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DESIGN METHODOLOGY FOR A LEAN PREMIXED PREVAPORIZED CAN COMBUSTOR

By
Marc R.J. Charest

A thesis submitted to
the Faculty of Graduate Studies and Research
in partial fulfillment of
the requirements for the degree of

Master of Applied Science

Ottawa-Carleton Institute for
Mechanical and Aerospace Engineering

Department of
Mechanical and Aerospace Engineering
Carleton University
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**Design Methodology for a Lean Premixed Prevaporized
Can Combustor**

submitted by **Marc R.J. Charest**
in partial fulfillment of the requirements for
the degree of Master of Applied Science.

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Abstract

This thesis documents the development of a design algorithm for a modern lean premixed prevaporized (LPP) combustor. The methodology was applied to a 1-MW marine gas turbine engine to illustrate the design procedure.

The algorithm includes a set of preliminary design procedures involving the use of empirical and semi-empirical models. These models capture complex processes such as droplet evaporation, chemical reaction, jet mixing, and heat transfer. Evaluation of the structural integrity of the design was also performed using the theory of solid mechanics.

The preliminary design procedures were verified using the advanced numerical techniques of computational fluid dynamics (CFD) and finite element analysis (FEA). These techniques are used to solve the swirling flowfield inside the premixer, the reacting flowfield inside the liner, and the complex stress state in the liner walls. Although CFD and FEA indicated that the preliminary design was successful, some large discrepancies existed between the predictions. These findings suggest the need for more complex numerical models and experimental testing to validate the preliminary design.

A three-dimensional solid model of the combustor and a complete set of engineering drawings were prepared and included as part of the mechanical design.

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To my family.

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List of Symbols

Variables

$\left(\frac{A_w}{A_c}\right)$	Outer liner wall to casing inner wall surface area ratio
$\left(\frac{A}{F}\right)$	The air to fuel mass ratio
A	Area
AR	Outlet to inlet diffuser area ratio
B_T	Spalding transfer number
B_{st}	Stoichiometric fuel mass-transfer number
$\left(\frac{C}{H}\right)$	Carbon to hydrogen ratio by mass
C	Convective heat flux
C_D	Drag coefficient or Hole discharge coefficient
C_p	Coefficient of static pressure recovery or specific heat
C_p^*	Maximum pressure recovery coefficient in a prescribed length
C_p^{**}	Maximum pressure recovery coefficient in a prescribed area ratio
C_S	Flameholder shape factor
c_v	Length
D	Diameter
D_c	Flameholder characteristic dimension
D_h	Hydraulic diameter
D_q	Quench distance
E	Activation energy
e	Trip strip height
E_{min}	Minimum ignition energy
F	Force
f_r	Friction factor
$\bar{g}_f^0$	Gibbs function of formation

G	Gibbs free energy or mass flux from droplet surface
G_ϕ	The axial flux of momentum
G_x	The axial flux of angular momentum
$(\frac{h}{c})$	Aspect ratio
$\bar{h}$	Enthalpy (molar basis)
h	Enthalpy (mass basis) or coefficient of convective heat transfer
H_0	Duct height
H_{eq}	Equivalent duct height
h_{fg}	Latent heat of vaporization
I	Combustor loading
J	Hole momentum flux ratio
K	Conductive heat flux or hole coefficient
k	Constant of proportionality
K_p	Equilibrium constant
k_{area}	Liner-to-casing area ratio, $= A_{liner}/A_{ref}$
K_{sw}	Swirler vane parameter
L	Length or luminosity factor
l_b	Beam length
$\dot{m}$	Mass flow rate
MW	The molecular weight
N	Diffuser axial length, total number of species, or number of moles
n	Quantity
Nu	Nusselt number
P	Pressure
p	Trip strip spacing
Pr	Prandtl number
Q	Volume flow rate or pattern factor
q	Dynamic velocity head, heat transfer, turbulent fluctuation, or fuel/air ratio by mass
$\bar{R}$	Universal gas constant, $=8.3144 \text{ J mol}^{-1} \text{ K}^{-1}$
R	Ideal gas constant, $\bar{R}/MW$
r	Radiative heat flux, radial location, or radius.
Re	Reynolds number

S	Circumferential hole spacing
S_L	Laminar flame speed
S_n	The swirl number
$\bar{T}$	Mean temperature
T	Temperature
t	Thickness or time
T_0	Unburned mixture temperature
T_f	Reacting mixture temperature
TI	Turbulence intensity
U	Overall heat transfer coefficient
u	Component of axial velocity
V	Velocity or volume
v	Component of radial velocity
V_{BO}	Blowout velocity
W	Mass flow fraction or power
w	Component of tangential velocity
$W_{1/2}$	Jet half width
We	Weber number
$[X]$	Molar concentration
X	Axial distance downstream of hole to jet centreline
X	Axial distance.
x_s	Stoichiometric coefficient
$\bar{Y}$	Mean mass fraction
Y	Mass fraction or radial location of jet centreline from liner wall

Greek Symbols

α	Coefficient of thermal expansion, ratio of hole cross-sectional area to annulus cross-sectional area, or spray angle
α_0	Thermal diffusivity upstream of the flameholder
σ_e	Von Mises equivalent stress
σ_p	Proof stress
σ_y	Yield stress
β	Ratio of hole mass flow rate to annulus mass flow rate

χ	Molar fraction
ΔG_T^0	Standard-state Gibbs function change
ΔP	Pressure loss
δ	Displacement
$\frac{\Delta P_{3-4}}{P_3}$	Pressure loss coefficient
$\frac{\Delta P_{3-4}}{q_{ref}}$	Pressure loss factor
ε	Emissivity or strain
ε_T	Thermal strain
η	Efficiency
η_c	Combustion efficiency
λ	Thermal conductivity or diffuser loss coefficient
μ	β/α
$\dot{\omega}$	Reaction rate
ϕ	Equivalence ratio
ρ	Density
σ	Surface tension, solidity, stress, or the Stefan-Boltzmann constant ($=5.67 \times 10^{-8} \text{ W}/(\text{m}^2 \cdot \text{K}^4)$)
τ	Particle time constant
θ	Temperature difference ratio, wall angle, or blade angle

Subscripts

0	Initial condition
1	Engine inlet or inner liner wall
2	Compressor exit or outer liner wall
3	Combustor inlet
4	Combustor exit
5	Gas generator exit
6	Power turbine exit
∞	Ambient condition
θ	Hoop direction
a	Property of air, or axial direction
act	Actual quantity
an	Annulus property

<i>avg</i>	Average value
<i>b</i>	Bulk flow
<i>bolt</i>	Bolt property
<i>c</i>	Jet centreline
<i>casing</i>	Casing
<i>contract</i>	Contraction
<i>conv</i>	Property of a conventional combustor
<i>diff</i>	Diffuser
<i>dilution</i>	Dilution holes
<i>dome</i>	Dome property
<i>dump</i>	Dump
<i>DZ</i>	Dilution zone
<i>evap</i>	Evaporation
<i>f</i>	Property evaluated at film temperature or property of fuel
<i>flame</i>	Flame property
<i>g</i>	Gas
<i>h</i>	Dilution hole property
<i>hub</i>	Hub
<i>i</i>	Interface between TBC and liner wall, or species
<i>ig</i>	Ignition
<i>in</i>	Inlet
<i>j</i>	Jet
<i>l</i>	Property of liquid fuel
<i>liner</i>	Liner property
<i>max</i>	Maximum value
<i>min</i>	Minimum value
<i>mix</i>	Mixture property or mixer swirlers
<i>mix, i</i>	Inner mixer swirler
<i>mix, o</i>	Outer mixer swirler
<i>n</i>	Fuel nozzle
<i>out</i>	Outlet
<i>overall</i>	Overall property.
<i>p</i>	Particle

<i>plenum</i>	Plenum
<i>prediff</i>	Prediffuser
<i>premixer</i>	Premixer
<i>pump</i>	Pump property
<i>PZ</i>	Primary zone
<i>R</i>	Droplet surface
<i>r</i>	Relative quantity or radial direction
<i>ref</i>	Reference parameter
<i>snout</i>	Combustor snout
<i>stoic</i>	Stoichiometric quantity
<i>sw</i>	Swirler or swirler tip
<i>swirl</i>	Combustor swirler
<i>t</i>	Tensile
<i>theo</i>	Theoretical quantity
<i>tot</i>	Total
<i>u</i>	Unburned mixture
<i>v</i>	Vane property or property of fuel vapour
<i>w</i>	Liner wall property

Acronyms

ABC	Augmented Backside Convection
AEC	Angled Effusion Cooling
CAD	Computer Aided Design
CFD	Computational Fluid Dynamics
DLE	Dry low emissions
DLN	Dry low NO _x
DZ	Dilution Zone
FEA	Finite Element Analysis
NO _x	Oxides of nitrogen (NO and NO ₂)
PFR	Plug Flow Reactor
PZ	Primary Zone
RQL	Rich-quench/lean-burn
SZ	Secondary Zone

TBC	Thermal Barrier Coating
TIT	Turbine Inlet Temperature
UHC	Unburned hydrocarbons
WSR	Well Stirred Reactor
YSZ	Yttria Partially Stabilized Zirconia

Chapter 1

Introduction

Governmental regulations for gas turbine emissions are becoming increasingly more stringent. The demand for new improved designs with ultra-low pollutant emissions is rapidly moving to the forefront of combustor development. Radical modern combustor designs have emerged to achieve emission requirements while maintaining the high combustion efficiency and good flame stability characteristics of conventional combustors. Lean premixed (LP) for gaseous and lean premixed prevaporized (LPP) for liquid fueled engines are two modern designs proven successful at meeting governmental regulations. Since published design methodologies for conventional combustors do not apply well to these modern designs, and current designs of these are typically regarded as proprietary, there is a need for the development of new design methodologies, particularly for LPP combustors. This thesis documents the development of these methods and then applies them to a 1-MW marine gas turbine.

1.1 Gas Turbine Emissions

The exhaust of gas turbine combustors contains several primary pollutants: oxides of nitrogen (NO_x), unburned hydrocarbons (UHC), carbon monoxide (CO), and particulate matter or smoke. The design of conventional diffusion flame combustors was first concerned with combustion efficiency, the actual heat released divided by the total heat potential of the fuel consumed, which meant reducing the UHC emissions (Saravanamuttoo *et al.* , 2001). Low combustion efficiency is characterized by the

occurrence of unburned hydrocarbons (UHC) and CO in the exhaust.

Efforts to minimize UHC emissions were followed by the elimination of visible smoke, a problem common to the diffusion (non-premixed) flames that are used in conventional combustors. Some of the fuel can pyrolyse to form fine soot particles that are visible as smoke. Pyrolysis is the thermal decomposition of fuel when heated in the absence of oxygen.

