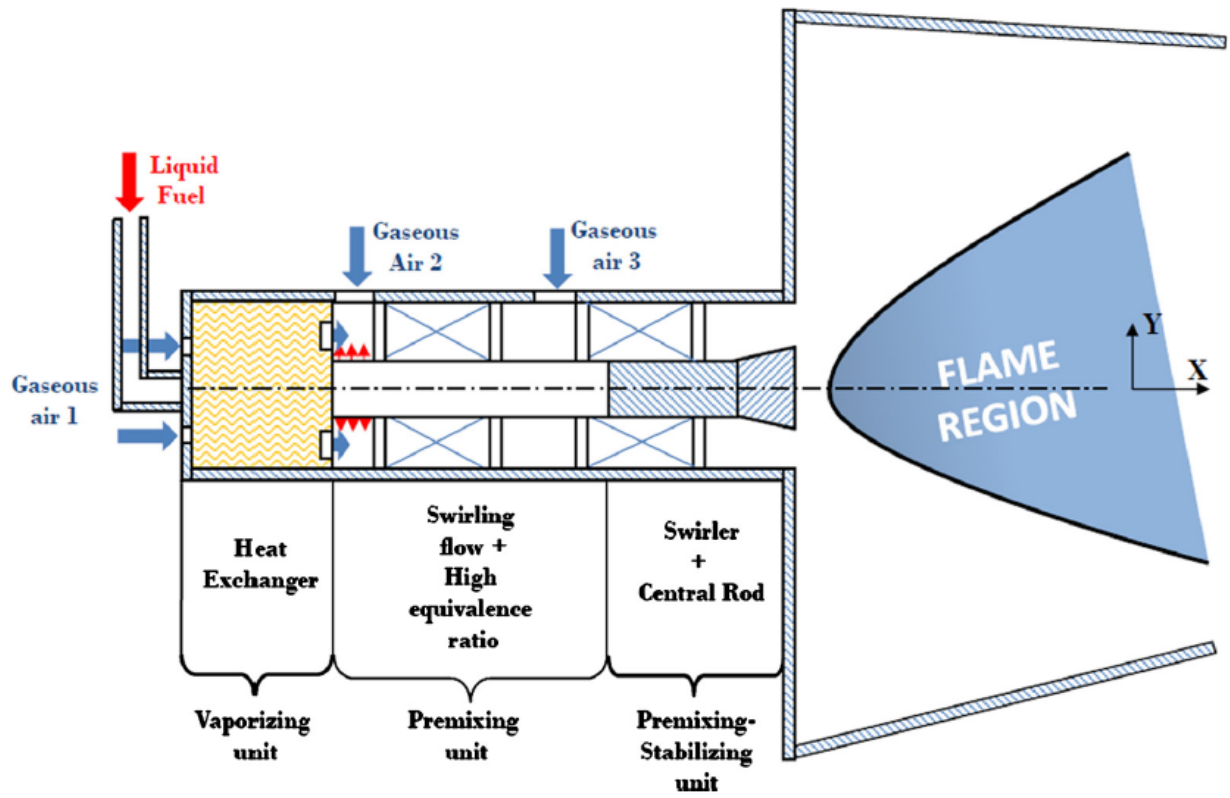


Figure 4: Schematic of the lean fully premixed injector/combustor concept introduced in Ref. [20].

Mixing Mechanisms for a Premixed Combustor

There are two major mixing processes that are described for the presently considered LFP combustor architecture. The first one consists of the mixing in the injector unit between the gaseous hydrogen stream and a portion of the air flow exiting the high-pressure compressor stage. The second one is the mixing between the dilution hole airstream with the high temperature flame products on the downstream side of the combustor. The full premixing of hydrogen and air in the injector unit targets a uniform, lean equivalence ratio in order to reduce the adiabatic flame temperature which limits the nitrous oxide formation. Prior to entering the turbine module of the engine, the burnt gas temperature from the combustion process must be reduced and the associated thermal energy from combustion translated into kinetic energy of the working fluid. The dilution airstream through the dilution holes of the combustor walls acts to reduce the combustion products temperature and increases the velocity of the flow by both mass and energy conservation.

The LFP injector architecture detailed in Ref. [31] and depicted in Fig. 4 has been conceptualized to generate a perfectly lean, swirl-stabilized premixed flame inside the combustion chamber for any fuel entering the first unit of the injector which consists of a heat exchanger. Three units are respectively at work in this configuration: the vaporizing unit (achieving liquid-fuel vaporization), the premixing unit (complete fuel/air premixing), and the premixing-stabilizing unit (flame stabilization). The first unit will feature cryogenic hydrogen fuel flowing into the heat exchanger where it will vaporize due to the heat transfer from the high-pressure compressor air through the heat exchanger's walls. In the next two units, the obtained gaseous hydrogen fuel will mix with high-pressure compressor airflow to achieve the targeted lean equivalence ratio. Finally, the chemical energy of the lean premixed mixture is converted to thermal and kinetic energy through a swirl-stabilized flame via the swirler unit at nearly constant pressure. Complex non-reacting and reacting flow phenomena will exist throughout the injector unit such as supercritical flow regimes, cryogenic fuel mixing with high-pressure air, and large temperature and pressure gradients that must be accounted

for in the modeling and design phases.

Extensive research documented in Refs. [32, 33, 34] has been conducted to analyze the injection of cryogenic fluids at sub-critical states into supercritical environments corresponding to elevated pressure and temperature. This type of flow regime is most common in liquid rocket engine applications but it is becoming more relevant to advanced combustion systems at take-off conditions with high overall pressure ratios, especially making use of clean fuels such as hydrogen. Beyond the critical point, a fluid can be characterized by a thermodynamic state with liquid-like density and gas-like diffusivity. During the transition from sub-critical to supercritical conditions, the fluid exhibits thermodynamic nonlinearities throughout the flow field [34]. In the context of developing and modeling steps towards advanced injection units for an aeroengine operating with cryogenic hydrogen fuel mixed with high-pressure air, accurate mass and heat transport models and descriptions are important. Overall, the complex flow environment highlights the need for specific modeling techniques to analyze the subsequent interaction between mixing species and assess various injection parameters and configurations. Characterizing both the species and heat mixing processes in the aforementioned LFP injector unit will be essential in predicting features such as thermodynamic conditions, mixture composition and temperature, local equivalence ratio, and autoignition time delay.

The works of Oefelein and Yang [35], Zong *et al.* [32], and Oswald *et al.* [36] explored the capabilities of computational fluid dynamics simulations and chemical kinetic mechanisms to model cryogenic fluid mixing and combustion under supercritical conditions. The treatment of sub-models for computations to predict processes in future aeroengines, particularly at take-off conditions (supercritical thermodynamic state of the fluid), causes challenges for large eddy simulation (LES) sub-grid scale models. In addition, real gas equations of state alongside proven chemical mechanisms need to be utilized [35]. Computational approaches for the modeling of supercritical fluid mixing needs to be taken into account for the design and analysis of the entire LFP injector system. The scope of this work focuses on the

premixing unit and corresponding combustor flow field.

An essential aspect of the premixing and injection analysis that should be considered in this study is the unique species and heat transport processes at work. While the separate streams of fuel and oxidizer species in a high pressure cryogenic injection scheme leads to diffusion flames such as in a rocket engine, the present work focuses on the case of premixing the cryogenic hydrogen fuel and the high-pressure air leading to a premixed flame, which is not documented to date. Several terms of the governing equations are thus required to be included for accurate description of species and heat transport in order to capture the flow mixing processes and its evolution. The multi-component turbulent reacting flow transport equation with N species, Eq. 1, and the equation that characterizes the species diffusion velocities, Eq. 2, taken from Williams *et al.*[37] are presented next. For species transport, one has:

$$\rho \left(\frac{\partial Y_i}{\partial t} + \mathbf{v} \cdot \nabla Y_i \right) = \dot{\omega}_i - \nabla \cdot (\rho Y_i \mathbf{V}_i) \quad (1)$$

Where Y_i are the mass fractions of species indexed i , ρ is the density of the fluid mixture, $\mathbf{v}$ is the fluid velocity vector, $\dot{\omega}_i$ is the mass reaction rate of the corresponding species by chemical reactions, and $\mathbf{V}_i$ is the species diffusion velocity vector. Equation 1 defines the transport of each species i in a flowfield. For the multi-component diffusion and species diffusion velocities, one has:

$$\begin{aligned} \nabla X_i = & \sum_{j=1}^N \frac{X_i X_j}{D_{ij}} (\mathbf{V}_j - \mathbf{V}_i) + (Y_i - X_i) \left(\frac{\nabla p}{p} \right) + \frac{\rho}{p} \sum_{j=1}^N Y_i Y_j (\mathbf{f}_i - \mathbf{f}_j) \\ & + \sum_{j=1}^N \left[\left(\frac{X_i X_j}{\rho D_{ij}} \right) \left(\frac{D_{T,j}}{Y_j} - \frac{D_{T,i}}{Y_i} \right) \right] \left(\frac{\nabla T}{T} \right) \quad (2) \end{aligned}$$

Where p , T and ρ are the thermodynamic variables (static pressure, temperature, and density), X_i is the mole fraction of species i , D_{ij} is the binary diffusion coefficient for species i and j , $\mathbf{f}_i$ is the external body force on species i , and $D_{T,i}$ the thermal diffusion coefficient for species i . From the individual terms of Eq. 2, it is important to note the $\nabla T/T$ term that may impose a large influence on the species diffusion due to the drastic temperature

gradient occurring when the cryogenic vaporized fuel and compressed air are mixed before reacting. Similarly, the pressure gradient term $\nabla p/p$ must be assessed to design the fuel injection scheme.

Prior to directly analyzing the mixing processes required to enable premixed combustion with hydrogen fuel, a performance map methodology is derived and applied to generate and identify a relevant premixed combustor operating point. This combustor performance map can be derived for any future aeroengine. The generation of this map and the presented methodology leverage two existing softwares: (1) the commercial gas turbine engine performance software GasTurb, to calculate thermodynamic-based data throughout the engine’s turbomachinery, and (2) the open source chemical kinetics code Cantera, to compute one-dimensional premixed flame characteristics such as flame transport properties and adiabatic flame temperature. Additionally, two key quantities, the flame equivalence ratio Φ and the premixed combustor flow split α are determined for the considered premixed hydrogen/air combustor. The presented method relies on a retrofitting procedure from hydrocarbon fuels to hydrogen fuel engine operation. The retrofit must therefore maintain consistency in terms of turbomachinery performance (stagnation pressure and temperatures throughout the engine), combustor thermal power, mass conservation, momentum conservation, and energy conservation. By taking into account these four major requirements, the energy density difference between hydrogen (120 MJ/kg) and Jet-A (43 MJ/kg) is accounted for performance-wise. The impact towards the aircraft overall mass and subsequent mass reduction is out of scope [38].

Chapter 1

Aeroengine Performance Analysis

1.1 GasTurb Software

To obtain aeroengine turbomachinery performance data, the commercial gas turbine simulation software, GasTurb, is employed. This software features both land-based and aeronautic gas turbine engines and offers a wide-range of modeling capabilities from basic thermodynamics to off-design performance analysis [39]. The engine design point calculation is utilized only for the current study. Within this model, the engine system operation, which includes basic input data, secondary flow paths, turbomachinery efficiencies, and nozzle/augmentor parameters, is defined with user inputs. Outside of the basic gas turbine engine data available from literature [40], the default input values are utilized which are consistent with contemporary aeroengine performance specifications. Based on a given aeroengine's operating characteristics such as total air mass flow, overall pressure ratio, and maximum turbine inlet temperature, GasTurb generates comprehensive thermodynamic data at each aeroengine station. The present case simulates static sea-level operation at 100% engine power which can be the most challenging regime. The underlying turbomachinery calculation procedure is next defined in further detail.

First, the air is assumed to behave like a half-ideal gas where the gas properties are functions of temperature and gas composition. The temperature rise from combustion is influenced by inlet temperature, pressure, and fuel-air-ratio, and the chemical interactions

are stored in tables created by the governing NASA CEA code [41]. Additionally, the effects of incomplete fuel burning is not included in the aeroengine modeling procedure. Lastly, the international standard atmosphere (ISA) approximation defines gas property variation with altitude that are applied to the inlet of the engine.

For the complete system, rather than directly utilizing conservation equations to describe the working fluid's thermodynamic transformation through each stage of the turbomachine, GasTurb employs engine component modeling in the form of component maps. The component maps define each engine module's operation such as the intake, compressor, and turbine. First, the intake map functions by taking the compressor relative corrected speed as an input argument and the flight mach number as the parameter in order to assess inlet pressure losses. However, the intake pressure ratio is set to unity in the design point calculation. Next, based on the intake system, the compressor map is defined by Fig. 1.1 in terms of corrected air mass flow, pressure ratio, and efficiency. Auxiliary coordinates, known as beta lines, are superimposed over the compressor map which are functions of corrected flow, pressure ratio, and compressor rotational speed. Lastly, similar to the compressor maps, the turbine map in Fig. 1.2 is defined in terms of corrected mass flow and pressure ratio and features beta lines represented by rotational speed and turbine efficiency.

These component maps drive the gas turbine performance calculations by accounting for shaft rotational speed, compressor and turbine pressure ratios, efficiencies, and corrected air mass flow at various operating conditions. Although the aeroengine modeling tool utilizes the component maps to allow for detailed design of each module in the system, default efficiency values are mostly employed for the static sea-level design point calculation as they represent realistic capabilities. Future work will explore the coupling of turbomachinery performance analysis to the combustor performance map in order to obtain a single calculation platform. Further, while GasTurb outputs data at each engine module, the static temperature and pressure at the inlet and outlet of the combustion chamber (T_3, P_3, T_4, P_4) are the key parameters for this thesis.

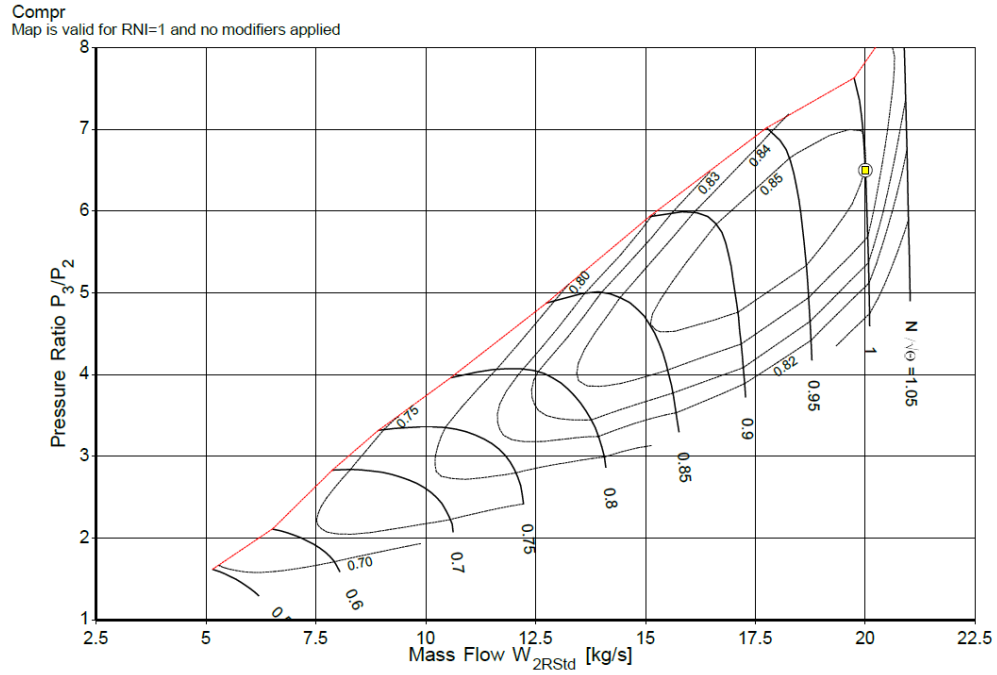


Figure 1.1: GasTurb compressor map defined in terms of corrected mass flow (x-axis) and pressure ratio (y-axis). The beta lines illustrate varying shaft speed as well as compressor efficiencies. Considered operating point is highlighted by the yellow square.

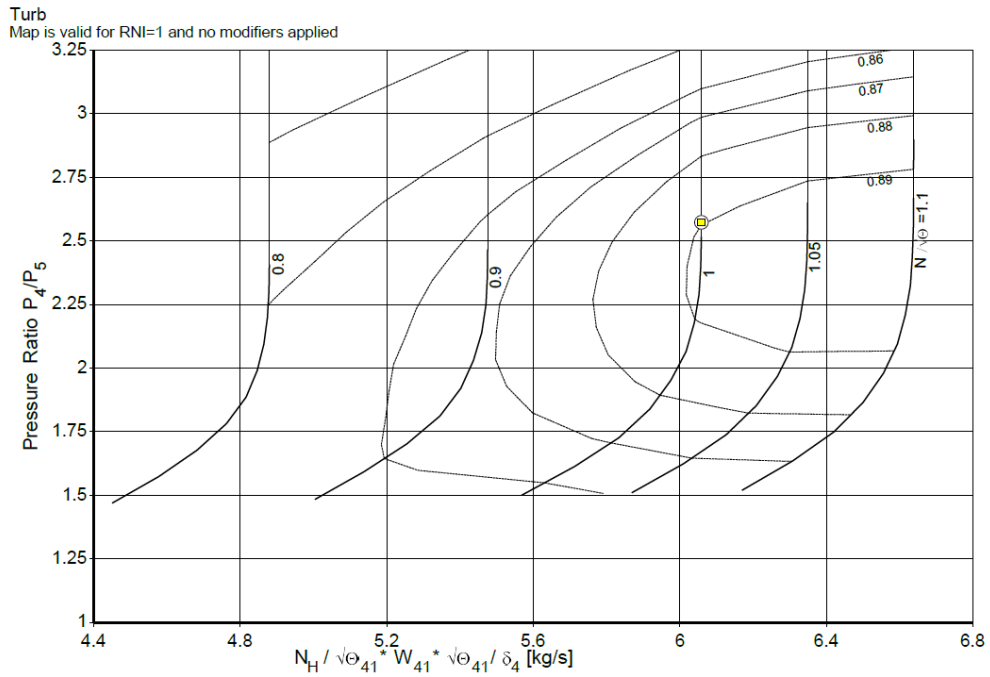


Figure 1.2: GasTurb turbine map defined as a function of corrected mass flow (x-axis), turbine pressure ratio (y-axis), and beta lines. Yellow square showcases current operating point.

1.2 Application to GE-CJ610 Aeroengine

The investigated aeroengine in this study is the General Electric CJ610 (GE-CJ610) turbojet engine, seen schematically in Fig. 1.3. This gas turbine engine does not feature an augmentor and was developed for subsonic propulsion. Additionally, it has logged over millions of flight hours [42]. Utilizing known public domain engine performance specifications from literature [40], the GE-CJ610 is modeled in GasTurb for a static sea-level operating point assuming full engine input power. The gas turbine input performance data is now detailed. Within the GasTurb design point user-interface, one can fully define the aeroengine in terms of basic data such as overall mass flow rates and pressure ratios, secondary air system data, ambient conditions, and component efficiencies. Figure 1.4 demonstrates the user-input window that is populated accordingly with the known GE-CJ610 engine data in terms of air mass flow rate, overall pressure ratio, and burner exit temperature. Outside of these parameters, appropriate values consistent with the CJ610 technology level are used for the component pressure ratios and efficiencies.

Next, input values are required for the engine secondary air system which corresponds to bleed flows and turbine cooling air. For the GE-CJ610 engine, the turbine cooling air flow is extracted prior to the combustor inlet and is redirected around the combustor and into the turbine module. In this calculation, it is assumed that 10% of the total engine air flow contributes to turbine and bleed flow cooling. Lastly, as mentioned previously, the ambient conditions are defined to be static sea-level. These input parameters are employed for the design point calculation for the considered engine. The results of the design point calculation are presented next.

1.3 Results of Thermodynamic Cycle Analysis

The GasTurb design point calculation provides engine performance data for each stage of the system at the specified operating condition. Based on the provided input data specific to the GE-CJ610 engine such as total mass flow of air, overall pressure ratio, and burner

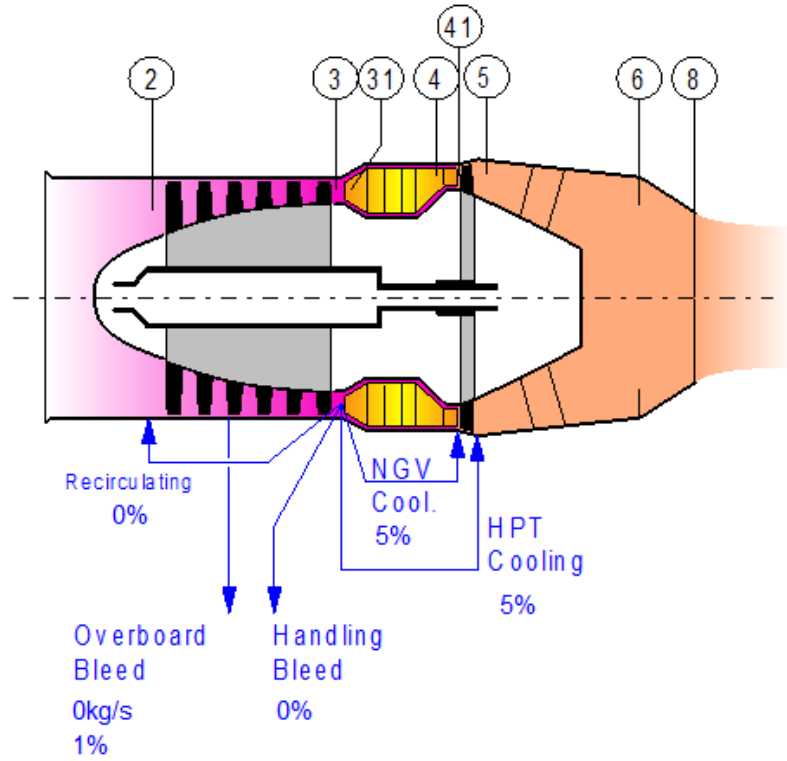


Figure 1.3: GasTurb turbojet engine diagram with station numbering and secondary flow parameters [39].

Basic Data	<i>Property</i>	<i>Unit</i>	<i>Value</i>	<i>Comment</i>
Secondary Air System	Inlet Corr. Flow W2Rstd	kg/s	20	
> Ambient Conditions	Intake Pressure Ratio		0.99	
> Comp Efficiency	Pressure Ratio		6.5	
> Comp Design	Burner Exit Temperature	K	1167	
> Turb Efficiency	Burner Design Efficiency		0.9999	
> Tip Clear.	Burner Partload Constant		1.6	used for off design only
> Reheat	Fuel Heating Value	MJ/kg	118.429	
> Nozzle Selection	Overboard Bleed	kg/s	0	
> Nozzle Calculation	Power Offtake	kW	0	
Stations	Mechanical Efficiency		0.9999	
	Burner Pressure Ratio		0.97	
	Turbine Exit Duct Press Ratio		0.98	

Figure 1.4: GasTurb software design point user interface [39]. The table lists the GE-CJ610 performance data from literature [40] retrofitted for hydrogen fuel.

exit temperature, the modeling tool describes the thermodynamic progression of the airflow throughout the engine. Figure 1.5 documents the GasTurb simulation resultant data at each component station within the turbojet. One can observe the transformations of the working fluid through compression (stations 2-3), combustion at constant pressure (stations 3-4), expansion in the turbine (stations 4-5), and acceleration out of the nozzle (stations 6-8). While there are six main turbomachine stations essential to the overall system, two are specifically important for the performance map in this work, station 3 (the compressor exit / combustor inlet) and station 4 (the combustor exit / turbine inlet). Specifically, the mass flow rate, static temperature, and static pressure values at these stations are required. These quantities are key to the calculations and modeling performed in this work and are utilized throughout the thesis.

Although the commercial gas turbine simulation software provides upstream and downstream combustor boundary conditions consistent with the GE-CJ610 turbojet, the combustion modeling procedure within GasTurb pertains to a diffusion flame-type reaction process which cannot accurately describe a premixed system. GasTurb employs NASA's CEA code which does not account for premixed combustor geometry or characteristics [39]. Because of this, an alternate approach is taken to determine the total engine fuel flow rate for premixed hydrogen. Equating the engine thermal power output between the hydrocarbon and hydrogen fuel types enables the determination of the fuel flow rate. The thermal power parameter is key in retrofitting existing aeroengines with alternate combustor architectures by ensuring consistency in terms of engine power output. To obtain the hydrogen fuel flow rate, the design point calculation is conducted for hydrocarbon fuel in order to observe the baseline fuel flow as well as the engine thermal power input. With the GE-CJ610 thermal power output, the value is divided by the hydrogen heat of reaction to estimate the corresponding total hydrogen fuel flow (0.10997 kg/s). Now, with the thermodynamic data corresponding to the GE-CJ610 turbojet, the premixed combustor performance map can be derived. Further, the GasTurb data at the compressor outlet (station 3) is employed to obtain one-dimensional hydrogen flame data.

	Units	St 2	St 3	St 4	St 5	St 6	St 8
Mass Flow	kg/s	19.8	19.8	17.928	19.908	19.908	19.908
Total Temperature	K	288.15	524.735	1167	903.471	903.471	903.471
Static Temperature	K	272.247	518.542	1159.71	879.763	897.484	771.575
Total Pressure	kPa	100.312	652.026	632.466	238.755	233.98	233.98
Static Pressure	kPa	82.2571	624.59	616.098	214.845	227.838	125.671
Velocity	m/s	178.633	113.493	132.312	232.377	117.279	546.162
Area	m ²	0.105305	0.041576	0.073213	0.1007	0.191938	0.06424
Mach Number		0.54	0.25	0.2	0.4	0.2	0.999928
Density	kg/m ³	1.05258	4.19617	1.85075	0.850757	0.884392	0.567416
Spec Heat @ T	J/(kg*K)	1004.52	1034.82	1203.32	1148.01	1148.01	1148.01
Spec Heat @ Ts	J/(kg*K)	1004.13	1033.49	1202.06	1142.43	1146.66	1116.17
Enthalpy @ T	J/kg	-10032.3	230525	963412	651244	651244	651244
Enthalpy @ Ts	J/kg	-25987.3	224084	954658	624244	644366	502097
Entropy Function @ T		-0.11924	2.00032	5.16668	4.1079	4.1079	4.1079
Entropy Function @ Ts		-0.317673	1.95733	5.14046	4.00238	4.0813	3.48633
Exergy	J/kg	-831.293	219233	687704	382533	380862	380862
Gas Constant	J/(kg*K)	287.05	287.05	287.047	287.048	287.048	287.048
Fuel-Air-Ratio		0	0	0.017365	0.015611	0.015611	0.015611
Water-Air-Ratio		0	0	0	0	0	0

Figure 1.5: GasTurb design point output at the relevant engine component stations. Stations 3 (combustor inlet) and 4 (combustor outlet) are most relevant to the current work.

Chapter 2

Premixed Combustor Performance Map Model Development and Derivation

2.1 Disclosure Statement

Elements presented in chapters 2, 3, and 4 are taken from Ref. [43] for the 2023 AIAA SciTech Forum. The thesis author was the lead author of the paper. Co-authors include: Mr. Chris Caulfield, Dr. Gan Xiao, and Dr. Paul Palies. Edits from the original SciTech publication include formatting, syntax, refined results, and additional derivation.

2.2 Performance Map Methodology

The premixed combustor performance map methodology is now presented and the map derivation conducted. This map originates from a set of six key requirements that characterize the combustor operating conditions. These requirements are linked to a corresponding set of unique equations that are utilized to build the map. The first requirement is with respect to the thermal power of the engine. The second, third, and fourth guidelines apply to conservation of mass, momentum, and energy throughout the combustor. The compressor-combustor-turbine power budget represents the fifth requirement. Lastly, the sixth requirement is a coupling relationship based on premixed combustor area ratio design. This set of requirements and corresponding equations enables the calculation

of realistic combustor operating points for a premixed system and allows for graphical representation of the performance map. The resultant map is a function of three parameters: the stagnation flame temperature $T_{t,f}$ (background color contour), the combustor air flow split ratio α (x -axis), and the premixed flame equivalence ratio Φ (y -axis). For each of the requirements, thermal power, mass conservation, momentum conservation, power and energy budgets, the equations are derived into the following form: $\Phi = f_k(\alpha, T_{t,f})$. This specific form allows for the graphical representation of the equations onto the performance map where the overall goal is to identify relevant premixed combustor operating points where all requirements are satisfied.

2.2.1 Engine Thermal Power Requirement

The thermal power of an aeroengine is classically expressed as the product of the total engine fuel flow rate $\dot{m}_F$ multiplied by the heat released by the fuel during combustion Δh . It writes:

$$\mathcal{P}_{th} = \dot{m}_F \times \Delta h \quad (2.1)$$

It is important to highlight that Eq. 2.1 serves as a key parameter for the retrofitting procedure in modifying contemporary aeroengines to run with alternative fuel types such as hydrogen. When considering alternate fuels for a given engine, the thermal power must be kept constant in order to maintain overall performance. For a premixed combustor, the premixed flame operates at an equivalence ratio Φ that must be determined consistently with the required thermal power for the engine. The equivalence ratio Φ is defined as:

$$\Phi = \frac{\dot{m}_F / \dot{m}_s}{s_a} \quad (2.2)$$

where s_a is the stoichiometric ratio of the single step chemical reaction between hydrogen and air and $\dot{m}_s$ is the air mass flow rate through the swirling injector units. For hydrogen/air flames, the ratio s_a is equal to 0.029. Next, the combustor air flow split ratio α is defined with the following expression [23]:

$$\alpha = \frac{\dot{m}_s}{\dot{m}_t} \quad (2.3)$$

where $\dot{m}_t$ is the total mass flow rate of air through the engine core. By combining the definitions of the equivalence ratio, Eq.2.2, and the flow split, Eq.2.3, one obtains a new definition:

$$\Phi = \frac{\dot{m}_F / (\alpha \dot{m}_t)}{s_a} \quad (2.4)$$

Equation (2.4) links the premixed flame equivalence ratio Φ to the air flow split ratio of the combustor α . The definition is of the form $\Phi = f_1(\alpha)$ and can be readily sketched on the performance map to represent the thermal power requirement.

2.2.2 Mass Conservation Requirement

The second requirement stems from the conservation of mass through the combustor which is defined as the total flow rate of air $\dot{m}_t$, plus the fuel flow rate $\dot{m}_F$. As the total air flow rate decomposes into $\dot{m}_s + \dot{m}_d$, the $\dot{m}_d$ term represents the dilution hole cooling air that bypasses the swirling injector and enters the combustion chamber downstream the flame. Further, in regards to the total mass flowing in the combustor, the fuel flow rate is neglected as it contributes less than 1% to the total flow ($\dot{m}_F \ll \dot{m}_t$). Thus, the inlet flow stream consists of only the air flow rates through the swirler (injector) units and the dilution holes. The mass conservation for a premixed combustor then writes $\dot{m}_t = \dot{m}_s + \dot{m}_d$. The swirler and dilution flow rate terms can be defined in alternate forms:

$$\dot{m}_s = \rho_s v_s A_s = \frac{\dot{m}_F}{\Phi s_a} \quad (2.5)$$

where ρ_s is the static air density at the swirler inlet, v_s the axial bulk velocity through the swirlers, and A_s is the total section area of the N swirlers around the engine azimuth. For the mass flow rate of air through the dilution holes, one has:

$$\dot{m}_d = \rho_d v_d A_d = (1 - \alpha) \dot{m}_t \quad (2.6)$$

where ρ_d is the static fluid density at the dilution hole entrance, v_d is the bulk velocity through the dilution holes, and A_d is the total section area of the M dilution holes. The

total air mass flow rate into the combustor writes:

$$\dot{m}_t = \rho_s v_s A_s + \rho_d v_d A_d \quad (2.7)$$

Equation 2.7 can also be expressed as follows by making use of the flow split and equivalence ratio definitions:

$$\dot{m}_t = \frac{\dot{m}_F}{\Phi s_a} + (1 - \alpha)\dot{m}_t \quad (2.8)$$

By summing the two equations, Eq. 2.7 and Eq. 2.8, one obtains:

$$2\dot{m}_t = \rho_s v_s A_s + \frac{\dot{m}_F}{\Phi s_a} + 2(1 - \alpha)\dot{m}_t \quad (2.9)$$

Finally, Eq. 2.9 can be rewritten in terms of Φ as:

$$\Phi = \frac{\dot{m}_F / s_a}{[2\dot{m}_t - \rho_s v_s A_s - 2(1 - \alpha)\dot{m}_t]} = \frac{\dot{m}_F / s_a}{2\alpha\dot{m}_t - \rho_s v_s A_s} \quad (2.10)$$

The resulting equation is in the form of $\Phi = f_2(\alpha)$ and represents mass conservation on the performance map. Not only does this final form demonstrate a relationship between Φ and α , but it also allows for the investigation of the bulk velocity of air into the swirler, v_s , which plays a key role in combustion dynamics and flame stabilization [44].

2.2.3 Momentum Conservation in a Premixed Combustor

The third requirement for the performance map applies the conservation of momentum to a control volume that encompasses the combustor volume. This control volume consists of the swirler inlet, the combustor outlet, and the dilution holes that are positioned perpendicularly to the centerline flow direction. The considered two-dimensional domain requires two equations along the x and y directions to assess the momentum flux across the control volume surfaces. Gravity and viscous terms are neglected in the present formulation and steady-state flow is assumed. The integral form of the momentum equation is defined with:

$$\int_A (\rho \mathbf{v}) \mathbf{v} \cdot \mathbf{n} dA = - \int_A p \mathbf{n} \cdot dA \quad (2.11)$$

Integrating Eq. 2.11 along the x and y directions leads to:

$$-\dot{m}_m v_m + \dot{m}_e v_e = P_s A_s - P_e A_e \quad (2.12)$$

$$\frac{1}{2} \dot{m}_d v_d - \frac{1}{2} \dot{m}_d v_d = -P_s A_d + P_s A_d \quad (2.13)$$

Equation 2.12 represents the x -momentum through the combustor where: $\dot{m}_m = \dot{m}_F + \dot{m}_s$ represents the mixture flow rate, v_m is the axial velocity of the mixture, $\dot{m}_e$ consists of the total flow rate exiting the combustor ($\dot{m}_e = \dot{m}_t + \dot{m}_F$), v_e is the axial velocity of the flow downstream the combustor and prior to the turbine module, P_s is the static pressure at the swirling injector, P_e the static pressure at the combustor outlet, and A_e the cross-sectional area of the combustor outlet.

The x -momentum equation features the axial flow of the premixture into the combustor as well as the axial flow of the combustion products exiting the combustor. Whereas the swirler unit will induce both axial and azimuthal velocity components on the flow, the control surface is defined just downstream of the swirler vanes where the premixture flows in the axial direction only. On the other hand, the y -momentum equation Eq. 2.13 defines the dilution hole flow paths in both the negative and positive y directions and assumes that the dilution air is split evenly between the inner and outer radii of the combustion chamber. Observing Eq. 2.13, it is observed that all terms cancel each other out; therefore, only the x -momentum equation is utilized. The control volume for the momentum requirement is illustrated in Fig. 2.1.