In the 1980s, the primary pollutant concern was shifted to the oxides of nitrogen (NO_x) as combustion efficiencies upwards of 99.8 percent became common and visible smoke emissions had been eliminated (Lefebvre, 1999). The Environmental Protection Agency's (EPA) "top-down approach" for determining the Best Available Control Technology (BACT) became a requirement in 1987 (Schorr, 1991) and forced limits of NO_x emissions for industrial gas turbines to be determined based on the current available technology. These regulations demanded the development of new designs such as water or steam injection, which lowered NO_x levels considerably by reducing the flame temperature. NO_x formation rates are high in conventional combustors due to the high peak local flame temperatures typical of diffusion flames.

The use of steam or water injection resulted in slight penalties in cycle efficiency that were initially accepted because of the corresponding reduction in NO_x emissions. Carbon monoxide (CO) emissions rose drastically as more and more water was injected to meet the continuously lowering NO_x limits. It was realized that radical new combustor designs would be required to satisfy the conflicting requirements for stable, efficient combustors with low NO_x emissions (Schorr, 1991). New modern designs included catalytic, rich-quench/lean-burn (RQL), and dry low NO_x (DLN) or dry low emissions (DLE) combustors. DLN combustors were developed first and evolved into DLE designs as the focus of emissions reduction turned towards ultra-low levels of NO_x , CO, and UHC. For example, Figure 1.1 illustrates the trend in emissions reduction as combustor designs developed for the Pratt & Whitney FT4 industrial engine.

Figure 1.1: Pratt & Whitney FT4 emissions. From Gauthier (2003).

DLE concepts have been successfully implemented by many large engine manufacturers such as General Electric (GE), Rolls Royce, Solar Turbines, Siemens-Westinghouse, the ABB company, and the European Gas Turbine Company (EGT) (Lefebvre, 1999).

1.1.1 Dry Low Emissions Concepts

DLE combustors reduce emissions by reducing the peak flame temperatures without resorting to steam or water injection (hence the term "dry"). Designs use one or a combination of several concepts: axial or radial staging, variable geometry, and premixing.

The stability of a flame is characterized by the ability to burn steadily at a fixed position without blowing out. Aircraft engines have strict stability requirements and therefore have only incorporated combustor staging and variable geometry with conventional combustor designs to avoid any unacceptable losses in ignition and blowout

Figure 1.2: Gas turbine combustion air/fuel ratios. Modified from Gauthier (2003).

performance. Since land-based engines have much more demanding emissions regulations and are less concerned with stability, modern industrial gas turbine combustors use premixed flames.

1.1.2 Lean Premixed Combustors

Premixed flames are distinguished from diffusion flames by the mixing of the fuel and air prior to ignition. Premixing allows careful control over the local flame temperatures in the combustion zone to avoid any potential NO_x -forming hotspots. Extremely well premixed flames exhibit ultra-low NO_x , CO, and UHC emissions if the flame temperature is maintained below 1900 K and given sufficient residence time (Leonard & Stegmaier, 1993). DLE designs using premixed flames are called lean premixed (LP) and lean premixed prevaporized (LPP) for gaseous and liquid fueled combustors respectively. These concepts use mixtures close to the lean limit of flammability, shown in Figure 1.2, to minimize flame temperatures and ensure that NO_x formation is limited.

Figure 1.3 illustrates the differences between conventional diffusion and modern premixed combustors. In both types, the air is divided into two streams. The first stream enters the primary zone (PZ) for combustion and the second is diverted into

(a) Conventional diffusion flame

(b) Premixed flame

Figure 1.3: Comparison of combustor types. Modified from Gauthier (2003).

the annulus for cooling and dilution purposes. The premixed design mixes the fuel and air prior to injection into the PZ, whereas mixing of the fuel and air for diffusion flames does not occur until inside the PZ. A smaller fraction of the air is diverted to the annulus in premixed designs as leaner fuel/air mixtures are required to reduce the flame temperature (Figure 1.2). This reduces the amount of air available for cooling purposes and necessitates the use of advanced techniques such as augmented backside cooling (ABC) to accomplish liner wall cooling. Conventional film cooling slot designs limit the amount of air available for leaner combustion. They must also be avoided as the admission of cooler air into the PZ can potentially reduce flame stability and increase CO emissions.

In conventional combustors additional air is admitted through holes in the liner into the secondary zone (SZ) to allow the complete oxidation of CO into CO_2 . Premixed combustors do not require a SZ as their lower peak flame temperature minimizes the dissociation of CO_2 into CO. The hot combustion products are then diluted with the remaining annulus air in the dilution zone (DZ). Crossflowing jets of cold air mix with the hot combustion products to lower the combustor exit temperature and trim its profile. Less time for mixing in the DZ is required for premixed combustors as the peak flame temperatures are significantly lower than those in conventional ones.

There are many industrial engines that have successfully incorporated LP and LPP combustors to achieve ultra-low emissions. The GE LM6000 and the Rolls Royce Industrial RB211 both use LP combustors that operate on natural gas and have demonstrated NO_x , CO, and UHC emissions less than 25 ppmvd each (Leonard & Stegmaier, 1993; Willis *et al.*, 1993). Nissan's Dry Low NO_x combustor achieves these emissions reductions as well (Hosoi *et al.*, 1996). Examples of engines that use LPP combustors are the OPRA OP16 (Klein *et al.*, 2002) and the Allison AGT100. The Solar Turbines family of engines use both LP and LPP combustors to achieve low emissions (Smith & Cowell, 1989; Smith *et al.*, 1986). Some recent design efforts of advanced LPP concepts include those of Wedlock *et al.* (1999), Liedtke *et al.* (2002), and Lin *et al.* (2004).

1.2 Motivation

The LP and LPP combustor designs described previously are considered modern designs and detailed information about them is typically proprietary (Dodds & Bahr, 1990). There exists no published detailed methodology on their design since they are relatively new technology and each design tends to be drastically different from the next. Therefore, it is difficult to formulate a general set of preliminary design procedures for premixed combustors since they have not yet converged on a widely accepted design. These procedures are the initial step in the overall design process, which precede complex numerical analysis using computational fluid dynamics (CFD) and finite element analysis (FEA). They provide the designer with a quick estimate of the combustor's performance.

The preliminary design published by Sawyer (1985) provides a starting point for gas turbine combustor designers. However, it is focused on aircraft engines and does not provide design guidelines for LP and LPP combustors. This complete design procedure for aircraft engines is dated and does not incorporate the latest empirical correlations or numerical models. There is a need to update Sawyer's design and modernize it to apply to the design of industrial LPP combustors.

1.3 Objectives and Tasks

The objective of this thesis is to develop a methodology for the design of a LPP gas turbine combustor. The purpose is to update the literature and to provide the reader with a simple algorithm that can be used to produce a stable, efficient, and feasible design. The specific tasks necessary to accomplish the objectives are:

1. development of a design concept,
2. formulation of a preliminary design procedure,
3. application of the design procedures to a 1-MW marine gas turbine, and
4. verification of the design with numerical analysis.

The work is applied to the design of a 1-MW marine gas turbine engine of twin-shaft configuration.

1.4 Outline

Included in this thesis is a survey of relevant publications, an overview of the theory used, a preliminary design, a detailed design with computational fluid dynamics (CFD) and finite element analysis (FEA), a complete mechanical design, a comparison between the results from the preliminary design and the numerical simulations, and a conclusion with suggestions for future work.

Chapter 2

Literature Review

This chapter provides an overview of gas turbine combustor design and focuses on modern can combustors for industrial applications. The sections will direct the reader to various models and correlations that will be utilized in subsequent chapters. This chapter discusses the following principles:

- combustor design,
- diffuser design,
- combustor aerodynamics,
- premixer design,
- combustion chemistry,
- combustion performance,
- droplet processes, and
- liner heat transfer.

The various components reviewed are illustrated in Figure 2.1.

2.1 Combustor Design

The simultaneous involvement of evaporation, turbulent mixing, ignition, and chemical reaction in gas turbine combustion is too complex for complete theoretical treatment. Instead, large engine manufacturers undertake expensive engine development programs to modify previously established designs through trial-and-error (Dodds & Bahr, 1990). They also develop their own proprietary combustor design rules from

Figure 2.1: Modern combustor components.

the experimental results of these programs. These design rules provide a means of specifying the combustor geometry to meet a set of requirements at the given inlet conditions.

Combustor designers without access to proprietary design procedures must derive their own methods from the literature or from experimentation. Numerous published empirical, semi-empirical, and analytical tools have been developed to reduce the need for costly experiments. The two extreme cases, empirical and analytical, differ by the method of derivation. Empirical design tools are correlations derived from experimental datasets whereas analytical ones are discretized versions of the governing equations. Simple empirical correlations provide accurate results quickly and are easily implemented into design codes, yet they are only applicable to cases for which the measured data was based on. Analytical methods, less accurate in comparison to empirical methods, are much more flexible as they are only restricted by the simplifying assumptions necessary to reduce their complexity and computation time. Hybrid semi-empirical tools combine both empirical and analytical methods to provide a reasonable balance between accuracy and computation time.

Authors such as Lefebvre (1983, 1999) and Mellor (1990) have compiled extensive

Figure 2.2: Reference dimensions.

lists of empirical, semi-empirical, and analytical models for the design of conventional aircraft combustors. The authors also provide information and references on new modern combustors that enable the formulation a of preliminary design.

2.2 Preliminary Design

Several authors have published preliminary designs for conventional aircraft gas turbines that provide a starting point for the development of modern industrial combustors (Sawyer, 1985; Dodds & Bahr, 1990; Mattingly *et al.* , 2002). The approaches made by these authors differ mainly by the specification of combustor size.

2.2.1 Combustor Sizing

Combustor sizing refers to the specification of the reference (casing) diameter and the cross-sectional area that defines the total ideal flow area inside the combustor, D_{ref} and A_{ref} respectively (Figure 2.2).

Varying the combustor size affects the residence time and stability characteristics by changing the reference velocity, V_{ref} . The reference velocity, based on the reference cross-sectional area and the combustor inlet conditions, is the effective average velocity through the entire combustor.

Combustor sizing can be performed based on the combustor pressure loss (Lefebvre, 1999), velocity (Dodds & Bahr, 1990), or stability (Mattingly *et al.* , 2002). A

newer design approach provided by Mellor & Fritsky (1990) selects the combustor diameter to meet a specified NO_x emission target.

Pressure Loss Method The pressure loss method selects a reference area A_{ref} to provide a reference velocity head q_{ref} typical of previous designs that exhibit similar pressure losses. This reference area is the maximum flow area between the casing walls. The velocity head or dynamic pressure is the difference between total and static pressure at the design point, defined based on the design point inlet air density and velocity. The reference velocity head is defined as

$$q_{ref} = \frac{1}{2} \rho_3 V_{ref}^2 \quad (2.2.1)$$

where the reference velocity V_{ref} is

$$V_{ref} = \frac{\dot{m}_3}{\rho_3 A_{ref}} \quad (2.2.2)$$

The difference in total pressure between the inlet and outlet of the combustor, called the overall total pressure loss ΔP_{3-4} , together with the reference velocity head determine the size of the combustor. These two quantities are of great importance to combustor design and are generally quoted in dimensionless form as the pressure loss coefficient $\frac{\Delta P_{3-4}}{P_3}$ and the pressure loss factor $\frac{\Delta P_{3-4}}{q_{ref}}$. The pressure loss factor denotes the resistance introduced into the stream by the combustor whereas the pressure loss coefficient dictates the percentage of the inlet total pressure lost. Typical values for the pressure loss coefficient and the pressure loss factor for aircraft engines are listed in Table 2.1. Pressure losses for industrial combustors are much lower, they are typically less than 2 percent (Saravanamuttoo *et al.*, 2001).