Next, further development of the x -momentum equation yields:

$$-\rho_m v_m^2 A_s + \rho_e v_e^2 A_e = P_s A_s - P_e A_e \quad (2.14)$$

where ρ_m is the static density of the mixture and ρ_e is the static density of the total flow exiting the combustion chamber. Substituting the two velocity terms with the corresponding mass flow rate definitions, one obtains:

$$-\rho_m \left(\frac{\dot{m}_F + \dot{m}_s}{\rho_m A_s} \right)^2 A_s + \rho_e \left(\frac{\dot{m}_F + \dot{m}_s + \dot{m}_d}{\rho_e A_e} \right)^2 A_e = P_s A_s - P_e A_e \quad (2.15)$$

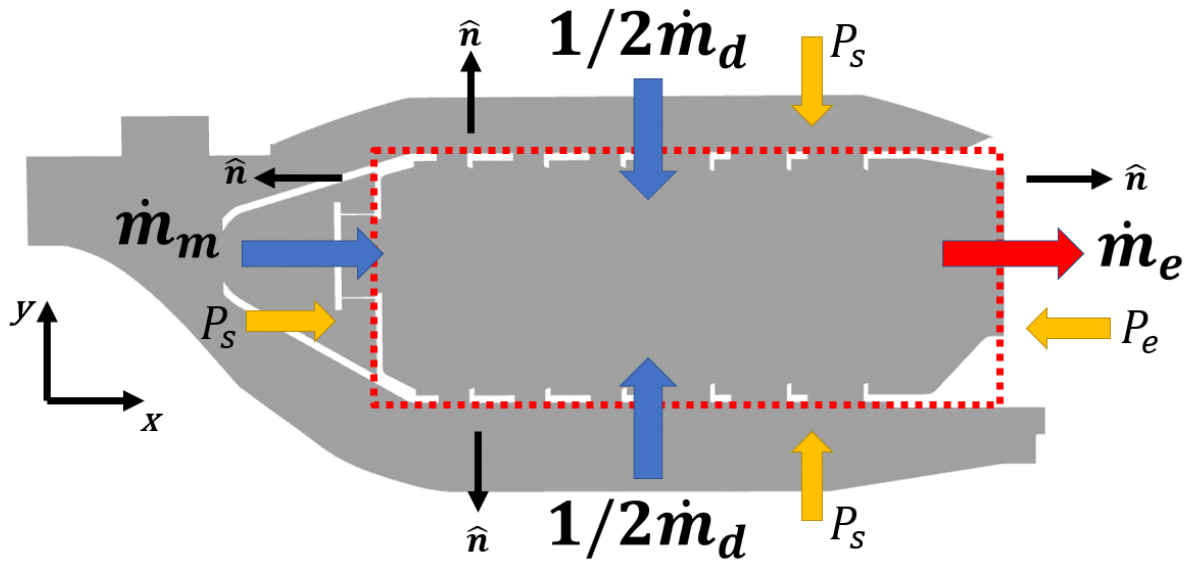


Figure 2.1: Control volume for momentum conservation requirement derivation. The original approach highlights equations for both the x and y directions.

Simplifying terms:

$$-\frac{(\dot{m}_F + \dot{m}_s)^2}{\rho_m A_s} + \frac{(\dot{m}_F + \dot{m}_s + \dot{m}_d)^2}{\rho_e A_e} = P_s A_s - P_e A_e \quad (2.16)$$

Employing relationships previously defined in order to express the momentum equation in terms of Φ and α , Eq. 2.16 translates to:

$$-\frac{(\Phi s_a \alpha \dot{m}_t + \alpha \dot{m}_t)^2}{\rho_m A_s} + \frac{(\Phi s_a \alpha \dot{m}_t + \alpha \dot{m}_t + (1 - \alpha) \dot{m}_t)^2}{\rho_e A_e} = P_s A_s - (P_s \Delta P) A_e \quad (2.17)$$

Notice that the combustor outlet pressure term, P_e , is replaced here with $P_s \Delta P$. This substitution directly incorporates the combustor pressure drop and removes the need to know the combustor outlet pressure in the given formulation. In the present study, the ΔP is neglected. Now, in order to proceed in solving Eq. 2.17 as a function of α and Φ , the equation is converted into a second order polynomial where Φ is the independent variable. The polynome thus writes:

$$a\Phi^2 + b\Phi + c = 0 \quad (2.18)$$

The polynomial coefficients, a , b , and c are as follows:

$$a = \left(\frac{s_a^2 \alpha^2 \dot{m}_t^2}{\rho_e A_e} - \frac{s_a^2 \alpha^2 \dot{m}_t^2}{\rho_m A_s} \right) \quad (2.19)$$

$$b = \left(\frac{2s_a \alpha \dot{m}_t^2}{\rho_e A_e} - \frac{2s_a \alpha^2 \dot{m}_t^2}{\rho_m A_s} \right) \quad (2.20)$$

$$c = \left(\frac{\dot{m}_t^2}{\rho_e A_e} - \frac{\alpha^2 \dot{m}_t^2}{\rho_m A_s} \right) - P_s A_s + (P_s \Delta P) A_e \quad (2.21)$$

Utilizing Eq. 2.18 alongside the three coefficients represented in Eqns. 2.19 - 2.21, the momentum conservation equation is now in the form $\Phi = f_3(\alpha)$ for the performance map. However, based on the final form, additional steps are required to identify physical solutions. This solution search approach will be discussed later in the thesis.

2.2.4 Combustor Energy Conservation Requirement

Within a fully premixed combustor architecture, the engine core airflow enters the combustion chamber via the swirler unit as well as through the dilution holes around the combustor's wall casing. The air through the swirl injector is also accompanied by the fuel flow, hydrogen in this work. Similar to the momentum equation, the description for the energy conservation corresponds to a control volume where two-dimensional flow is assumed between the inlets and outlet of the combustor. The compressed air and fuel enter axially into the combustion chamber while the dilution hole air flows perpendicularly to the swirler stream. The two separate streams mix together and exit the combustor as a single mixed airflow consisting of hot combustion products and dilution cooling air. Because the flow is three-dimensional, an analytical model with a one-dimensional energy conservation equation is not suitable to replicate features such as the dilution hole streams. Accordingly, the analytical description from the energy equation can be developed in two dimensions, while also keeping the derivation tractable compared to three dimensions. In addition, the azimuthal swirl flow that may occur inside the combustor is here neglected. The derivation is based on the conservation of total energy applied and integrated over the control volume defined inside the combustor from the downstream side of the flame to the combustor outlet. The total energy budget, including both internal thermal energy e and kinetic energy $\mathbf{v}^2/2$, assuming (1) no body forces, (2) no viscous work, (3) adiabatic system, (4) and no external work done, writes:

$$\int_V \frac{\partial}{\partial t} \left[\rho \left(e + \frac{\mathbf{v}^2}{2} \right) \right] dV + \int_S \rho \left(e + \frac{\mathbf{v}^2}{2} \right) \mathbf{v} \cdot \mathbf{dS} = \int_V \dot{q} dV - \int_S p \mathbf{v} \cdot \mathbf{dS} \quad (2.22)$$

While Eq. 2.22 includes the total heat released in the combustor by the flame $\dot{q}$, the choice is made here to analyze a reduced control volume that begins once the heat addition by the flame has been made to the working fluid. Furthermore, the assumption of steady flow reduces the total energy equation to:

$$\int_S \rho \left(e + \frac{\mathbf{v}^2}{2} \right) \mathbf{v} \cdot \mathbf{dS} = - \int_S p \mathbf{v} \cdot \mathbf{dS} \quad (2.23)$$

As observed in Fig. 2.2, splitting the control volume of Eq. 2.23 into three surfaces corresponding to the premixed flame location (burnt flame side) with subscript index f , dilution hole inlet with subscript index d , and combustor outlet with index e leads to:

$$\int_{A_f} \rho \left(e + \frac{\mathbf{v}^2}{2} \right) \mathbf{v} \cdot d\mathbf{S}_f + \int_{A_e} \rho \left(e + \frac{\mathbf{v}^2}{2} \right) \mathbf{v} \cdot d\mathbf{S}_e + \int_{A_d} \rho \left(e + \frac{\mathbf{v}^2}{2} \right) \mathbf{v} \cdot d\mathbf{S}_d = - \int_{A_f} p \mathbf{v} \cdot d\mathbf{S}_f - \int_{A_e} p \mathbf{v} \cdot d\mathbf{S}_e - \int_{A_d} p \mathbf{v} \cdot d\mathbf{S}_d \quad (2.24)$$

Evaluating the surface integrals over both the left and right-hand sides of the equation and defining $d\mathbf{S} = \mathbf{n} \cdot dS$, one obtains:

$$\left[-\rho_f \left(e_f + \frac{v_f^2}{2} \right) v_f A_f \right] + \left[\rho_e \left(e_e + \frac{v_e^2}{2} \right) v_e A_e \right] + \left[-\rho_d \left(e_d + \frac{v_d^2}{2} \right) v_d A_d \right] = - \left[-p_f v_f A_f \right] - \left[p_e v_e A_e \right] - \left[-p_d v_d A_d \right] \quad (2.25)$$

Next, utilizing the definition of the combustor air flow split α as it relates to the total engine air flow rate of the engine core $\dot{m}_t$, the following expression is obtained:

$$\dot{m}_t = \frac{\rho_f v_f A_f}{\alpha} = \frac{\rho_d v_d A_d}{(1 - \alpha)} = \rho_e v_e A_e \quad (2.26)$$

where ρ_f is the static density of the flame gas products mixture, v_f is the bulk flow velocity downstream the flame, and A_f is the cross-sectional area of the combustion chamber ($A_f = A_e$). Dividing each term of Eq. 2.25 by $\dot{m}_t$ and substituting the internal energy e term with $h - p/\rho$ where the enthalpy h equals $c_p T$, Eq. 2.25 then reduces to:

$$-\alpha \left(c_{p,f} T_f - \frac{p_f}{\rho_f} + \frac{v_f^2}{2} \right) + \left(c_{p,e} T_e - \frac{p_e}{\rho_e} + \frac{v_e^2}{2} \right) - (1 - \alpha) \left(c_{p,d} T_d - \frac{p_d}{\rho_d} + \frac{v_d^2}{2} \right) = \alpha \frac{p_f}{\rho_f} - \frac{p_e}{\rho_e} + (1 - \alpha) \frac{p_d}{\rho_d} \quad (2.27)$$

Simplifying the pressure terms on both sides of Eq. 2.27 leads to:

$$-\alpha c_{p,f} T_f - \alpha \frac{v_f^2}{2} + c_{p,e} T_e + \frac{v_e^2}{2} + (\alpha - 1) c_{p,d} T_d + (\alpha - 1) \frac{v_d^2}{2} = 0 \quad (2.28)$$

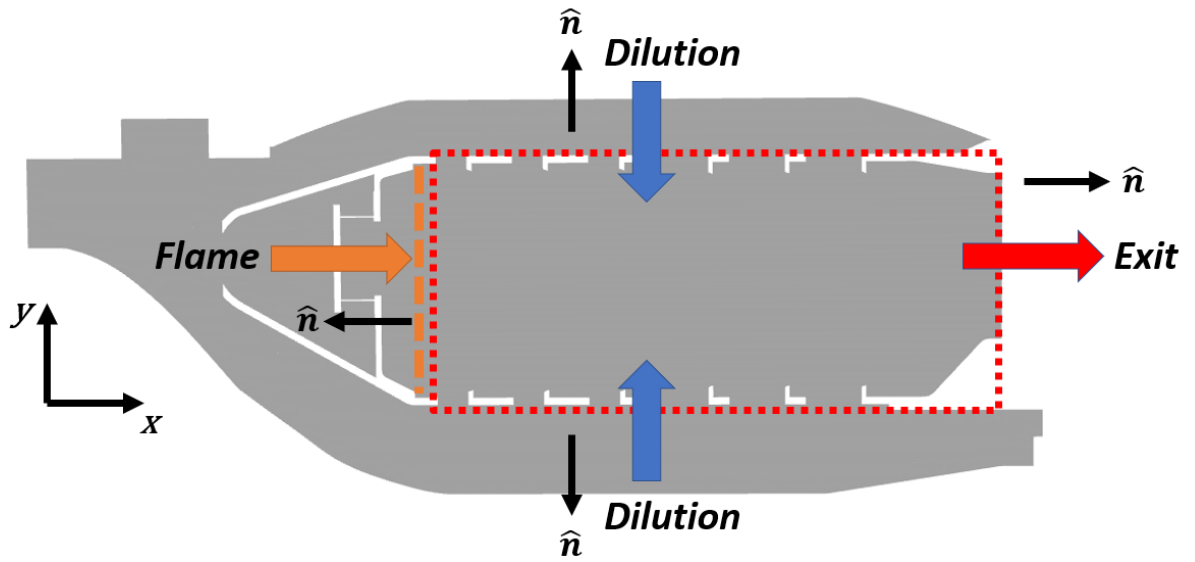


Figure 2.2: Control volume for energy conservation requirement derivation. The orange dashed line represents the premixed flame downstream location.

with $c_{p,f}$ as the specific heat at the flame temperature at constant pressure, T_f being the static flame temperature, $c_{p,e}$ the fluid specific heat at the combustor outlet, T_e the outlet static temperature, $c_{p,d}$ the air specific heat at the dilution hole flow temperature, and T_d the static temperature at the dilution holes. Next, expressions for the flow velocities v_f , v_e , and v_d are inserted into Eq. 2.28 using the mass flow rate definitions. As subscript f denotes the location immediately downstream the flame which is assumed here as a one-dimensional interface, the mass flow rate combines air flowing through the swirler $\dot{m}_s$ and added fuel flow rate $\dot{m}_f$. However, it is important to clarify that $v_f \neq v_m$ as the term in this case denotes the burnt mixture. Because this interface is located upstream the dilution holes, the burnt gas flow velocity can write:

$$v_f = \frac{\dot{m}_F + \dot{m}_s}{\rho_f A_f} = \frac{\Phi s_a \alpha \dot{m}_t + \alpha \dot{m}_t}{\rho_f A_f} \quad (2.29)$$

The combustor outlet bulk velocity v_e writes:

$$v_e = \frac{\dot{m}_F + \dot{m}_s + \dot{m}_d}{\rho_e A_e} = \frac{\Phi s_a \alpha \dot{m}_t + \alpha \dot{m}_t + (1 - \alpha) \dot{m}_t}{\rho_e A_e} = \frac{\Phi s_a \alpha \dot{m}_t + \dot{m}_t}{\rho_e A_e} \quad (2.30)$$

For the dilution hole bulk velocity v_d , one has:

$$v_d = \frac{\dot{m}_d}{\rho_d A_d} = \frac{(1 - \alpha) \dot{m}_t}{\rho_d A_d} \quad (2.31)$$

Substituting Eq. 2.29, Eq. 2.30, and Eq. 2.31 into the velocity terms of Eq. 2.28 leads to:

$$\begin{aligned} -\alpha c_{p,f} T_f - \frac{\alpha}{2} \left(\frac{\Phi s_a \alpha \dot{m}_t + \alpha \dot{m}_t}{\rho_f A_f} \right)^2 + c_{p,e} T_e + \frac{1}{2} \left(\frac{\Phi s_a \alpha \dot{m}_t + \dot{m}_t}{\rho_e A_e} \right)^2 + (\alpha - 1) c_{p,d} T_d \\ + \frac{(\alpha - 1)}{2} \left(\frac{(1 - \alpha) \dot{m}_t}{\rho_d A_d} \right)^2 = 0 \end{aligned} \quad (2.32)$$

Based on this form, following the methodology of the momentum conservation requirement, Eq. 2.32 is converted into a second order polynomial with Φ being the independent variable. The characteristic polynome for the energy equation writes:

$$r\Phi^2 + s\Phi + t = 0 \quad (2.33)$$

The polynomial coefficients, t , r , and s are then represented with:

$$r = \left(\frac{s_a^2 \alpha^3 \dot{m}_t^2}{2\rho_f^2 A_f^2} \right) - \left(\frac{s_a^2 \alpha^2 \dot{m}_t^2}{2\rho_e^2 A_e^2} \right) \quad (2.34)$$

$$s = \left(\frac{s_a \alpha^3 \dot{m}_t^2}{\rho_f^2 A_f^2} \right) - \left(\frac{s_a \alpha \dot{m}_t^2}{\rho_e^2 A_e^2} \right) \quad (2.35)$$

$$t = \left(\frac{\alpha^3 \dot{m}_t^2}{2\rho_f^2 A_f^2} \right) - \left(\frac{\dot{m}_t^2}{2\rho_e^2 A_e^2} \right) + \alpha c_{p,f} T_f - c_{p,e} T_e + (1 - \alpha) c_{p,d} T_d + \frac{(1 - \alpha)}{2} \left(\frac{(1 - \alpha) \dot{m}_t}{\rho_d A_d} \right)^2 \quad (2.36)$$

Eq. 2.33 and the three coefficient expressions in Eq. 2.34, Eq. 2.35 and Eq. 2.36 constitute the formed energy equation which is a direct function of two key parameters: the combustor flow split α and the equivalence ratio Φ . With this form, the energy equation is now defined with $\Phi = f_4(\alpha)$ to plot on the performance map. Similarly to the momentum equation Eq. 2.18, a procedure is required to extract solutions from Eq. 2.33.

2.2.5 Compressor-Combustor-Turbine Power Budget

The combustor power budget, commonly expressed via the compressor-turbine power balance for a RQL combustor architecture [25], takes a specific form for a premixed combustor [23] with the introduction of the air flow split ratio α . In a premixed combustor, the combustion region is solely located on the most upstream side contrary to a RQL type of combustion system where burning occurs throughout. In both cases, this budget expresses the balance between the power available from the fuel as the difference between the power received on the turbine and extracted from the compressor. The premixed case is characterized with the following form:

$$\dot{m}_F \Delta h = (\alpha \dot{m}_t + \dot{m}_F) c_{p,f} T_{t,f} + (1 - \alpha) \dot{m}_t c_{p,d} T_{t,d} - \dot{m}_t c_{p,s} T_{t,s} \quad (2.37)$$

where $T_{t,f}$ is the stagnation flame temperature, $T_{t,d}$ is the air stagnation temperature at the dilution holes, $c_{p,s}$ is the specific heat of air at the compressor exit and swirler inlet, and $T_{t,s}$ is the air stagnation temperature at the swirler inlet. The next steps seek to simplify and

rearrange Eq. 2.37 to a form suitable for the performance map:

$$T_{t,f} = \frac{\dot{m}_F \Delta h + \dot{m}_t c_{p,s} T_{t,s} + (\alpha - 1) \dot{m}_t c_{p,d} T_{t,d}}{(\alpha \dot{m}_t + \dot{m}_F) c_{p,f}} \quad (2.38)$$

Employing Eq. 2.38 and substituting for $\dot{m}_f$ with Eq. 2.4, as well as considering that the thermodynamic conditions upstream the swirler and dilution hole inlets are the same, one gets:

$$T_{t,f} = \frac{\Phi s_a \alpha \dot{m}_t \Delta h + \alpha \dot{m}_t c_{p,s} T_{t,s}}{(\alpha \dot{m}_t + \Phi s_a \alpha \dot{m}_t) c_{p,f}} \quad (2.39)$$

Finally, dividing by $\alpha \dot{m}_t$ and rearranging the equation leads to an expression for the premixed total flame temperature $T_{t,f}$ as a function of Φ :

$$T_{t,f} = \frac{\Phi s_a \Delta h + c_{p,s} T_{t,s}}{(1 + \Phi s_a) c_{p,f}} \quad (2.40)$$

Equation 2.40 is a function of Φ only and can be plotted on the performance map to serve as the background color contour.