The overall combustor pressure loss is the sum of the losses through several components: the diffuser, the swirler, and the liner. The losses through the liner can be further broken down into the cold losses and the hot losses due to combustion. The cold losses arising from turbulence and frictional effects are much larger in comparison with the fundamental losses incurred by the expansion of hot gas. In combination with these losses, those incurred across the swirler benefit combustion and dilution

Table 2.1: Typical pressure losses in aircraft combustion chambers. From Lefebvre (1999).

Type of chamber	$\Delta P_{3-4}/P_3$	$\Delta P_{3-4}/q_{ref}$
Can	0.07	37
Can-Annular	0.06	36
Annular	0.06	20

by increasing air injection velocities, creating steep penetration angles, and promoting turbulence. The goal is to design an aerodynamically efficient combustor that minimizes parasitic losses and maximizes the benefits from pressure loss.

2.3 Diffuser Design

Many processes benefit from low flow velocities in addition to flame stabilization. Lower annulus velocities promote flow uniformity, improved jet penetration at steeper angles, lower skin-friction losses, and lower sudden-expansion losses downstream.

The goal of diffuser design is to minimize the total pressure loss incurred while recovering as much dynamic velocity head as possible. A good design achieves a high static pressure recovery with low pressure losses, is stable, insensitive to fluctuations in inlet conditions or manufacturing tolerances, and short in length (Klein, 1995). Diffusers must also discharge to provide the necessary airflow distributions without any adverse effects from changes in mass flow splits, flow asymmetry, or wakes produced by objects in the flow path (injector fuel lines, struts, etc).

The two main types that are used in gas turbines are faired and dump diffusers (Figure 2.3). Vortex controlled and hybrid diffusers are other types of diffusers that have been proposed for use in gas turbines, yet remain in the research and development phase.

Figure 2.3: Diffuser types.

2.3.1 Dump Diffusers

The dump diffuser is much simpler than faired types and provides a larger degree of diffusion over a shorter distance. Dump diffusers are relatively insensitive to inlet conditions, manufacturing tolerances, and dimensional changes due to thermal expansion, but incur higher pressure losses than the faired designs.

Dump diffusers consist of two sections: a faired prediffuser and a step region. The prediffuser reduces the velocity of the flow by approximately 60 percent (Lefebvre, 1999) before the flow is dumped into the highly separated step region. This initial reduction in velocity helps minimize the dumping losses, which are proportional to the square of the velocity entering the step. Design involves optimizing the division of diffusion between the two sections to accomplish a suitable compromise between pressure loss and overall diffuser length.

Prediffuser The design of prediffusers requires the estimation of their effectiveness at converting kinetic energy into static pressure. This conversion is quantified by the static pressure recovery coefficient, C_p . Another important parameter for diffuser design is the loss coefficient, λ . It is a measure of the total pressure losses incurred by the diffuser due to stalling and frictional effects.

The actual coefficient of static pressure recovery can be determined using empirical or analytical models. The experimental work of Kline *et al.* (1959), Reneau *et al.* (1967), and Sovran & Klomp (1967) on faired diffusers provides designers with a means of selecting an optimal geometry and predicting its performance. These experiments do not take into account the inlet conditions that combustor diffusers must accommodate such as high levels of turbulence, Reynolds and Mach number effects, compressor exit velocity profile non-uniformities, and swirl (Little & Manners, 1993; Agrawal *et al.*, 1996). Nonetheless, the experimental data discussed above provides a suitable preliminary design despite its limitations.

The widely accepted performance chart of Figure 2.4 for conical diffusers created by Sovran & Klomp (1967) provides a means of estimating the static pressure recovery coefficient based on the area ratio AR , length N , and inner radius of the diffuser R_{in} .

Figure 2.4: Performance chart for conical diffusers. Modified from Sovran & Klomp (1967)

The optimum geometry lies between two lines marked C_p^* and C_p^{**} which denote the maximum pressure recovery for a prescribed area ratio and length, respectively. The divergence angle, 2θ , remains approximately constant along the C_p^{**} line with a value of 12° .

Step Region Minimal losses in the step region can be expected if enough diffusion has already occurred in the prediffuser. A large portion of the pressure losses in this region are due to mixing and can be difficult to estimate because of the complicated geometry. Dodds & Bahr (1990) stated that experimental data for these losses corresponds well with the theoretical losses for a step increase in area for uniform, steady, frictionless, incompressible flow.

Difficulty arises when one attempts to estimate the cross-sectional area at the outlet of the dump where the flow reattaches (Figure 2.3(b)). The location of this reattachment point is unknown during the preliminary design phase and requires complex CFD analysis or experimental testing to determine. Dumping losses are also affected by inlet conditions such as turbulence, inlet velocity profile, and the spacing

between the dump and the combustor dome. Accurate estimations of the pressure losses in the dump region require experimentation to fully characterize the diffuser losses due to these complications.

Appreciable losses are unavoidable in the diffuser due to the nature of fluid flows in adverse pressure gradients. Design must focus on minimizing these parasitic losses as they degrade the thermodynamic efficiency of the engine.

2.4 Combustion Aerodynamics

The design and performance of a combustor is strongly affected by aerodynamic processes (Lefebvre, 1999). The performance of many designs differ mainly due to the aerodynamic efficiency, a measure of the effectiveness at introducing and distributing air in the liner (Scull & Mickelsen, 1957). The achievement of aerodynamically efficient designs, characterized by good mixing and stable flow patterns with minimal parasitic losses, is one of the primary design objectives.

2.4.1 Central Recirculation

Anchored flames cannot be established in flows with velocities significantly above the laminar flame speed. Fuel and air must move slowly enough for the flame to propagate upstream and ignite fresh mixture. The point at which the flame can no longer propagate back through the flow is the stabilization point or anchor. Zones of flow reversal help stabilize the flame by creating localized regions of low velocity flow called flameholders.

Large scale central recirculation zones, as shown in Figure 2.5, serve many other purposes as well. Hot combustion products become trapped in the recirculating mass and are returned to the combustor dome inlet. This hot gas helps stabilize the flame by providing a continual source of ignition to the incoming fuel. It also serves as a zone of intense mixing within the combustor by promoting turbulence through high levels of shear between the forward and reverse flows. Lastly, CO, unburned fuel, and other intermediate species are able to reside within the combustor longer

Figure 2.5: Combustor streamlines.

and react to completion. Figure 2.5 illustrates the process of recirculation in a gas turbine combustor using streamlines. These streamlines indicate the path which a fluid particle would follow.

Flame stability, combustion intensity, and performance are directly associated with the size and shape of this recirculation vortex or bubble (Gupta *et al.*, 1984). It forms at the onset of flow reversal when an adverse axial pressure gradient exceeds the kinetic energy of the incoming flow (Beer & Chigier, 1972). Adverse pressure gradients may be introduced by creating high degrees of swirl or angular momentum at the inlet of the combustor and large sudden expansions in areas such as dumps or bluff bodies.

Figure 2.6: Two main swirler types. From Dodds & Bahr (1990).

2.4.2 Swirlers

Swirlers are static mixing devices used to impart swirl to the flow. The goal of swirler design is to maximize the benefits of recirculation by imparting sufficient swirl to the flow while minimizing the incurred pressure losses. Modern combustors also use swirlers to promote mixing of the fuel and air in the premixer prior to combustion.

There are many methods of producing swirl. These include axial, radial, tangential, and discrete jet swirlers. The first two, displayed in Figure 2.6, are typically used in gas turbine engines. Hallett (2003) has provided a brief summary of these different types of swirlers; Gupta *et al.* (1984) and Basu *et al.* (1999) have surveyed existing designs for industrial burners and gas turbine combustors. Syred & Beer (1974) and Lilley (1977) have reviewed much of the earlier work conducted on swirling flows.

2.4.2.1 Axial Swirlers

Axial swirlers tend to have higher pressure losses than the radial type but are much simpler to manufacture. Parameters of interest to axial swirler designers are depicted by Figure 2.7. They include: the vane angle θ_v , the inner hub radius R_{hub} , the outer

Figure 2.7: Design parameters of axial swirlers.

swirler radius R_{sw} , the vane thickness t_v , the vane length c_v , and the number of vanes n_v . Typical axial swirler designs have vane angles between 30° and 60° , vane thicknesses between 0.75 and 1.5 mm, and between 8 and 16 vanes (Dodds & Bahr, 1990).

A useful parameter for design is the Swirl number, S_n . The swirl number is a measure of the ratio of angular momentum flux to axial momentum flux and defined by (Chigier & Beer, 1964)

$$S_n = \frac{G_\phi}{G_x R_{sw}} \quad (2.4.1)$$

where G_ϕ = the axial flux of angular momentum

G_x = the axial flux of momentum

The swirl number determines the criterion for recirculation. Little or no recirculation occurs below the critical value of 0.6 (Lilley, 1977). Above this value a central recirculation zone forms.

In unconfined swirling jets, the recirculation zone increases in length and diameter as the swirl number is increased to a value of 1.5 (Lilley, 1977). The zone continues to increase in diameter beyond this value, however, its length begins to decrease.

Figure 2.8: Flow through liner hole.

Also of interest to the designer is the pressure drop incurred across the swirler. Knight & Walker (1957) and Crocker & Smith (2001) have provided simple correlations for the prediction of the pressure drop. However, the development of more accurate correlations is limited by the lack of available experimental data and accurate computational fluid dynamics models (Dodds & Bahr, 1990).

Other important work includes the experiments of Mathur & Maccallum (1967) and Martin (1988) on vaned swirlers.

2.4.3 Air Admission

Satisfactory mixing performance in both the combustion and dilution zones is vital to good combustion performance, low emissions, and a satisfactory exit temperature profile (Lefebvre & Norster, 1969). Mixing is intensified by admitting high velocity jets of air at oblique angles through circumferential rows of holes into the liner. The process is illustrated in Figure 2.8.

In the combustion zone (i.e. the PZ), these jets serve to strengthen the stability-promoting recirculation zone. Jets issuing from holes in the aft end of the combustor,

in the dilution zone, dilute the hot combustion products and trim the outlet temperature to provide a uniform profile desirable for downstream components. Here, the exit temperature distribution depends primarily on the number of jets and the penetration of the relatively cooler jet air into the hot combustion gases.