2.2.6 Performance Map Coupling Relationship

A useful relationship to verify the derived analytical approach consists of the balance between the bulk velocity of air into the swirling injector and the air velocity through the dilution holes to serve as an essential point of cross-validation for the combustor operating point identified by the performance map. This relationship, driven by the combustor flow split ratio α , can have a substantial impact on the combustor operating point and overall burner performance. Utilizing the swirler and dilution hole mass flow rate terms derived from the mass conservation requirement (Eqns. 2.5 and 2.6), the following expression is obtained:

$$\frac{\dot{m}_s}{\dot{m}_d} = \frac{\alpha \dot{m}_t}{(1 - \alpha) \dot{m}_t} = \frac{\rho_s v_s A_s}{\rho_d v_d A_d} \quad (2.41)$$

Simplifying terms and assuming that the air density at the swirler is equal to the air density at the dilution hole entrances ($\rho_s = \rho_d$), the ratio of swirler velocity to dilution hole velocity

writes:

$$\frac{v_s}{v_d} = \frac{\alpha A_d}{(1 - \alpha) A_s} \quad (2.42)$$

Equation 2.42 reveals that the velocity relationship is merely a function of the corresponding surface areas as well as the combustor flow split α . Now, as the v_s term is parameterized within the mass conservation requirement of Eq. 2.10 and is influenced by the overall annular injection scheme as described earlier, the calculated bulk velocity into the swirler alongside the determined α value from the performance map are used to solve for the resultant dilution hole velocity v_d . Then, the output dilution hole velocity can be compared with the velocity term from Eq. 2.6 for a point of cross-validation. Along a similar approach, to assess any discrepancies between the performance map and CFD, Eq. 2.42 can be employed to observe the influence of the combustor area ratio A_d/A_s (total dilution hole surface area to total swirling injector surface area) on the resultant velocity terms.

2.3 Momentum and Energy Polynome Solution Search

This section outlines the additional steps taken to extract relevant solutions from both the momentum (Eq. 2.18) and energy conservation (Eq. 2.33) requirements. The solution search approach relies in determining Φ for a given α value. As highlighted previously, these two requirements have been developed into second order polynomials and cannot be readily applied to the combustor performance map in their current forms. To determine physical solutions and plot those onto the performance map, the discriminant values of the polynomials, $\Delta = b^2 - 4ac$, must be positive or null to verify that the Φ solutions exist at a given α -value. Therefore, the Φ solutions must correspond to this condition for the discriminant value while also existing in the relevant equivalence ratio range ($0.4 < \Phi < 1$). Moving forward, the quadratic formula is then utilized for both equations to find the roots of the polynomials:

$$\Phi_{1,2} = \frac{-b \pm \sqrt{b^2 - 4ac}}{2a} \quad (2.43)$$

where a , b , and c represent the coefficients of the second order polynomial. This yields two solutions, Φ_1 and Φ_2 , for each value of flow split ratio α ranging from 0 to 1.

For the momentum conservation requirement, the discriminant values of Eq. 2.18 equate to all positive values across the domain of α . Based on this finding, the Φ roots from the momentum equation are all physical solutions and can be applied to the performance map directly. On the other hand, for the energy conservation requirement of Eq. 2.33, the discriminant values are both positive and negative across the α domain. To identify possible solutions, the discriminant Δ of the energy equation is plotted as an intermediate contour plot as a function of both α and Φ . Additionally, the corresponding roots Φ_1 and Φ_2 are superimposed over this contour to verify that the energy solutions are indeed real values and in the positive discriminant domain region. Figure 2.3 illustrates an example of the solution search method where the yellow region represents positive discriminant Δ values, and the vertical lines correspond to the two roots, Φ_1 and Φ_2 , of Eq. 2.33 at given values of α .

Although the two plotted Φ solutions in the example reveal satisfactory intersection ($\Delta > 0$), the results must be assessed further to verify convergence on the 0.4 to 1 equivalence ratio range. In Fig. 2.3, the closest resolved Φ roots are outside of the plot's range which causes the solution roots to appear as lines due to the original grid resolution (51 intermediate values for Φ and 41 for α). To ensure that the polynomial roots of the energy requirement do in fact intersect at relevant equivalence ratios, the discretization step sizes of both α and Φ are decreased so that the calculation resolution is enhanced. The outcome of this modification reveals that even though the crossing of the Φ_1 line appears ideal on the plot, the two closest Φ_1 roots to the 0.4 to 1 range diverge about the given value of α ; whereas the Φ_2 solutions do converge on the targeted range. Therefore, the Φ_2 results are utilized only to obtain a physical solution for the energy conservation requirement.

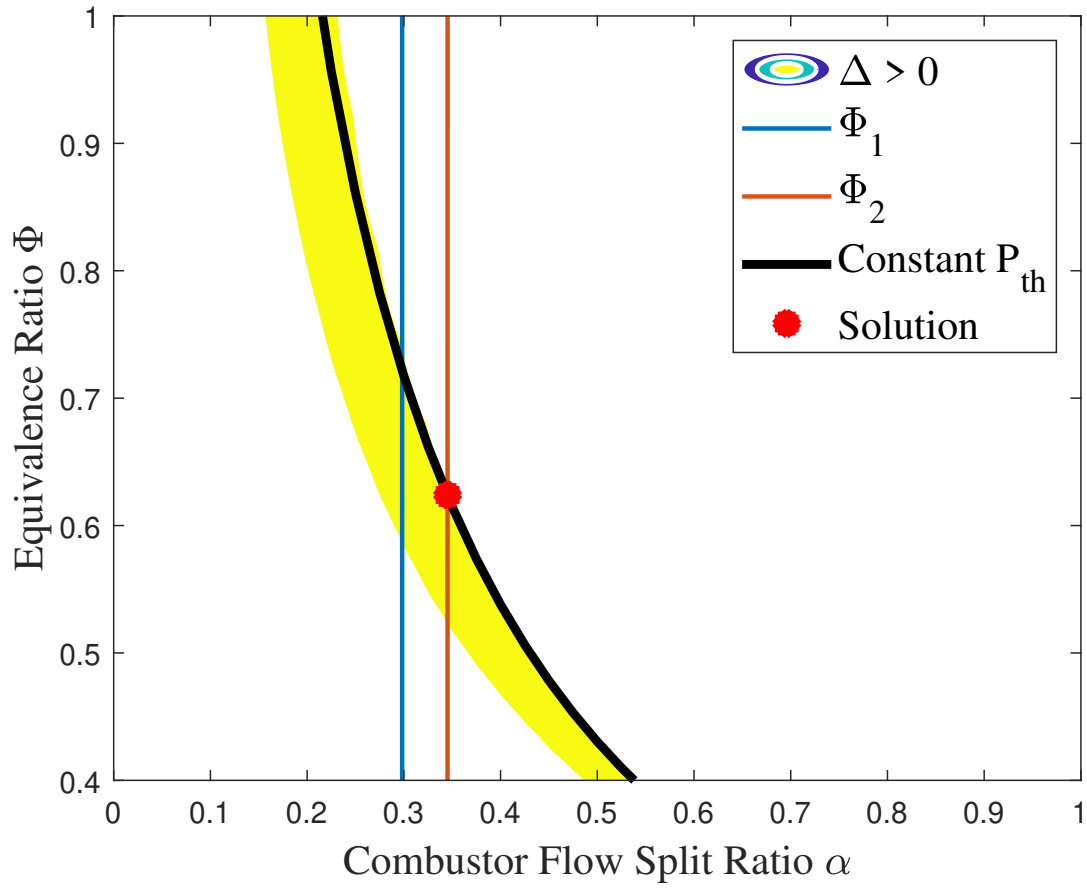


Figure 2.3: Graphical representation of the solution search procedure for the energy conservation budget of Eq. 2.33.

With the increased discretization resolution, there exists several Φ_2 solutions on the required equivalence ratio range at given α values, but only one solution curve is required: the one that intersects with the other requirements. To find this agreement, the constant thermal power line is incorporated into Fig. 2.3 to obtain a single intersecting point between thermal power requirement and the energy conservation requirement with the Φ_2 root. The final Φ_2 solution is the one that coincides with the thermal power budget as shown in Fig. 2.3. As previously highlighted, this operating point is not the only solution to the energy equation as several exist at various values of α . Furthermore, it is worth noting that while the polynomials of Eqns. 2.18 and 2.33 define Φ as the unknown variable, there are parameters in the coefficients that are implicitly functions of equivalence ratio such as ρ , c_p , and T . To account for this potential implicit dependency, the obtained Φ roots are inserted back into Eqns. 2.18 and 2.33 alongside their corresponding coefficients at (α, Φ) separately from the graphical approach. The outcome of the procedure confirms that the roots satisfy the polynomial expressions and thus validates the proposed solution search method.

2.4 Chemical Kinetics Data

To obtain the relevant combustion conditions for the lean premixed combustor operating with hydrogen, chemical kinetics calculations are employed. Specifically, one-dimensional hydrogen/air flame simulations are conducted at premixed equivalence ratios from 0.4 to 1 at the relevant engine pressure and temperature values computed from thermodynamics-based performance analysis. This range is selected to cover the lean premixed mode operation to minimize nitrous oxide emissions. The 1D flame simulation results include adiabatic flame temperatures, burnt and unburnt mixture densities, and specific heats to be utilized for the performance map. Additionally, the flame displacement speeds are calculated as a metric for convergence of the one-dimensional computational mesh.

An essential aspect of the hydrogen/air one-dimensional flame simulations is the chemical kinetic mechanism utilized. The selected mechanism is now described, see Ref. [45]. The detailed $\text{H}_2/\text{O}_2/\text{N}_2$ mechanism contains 33 species and 224 reactions, which can be used to

predict the nitrous oxide formation and emissions for gas turbine engine conditions. The mechanism is developed based on existing sets of experimental data available in literature and includes laminar flame speed, ignition time delay, and in-flame nitrous oxide data. This comprehensive mechanism includes two main components: an updated H_2/O_2 core mechanism and a validated NO_x sub-mechanism. The first component was obtained by updating the H_2/O_2 core mechanism from Keromnes *et al.* with 11 recent available data-sets. The second component of the mechanism assesses the NO_x chemistry sub-mechanism by comparing four existing models: GRI 3.0 [46], Glarborg *et al.* [47], Zhang *et al.* [48], and Konnov *et al.* [49]. The developed mechanism [45] was validated against an experimental database under a wide range of operating conditions of pressure (1-10 atm), inlet temperature (300-650 K), and equivalence ratio (0.4-1.0). It is thus used for the present study to calculate the subsequent adiabatic flame temperature and species transport properties.

The 1D flame calculations are solved with the multi-component transport model in Cantera [50]. For the hydrogen/air mixture, the solution process required a grid spacing $\Delta x = 1\mu\text{m}$ to perfectly resolve the flame displacement speed throughout the entire flame structure. However, a more coarse grid could be used to capture the adiabatic flame temperature only. These two parameters are important quantities in premixed combustion as the flame temperature impacts nitrous oxide production levels, and the flame displacement speed contributes in defining where a flame will reside/stabilize in the combustor. The flame displacement speed is defined from Refs. [20, 51]. The adiabatic flame temperature and flame displacement speed are next plotted as a function of equivalence ratio. Figure 2.4 illustrates the hydrogen/air adiabatic premixed flame temperature in the lean regime up to the stoichiometric condition. The burnt side static flame temperature ranges from 1600 K in the leanest regime up to 2500 K at an equivalence ratio of one. High flame temperatures are likely not to be used because of associated thermal nitrous oxide production. Figure 2.5 points next to the influence of the equivalence ratio Φ on the hydrogen/air flame displacement speed S_d throughout the flame front structure. Figure 2.4 and Fig. 2.5 highlight the passage from unburnt to burnt gas in the premixed flame domain.

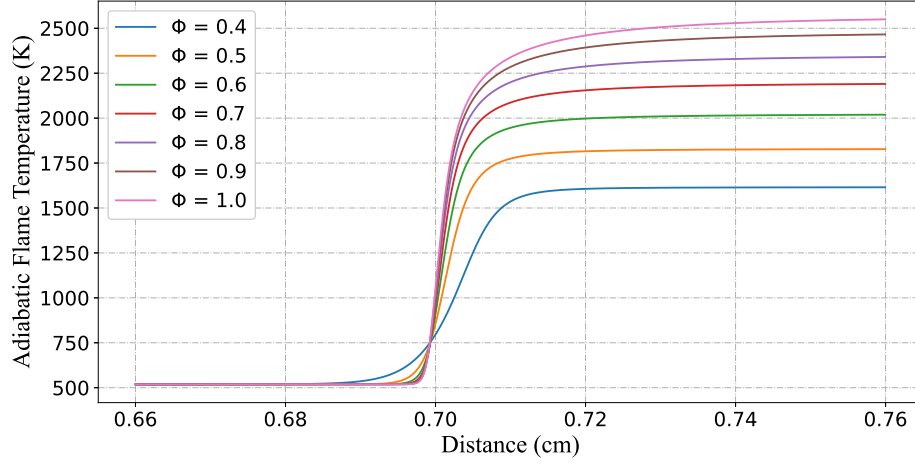


Figure 2.4: Influence of equivalence ratio on adiabatic flame temperature for premixed flames. As the equivalence ratio increases, it is observed that the flame temperature also increases. Results obtained with one-dimensional premixed flame calculations utilizing the Cantera software with the chemical mechanism [45].

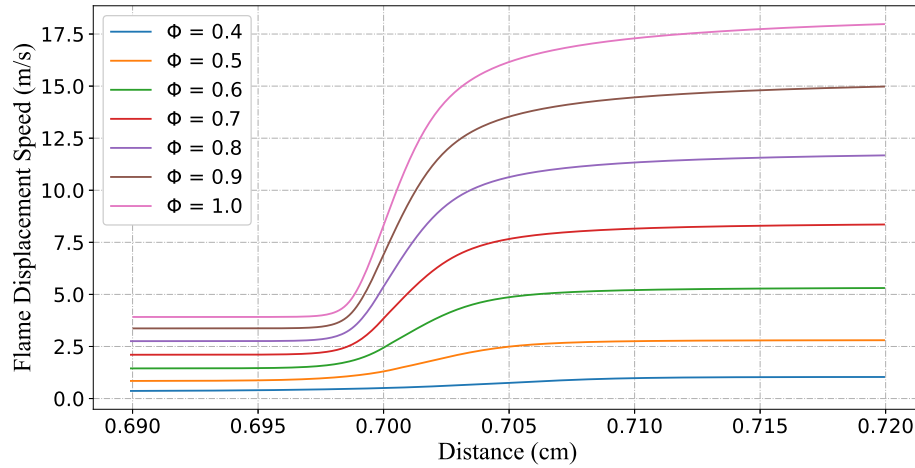


Figure 2.5: Premixed flame displacement speed as a function of equivalence ratio. As the equivalence ratio is increased the flame speed increases throughout the flame front.

2.5 Performance Map Results

The results of the premixed combustor performance map are now presented. The map can be applied to a wide range of gas turbine aeroengines as long as performance specifications are obtained with GasTurb or an in-house code [52]. The main parameters influencing the combustor operating point are the overall engine thrust class, engine thermodynamics, injector count, and the swirler bulk velocity. With the GasTurb and Cantera data serving as inputs, the requirements are plotted as functions of combustor flow split α , equivalence ratio Φ , and stagnation flame temperature $T_{t,f}$. Each requirement is plotted uniquely with the goal of obtaining intersection at a single location. Figure 2.6 illustrates the graphical display of the combustor performance map calculation procedure for the GE-CJ610. The constant thermal power requirement is plotted as a black curve. The mass conservation is represented by a magenta curve and implicitly contains the swirler bulk velocity within its formulation. The momentum requirement is plotted as an amber curve, and the energy equation is displayed as a dashed gray line. Finally, the compressor-combustor-turbine power budget is illustrated as a background color contour representing the stagnation flame temperature.

The identified premixed operating point can be observed at the α and Φ values where the governing equations coincide. The intersection point corresponds to a combustor flow split ratio of 0.445, an equivalence ratio equal to 0.485, and a stagnation flame temperature of 1770 Kelvin. This is an ideal target in mitigating thermal nitrous oxide formation, and the obtained equivalence ratio value is well above the lean flammability limit for hydrogen. However, it is important to note that this is not the only possible operating point as the mass, momentum, and energy conservation requirements contain several solutions along the constant thermal power curve. Finally, it is worth noting that while the operating point identified in Fig. 2.6 appears fairly exact, there is a relative amount of error or uncertainty based on the assumptions made in the governing equations.