The intense mixing between the cross flow and jet is a result of the development of a turbulent shear layer around the periphery of the jet and eddies formed in the jet wakes (Carrotte & Stevens, 1990). This rate of mixing is affected by (Lefebvre, 1999):

- the size and shape of the issuing orifice,
- the initial jet penetration angle,
- the jet-to-mainstream momentum-flux ratio,
- the presence of adjacent and opposed jets,
- the distance downstream,
- the proximity of the liner walls, and
- the inlet velocity and temperature profiles of the jet and hot gases.

Holdeman *et al.* (1984) stated that the momentum-flux ratio is the most significant parameter involved. Together with the hole geometry, this parameter has been experimentally proven by Hatch *et al.* (1995) to determine the degree of penetration and the extent of mixing downstream. Thus, successful hole design entails the selection of a suitable orifice shape, size, and number to provide the desired penetration and mixing characteristics of the jets. Hatch *et al.* (1995) suggested that moderate penetration to the centreline is desirable, which was confirmed by Lefebvre (1999).

Of interest to the designer is the effect of orifice geometry on hole discharge coefficient C_D and jet penetration distance Y_{max} .

2.4.3.1 Orifice Configuration and Shape

Orifices have been successfully designed using different sizes and shapes, each configuration with their own advantages and disadvantages. Dodds & Bahr (1990) provided

Figure 2.9: Example orifice configurations. From Dodds & Bahr (1990).

a brief summary on the effect of hole configuration and shape on the discharge coefficient and penetration angle.

Of the different types of holes (Figure 2.9), the round punched hole is the simplest to manufacture, however, it performs poorly when the dynamic pressure in the annulus is large compared to that of the jet. The resulting shallow penetration angle does not promote mixing near the combustor centreline where the temperature profile peaks. However, round holes operate well when the annulus dynamic pressure is low. A thimble or a scoop may be used to help guide the flow and overcome the shortcomings of round punched holes.

Hole shape can also be important when the available space is limited by cooling slots or other liner design features. In this case, elliptical or rectangular holes with the major axis aligned along the circumferential direction can be used. Rectangular holes are seldom used as they introduce large stress concentrations near the hole corners. The use of these non-circular holes has proven to have negligible effect on turbulent mixing and jet trajectories (Liscinsky *et al.* , 1996).

2.4.3.2 Jet Trajectories

Modeling of the jets issuing from liner holes is essential to their design. This requires accurate prediction of the hole discharge coefficient, the jet penetration distance, and its trajectory.

A simple correlation for estimating the discharge coefficient for round sharp-edged holes has been provided by Kaddah (1964). Adkins & Gueroui (1986) presented a more accurate empirical model derived from theory. These models, however, do not

capture the strong effect of annulus flow details on jet characteristics. While complex numerical models are required to predict these effects, their use is limited by the resolution required in near-hole regions and the expense of time-consuming numerical modeling techniques (McGuirk & Spencer, 2000).

Modeling of the jet trajectory is essential to ensure that good mixing and a suitable exit pattern factor are provided. The pattern factor is defined as (Lefebvre, 1999)

$$\text{Pattern factor} = \frac{T_{max} - T_4}{T_4 - T_3} \quad (2.4.2)$$

where T_{max} = the maximum recorded combustor exhaust temperature

T_3 = the combustor inlet air temperature

T_4 = the mean combustor exhaust temperature

Holdeman *et al.* (1987) and Holdeman (1993) provided empirical correlations for the determination of the jet centreline trajectory, jet temperature profile, and the jet width for confined ducts with single-sided and opposed rows of jets. These models correspond well with experimental and numerical data. Much of the work on single and multiple jets in a confined crossflow has been summarized by Lefebvre (1999).

2.5 Premixer Design

Premixers play an important role in modern combustors. Premixers are devices composed of one or more swirlers designed to mix the fuel and air prior to combustion, as shown in Figure 2.10. The performance of these devices is quantified by the mixedness or the homogeneity of the discharged mixture. The detrimental effects of unmixedness on NO_x emissions have been investigated by Fric (1992).

Design must also ensure that the fuel/air mixture does not reside in the premixer for too long and autoignite. The mixture must also move fast enough to ensure that flashback does not occur (Poeschl *et al.*, 1994). Autoignition is the spontaneous ignition of a fuel/air mixture after a certain time lapse above the autoignition temperature. Flashback occurs when the flame propagates along boundary layers or slow moving flows to ignite the incoming fuel/air mixture.

Figure 2.10: Premixer concept.

Many premixer designs exist and various configurations have been investigated by Li & Gutmark (2004). Joshi *et al.* (1994) documented the design of a fuel air premixer for General Electric's aero-derivative dry low NO_x combustors whereas Lin *et al.* (2004) experimented with a multi-holed concept.

2.6 Combustion Chemistry

Combustion is an exothermic reaction in which fuel and oxidizer, air in most cases, react to release heat. Stoichiometric combustion occurs when the exact quantity of oxidizer required to completely burn the fuel is supplied to the combustion reaction. The mixture is said to be lean if more than the stoichiometric quantity of oxidizer is supplied and rich if the amount of oxidizer is less than stoichiometric. The ratio of this stoichiometric quantity of air to fuel mass is determined from an atom balance. The stoichiometric relation for the combustion of a hydrocarbon fuel given by $C_\alpha H_\beta$ in air is

where $a = \alpha + \frac{\beta}{4}$. Thus, the stoichiometric air-fuel ratio

$$\left(\frac{A}{F}\right)_{stoic} = \left(\frac{\dot{m}_a}{\dot{m}_f}\right)_{stoic} = \frac{4.76a}{1} \frac{MW_a}{MW_f} \quad (2.6.2)$$

where MW_a and MW_f are the molecular weights of fuel and air respectively.

The equivalence ratio ϕ is used to quantify whether the mixture is rich, lean, or stoichiometric. It is defined as

$$\phi = \frac{\left(\frac{A}{F}\right)_{stoic}}{\left(\frac{A}{F}\right)_{act}} \quad (2.6.3)$$

The equivalence ratio is unity for a stoichiometric mixture. Values greater than unity indicate that the mixture is fuel rich, whereas values less than unity indicate a lean mixture.

2.6.1 Chemical Equilibrium

Equation 2.6.1 is an idealization as the products of combustion differ in high temperature combustion. CO_2 , H_2O , O_2 , N_2 , and other major species dissociate into minor species such as H_2 , OH , CO , H , O , N , and NO . Calculation of the mole fractions of these species may be performed using several approaches.

One approach, used by Olikara & Borman (1975) and discussed in detail by Turns (2000), is called the equilibrium-constant approach. This approach is based on the second law of thermodynamics, which states that the system will spontaneously shift towards the point of maximum entropy. Once this point is reached, no more compositional changes occur and the system is said to be in equilibrium. Thus, the equilibrium temperature, pressure, and chemical composition may be determined for a fixed internal energy, volume, and mass by applying the first and second laws of thermodynamics with the equation of state.

2.6.2 Chemical Kinetics

Chemical kinetics involves the study of chemical reactions and their rates. In many combustion processes, the rate at which the chemical reaction proceeds towards equilibrium controls the overall rate of combustion. In this case, other factors including

mixing and evaporation occur fast enough that they can be considered insignificant. Chemical reaction rates also determine the speed of pollutants formation and destruction, moreover, they are related to ignition and flame extinction. Lastly, the minimum volume required for combustion is dictated by the chemical reaction rates (Longwell & Weiss, 1955).

The global reaction of Equation 2.6.1 can be written as

Equation 2.6.4 describes the reaction of 1 mole of fuel F with a moles of oxidizer Ox to form b moles of product Pr . Experimental results show that the rate of fuel consumption can be expressed by (Turns, 2000)

$$\frac{d[X_F]}{dt} = -k_G(T)[X_F]^n[X_{Ox}]^m \quad (2.6.5)$$

where $[X_i]$ denotes the molar concentration of the i th species, k_G is the global rate coefficient, and the exponents n and m describe the reaction order. The constants k_G , n , and m for global reactions are derived from curvefitting of experimental data and usually only hold their validity over a limited range of temperatures.

The global reaction of Equation 2.6.1 does not fully describe the complex combustion process. The overall reaction of fuel and air consists of many sequential steps involving intermediate species that are formed and destroyed. These individual reactions that occur are called elementary reactions. A mechanism is the collection of elementary reactions needed to describe the global reaction. Reaction mechanisms may involve a few steps (as in H_2 - O_2 combustion) or several hundred (as in hydrocarbon-air combustion).

Reaction Mechanisms The oxidation of higher order paraffins has been studied by many researchers. Subsequently, many empirical models have been derived consisting of either a single global step or several quasi-global steps. Examples include the important global reaction scheme for hydrocarbons of Westbrook & Dryer (1981) and multi-step reaction schemes by Hautman *et al.* (1981) and Jones & Lindstedt (1988).

Figure 2.11: Constant pressure reactors.

A detailed 30-step reaction mechanism for "jet fuel" was devised by Kollrack (1976). Turns (2000) reviewed other important reaction mechanisms including the extended Zeldovich mechanism for thermal NO_x formation.

2.6.3 Combustion Modeling and Emissions Prediction

Many researchers have made improvements to the accuracy of emissions predictions and combustor simulation over early empirical models by applying reactor theory. Examples of early empirical models include those of Mellor (1976) and Lefebvre (1984).

Reactor theory couples chemical and thermal analysis to describe the detailed evolution of a system using simplified thermodynamic models called reactors. Two types of reactors are well-stirred reactors (WSR) and plug-flow reactors (PFR), illustrated in Figure 2.11. In a WSR, fuel and air are supplied at a steady rate and instantly mix and form a homogeneous mixture. The mixture burns at a rate prescribed by chemical kinetics and the products are expelled at the same rate. The pressure, temperature, and species concentrations remain constant throughout the reactor as a result of the perfectly mixed assumption. PFRs model one-dimensional flow that reacts as it moves downstream. Here, mixing is ignored and the flow is assumed perfectly mixed in the radial direction perpendicular to the flow.

The simple model of Rizk & Mongia (1995b), as shown in Figure 2.12, illustrates the use of reactor theory to simulate gas turbine combustion. The model uses a

Figure 2.12: Combustor reactor model. Modified from Rizk & Mongia (1995b)

network of two reactors and is applicable to both conventional and modern combustors. The WSR models combustion in the primary zone whereas a downstream PFR models the reaction of any escaped fuel and dilution of combustion products. Others have used more complex networks to improve the accuracy of these models by adding reactors for the pilot, cooling slots, and other features of interest (Rizk & Mongia, 1993b,a, 1995a; Tonouchi *et al.*, 1998; Andreini & Facchini, 2002).

While capable of describing the overall combustion trends, the reactor models discussed above cannot fully describe the complex processes such as swirl, recirculation, fuel injection, atomization, fuel evaporation, mixing, convective and radiative heat transfer. Empirical or semi-empirical models can be combined with analytical CFD models to provide a qualitative understanding of these processes with reasonably accurate quantitative results. The three-dimensional model of a gas turbine combustor developed by Rizk & Mongia (1991) is one example. The model uses three-dimensional finite-volume CFD to solve the flow-field and is followed by the application of empirical correlations by Lefebvre (1984, 1985) to yield the quantities of interest. Rodriguez & O'Brien (1999) describe an unsteady, finite-rate model that combines one-dimensional finite-volume methods with the global reaction mechanism of Westbrook & Dryer (1981). Figure 2.13 illustrates a CFD solution of a combustor using finite-rate chemistry.

Figure 2.13: Combustor temperature and NO distribution using CFD and finite-rate chemistry.

2.7 Combustion Performance

Gas turbine combustors must operate with stability over a wide range of operating conditions (temperature, pressure, velocity, etc.) while maintaining good combustion efficiency. They must also be capable of igniting with ease and re-initiating combustion during operation in the event of a flameout.

2.7.1 Combustion Efficiency

Combustors operating at low combustion efficiencies are regarded as products of poor design (Lefebvre, 1999). These low efficiencies may be directly correlated to wasted fuel and high pollutant emissions. Typically governmental regulations require that combustors operate at efficiencies of 99 percent or greater.

Combustion efficiency is dependant on the rate at which heat is released by the chemical reaction. It is the ratio of the theoretical mass flow rate of fuel divided by the actual value necessary to provide the same heat release. The combustion efficiency is defined as,

$$\eta_c = \frac{\dot{m}_{f_{theo}}}{\dot{m}_{f_{act}}} \quad (2.7.1)$$

where $\dot{m}_f$ is the mass flow rate of fuel.

The efficiency is governed by the time it takes to complete evaporation, mixing, and the chemical reaction. Only one of these processes typically governs the system at a given time. Childs (1950) has provided empirical relations for the estimation of combustion efficiency based on engine data correlations. These relations assume that combustion is governed by chemical reaction whereas the empirical correlations of Lefebvre (1985) assume that both chemical reaction and evaporation processes control the system.

2.7.2 Combustion Stability

Combustion must be sustained in highly turbulent, high velocity air streams with extreme conditions including the ingestion of ice and rain. Stable operation without blowing out, easy ignition when blowout occurs, and the absence of combustion

oscillations is essential during all operating conditions.

Static Stability Until now, the issue of static stability has been referred to as stability. Static instability is alleviated by creating sheltered zones of low velocity behind flameholders such as the dome or some other bluff body where the flame can exist in the wake. Some important work on these flameholders and flame stabilization is that of Longwell *et al.* (1953), Longwell (1953), Spalding (1953), Rao & Lefebvre (1973), Herbert (1960), Ballal & Lefebvre (1981), Rizk & Lefebvre (1986), and Baxter & Lefebvre (1992).

Stability is measured by running tests at different operating conditions until flame extinction is observed. The resulting range of stable operation is bounded by converging rich and weak limits, as illustrated in Figure 2.14. The parameter along the horizontal axis, the combustor loading I , reflects the influence of the most important parameters on combustion stability: air flow $\dot{m}_a$, combustor volume V , and pressure P . It is defined as

$$I = \frac{\dot{m}_a}{VP^n} \quad (2.7.2)$$

where the exponent n , determined experimentally, is typically chosen to be 2 (Mattingly *et al.*, 2002).

Of particular interest for modern combustor design is the point of weak extinction, as they tend to operate near the lean limits of flammability. Any slight variation in operating conditions could cause the air/fuel ratio to exceed the lean limit and result in sudden extinction of the flame. This problem is typically solved by the addition of a pilot operating at near-stoichiometric fuel/air ratios or using variable geometry to keep the PZ away from the lean limit.

Combustion Noise / Dynamic Instability Operating near the lean limit creates additional challenges to those stated above. Designers of modern combustors must be concerned with the onset of dynamic instability, the occurrence of combustion oscillations (Richards & Janus, 1997). These oscillations are a result of the coupling between acoustics and the combustion process (Bragg, 1963).

Figure 2.14: Typical stability loop.

Noise is emitted when a volume of gas expands at constant pressure due to heating from combustion. The resulting expansion produces a sound wave that propagates through the combustor. These waves are returned back to the combustion zone by the liner walls some time after their creation. The time for the wave to travel through the combustor and back to its origin is called the delay time.

Pressure waves are formed when localized areas of the flame are extinguished and reignited due to changes in heat release, which are caused by fluctuations in the air supply, aerodynamic disturbances, and variations in the fuel supply. If fluctuations in the heat release process are periodic, the pressure waves emanating from the ignition of these small volumes of gas will have the same frequency. Energy is added to the system if the heat release and pressure fluctuations are in phase, resulting in unstable pressure oscillations that grow in amplitude.

This phenomenon is more severe for modern premixed combustors because the heat release occurs more abruptly across the flame front and variations in the reaction rate are more severe. These oscillations are illustrated in Figure 2.15. Their control was reviewed by Lefebvre (1999).

Figure 2.15: Oscillations in DLE combustors. From Gauthier (2003).

2.7.3 Ignition

Design of a gas turbine combustor requires the selection of an ignition source capable of supplying a sufficient amount of energy to initiate combustion during ground startups.

The influx of energy required to ignite the fuel can be delivered by means of torch igniters, glow plugs, hot surfaces, plasma jets, lasers, chemical ignition, gas addition, and oxygen injection. Spark ignition is thus far the most widely used method for ignition (Lefebvre, 1999). Figure 2.16 illustrates a typical spark igniter.

Ignition Energy To substantiate ignition, sufficient energy to create a self-propagating volume or kernel of hot gas is required. The size of the smallest volume which satisfies this criterion is termed the quenching distance. The minimum ignition energy is the amount of energy required to produce a kernel of this critical size. This concept is depicted in Figure 2.17.

The two parameters have been experimentally studied by Ballal & Lefebvre (1977), Rao & Lefebvre (1973), and Ballal & Lefebvre (1979), who provided models for their prediction. Lefebvre (1999) summarized much of the relevant work on ignition performance.

Figure 2.16: Surface discharge igniter.

Figure 2.17: Ignition kernel.

Autoignition Ignition of fuels can also occur spontaneously after sufficient time in contact with heated air. Spontaneous or autoignition is more of a concern in modern combustors where the fuel and air are mixed prior to combustion. Before entering the PZ, the mixture can react, causing damage to the premixer or other components as well as producing high levels of NO_x . The time between the creation of a combustible mixture and its ignition is called the ignition delay. Correlations for ignition delay times of aircraft fuels are provided by Spadaccini & Tevelde (1982) and Lefebvre (1999).

2.8 Droplet Processes

Special consideration must be taken for liquid fuel combustion. The fuel must be atomized into a fine spray, evaporated, and then burned.

2.8.1 Atomization

Injectors are used to atomize the liquid into a cloud of fine droplets thereby increasing the rate at which the fuel evaporates. Atomization occurs when the fuel is passed through a suitable nozzle to form a jet or thin film that becomes unstable and breaks up into many small droplets (Figure 2.18). This instability is caused by frictional forces or shear developed between the fluid and neighbouring gas stream.

The two common methods of atomizing liquids are pressure and twin-fluid injectors. Pressure nozzles force the flow through a small orifice to emit a high velocity jet in which the shear produced is almost entirely a result of liquid motion. Alternatively, a high velocity jet of steam or air is used to shear and break up the liquid in twin-fluid atomizers.

Designers must select the appropriate atomizer to provide the necessary characteristics for their application. One important parameter in these considerations is the Weber number. The Weber number quantifies the processes that tend to produce

Figure 2.18: Schematic of sheet breakup.

instability in a liquid sheet, and is defined as

$$We = \frac{\rho V_r D_0}{\sigma} \quad (2.8.1)$$

where ρ = the density of the gas stream
 V_r = the gas velocity relative to the liquid
 σ = the surface tension of the liquid
 D_0 = a length dimension characterizing the liquid jet

Thus, increasing the Weber number results in more rapid instability and produces smaller droplets.

The size distribution of these droplets within the spray and the geometry of the spray strongly influences the performance of liquid fueled combustors (Hallett, 2003). Quantification of droplet size distributions is difficult due to the randomness of the atomization process. Atomizers produce drop sizes ranging from a few microns to several hundred microns (Lefebvre, 1999). The wide range in drop size within the spray is characterized by another important parameter known as the distribution. The most widely used expression for droplet size distributions is that developed by

Rosin & Rammler (1933). It is expressed as

$$1 - Q = \exp[-(D/X)^q] \quad (2.8.2)$$

where Q = the fraction of the total volume contained in drops of diameter less than D

X, q = experimentally determined constants

The diameter of the drops within the distribution can be averaged using the Sauter Mean Diameter (SMD). It is the diameter of a drop that has the same volume-to-surface area ratio as the entire spray. The SMD is important because the droplet evaporation/burning rate is proportional to the droplet surface area, and the mass evaporated/burned is proportional to the volume.

Mean drop sizes are dependant on the atomizer size, design features, and operating conditions. The physical properties of the fuel such as density, surface tension, and viscosity are also factors that determine the quality of atomization (Lefebvre, 1999). Because the physical processes involved with atomization are not completely understood, many authors have performed empirical studies to develop correlations for the prediction of the SMD. Lefebvre (1989) and Winterfeld *et al.* (1990) have provided equations for the prediction of the SMD in fuel sprays for various nozzle types.

Pressure-Swirl Nozzles Pressure-swirl, or simplex nozzles, impart angular momentum to the fuel with a swirl chamber and then force the liquid fuel through a small orifice. Centrifugal forces cause the fuel jet to spread out in a conical sheet with high angular velocity. The conical sheet breaks up into small drops to form a wide angled hollow cone spray (Figure 2.19).

Although simple and inexpensive, these atomizers only exhibit good atomization over a narrow turn-down ratio when compared to twin-fluid atomizers. The turn-down ratio is the ratio of the maximum to minimum fuel flow rate. Good atomizers provide sufficiently fine droplets that ignite and burn easily over wide turn-down ratios.

Figure 2.19: Hollow cone spray. From Delevan Spray Technologies (2005).

2.8.2 Droplet Evaporation

The evaporation of drops in a spray relies on heat and mass transfer processes. Heat for evaporation is transferred to the droplet surface from the surrounding gas by conduction and convection while vapour simultaneously diffuses and convects into the surrounding gas (Figure 2.20).

These heat and mass transfer processes are closely coupled. The vaporization rate is determined by the rate of heat flux to the droplet surface, which is dependent on the rate of vaporization. Any increase in the rate of heat transfer above the rate required for vaporization is regulated by the increase in droplet temperature. The increased rate of vaporization resulting from a higher liquid temperature effectively cools the droplet by transporting heat away from the droplet with the fuel vapour.

Initially, when the droplet is injected into a hot environment, the heat transferred to the droplet serves only to increase its temperature. Vaporization only occurs later on as the temperature increases asymptotically towards its steady state value, as shown in Figure 2.21. At steady state, this heat transferred to the droplet is used solely for vaporization.

As the droplets evaporate, they decrease in size due to the loss of mass. The relationship between the square of drop diameter and the time required for evaporation is illustrated by Figure 2.22. The slope of the curve depicted, called the evaporation constant, increases from zero to its steady state value as the droplet heats up. It is

Figure 2.20: Droplet vaporization phenomena.