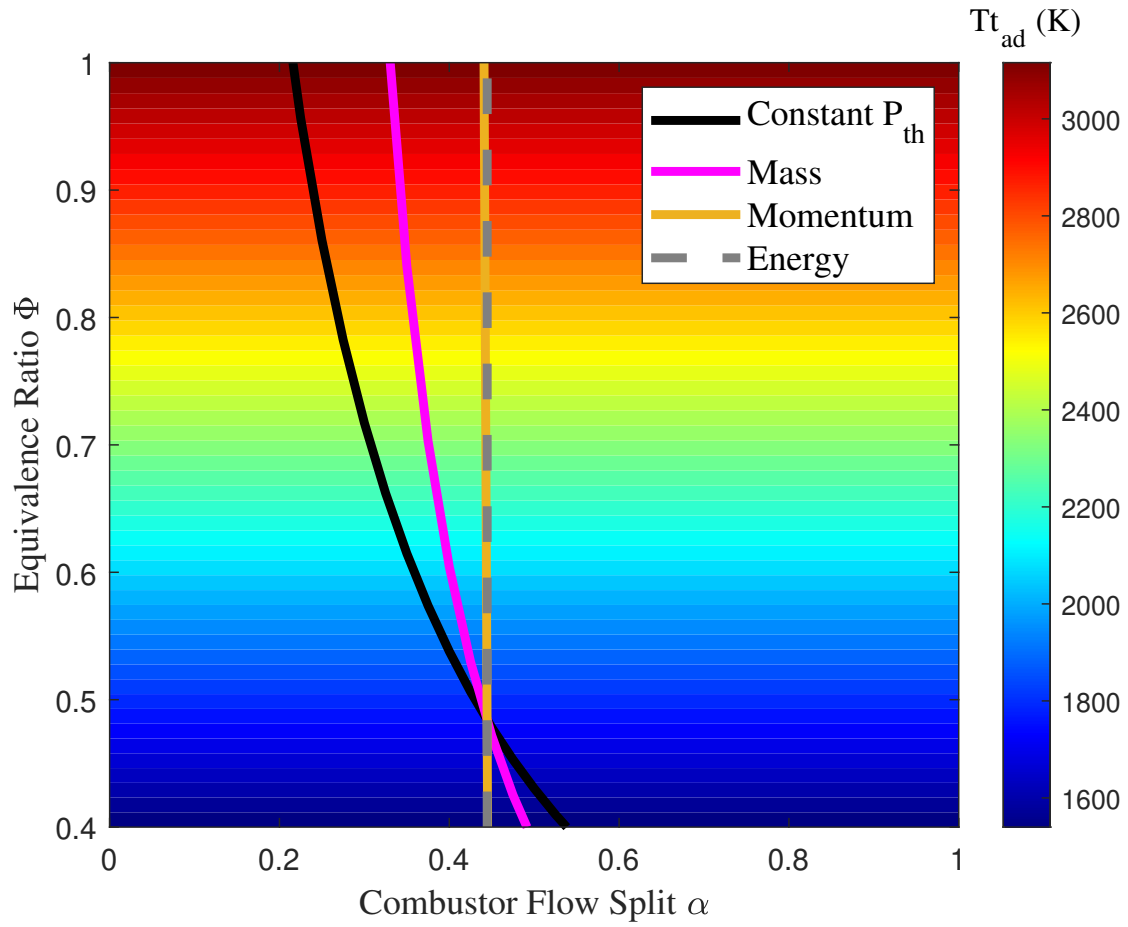


Figure 2.6: Premixed combustor performance map applied to the GE-CJ610 aeroengine. Operating point identified at the intersection of each requirement. A level of uncertainty is present due to assumptions made in the governing equations.

Based on the results from Fig. 2.6, although the momentum conservation equation is featured, simplifications were made to Eq. 2.18. Specifically, the pressure terms from Eq. 2.21 are neglected. From Chapter 2 section 2.3, a control volume was applied to the premixed combustor domain which resulted in a momentum flux in the x-direction only. This formulation led to a term for the hydrogen/air premixture at the swirler exit and a term for the combustion products alongside the dilution flow at the combustor exit. Given the one-dimensional nature of the equation, the cross-sectional area terms at the upstream and downstream ends of the control volume are essentially equal. Additionally, the isobaric combustion process reveals relatively constant pressure across the domain which ultimately cancels out both of the pressure terms.

The total GE-CJ610 engine injector count is now analyzed. The full injection scheme of the CJ610 engine is represented as a two-dimensional annulus with the inner and outer radii measured from the combustion chamber inlet of the original assembly. The swirling injector units are positioned inside the annulus to illustrate the complete architecture where the flow is towards the reader. Figure 2.7A illustrates the nominal annular injection scheme of the GE-CJ610 engine where the injectors are positioned in pairs around the combustor azimuth. This configuration will be referred to as Case A. One can discern the available space for additional injector ports as well as enhanced swirler surface area. Figure 2.7B presents Case B with eighteen premixed injectors in the annular combustion chamber. Figure 2.7C shows Case C with the maximum number of swirlers possible for the CJ610 geometry with the original swirler sizing of 46 mm outer diameter. Finally, by optimizing both the total number and size of the swirling injectors around the engine annulus for Case D, Figure 2.7D shows the resultant distribution. The four cases (A, B, C and D) are next applied to the combustor performance map to select a relevant premixed combustor operating point in terms of equivalence ratio, flow split ratio, and bulk velocity at the swirler inlet.

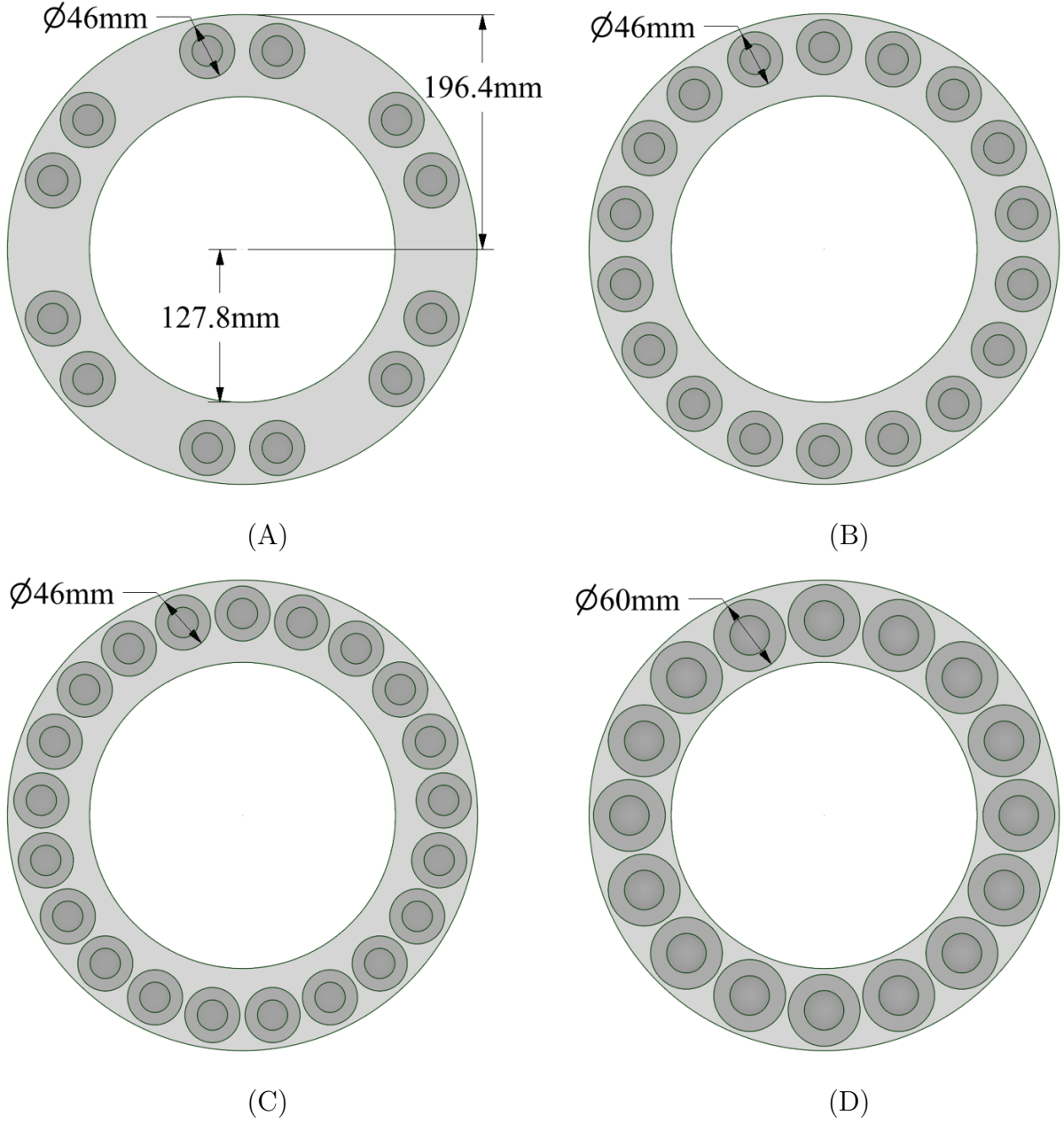


Figure 2.7: Depiction of the considered injector configurations for the GE-CJ610 engine under operation with swirled premixed hydrogen/air injector units. Number of injectors and swirler diameters are assessed. (A): baseline existing configuration. (B): 18 injector configuration. (C): 21 injector configuration. (D): 16 injector optimized configuration. In cases A, B, and C, injector units are not centered within the annulus which is consistent with the original architecture.

The identification of a combustor operating point is a critical step prior to progressing forward with additional analyses such as experimental testing or computational fluid dynamics (CFD) simulations. The performance maps corresponding to each case were generated where differences exist in the number of injectors, bulk flow velocity into the injector system, and swirler surface areas.

The outcome of the injector count parameterization is now detailed. While only one map is shown from Fig. 2.6, each of the four maps appear identical at their points of intersection. As the stagnation flame temperature, thermal power, and energy conservation in the combustor remain similar for each case, the mass conservation into the combustion chamber varies via the bulk flow velocity that is unique to the different injector architectures. Table 2.1 tabulates the quantities of interest extracted from the performance map for the four cases. Because the thermal power and energy conservation remain constant in each case, the operating point identification becomes a function of upstream bulk flow velocity for the considered aeroengine geometry and operating regime (static sea-level). It is worthwhile to note that although Tab. 2.1 reports unique bulk velocity values for each case, these values are not the absolute minimum for the given geometry. Modification of the dilution hole surface areas could lead to even lower bulk velocities.

The values from the combustor performance map as well as Table 2.1 pertain to the performance specifications of the GE-CJ610 engine as well as the geometrical aspects of the premixed combustor design. As experimental ground testing is not available, computational fluid dynamics (CFD) simulations can serve as validation tool for the performance map. Case A is the focal point of the computational modeling; however, a more optimized geometry will be the subject of future work. Between the analytical and computational models, the most relevant parameters to compare are the combustor flow split and the swirler bulk velocity. The correct combustor flow split value should yield the suitable equivalence ratio and stagnation flame temperature for the reacting simulations. Prior to the CFD calculations, however, the premixed combustor model and the computational settings are detailed.

Table 2.1: Results from GE-CJ610 injector parameterization.

CJ610 Combustor Operating Points						
Case	N_{inj}	d_{sw} (m)	α	Φ	Tt_{ad} (K)	U_b (m s ⁻¹)
A	12	0.0253	0.445	0.485	1770	309
B	18	0.0253	0.445	0.485	1770	206
C	21	0.0253	0.445	0.485	1770	176
D	16	0.0330	0.445	0.485	1770	136

Chapter 3

Partially-Premixed Aeroengine Combustor Geometry and Computational Settings

3.1 GE-CJ610 Engine Geometry

In this section, the computational model of the GE-CJ610 turbojet engine is introduced alongside the employed methodologies for the CFD simulations of the premixed combustor geometry. The starting point of the modeled combustor geometry in this work is the public domain GE-CJ610, which features all turbomachinery components of the turbojet engine seen in Fig. 3.1. Because the combustion chamber is the central focus in the present work, the geometry is reduced to include only the final high-pressure compressor stage, the combustor, and the first stage of the high-pressure turbine. While the upstream and downstream blade rows can provide insight into acoustics and combustion dynamics within the system, the modules are omitted in the preliminary calculations. Additionally, computational expense is taken into account by further reducing the digital geometry to only a sixty-degree sector. The symmetry of the system is kept consistent for geometrical aspects such as combustor dilution holes and injector count. Next, given the nominal combustor architecture of the baseline aeroengine model, the new design of the premixed scheme is presented.

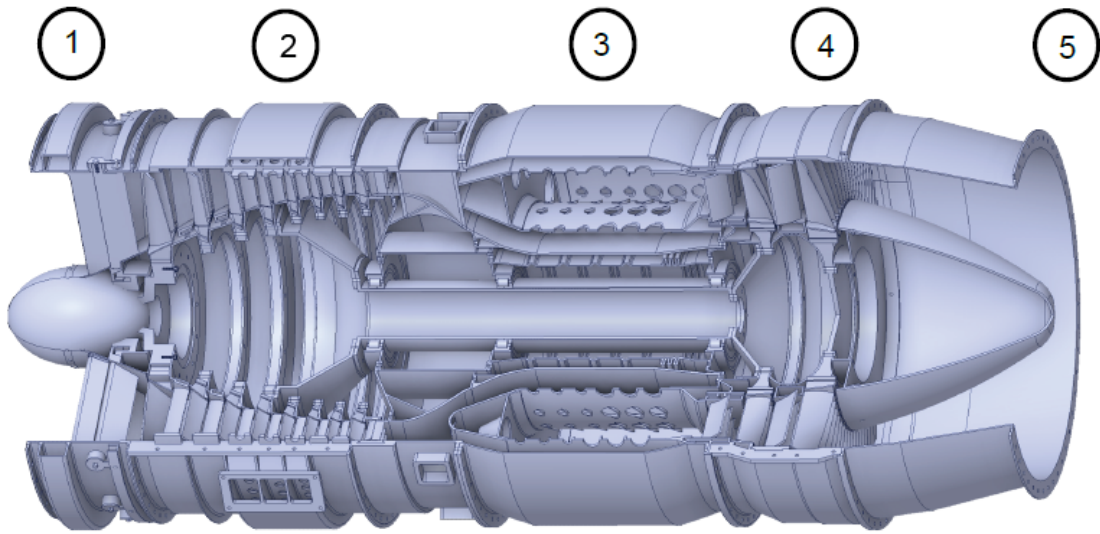


Figure 3.1: Complete computational model of the General Electric (GE) CJ610 turbojet engine to be retrofitted for premixed hydrogen combustion. Featured engine modules include: (1) inlet, (2) compressor, (3) combustor, (4) turbine, and (5) nozzle/exhaust. The assembly was obtained through a public online source.

3.2 Combustor Geometry

The baseline digital model of the CJ610 aeroengine combustor consists of twelve total injectors distributed around the engine azimuth. The injectors are positioned in pairs throughout the annular combustion chamber, and the pairs are stationed sixty-degrees apart from one another. Additionally, the annular chamber features a dilution hole cooling scheme in the combustor lining where high pressure air from the compressor bypasses the injector units and provides cooling to the hot combustion products before entering the turbine module. This bypass stream participates to combustion in an RQL combustor but not in a LFP architecture. The dilution hole rows are located on the inner and outer diameters of the combustor, and they are present throughout the annulus, unlike the sixty-degree injector unit spacing. Figure 3.2 presents the complex geometry of the baseline RQL combustor and reduced model.

In order to address the constraint of overall computational time and cost for the initial computations, the annular combustor model is reduced to a sixty-degree sector consisting of two injectors and the corresponding dilution hole rows. As a result, both sides of the combustor sector are defined as periodic boundaries for meshing and simulation purposes. In order to satisfy the periodic boundary condition requirement during the combustor fluid volume extraction of the sector, the symmetry of the sector must be maintained, which is a particular challenge for the dilution hole row spacing. The periodic requirement is fulfilled here by adjusting the spacing of the dilution holes about the engine central axis. However, by modifying the spacing between dilution hole rows, a single row is removed from the sector. To compensate for the removed single row's effective cooling area resulting from the dilution hole shift, each of the dilution hole surface areas are slightly scaled to account for the missing row while maintaining consistency in the area ratios between the large and small dilution holes. This process satisfies the periodic boundary conditions of the sixty-degree sector while ensuring geometrical accuracy with the full engine annulus.

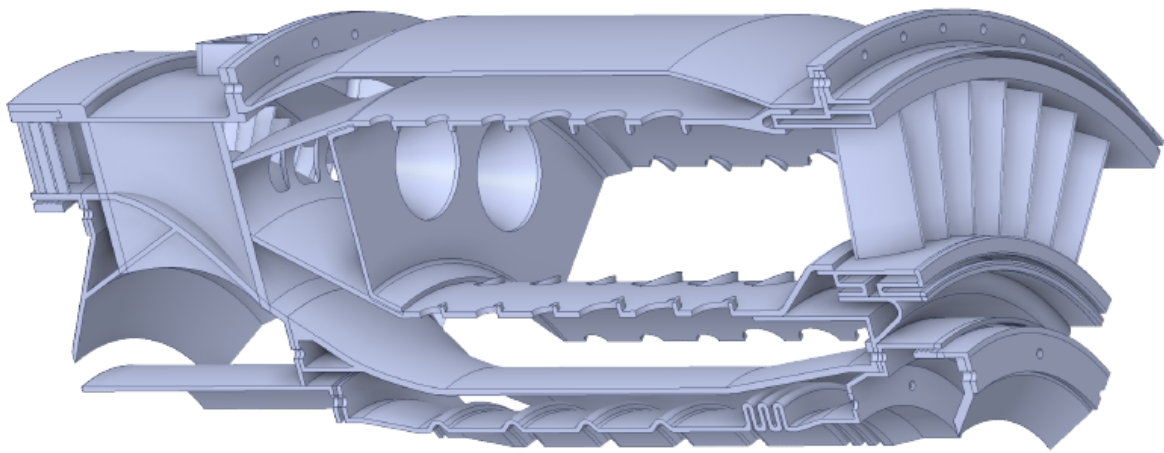


Figure 3.2: Sixty-degree sector of the baseline combustor architecture. Two injector ports are observed for the RQL system, and the flow is from left to right.

Given that the high-pressure airflow exiting the compressor stator stage will convect around the swirler unit and through the vanes into the combustion chamber, this leaves the premixed fuel injection scheme as yet to be defined and optimized. Again, the central goal of the lean fully premixed injector design is to generate a fully premixed swirl-stabilized flame via the swirling mixing interaction between gaseous hydrogen and air prior to their combustion. Therefore, the fuel inlet design and location must accomplish this challenge. It is worth acknowledging that both the fundamental studies and realistic combustor geometry must complement one another to reach this goal as pioneered in this thesis. The complete modeling of the fully premixed injector system conceptualized in Ref. [53] where the liquid fuel experiences supercritical mixing and passes through a heat exchanger is outside the scope of the current research and subject of future work. Therefore, the fuel injection thermodynamics in the computational modeling aim to mimic a partially-premixed injection rather than the fully premixed injector.

As the baseline configuration of the CJ610 combustor corresponds to a widely-used RQL architecture, both the fuel and air injection schemes of the premixed system differ from that of the original design and are discussed. First, the swirling injector units are incorporated into the geometry to target a premixed fuel-air mixture upstream the combustion chamber and promote flame stabilization as seen in Fig. 3.3. Referring to the injector body, radial swirler units consisting of eighteen swirler vanes each with NACA 8411 airfoil profiles [54] are employed. The swirler geometry dimensions are scaled-up by a factor of 1.15 in order to match the sizing of the given combustor. This swirler design corresponds to a swirl number that ensures the formation of a strong recirculation zone. Additionally, this design has demonstrated a stable methane/air fully premixed flame for a set of experimental conditions [20] and is utilized for the current study as a starting point to build upon. It is worthwhile to provide additional insight into the premixed injector scheme presented in this work.

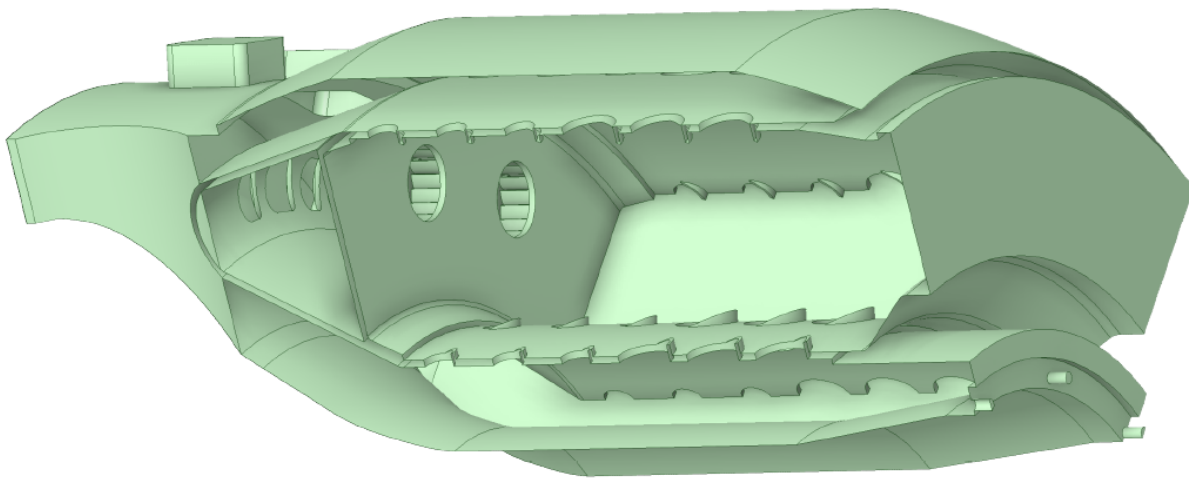


Figure 3.3: Partially-premixed GE-CJ610 combustor fluid volume equipped with swirling injector unit design. The upstream and downstream turbomachinery blades are not included.

3.3 Swirling Injector Unit Design

Figures 3.4 and 3.5 present the preliminary premixed hydrogen/air injection scheme that is investigated in this thesis. The modeling of the complete lean fully premixed injector design is outside the scope of this work. For the initial design, the current injector model consists of a radial swirler where eighteen fuel ports are positioned around the outer perimeter of each swirler unit between the leading edges of the swirler vanes. The swirler vane profiles consist of NACA 8411 airfoils designed to obtain a high swirl number and promote a swirl-stabilized flow regime [20]. With this architecture, the fuel flow must pass through the swirler vanes alongside the air flow which promotes a swirling premixture into the combustion chamber.

Illustrated by Fig. 3.5, the dimensions of both the fuel ports and the swirler vanes must be taken into account during the injector design process. Given the low density and high diffusivity of hydrogen, the fuel port outlet surface areas must be large enough to limit the fluid velocity. The hydrogen fuel velocity and its temperature will have a substantial influence on mixing and combustion dynamics. In terms of the swirler vane dimensions, the vanes can be parameterized in regards to the total vane count, height, and width. For the initial swirling injector design, the baseline 18 swirler vanes are utilized. However, the vane height is increased from the original geometry in order to provide additional mixing time between the fuel and air prior to injection. Additionally, the width of the swirler vanes must be accounted for to prevent blockage and allow the premixture to flow freely into the combustion chamber while also being large enough to induce a strong swirling motion on the flow. While all of these aspects are not fully optimized in the current design, the overall swirling injector architecture allows for a first glance into the partially-premixed swirl-stabilized injection scheme. For the most part, the goal of this novel injector unit in the CFD simulations is to observe successful premixing while targeting a specific combustor flow split ratio based on the performance map.

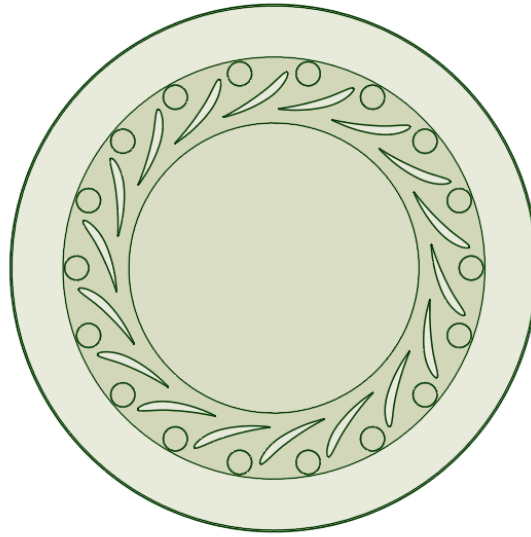


Figure 3.4: Two-dimensional front view of the swirling injector design. Fuel port and swirler vane geometric scheme can be observed. Flow is towards the reader.

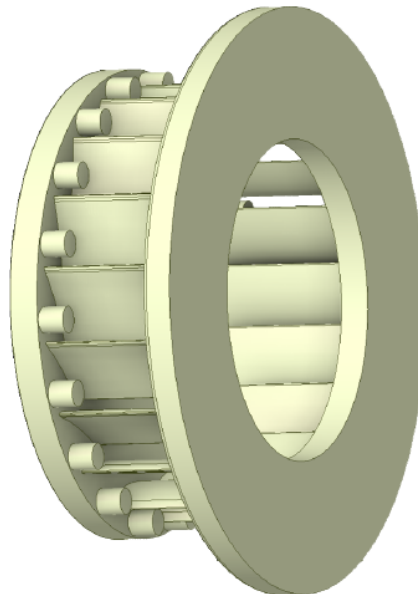


Figure 3.5: Three-dimensional side view of the swirling injector. Both the fuel and air flow exits the injector through the central hole only. Flow is from left to right.

3.4 Computational Mesh

The meshing methodology for the extracted sixty-degree GE-CJ610 combustor sector is now described. A quality numerical mesh is essential in computational modeling in order to properly resolve the flowfield and its characteristics. The meshing procedure follows the Watertight Geometry Workflow from Ansys Fluent which includes all of the required mesh generation steps. Three key steps specific to the mesh are: (1) the local grid refinement, (2) the boundary layer creation, and (3) the surface and volumetric mesh generation. The mesh for the CJ610 premixed combustor architecture features local sizing around the swirler units and inside the combustion chamber including the dilution holes. In these regions of interest where the mixing between air and hydrogen is strong, the enhanced computational grid resolution allows for more detailed effects of these mixing processes to be captured such as those from convective and diffusive natures for the mean flow terms. The local refinement around the swirler units features cell sizes of 0.4 mm whereas the combustion chamber contains elements equal to 0.8 mm. Regarding the fluid to solid interface, the boundary layers on the combustor walls are not of specific interest for the present article and are kept with default values that enable suitable turbulence modeling in these regions. Turbulence in the swirler units and dilution hole flow path is likely to dominate the flowfield.

The computational meshing process performs spatial discretization on both the surface and the inner volume of the premixed combustor geometry. For the surface mesh, the minimum cell size is 0.025 mm and the maximum cell size is 1.6 mm. These cell sizes follow the standard octree law which is defined to parameterize the grid sizing and establish the maximum volume cell size as 2^N the minimum cell size. For the current mesh, the parameter N is set to 6. This scaling is commonly employed in computational mesh generation to promote overall quality and enhance solver robustness. An important parameter in surface mesh quality is the maximum cell skewness which is 0.72 for the current geometry and a satisfactory value. Figure 3.6 provides mesh views along the length of the combustor to illustrate the overall cell elements.

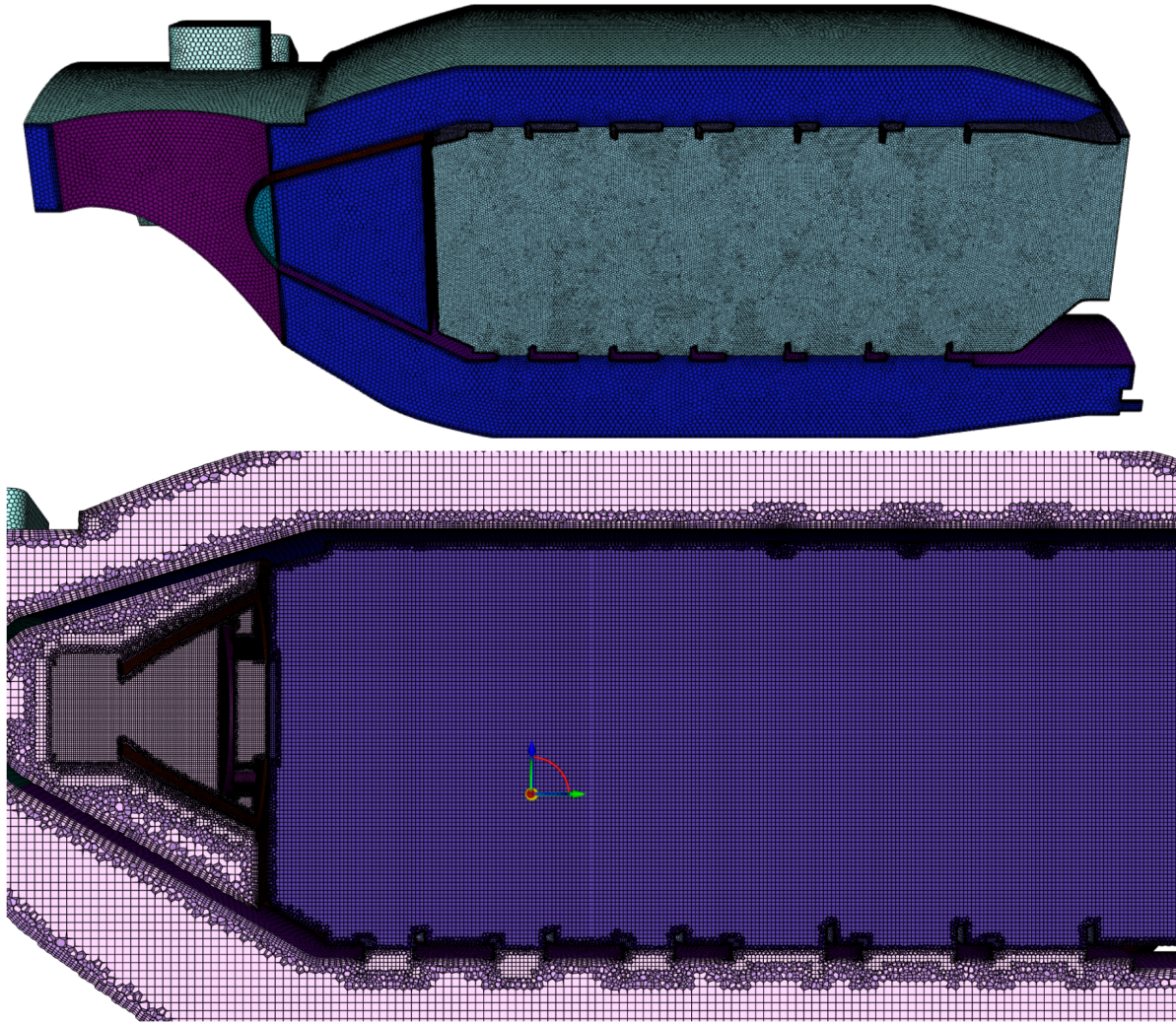


Figure 3.6: Computational mesh of the combustor architecture. The top image illustrates the surface mesh whereas the bottom image reveals the fluid volume mesh. Mesh refinement can be observed in the combustion chamber as well as around the swirler body.

The combustor fluid volume mesh is defined with Ansys Fluent’s Mosaic Poly-hexcore method. This methodology acts to accelerate the meshing process through a reduced face count, higher-quality cells, and efficient parallel scalability [55]. The mosaic poly-hexcore mesh is a combination of hexahedral, isotropic poly prism, and polyhedral elements. The hexahedral cell elements are characterized by their accuracy and efficiency, and they makeup the core volume mesh. Additionally, this mesh generation technique enables a reduced overall computation time with lower memory and disk space requirements. The cell sizes for the fluid volume feature 0.025 mm for the minimum and 6.4 mm for the maximum, and the final cell count for the geometry is approximately 13 million volume cells. In terms of quality, the volumetric mesh results in a minimum orthogonal quality of 0.25 which points to a well-defined mesh. Given the surface and volumetric mesh quality characteristics, a reliable calculation procedure should be obtained while also properly resolving dominant small-scale flow features.

Given the relatively high cell element count for the computational meshing procedure of the fluid volume, it is worth noting that a mesh grid dependency analysis was not conducted. It is possible that a relatively reduced mesh cell count could provide identical flowfield data while lowering the required computational power. However, the presented meshing strategy and ensuing computations benefited from available computational resources in order to produce high-quality cells for the baseline discretization.

3.5 CFD Modeling with Ansys Fluent

3.5.1 Numerical Settings and Boundary Conditions

The numerical setup for the premixed combustor CFD calculations is now presented. For all simulation cases conducted, the calculations are performed with the pressure-based solver where the static pressure is obtained from a combination of continuity and momentum equations unlike the density-based approach where the mass conservation is used to retrieve

the density and the pressure is calculated from the equation of state [56]. Additionally, the choice to utilize the pressure-based solver is supported by the low Mach conditions in the combustor as well as the high speed of sound value for hydrogen [57]. Within the solver, Ansys Fluent’s SIMPLEC pressure-velocity coupling method is utilized. There are three boundary conditions applied to the combustor geometry: air mass flow at the combustor inlet, hydrogen mass flow at the fuel port outlets, and a pressure outlet condition at the chamber exit. For both mass flow inlet conditions, which were obtained from literature and the thermal power balance respectively, the values are scaled to pertain to the sixty-degree fluid sector. The pressure outlet condition is set to zero where the operating pressure outside of the model is set to the combustor exit static pressure from GasTurb.

Although both non-reacting and reacting simulations are conducted for the premixed CJ610 geometry, hydrogen fuel flow is present in all calculations which requires the use of Ansys Fluent’s Species model. The Species model configuration and setup is identical for all cases. While the current work aims at establishing a fully premixed flame by design, the modeling procedure incorporates the partially premixed combustion model to simulate the combustion process.

Within the partially premixed combustion modeling methodology, there are several inputs to be defined. First, the turbulence-chemistry interaction process has to be set and is here modeled with the flamelet-generated-manifold (FGM) model, which represents thermo-chemistry via tabulated mixture fraction and a reaction progress variable [56]. The flamelets are composed with a validated chemical kinetics mechanism for hydrogen [58]. This mechanism has been validated for a wide range of temperature, pressure and equivalence ratios. Alongside this kinetic mechanism, the thermodynamic and chemical transport databases corresponding to the combustor operating pressure of 616 kPa are generated for the simulation. This is especially important for the mass transport and mixing of both air and hydrogen considered here. The turbulent combustion process is also modeled with the probability density function (PDF) method. Coupled to the flamelet parameters, the generated PDF table defines the combustion characteristics in terms of heat and species.

At the fuel inlet of the combustor, the mass fraction of hydrogen fuel is defined as unity and its temperature set to 300 K while the pressure is set near the combustor operating pressure. Whereas future thermal-powered aircraft will likely operate with hydrogen fuel stored cryogenically, the liquid fuel will be heated up to gas phase through the heat exchanger unit of the LFP injector with temperature of the order of 100 K maxi. For the present work, because the design of the heat exchanger and the exact thermodynamic conditions are not known, a near-atmospheric temperature of 300 K at the fuel injection outlet is assumed as a starting point.

For the reacting numerical simulations, a virtual ignition process is applied in the combustion chamber. To ignite the mixture, the GE-CJ610 fluid domain is split into two cell zones: the diffuser and the combustor. The diffuser zone consists of the entire domain upstream the combustion chamber as well as the dilution flow path around the combustor casing. The combustor zone solely features the combustion chamber. By dividing the fluid volume into two sections, the ignition is applied in the relevant zone only. With the FGM model that is characterized by species mixture fraction and progress variable, a progress variable equal to zero represents a unburnt mixture whereas a value of one corresponds to a burnt mixture. After establishing a well-developed, non-reacting flowfield, for both the time-averaged case with both air and fuel streams, the progress variable in the combustor cell zone is set to one to ignite the mixture. A swirling premixed flame is then present in the CJ610 combustor via this virtual ignition strategy. It is worth noting that the flame structure and flame dynamics are not of interest at this stage.

3.5.2 RANS and SBES Turbulence Modeling

The fluid dynamic modeling procedures for the steady-state and transient numerical simulations are now described. The fluid physics and turbulence modeling described in this section originate from Ref. [59]. The steady-state modeling approach is based on a Reynolds-Averaged-Navier-Stokes (RANS) formulation to obtain the mean flowfield in the domain.

Specifically, the turbulence closure for this RANS formulation is based on the realizable k-epsilon model. The realizable method differs from the standard k-epsilon model in two ways: it contains a formulation for the turbulent viscosity and it includes a transport equation for the turbulent eddy dissipation rate. Further, in regards to the current application, the benefits of the realizable model include more accurate spreading rate prediction of planar and round jets and good performance for rotational flows. On the other hand, a limitation of the realizable k-epsilon model relates to the production of non-physical turbulent viscosity in domains with both rotating and stationary fluid zones.

Next, the time-dependent flow modeling technique utilized is introduced. The transient combustor modeling approach employs stress-blended eddy simulation (SBES) for turbulent flow closure. SBES is derived from a hybrid formulation of the governing equations with both large eddy simulation (LES) and Unsteady-RANS (URANS) elements. This hybrid turbulence model offers a practical solution to resolve most scales in numerical simulations with efficient computation by employing RANS near the walls and limiting modeled scales versus resolved scales.