Figure 2.21: Droplet heating.

Figure 2.22: Evaporation rate curves.

defined as

$$\lambda = \frac{d(D)^2}{dt} \quad (2.8.3)$$

It is common to assume that the evaporation rate remains constant over the entire life of the droplet, as the droplet heat-up period is usually small in comparison. Chin & Lefebvre (1982) provided a means of estimating an effective value for the evaporation constant to account for transient effects.

The rate of evaporation of a single droplet is then described by

$$\dot{m}_f = \frac{\pi}{4} \rho_f \lambda D \quad (2.8.4)$$

where ρ_f is the density of fuel.

2.8.3 Droplet Combustion

The radially outward fluxing fuel vapour envelopes the droplets and mixes them with the surrounding air, forming a combustible mixture. A resulting diffusion flame surrounds the individual droplets where the fuel and air quantities are near-stoichiometric. This process is depicted by Figure 2.23.

Figure 2.23: Droplet combustion.

Modeling droplet evaporation and combustion is reviewed by Hallett (2003). A more in-depth explanation and discussion is available from Kuo (1986), Lefebvre (1989), and Williams (1990). Some important work includes the detailed droplet combustion models of Law (1976), Law & Sirignano (1977), and Law & Law (1982).

2.8.4 Spray Flames

In practical combustion devices, droplets usually do not burn individually. They are atomized into a fine spray where they evaporate quickly to produce a cloud of vapour with cooler droplets at the core. This vapour mixes with the air and burns much like a gaseous flame.

Modeling of evaporation and sprays is much more complex than for single droplets. Faeth (1983) reviewed some of the models used and their comparisons with measurements.

2.9 Heat Transfer

The liner serves several key purposes. It must contain the burning fuel/air mixture while metering the admission of cooling air into the combustion zones. Structural loads are also carried by the liner and it protects other components from the harsh environment in which they are submerged.

For these reasons, liners are typically fabricated from high-strength, oxidation resistant nickel and cobalt-based alloys. Examples include Hastelloy X and HS 188. They are capable of operating at temperatures up to 1150 K for extended periods of time (Dodds & Bahr, 1990). Above these temperatures their life is compromised severely. Oxidation occurs rapidly at temperatures above 1400 K and incipient melting commences between 1550 and 1750 K (Dodds & Bahr, 1990).

For engines operating with high pressure ratios and high turbine inlet temperatures, it is common for peak gas temperatures within the liner to exceed 2500 K (Dodds & Bahr, 1990). This necessitates the use of available air to cool the liner to temperatures within material limits.

Despite operating close to material limitations, engine requirements are becoming more and more demanding. Manufacturers have increased the time between overhauls from several hundred hours to several thousand hours (Dodds & Bahr, 1990). These increased life requirements are complicated by the desire to improve engine performance and reduce emissions using raised pressure ratios, turbine inlet temperatures, and bypass ratios. The resulting combination of higher annulus air temperatures and reduced amounts of air for cooling has necessitated the use of specialized techniques for the removal of heat from the liner.

2.9.1 Liner Cooling

An effective cooling scheme is one which provides the maximum heat sink capability while minimizing the amount of coolant flow. For initial design, the engineer requires an accurate prediction of the liner wall temperature to judiciously select the appropriate cooling scheme. This selection is typically based on the expected operating

Figure 2.24: Schematic of heat transfer processes in gas turbine combustion.

life of the liner. Small inaccuracies in the liner wall temperature can result in large discrepancies between predicted and actual operating life. Sturgess (1978) found that predictions that are inaccurate by more than 25 °F (14 °C) are unacceptable from a life estimation point of view.

Nonetheless, Holman (1997) provides a comprehensive understanding of the heat transfer processes involved and many useful correlations for the estimation of the wall temperature. Early work on the heat transfer processes applied to gas turbine combustion by Lefebvre & Herbert (1960) has led to many publications providing methods of estimating the liner wall temperature. Ballal & Lefebvre (1972) and Kretschmer & Odgers (1978) are two of these publications.

The individual processes by which heat is transferred to the liner are illustrated by Figure 2.24. The modes of heat transfer involved include conduction, convection, and radiation. An equilibrium balance gives

$$K_{1-2} = R_1 + C_1 = R_2 + C_2 \quad (2.9.1)$$

where K_{1-2} = the heat transferred by radial conduction through the liner wall

in the radial direction

R_1	=	the heat transferred internally by radiation
R_2	=	the heat transferred externally by radiation
C_1	=	the heat transferred internally by convection
C_2	=	the heat transferred externally by convection

The heat conducted along the liner wall, K , is considered negligible as it is small in comparison to the heat flux by radiation and conduction (Lefebvre, 1999).

Solution of Equation 2.9.1 yields the average liner surface wall temperature, not the peak values. The accuracy of this quasi one-dimensional model is limited by the gross assumptions made to specify the mainstream conditions. The resulting predictions are inaccurate in the reaction and primary zones due to difficulties in estimating the effective gas temperature and the radiant heat flux to the walls (Kretschmer & Odgers, 1978). Furthermore, precise calculation of the wall temperature in the dilution zone is prevented by the large temperature difference between the hot gas flow and the cool jets of air (Ballal & Lefebvre, 1972).

Rizk (1994) developed a more modern calculation procedure that simulates the combustor flow-field with CFD in combination with a radiation model to predict the liner wall temperature. One is able to model specific geometric disturbances that may cause local hot spots. These hot spots cause liner durability issues and result in the overall failure by buckling, creep damage, local burning, or stress rupture (Sturgess, 1978).

2.9.2 Cooling Schemes

Various cooling schemes are used to eliminate hot spots and maintain the liner wall temperature below material limits. Attention must be made not to excessively cool the liner, as it may extinguish the reaction near the walls and lead to production of CO and UHC in large quantities. The severity of the problem is increased in premixed combustion as slight temperature variances incite large changes in reaction rate.

Conventional cooling slots provide protection by means of a cool insulating barrier

Figure 2.25: ABC cooling examples.

of cool air along the liner inside surface. These slots are considered to be inefficient and highly wasteful of air that could otherwise be used for combustion (Kretschmer & Odgers, 1978). More advanced cooling schemes have been developed over the years to achieve the equal or greater degrees of cooling using less air. Nealy (1978) reviewed some of the approaches to liner cooling.

Angled effusion cooling (AEC), transpiration cooling, quasi-transpiration cooling, and augmented backside convection (ABC) cooling are all advanced methods that maximize the heat transfer potential of the cool annulus air. Additional protection from the hot gases may be obtained from segmented wall construction and thermal barrier coatings (TBC). The use of ABC cooling in combination with a TBC has proven effective for modern combustors (Smith & Fahme, 1999; Arellano *et al.*, 2001).

2.9.2.1 Augmented Backside Convection

ABC cooling involves increasing the rate of convective heat transfer from the backside surface of the liner to increase its heat sink potential. The advantage of this type of cooling over others is the reduction in required overall cooling flow when compared with conventional cooling slots. Several methods of augmentation have been successfully used: forced convection (Figure 2.25a), cooling fins or pins (Figure 2.25b), impingement cooling (Figure 2.25c), and trip strips.

Forced convection methods use a double wall construction where annulus air is forced through small channels to increase its velocity, improving the convective heat

transfer coefficient. Likewise, impingement cooling requires a double wall construction to meter annulus air through small holes drilled normal to the liner. Air jets issuing from these holes impinge on the backside of the liner and remove heat. The advantage of this technique over forced convection is that holes can be drilled to locally adjust the liner wall temperature. Furthermore, the jets form a thin film after impinging on the liner surface that may be admitted to the inside of the liner to provide an additional protective barrier.

High manufacturing costs and the requirement for a double wall construction are among the major drawbacks of forced convection and impingement cooling schemes. Double-walled liners have higher material costs, are heavier, and are susceptible to buckling due to differences in thermal expansion between the two walls.

Increasing the surface area on the outside of the liner using pins, fins, pedestals, or ribs are other methods of ABC cooling. They are usually configured axially along the liner, but circumferential ribs or fins may be used. These techniques have been used on heat shields for combustor domes for some time. A correlation for modeling these types of devices is provided by Rizk (1994).

Trip strips augment cooling effectiveness by resetting the boundary layer in the annulus with thin circumferential fins. Boundary layers are thin films of slow moving fluid along the wall that grow asymptotically as they progress downstream. The rate of convective heat transfer from the surface is much higher when these films are thin compared to their fully developed thickness. The fins disrupt the flow and force boundary layers to redevelop in order to take advantage of the favourable heat transfer properties of thin boundary layers, as shown in Figure 2.26. Early work by Norris (1970) on turbulent flow in ducts over rough surfaces led to the procedures on the design of these strips for gas turbine combustors.

2.9.2.2 Thermal Barrier Coatings

Thermal barrier coatings (TBC) are widely used on combustors as their application is relatively easy. They are thin coatings of ceramics with high thermal resistance and low emissivity applied to the inside surface of the liner by plasma spraying or

Figure 2.26: Illustration of boundary layer resetting using trip strips.

Figure 2.27: TBC microstructure. Modified from Padture *et al.* (2002)

electron beam deposition (Wang *et al.* , 2004). Figure 2.27 illustrates the structure of a TBC. It consists of a metallic bond coat to increase the base metal adhesive properties, followed by a thermally grown oxide layer to provide bonding, and finally a ceramic top layer of yttria partially stabilized zirconia (YPSZ). One can expect reductions in the liner wall temperature of approximately 50 K, depending on the thickness and materials used, using TBCs Lefebvre (1999). The development of more advanced thicker TBCs capable of larger reductions in base metal temperature are discussed by Price (2004).

2.10 Concluding Remarks

No set of algorithms exists in currently available literature for the design of modern LPP combustors. However, formulation of one is possible using the many correlations and models discussed in this chapter. Such a model should be simple enough for implementation using computer programs and also provide varying degrees of complexity, depending on the application. Based on these considerations, a new design methodology for LPP combustors is devised and presented in the proceeding chapter.

Chapter 3

Preliminary Design Strategy

This chapter documents the preliminary considerations that take place during the early design stages for a LPP gas turbine combustor. The overall combustor design involves three steps: preliminary design, detailed design, and testing (Figure 3.1). The overall dimensions of the concept are determined during preliminary design, after which its performance is verified and validated by detailed design and testing, respectively. Experimental testing is not included in the scope of this thesis.

3.1 Preliminary Design

In order to formulate a preliminary design, a compromise must be made between the convenience of simple algorithms and the accuracy of complex numerical models that require vastly larger amounts of computing resources. Simple algorithms can be quickly and easily implemented into computer programs whereas numerical modeling of gas turbine combustion requires sufficient resolution in the model to accurately capture the complexity of the processes involved. Theory is used whenever possible to yield a versatile algorithm that may be applied to many different situations.

The following sections will discuss the individual steps involved in the preliminary design. These steps include:

- concept creation,
- combustor sizing,
- diffuser design,

Figure 3.1: Overall design flowchart.

- premixer design,
- dome region design,
- combustor length selection,
- dilution hole design,
- igniter specification,
- liner heat transfer analysis, and
- preliminary structural analysis.

3.2 The Concept

Design begins with a concept. Figure 3.2 illustrates the concept chosen. Air discharged from the compressor enters the combustor at Station 3.0. The air passes through a faired prediffuser before it is dumped into the annulus. In this region, just downstream of the sudden expansion, the cross-sectional flow area in the annulus is large to ensure that any asymmetry in the flow is avoided. As the air accelerates through the annulus, a portion is admitted by holes in the liner to dilute the hot combustion gases. The other portion of the air flows through the annulus where it cools the outside of the liner wall. This cooling effect is enhanced by the use of trip strips. Annulus air used for cooling is then dumped into a plenum and enters the premixer. Inside the premixer, the air passes through two concentric, counter-rotating axial swirlers to mix with an evaporating liquid fuel spray. The exiting fuel and air mixture is dumped into the combustor PZ by another axial swirler where it ignites and burns. The resulting hot products are diluted with relatively cooler air and accelerated out of the combustor by a converging nozzle.

3.3 Combustor Sizing

Combustor sizing refers to the definition of the reference area (or diameter) to provide sufficient stability without incurring excessive pressure losses. All operating points (i.e., idle, full power) are considered and the smallest size that provides stability over

Figure 3.2: LPP concept.

Figure 3.3: Sizing parameters.

the entire range of operation is chosen.

Specification of the size for a premixed combustor is difficult as design data similar to that provided in Table 2.1 is not currently available. Thus, a premixed combustor must be scaled from an equivalent conventional combustor. The scaling parameter chosen for this algorithm is the lean blowout velocity, which is related to flame stability.

The parameters involved in sizing the premixed combustor are illustrated in Figure 3.3.

3.3.1 Operating Parameters

For simplicity, the static conditions are assumed equal to those at stagnation for both temperature and pressure. This is a reasonable assumption since the Mach number everywhere is below 0.25. For a Mach number of 0.25, the static pressure and temperature deviate from stagnation by only 4 percent and 1 percent, respectively (White, 1999).

Thus, for each point the following conditions are known:

- the air flow rate, $\dot{m}_3$
- the inlet temperature, T_3
- the outlet temperature, T_4
- the inlet pressure, P_3

And from the ideal gas law, we know the inlet density,

$$\rho_3 = \frac{P_3}{R_a T_3} \quad (3.3.1)$$

where R_a is the ideal gas constant for air.

3.3.2 Size Conventional Combustor

The casing size for most industrial combustors is dictated by the overall pressure loss ΔP_{3-4} . The cross-sectional area of the casing, A_{ref} , is determined from (Lefebvre, 1999)

$$A_{ref} = \left[\frac{R_a}{2} \left(\frac{\dot{m}_3 T_3^{0.5}}{P_3} \right)^2 \left(\frac{\Delta P_{3-4}}{q_{ref}} \right) \left(\frac{\Delta P_{3-4}}{P_3} \right)^{-1} \right]^{0.5} \quad (3.3.2)$$

Suitable values for $\Delta P_{3-4}/P_3$ and $\Delta P_{3-4}/q_{ref}$ are chosen by the designer based on Table 2.1.

The casing diameter for a can may then be derived from geometry

$$D_{ref} = \sqrt{\left(\frac{4}{\pi} \right) A_{ref} + 2t_{liner}} \quad (3.3.3)$$

The liner diameter must be chosen carefully. It would appear desirable to select the largest diameter possible and reduce the velocity of the flow within the liner. This increases the residence time in the combustor and promotes stability. However, for any given casing area, increasing the liner diameter results in a reduction of the annulus flow area A_{an} . The smaller area results in higher annulus velocities that decreases the static pressure in the annulus. Therefore, a larger liner diameter is undesirable since a high static pressure drop across the liner admission holes is necessary to provide adequate penetration of the jets.

Sawyer (1985) provided a general rule of thumb for selecting the liner diameter of conventional combustors. The author stated that the liner cross-sectional area should

be kept within 60-72 percent of the casing area for conventional combustors. A value of 70 percent is chosen and hence, the liner cross-sectional area is

$$A_{liner} = 0.7A_{ref} \quad (3.3.4)$$

The diameter of the liner is

$$D_{liner} = \sqrt{\frac{4}{\pi} A_{liner}} \quad (3.3.5)$$

and the annulus cross-sectional flow area is

$$A_{an} = \frac{\pi}{4} [D_{ref}^2 - (D_{liner} + 2t_{liner})^2] \quad (3.3.6)$$

3.3.3 Equivalence Ratio

The overall equivalence ratio $\phi_{overall}$ necessary to achieve the specified temperature rise ($\Delta T = T_4 - T_3$) is computed by assuming equilibrium combustion and using the equilibrium constant approach. While the combustion process will not generally achieve equilibrium in practice, the assumption is valid since the heat release rate and the formation of CO occurs fast when compared to the formation of NO_x .

3.3.3.1 Equilibrium Constant Approach

The equilibrium composition of a mixture at a given temperature T and pressure P is calculated using the principle of Gibbs free energy. For a reacting system, the Gibbs free energy attains a minimum when equilibrium is reached.

For the general system, where

chemical equilibrium occurs when

$$\Delta G_T^0 = -\bar{R}T \ln K_p \quad (3.3.8)$$

where $\bar{R}$ is the universal gas constant ($=8.3144 \text{ J mol}^{-1} \text{ K}^{-1}$). The term on the left-hand-side of Equation 3.3.8 is called the standard-state Gibbs function change. It is determined by

$$\Delta G_T^0 = (e\bar{g}_{f,E}^0 + f\bar{g}_{f,F}^0 + \dots - a\bar{g}_{f,A}^0 - b\bar{g}_{f,B}^0 - \dots) \quad (3.3.9)$$

where $\bar{g}_{f,i}^0$ is the Gibbs function of formation for the i th species.

The equilibrium constant, K_p , is defined as

$$K_p = \frac{(P_E/P^0)^e \cdot (P_F/P^0)^f \cdot etc}{(P_A/P^0)^a \cdot (P_B/P^0)^b \cdot etc} \quad (3.3.10)$$

where P_i is the partial pressure of the i th species and P^0 is the standard state pressure (1 atm).

Substituting Equation 3.3.10 into Equation 3.3.8 and noting that $P_i = \chi_i P$ yields

$$\frac{\chi_E^e \cdot \chi_F^f \cdot etc}{\chi_A^a \cdot \chi_B^b \cdot etc} \cdot (P/P^0)^{(e+f+\dots-a-b-\dots)} = \exp \left[\frac{-\Delta G_T^0}{RT} \right] \quad (3.3.11)$$

where χ_i is the molar fraction of the i th species.

Additional equations are created using the conservation of elements (Equation 3.3.7) and the requirement that the mole fractions sum to unity,

$$\sum_i \chi_i = 1 \quad (3.3.12)$$

Simultaneous solution of Equations 3.3.7, 3.3.11, and 3.3.12 yields the product composition. The solution becomes difficult when applied to the complex systems found in practical combustion involving many species and multiple concurrent equilibrium reactions. An example application of the equilibrium constant approach to a multi-reaction system is that of Olikara & Borman (1975).

3.3.3.2 HPFLAME

The FORTRAN program developed by Olikara & Borman (1975), HPFLAME, applies the equilibrium constant approach to the C, H, N, O system. The code solves for 12 species (H, O, N, H₂, OH, CO, NO, O₂, H₂O, CO₂, N₂, and fuel) in a system with 7 equilibrium reactions and four atom-conservation equations. It computes the adiabatic flame temperature and product concentrations χ_i for constant-pressure P combustion of a fuel, represented by $C_\alpha H_\beta O_\gamma N_\delta$, with reactant enthalpy H_{reac} and equivalence ratio ϕ . Thus, iteration over the adiabatic flame temperature is necessary to determine the equivalence ratio that yields a value equal to T_4 .

An atom balance for a given equivalence ratio gives

$$C_\alpha H_\beta O_\gamma N_\delta + \frac{x_s}{\phi} (O_2 + 3.76 N_2) \rightarrow PRODUCTS \quad (3.3.13)$$

where the stoichiometric coefficient is

$$x_s = \alpha + \frac{\beta}{4} - \frac{\gamma}{2}$$

The enthalpy of reactants is computed using

$$\begin{aligned} H_{reac} &= \sum_i N_i \bar{h}_i \\ &= 1 \cdot \bar{h}_{C_\alpha H_\beta O_\gamma N_\delta} + \frac{x_s}{\phi} \cdot \bar{h}_{O_2} + 3.76 \frac{x_s}{\phi} \cdot \bar{h}_{N_2} \end{aligned} \quad (3.3.14)$$

where $\bar{h}_i$ is the absolute enthalpy on a molar basis and N_i is the number of moles of the i th species. Curve fits for the enthalpy of various gases and fuels are provided in Table A.1 and Table A.2, respectively.

3.3.4 Fuel Flow Rate

Resolution of the overall equivalence ratio is followed by the computation of the corresponding fuel mass flow rate. This is derived from the stoichiometric air-fuel ratio.

For a fuel $C_\alpha H_\beta$, the stoichiometric air-fuel ratio is defined as in Equation 2.6.2:

$$\left(\frac{A}{F} \right)_{stoic} = \frac{4.76a}{1} \frac{MW_a}{MW_f}$$

where $a = \alpha + \frac{\beta}{4}$.

The actual air-fuel ratio,

$$\left(\frac{A}{F} \right)_{act} = \frac{\left(\frac{A}{F} \right)_{stoic}}{\phi_{overall}} \quad (3.3.15)$$

Thus, the fuel flow rate:

$$\dot{m}_f = \frac{\dot{m}_3}{\left(\frac{A}{F} \right)_{act}} \quad (3.3.16)$$

Table A.2 lists the molecular weights of various fuels.

3.3.5 Airflow Distribution of Premixed Combustor

The airflow distribution is selected to provide the desired equivalence ratios in the primary and dilution zones. For the primary zone, the equivalence ratio is determined by the flame temperature.