To switch between RANS and LES under the SBES model, a blending function is employed. This function is implemented in both the turbulent viscosity and the viscous stress tensor. Specifically, the URANS regions are computed with the SST k-omega model while the LES regions are based on the dynamic Smagorinsky subgrid scale model. As the combustion process and species transport are modeled with the FGM approach in this study, the SBES method is required to solve several additional scalars: the mean mixture fraction, the mean reaction progress variable, and their respective variances. The impact of SBES is observed through the diffusion term while the mean equations are solved identically between LES and SBES. However, for modeling variances, the two models utilize separate approaches. In the case of LES, an algebraic formulation models the scalar variances at the subgrid scale. With SBES, the variances are obtained by solving transport equations as the equilibrium approach is not accurate in balancing the production and dissipation rates at the integral

length scale.

By employing both RANS and SBES turbulence modeling, unique flowfield data can be retrieved. The RANS calculation can provide mean flowfield profiles in order to observe the dominant mixing patterns in the geometry and flow path. Additionally, the RANS model serves as an essential validation tool between the analytical approach in the performance map and the computational approach with CFD. Defined mostly with LES theory, the SBES modeling approach highlights the scales of turbulent vortical structures within the combustor domain. The turbulent eddies help to pinpoint regions of strong mixing and swirling flow which is expected around the swirler unit as well as between the dilution air flow and combustion products.

Chapter 4

Partially-Premixed Aeroengine Combustor Simulation Results

The outcomes from the computational fluid dynamics simulations of the premixed combustor architecture are now illustrated. The resultant flowfields include non-reacting and reacting premixed hydrogen/air flowfield cases with both RANS and SBES turbulence modeling. To visualize the data, two-dimensional slices are made on the three-dimensional combustor domain where solid components such as the swirler and dilution holes are resolved. The flowfield data are interrogated in terms of physical quantities such as equivalence ratio field. The consistency between the GasTurb boundary conditions compared to the analytical performance map model is discussed. The non-reacting combustor simulations provide insightful data regarding overall design and implementation whereas the reacting flowfields demonstrate initial scenarios of premixed hydrogen combustion at aeroengine operating conditions. These results serve to assess the combustor performance map results with the numerical simulation data while also analyzing the swirling injector design.

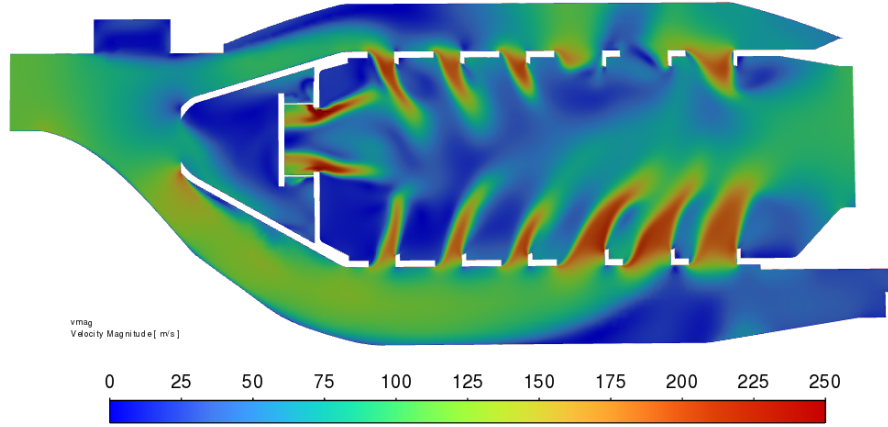
4.1 Non-Reacting Combustor Flowfield

The non-reacting flowfield data consists of premixed hydrogen/air isothermal flow. This section presents the results of the non-reacting simulations obtained with both the RANS

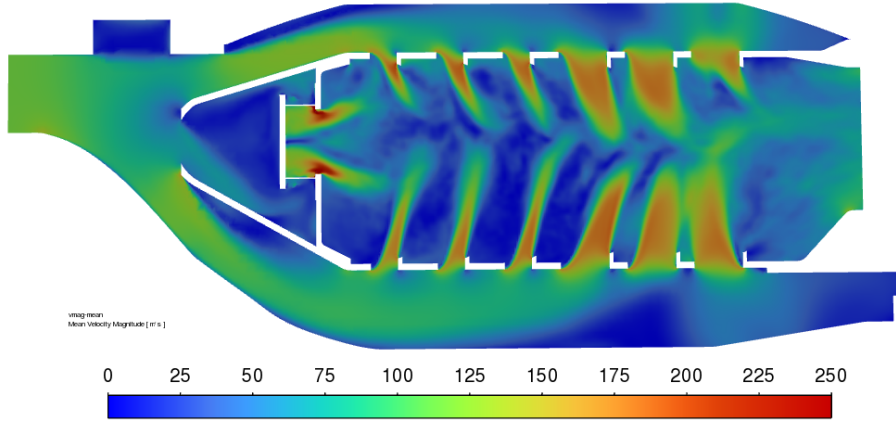
and SBES turbulence closure models. With the goal of comparing the flowfield against the performance map as well as investigating the mixing processes inside the premixed combustor architecture, the results are analyzed through the velocity and equivalence ratio flow profiles. In order to visualize the CFD results, two separate planar-cuts are made in the fluid volume. The first cut acts in the longitudinal direction of the combustor and intersects the geometry at a point where the flow through a swirling injector and row of dilution holes can be observed. The second planar-cut is oriented along the azimuthal direction and offers a direct look at the swirling injector to properly visualize the fuel-air mixing at the injection points. Further, based on the resultant flowfields of both the time-averaged and transient non-reacting numerical simulations, additional results are included into the thesis to discuss the premixed combustor design challenge.

For both the steady state and transient numerical simulations, the outcome of the calculation procedures reveal net mass flux and net heat transfer values on the order of 10^{-3} . These metrics point to well-converged calculations although full convergence is not obtained. Figure 4.1 provides color contour fields of the time-averaged RANS, statistically averaged SBES, and instantaneous SBES calculations in terms of the velocity magnitude in the lengthwise direction. One can observe the complete combustor domain from left to right: the combustor inlet, the swirling injector unit immediately upstream the combustion chamber, the dilution hole row flow paths into the chamber, and the combustor exit. It is important to note that the only outlet is located at the combustion chamber exit as the cooling dilution flow must convect downstream through this outlet as well. Referring to the regions of specific interest at the swirling injection location and the dilution flow paths, strong mixing is observed via the turbulent sheared flow. Additionally, the impact of the swirler unit and its swirl number is observed as the hydrogen-air mixture attaches to the upstream combustor wall which indicates a strong tangential velocity component and rapid mixing. The magnitudes of the dilution flow streams indicate potential rapid cooling as well as promising energy conversion from chemical to kinetic.

(A) : RANS



(B): Averaged SBES



(C): Instantaneous SBES

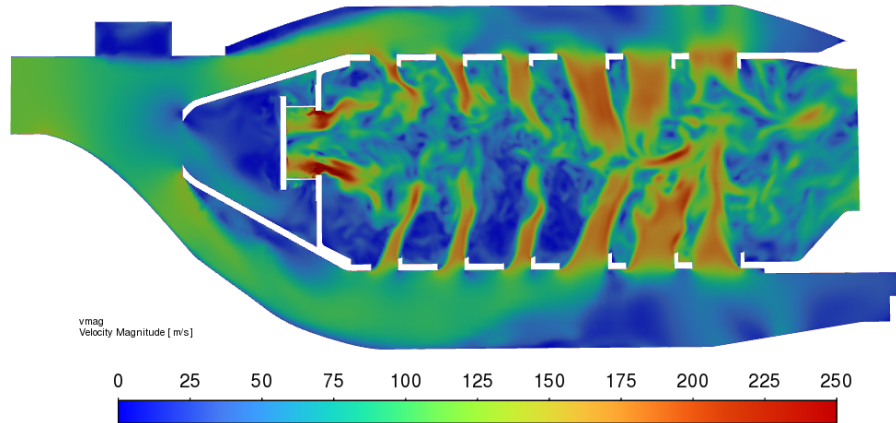


Figure 4.1: Resultant velocity magnitude (in m s^{-1}) flowfields computed with corresponding formulations. Swirling injector unit and dilution hole flow paths are observed. Flow is from left to right.

As Figures 4.1A and Fig. 4.1B highlight the time and spatially-averaged flowfield characteristics, Fig. 4.1C demonstrates an instantaneous snapshot of the fluid domain under the SBES modeling approach. Similar to the averaged results, the greatest magnitudes of velocity are found through the swirling injector and the dilution holes. However, the transient and spatially-averaged methodologies reveal detailed structures of the turbulent flowfield, which gives additional insight to the mixing processes. At the premixed injection point, strong mixing and rotating flow is once again observed around the outer radius of the combustion chamber inlet. Towards the center of the chamber inlet, the swirling flow is also noted which can be attributed to the development of the IRZ due to the high swirl number. Further, the interaction between the dilution flow and injected mixture signals three-dimensional mixing alongside vortical structures at various scales.

Next, Fig. 4.2 compares the results of Fig. 4.1 at the swirling injector outlet plane. Looking at the injector systems with the axial bulk flow direction towards the reader, one can discern the swirler vanes whereas the hydrogen fuel ports are located just behind the vanes. The premixture is obtained as the airflow travels around the outer perimeter of the swirler body where it traverses through the vanes alongside the fuel flow. The strength of the swirling premixture is illustrated through the compact annulus just inside of the vanes as well as the velocity magnitude of the annulus. Figure 4.2A depicts the RANS profile, and Fig. 4.2B displays a statistically-averaged SBES flowfield, and Fig. 4.2C the instantaneous SBES. In all three cases, although not illustrated, the hydrogen fuel ports represent some of the most intense velocity magnitudes which is expected due to the hydrogen species (taken at 300 K for the simulations, see Section 3.5.1) and reduced surface areas of the fuel ports. The overall dimensions of the fuel ports will be the focus of future work as the fuel velocity plays a key role in the mixing processes. As the averaged fields point to a relatively symmetrical flowfield around the injector circumference, the instantaneous SBES in Fig. 4.2C shows varying velocity towards the center of the injector. The resultant velocity profiles throughout the premixed combustor architecture provide valuable insight into characteristics of the present design of a premixed system.

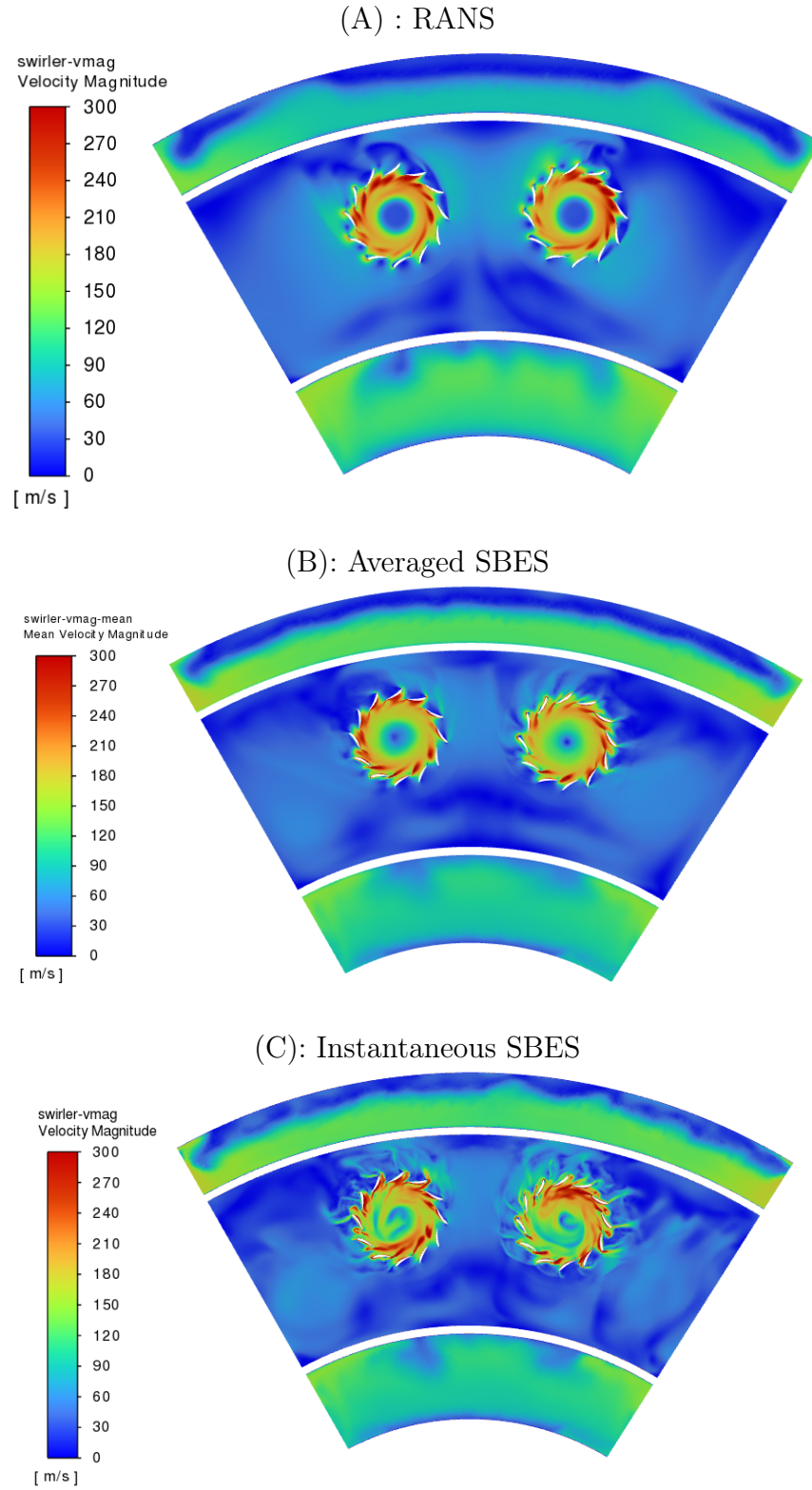


Figure 4.2: Depictions of the velocity magnitude at the swirling injector exit plane. Flow is towards the reader.

With the analysis of the inherent mixing processes being important to this study, another key focus of lean premixed combustion is to mitigate nitrous oxide production and emissions. This is investigated via the equivalence ratio field as seen in Fig. 4.3. Again, the equivalence ratio can be defined with the following expression: $\Phi = \frac{Y_{H_2}/Y_{Air}}{0.029}$, where Y_{H_2} is the mass fraction of hydrogen, Y_{Air} is the mass fraction of air in the mixture, and the constant value is the stoichiometric value for a hydrogen/air mixture. Based on the initial target of lean premixed combustion as well as the results from the performance calculation, an equivalence ratio in the 0.4 - 0.65 range is required. As observed in the equivalence ratio flowfield, the mixture at the expected flame location yields a value of nearly three. This result points to the need for additional analysis as the air and fuel mass flow rate boundary conditions were defined to promote a lean premixture.

Based on the rich equivalence ratio profile at the swirling injector outlet, the first parameter to interrogate is the combustor flow split ratio α . The combustor performance map highlighted a flow split ratio around 0.3 - 0.5 for the given model geometry and lean regime target. The flowfield in Fig. 4.3 reveals a total air mass flow rate of ≈ 0.245 kg/s through the two injector units which corresponds to a flow split ratio of 0.083. This outcome indicates a lack of airflow through the injection ports. Specifically, the regions of interest relate to the swirling injector unit and the upstream combustor geometry. In terms of the swirler design, some parameterization was conducted in order to promote a higher flow split ratio into the combustion chamber. The injector unit modifications included reducing the swirl vane count, scaling down the vane width, and increasing the height of the swirler vanes. These aspects were targeted to increase the effective surface area for the air flow through the swirler vanes. The highest obtained combustor flow split equated to roughly 0.23 where the swirler vane count was decreased and the vane sizes scaled to one half of the original design. However, this design did not mimic the targeted premixed swirl-stabilized regime and was not included in the results. Additionally, the results point to the need for a refined fuel injection scheme as fuel collides with the inlet combustor wall.

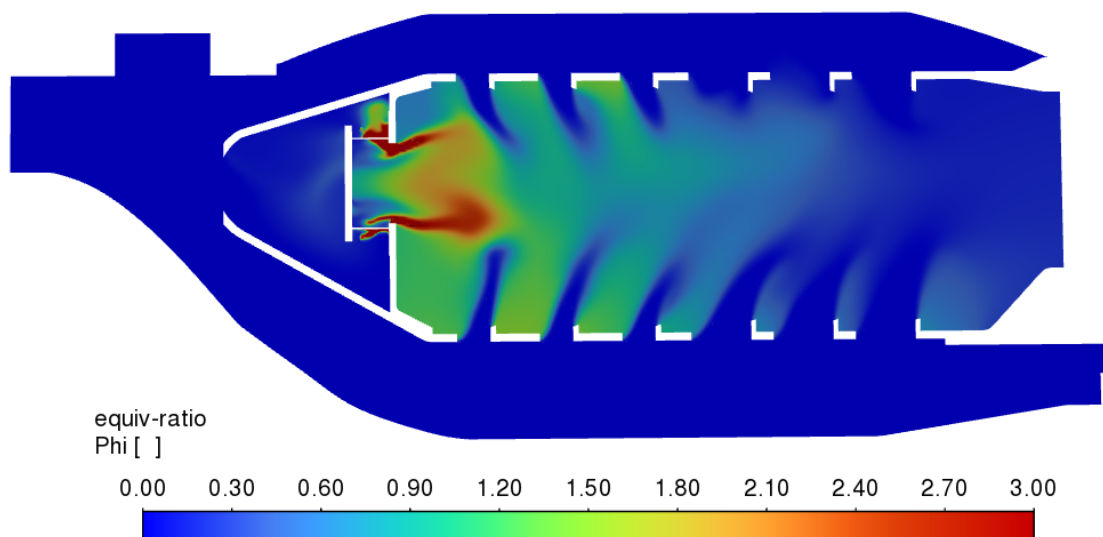


Figure 4.3: Combustor equivalence ratio flowfield. A rich equivalence ratio is observed due to a low combustor flow split value.

To observe the influence of the swirler unit on the air flow split, the swirler bodies were removed from the numerical model assuming a jet flow without swirler. Rather than two swirler bodies positioned at the combustion chamber inlet, there were two holes of the same outer diameter as the swirlers (≈ 46 mm) which resulted in a flow split around 0.25. While this value does appear on the performance map around an equivalence ratio of unity with the constant thermal power curve, it is different from the performance map flow split ratio that was calculated. Ultimately, the result of the combustor flow split investigation points to the need for a refined structural combustor design to guide more airflow into the injector ports. Although a new combustor design is outside the scope of the current work, the current results still provide quality insight into premixed hydrogen combustor flowfield characteristics. For the comparison between the performance map and computational results, more information is detailed in Section 4.3.

4.2 Reacting Flowfields

A reacting case with injected hydrogen fuel is now discussed. Although the results of the non-reacting flowfields highlight an equivalence ratio higher than the performance map prediction, the reacting simulation demonstrates the application of the ignition method described previously and the resultant temperature field. The modeling procedure with the combustion process utilizes RANS to capture the mean temperature profile inside the domain. Figure 4.4 reveals the mean temperature field of the reacting premixed combustor architecture with hydrogen fuel. Compared to the results of Section 4.1, one can observe the structural difference in the two-dimensional planar cut. The swirler vane height is shorter compared to the non-reacting results, and the cone structure surrounding the injector unit is removed. These modifications were made to promote a higher combustor flow split ratio. The high temperature regions around 2500 K in Fig. 4.4 can be explained by the low combustor air flow split which leads to a high equivalence ratio.

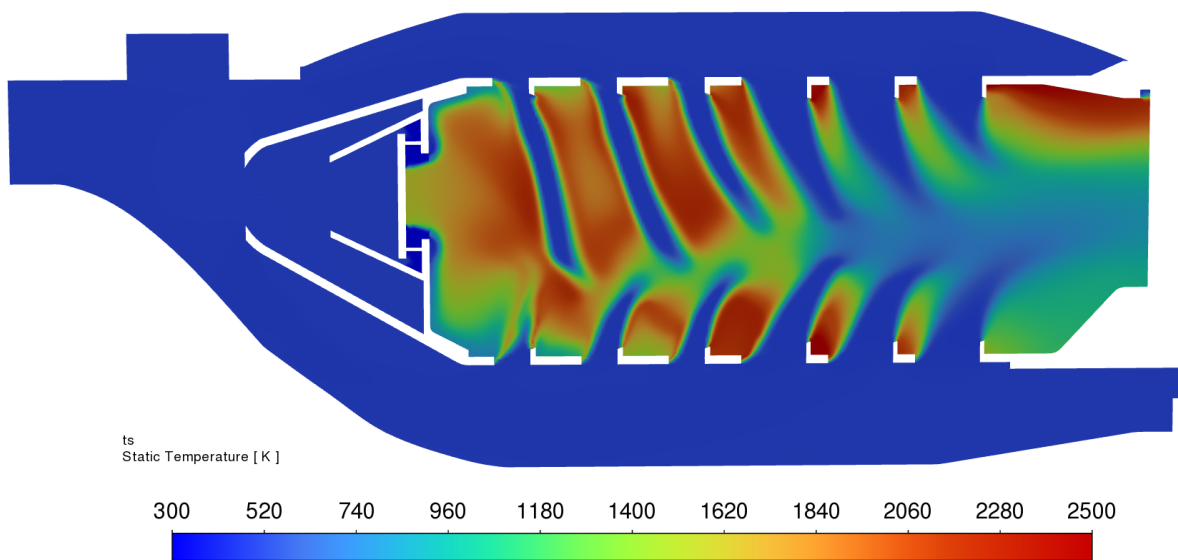


Figure 4.4: Mean temperature field of the combustor.

Although the reacting flowfield does not demonstrate the lean premixed hydrogen/air design point, there is still useful information to be extracted. First, the computational method for the ignition process successfully ignites the mixture which occurs in the combustion chamber only. Three aspects can be highlighted in the reacting flowfield: the high-swirl flame front at the chamber inlet, the cooling dilution hole flow paths, and the temperature of the combustion products at the combustor exit. The adiabatic flame temperature points to a value higher than the 1770 K predicted by the performance map due to the higher equivalence ratio. In regards to the dilution cooling flow, the interaction between burnt gases and the dilution stream enables the maximum turbine inlet temperature of 1170 K to be satisfied. This finding suggests that the total dilution hole count could be optimized to enhance the combustor flow split while accounting for downstream material constraints.

4.3 Performance Map Validation and Next Steps

This section serves to compare the analytical model in the combustor performance map to the computational model from the CFD simulation results of the premixed GE-CJ610 combustor. The main parameters to be interrogated mostly originate from the GasTurb design point calculation such as the mass flow rates and upstream/downstream thermodynamic conditions in temperature and pressure. These values are defined identically in both models for cross-validation as engine experimental data cannot be obtained. Specifically, the validation process focuses on three performance map parameters, combustor flow split, equivalence ratio, and bulk swirler velocity. Based on the derivations and assumptions behind the performance map requirements, the axial velocity flowfield of the combustor domain serves as a comparison point and the bulk swirler velocity magnitude is extracted from the CFD data. Figure 4.5 showcases this resultant flowfield.

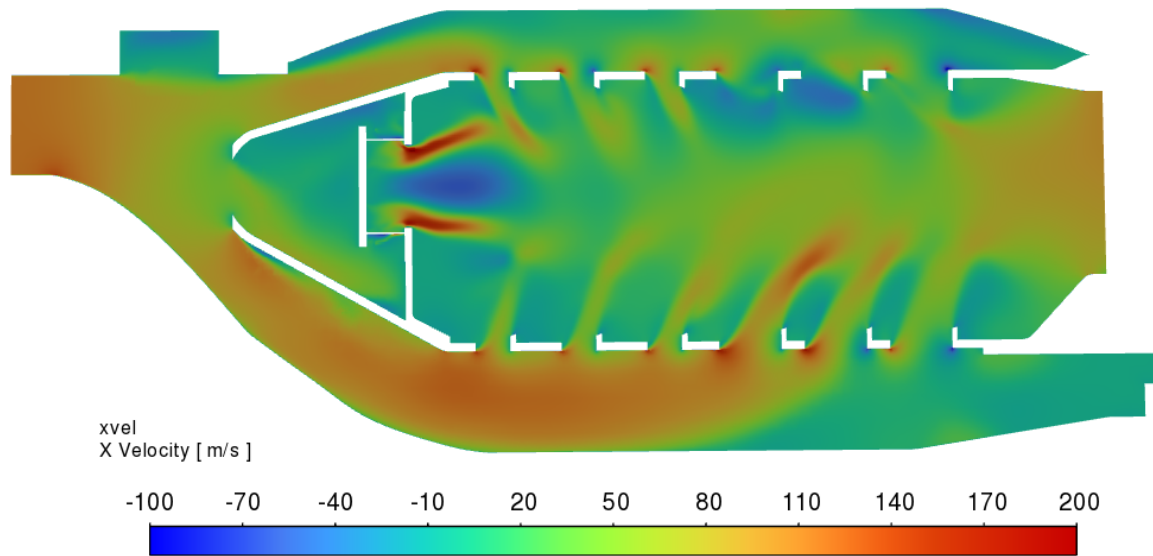


Figure 4.5: Axial velocity profile computed with RANS. The swirling injector unit induces an inner recirculation zone at the injection point which can be noted by the negative velocity field.

The axial velocity flow profile provides information related to the bulk velocity of air through the swirler unit. The premixture velocity into the combustion chamber is an essential parameter to study for mixing and combustion stability purposes as well as in the performance map operating point identification. Looking back to Fig. 4.5, one notes the axial flowfield at the chamber injection point which contains a strong velocity gradient and shear layer between the central and outer injection zone. This three-dimensional flow feature is attributed to the swirler geometry where the outer swirling flow in the positive direction is accompanied by an inner recirculation zone towards the injector core to promote a swirl-stabilized flame. The axial velocity magnitudes in this region vary strongly. The bulk axial velocity through the injector is next post-processed for the purpose of comparing against the performance map's estimation.

With post-processing tools, the average swirler bulk velocity into the swirler unit is determined to be 126.6 m/s. This value is substantially lower than the 309 m/s predicted from Table 2.1A. The area ratio A_d/A_s of the dilution holes and swirlers is now parameterized to aim for a reduction of the bulk swirler velocity. The parameterization process consists of comparing the area ratio of the CJ610 combustor architecture to a conceptual geometry that reduces the bulk velocity while satisfying the determined operating point from the performance map. To do so, Eq. 2.42 is first recalled from previously to serve as a coupling relationship between the performance map and CFD modeling. This equation is a function of both the combustor flow split α and combustor area ratio A_d/A_s that can highlight the observed discrepancies between the analytical and computational models. The area ratio of the current premixed CJ610 combustor, $A_d/A_s = 6.62$, is plotted on the performance map of Fig. 4.6 as a function of α . Although the green curve on the y-axis yields v_s/v_d rather than Φ , it still maintains traction with the combustor flow split on the x-axis. With the initial observation, it is apparent that a combustor flow split ratio greater than 0.2 cannot be obtained with the current CJ610 area ratio and concludes that the geometry is lacking the correct A_d/A_s for the premixed design point.

Next, the area ratio of a conceptual combustor geometry is analyzed via the performance map in Fig. 4.6. Again, the goal is to determine an area ratio that substantially reduces the bulk swirler velocity while promoting a higher and more consistent combustor flow split ratio according to the performance map. As observed in the plot, the area ratio of the GE-CJ610 geometry does not coincide with the performance map operating point in terms of α . The conceptual area ratio is parameterized until agreement with the performance map requirements is obtained as a function of α . In order to obtain the 0.445 combustor flow split from Table 2.1, an area ratio A_d/A_s equal to 0.625 is required according to the map. This combustor geometry then corresponds to a velocity ratio v_s/v_d of 0.5 and a substantially reduced bulk velocity v_s of less than 100 m/s. The conceptual premixed combustor design requires an area ratio reduction of nearly an order of magnitude which will be undertaken in future work. In order to achieve the determined area ratio, the swirling injector surface area will have to be increased while decreasing the dilution hole surface areas. These modifications will require careful consideration in terms of the fuel/air premixed process and the dilution hole cooling scheme. While the validation procedure has shown discrepancies between the analytical performance map and simulation results, the performance map requirement from Eq. 2.42 demonstrates that the root cause lies in the dilution to swirler surface area ratio.

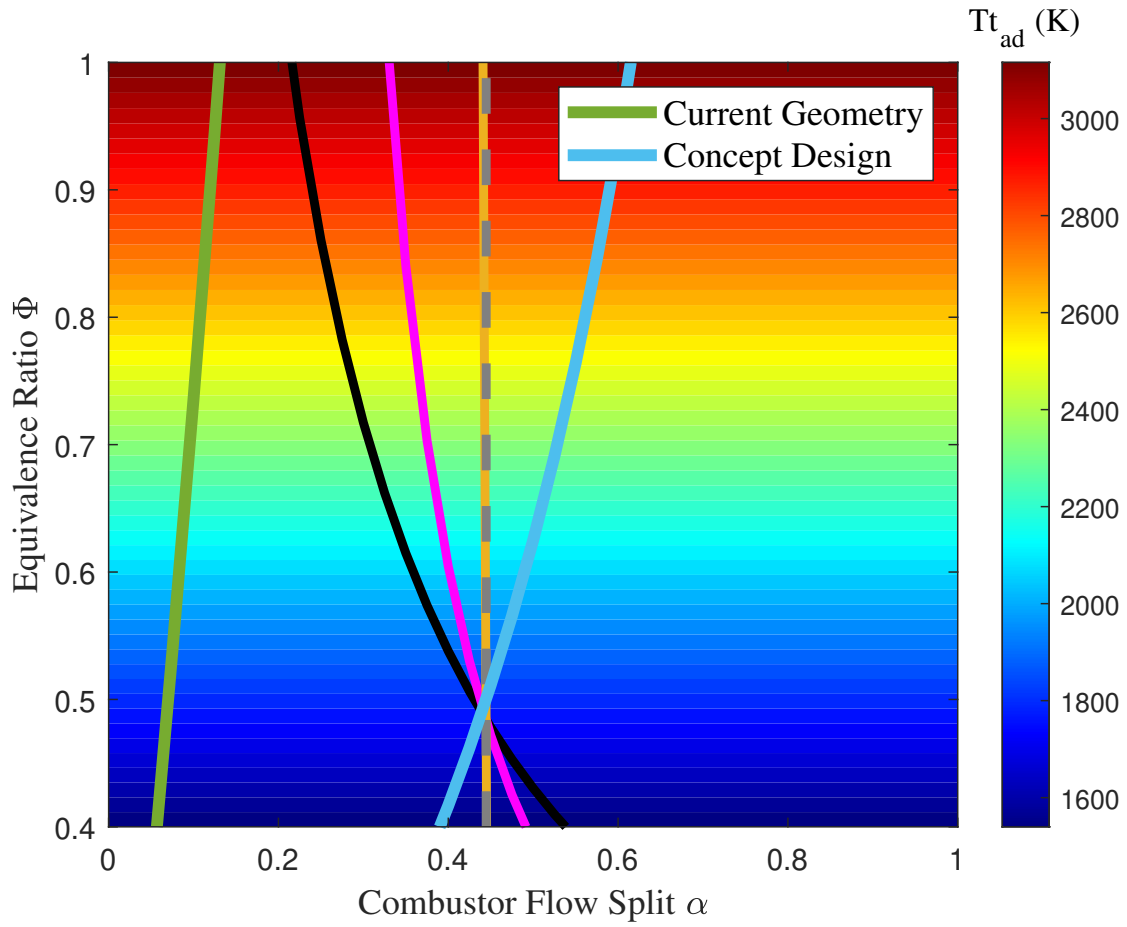


Figure 4.6: Combustor performance map coupling relationship with parameterization of combustor area ratio from Eq. 2.42. It is observed that the current surface areas of both the swirler and dilution holes need to be adapted for retrofit. The requirement of Eq. 2.42 provides an additional key requirement for accurate operating point agreement between the two models.

Conclusion

The decarbonization of the civil aeronautic transport sector is a major task facing today's airline industry and governmental agencies across the globe. Although civil air travel currently accounts for less than 5% of global greenhouse gas emissions, the worldwide demand for air travel is expected to steadily increase in the coming decades. As aircraft and aeroengine manufacturers constantly reduce fuel consumption by improving designs and turbomachinery efficiencies, these means of emission reduction cannot fully achieve the net-zero goal due to the hydrocarbon fuel types utilized. Therefore, alternate fuel types and propulsion methods are at the forefront of aeroengine design and innovation. A leading candidate for the decarbonization of aircraft gas turbine engines and hence the civil transport sector is hydrogen fuel. Hydrogen combustion eliminates carbon dioxide production; however, the hydrogen/air reaction can emit nitrous oxides as well as soot particulates. To address these concerns, premixed hydrogen combustion in the lean regime promotes complete fuel burn efficiency and the minimization of nitrous oxide production. Prior to the implementation of premixed hydrogen/air in civilian transport, there are many aspects that must be considered such as fuel storage and supply, injector systems, and premixed combustion characteristics. The work conducted in this study presents a novel approach at determining suitable operating points for premixed aeroengine combustors at relevant engine conditions.

In this thesis, a premixed combustor performance map is derived and obtained by correlating aeroengine performance specifications to premixed combustion characteristics. The map leverages thermodynamic turbomachinery data alongside chemical kinetics data in order to determine a realistic combustor operating point with premixed hydrogen. The

thermodynamic aeroengine station data comes from a commercial gas turbine performance software whereas the chemical kinetics data corresponds to one-dimensional hydrogen flame simulations. With the two datasets, classical conservation laws are applied and uniquely defined to relate to a premixed combustor geometry. These governing equations are defined in terms of combustor flow split α , equivalence ratio Φ , and stagnation flame temperature $T_{t,f}$ and serve as key requirements in the operating point identification. Based on a given aeroengine's general parameters, the combustor performance map utilizes the five requirements to highlight a unique α and Φ value to describe the combustion chamber conditions. This methodology allows for efficient calculation of a premixed operating point for a wide range of aeroengine types.

This work utilizes computational fluid dynamics as a means of supporting and validating the analytical performance map model. Not only does the CFD data serve as a comparison point to the performance map, but it also provides initial insight into premixed hydrogen combustor design and implementation. The CFD simulations present non-reacting steady state and transient premixed combustor flowfields which highlight key features within the domain in regards to the mixing processes, thermodynamic flow profiles, and design efficacy at realistic engine conditions. Additionally, a reacting case is conducted to demonstrate an artificial ignition method and observe the effects of the dilution hole cooling flow. Towards verifying the analytical model with the computational results, the axial velocity flowfield serves to compare the two as the bulk swirler velocity is a relevant parameter in the performance map methodology. After comparing the velocity term between both models, it is determined that the computational result does not match the performance map value due to an inaccurate combustor flow split ratio.

The resultant CFD flowfields reveal a combustor flow split relatively lower than predicted by the performance map. This is evident in the equivalence ratio combustor profile that illustrates a premixture more rich than anticipated. Additionally, the lack of air mass flow through the swirling injector units contributes to a bulk velocity that is inconsistent with the analytical model. In order to resolve the discrepancies between the performance

map and computational modeling, an additional requirement is introduced that defines the combustor flow split as a function of the swirler to dilution hole velocity and area ratios. This relationship is investigated for the GE-CJ610 combustor geometry as well as a conceptual future geometry. The conceptual geometry highlights that by scaling the swirler to dilution hole area ratio by an order of magnitude greater, an accurate combustor flow split can be achieved while also decreasing the swirler bulk velocity to a suitable level for combustion purposes.

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Vita

Tate Prater is from Chattanooga, TN and obtained his Bachelor's degree from the University of Tennessee, Knoxville in Aerospace Engineering. Following his undergraduate education, Tate attended the University of Tennessee Space Institute to pursue a Master's degree and was a Graduate Research Assistant for the Combustion and Propulsion for Aviation Research Center (C-PARC). Tate will be joining The Boeing Company as a propulsion engineer following graduate school.