Premixed combustors operating below 1900 K were determined to exhibit little NO_x formation if the fuel and air was uniformly mixed (Leonard & Stegmaier, 1993). A slightly lower flame temperature of 1800 K is selected to minimize the peak value in any local hot spots caused by unmixedness. Operating with a flame temperature any closer to the weak extinction limit (approximately 1600 K) leads to inherent issues with combustor stability and excessive CO (Gauthier, 2003). Thus,

$$T_{flame} = 1800 \text{ K} \quad (3.3.17)$$

The primary zone equivalence ratio is computed using the procedure of Section 3.3.3.

$$\phi_{PZ} = \phi(T_{flame}) \quad (3.3.18)$$

Furthermore, the dilution zone equivalence ratio is equal to the overall equivalence ratio

$$\phi_{DZ} = \phi_{overall} \quad (3.3.19)$$

The airflow distribution is determined by computing the air fractions in the various combustor zones. Within the liner,

$$W_{PZ} = \frac{\phi_{overall}}{\phi_{PZ}} \quad (3.3.20)$$

$$W_{DZ} = \frac{\phi_{overall}}{\phi_{DZ}} \quad (3.3.21)$$

and in the remaining passages,

$$W_{plenum} = W_{premixer} = W_{PZ} \quad (3.3.22)$$

$$W_{dilution} = W_{DZ} - W_{PZ} \quad (3.3.23)$$

$$W_{an} = 1.0 - W_{dilution} \quad (3.3.24)$$

where the fraction of air in location i is defined by

$$W_i = \frac{\dot{m}_i}{\dot{m}_3} \quad (3.3.25)$$

3.3.6 Resizing for Premixed Combustion

The combustor is resized to maintain the stability of an equivalent conventional combustor (Gauthier, 2005). Leonard & Stegmaier (1993) stated that the required combustion volume for stable premixed operation is roughly twice that of a conventional combustor.

The scaling parameter used to measure stability is the lean blowout velocity. For a bluff-body flame holder, the lean blowout velocity for homogeneous fuel-air mixtures is (Ballal & Lefebvre, 1981)

$$V_{B0} = C_s (D_c S_L^2 / \alpha_0) \quad (3.3.26)$$

where C_s = a shape factor that depends upon the flameholder

D_c = the characteristic dimension of the flameholder

α_0 = the thermal diffusivity upstream of the flameholder, $\lambda / C_p \rho$

S_L = the laminar flame speed

The flow through a pipe is governed by

$$\dot{m} = \rho A V = \rho \left(\frac{\pi}{4} D^2 \right) V \quad (3.3.27)$$

Non-dimensionalizing Equation 3.3.26 with velocity gives

$$\begin{aligned} \frac{V_{B0}}{V} &= \frac{C_s D_c S_L^2}{\alpha_0} \times \frac{\rho \frac{\pi}{4} D^2}{\dot{m}} \\ &= \frac{\pi}{4} \frac{C_s \rho D^3 S_L^2}{\alpha_0 \dot{m}} \end{aligned} \quad (3.3.28)$$

Assuming the shape factor is constant, Equation 3.3.28 becomes

$$\frac{V_{B0}}{V} \propto \frac{\rho D^3 S_L^2}{\alpha_0 \dot{m}} \quad (3.3.29)$$

where the density obeys the ideal gas law,

$$\rho = \frac{P}{RT} \quad (3.3.30)$$

The flame temperature can be assumed to vary linearly with the equivalence ratio for very weak mixtures, as shown in Figure 3.4. Using this assumption, selecting the

Figure 3.4: Temperature vs. equivalence ratio relationship.

reference velocity as the characteristic velocity, and noting that α_0 and $\dot{m}$ remain constant

$$\frac{V_{B0}}{V_{ref}} \propto \frac{D^3 S_L^2}{\phi} \quad (3.3.31)$$

Rearranging yields the diameter of the liner necessary for premixed combustion

$$D_{liner_2} = D_{liner_1} \times \left(\frac{\phi_{PZ_2}}{\phi_{PZ_1}} \right)^{\frac{1}{3}} \left(\frac{S_{L_1}}{S_{L_2}} \right)^{\frac{2}{3}} \quad (3.3.32)$$

where 1 and 2 denote the conventional and premixed combustors, respectively.

A primary zone equivalence ratio 0.8 was used for the conventional combustor since it is for an industrial application where stability is less critical (Gauthier, 2005).

3.3.6.1 Laminar Flame Speed

The laminar flame speed S_L is the maximum homogeneous burning velocity, as determined by the Semenov equation. Use of the Semenov equation is discussed in detail by Gauthier *et al.* (1996). Nonetheless, the laminar flame speed

$$S_L = \sqrt{\frac{k T_u^2 T_g^5 \exp \left[-\frac{E}{RT_g} \right]}{(T_g - T_u)^3}} \quad (3.3.33)$$

- where k = a constant of proportionality
 T_u = the temperature of the unburned mixture, $= T_3$
 T_g = the temperature of the reacting mixture, $= T_{flame}$
 $\bar{R}$ = the universal gas constant, 8.314 kJ / kmol K
 E = the activation energy, 92 kJ / mole (Gauthier *et al.* , 1996)

The constant of proportionality k is calculated using the data at a known condition. If one is not known, Gauthier *et al.* (1996) stated that the maximum homogeneous burning velocity at ambient conditions (1 atmosphere and 293 K) may be used. This value is approximately 43 cm s⁻¹ for alkanes and occurs when the temperature of the reacting mixture is a maximum, at $\phi = 1.05$. The flame temperature is computed using the procedures discussed in Section 3.3.3.

Both the conventional and premixed laminar flame speeds can be computed once the constant k is known. An equivalence ratio of unity is used for the conventional case since combustion occurs in stoichiometric quantities. The value of the equivalence ratio for the premixed case is that set for the primary zone in Equation 3.3.18.

3.3.6.2 Casing Diameter

The casing diameter is then reworked using

$$A_{liner} = k_{area} A_{ref} \quad (3.3.34)$$

$$D_{ref} = \sqrt{\frac{D_{liner}^2}{k_{area}}} \quad (3.3.35)$$

The ratio of liner to casing cross-sectional area, k_{area} , is selected using the annulus velocity considerations discussed previously in Section 3.3.2. In addition to these considerations, the high annulus velocities required for ABC liner cooling complicate the decision.

3.4 Contraction

A small contraction in the aft section of the liner is used to accelerate the flow and end the recirculation zone. A 30 percent increase in velocity is chosen based on a survey

Figure 3.5: Nozzle dimensions.

of many existing designs. The casing must contract as well in order to maintain a constant annulus area. Figure 3.5 illustrates the contraction.

The liner diameter of the contracted section is

$$D_{liner,2} = \sqrt{0.7} D_{liner} \quad (3.4.1)$$

and the length of the contraction is

$$L_2 = \frac{D_{liner} - D_{liner,2}}{2 \tan \theta_{contract}} \quad (3.4.2)$$

The wall angle $\theta_{contract}$ is chosen to be 30° .

To maintain a constant annulus area, the casing diameter shall be

$$D_{ref,2} = \sqrt{\left(D_{liner,2} + 2t_{liner} + \frac{4}{\pi} A_{an} \right)^2} \quad (3.4.3)$$

3.5 Diffuser Design

This section details the design of the diffuser, consisting of a conical prediffuser followed by a sudden expansion. The optimum geometry for these components is selected and a preliminary performance analysis is conducted. This analysis is performed to ensure that the overall pressure loss incurred by the diffuser is not large. The overall losses are equal to the sum of the losses for each component

$$\Delta P_{diff} = \Delta P_{prediff} + \Delta P_{dump} \quad (3.5.1)$$

Figure 3.6: Prediffuser geometry.

3.5.1 Prediffuser

The prediffuser, as illustrated in Figure 3.6, is designed to provide a 60 percent reduction in velocity. The required area ratio

$$AR_{prediff} = \frac{A_{3.1}}{A_3} = \frac{1}{0.6} = 1.67 \quad (3.5.2)$$

The pressure losses are measured in terms of the static pressure recovery coefficient, C_p , which is defined as

$$C_p = \frac{(P_{3.1} - P_3)}{q_3} \quad (3.5.3)$$

where 3 is the diffuser inlet and 3.1 is the prediffuser exit. Experimental values of C_p based on the diffuser area ratio and the diffuser length to inlet radius ratio are available in Figure 2.4. For a conical diffuser with a prescribed area ratio, the maximum pressure recovery lies along the line of C_p^{**} where the divergence angle is approximately 12° . Choosing $2\theta_{prediff} = 12^\circ$, for $AR = 1.67$

$$C_p = 0.5 \quad (3.5.4)$$

The total pressure loss across the prediffuser

$$\Delta P_{prediff} = \left[\left(1 - \frac{1}{AR_{prediff}^2} \right) - C_p \right] \cdot q_3 \quad (3.5.5)$$

where inlet dynamic pressure is calculated using the following relation

$$q_3 = \frac{1}{2\rho_3} \left(\frac{\dot{m}_3}{A_3} \right)^2 \quad (3.5.6)$$

Figure 3.7: Dump geometry.

Finally, the length of the prediffuser

$$L_{prediff} = \frac{D_{3.1} - D_3}{2 \tan \theta_{prediff}} \quad (3.5.7)$$

where the prediffuser outlet diameter $D_{3.1}$ is

$$A_{3.1} = AR \cdot A_3 \quad (3.5.8)$$

$$D_{3.1} = \sqrt{\frac{4}{\pi} A_{3.1}} \quad (3.5.9)$$

3.5.2 Sudden Expansion

The losses in the sudden expansion, illustrated in Figure 3.7, correspond well with theoretical dumping losses (Dodds & Bahr, 1990). Applying the momentum equation with the assumption of uniform, steady, incompressible flow, with negligible frictional effects

$$\Delta P_{dump} = \left[1 - \frac{1}{AR_{dump}} \right]^2 \cdot q_{3.1} \quad (3.5.10)$$

These losses are doubled to provide a conservative estimate.

Computation of the dump area ratio AR_{dump} has proven to be somewhat problematic. The flow in the dump region is highly separated and an effective flow area is difficult to define. It is therefore assumed in the present design that the flow

Figure 3.8: Premixer dimensions.

reattaches in the annulus and thus, the dump diffuser area ratio is

$$AR_{dump} = \frac{A_{3,2}}{A_{3,1}} \quad (3.5.11)$$

where the area at the exit of the dump region $A_{3,2} \approx A_{an}$.

The dynamic pressure at the inlet to the sudden expansion is

$$q_{3,1} = \frac{q_3}{AR_{dump}^2} \quad (3.5.12)$$

3.6 Premixer Design

The premixer is designed to evaporate the liquid fuel and mix the fuel vapour with air. Figure 3.8 illustrates the premixer concept and indicates the relevant parameters.

The air enters the premixer and passes through two concentric, counter-rotating swirlers where liquid fuel is injected into the air. Injection is accomplished using pressure nozzles that produce an atomized cone spray of fine droplets. The droplets evaporate and the resultant vapour mixes with the air to form a combustible mixture.

The shear layer created between the two counter-rotating streams helps mix the fuel vapour supplied by the evaporating droplets with the air. The rate of mixing,

combined with the rate of evaporation, determine the premixer length; the premixer must be sufficiently long to allow both to progress to completion.

The fuel/air mixture passes through a third swirler before entering the combustion chamber and reacting. This final swirler ensures that the flow has sufficient swirl to produce a strong recirculation zone. It also prevents radiation from entering the premixer and potentially igniting the the fuel/air mixture.

Precautions Many precautions must be taken to ensure that the fuel/air mixture does not react prior to entering the primary zone of the combustor. Any potential flameholders, such as struts, fuel lines, hubs, or any other bluff bodies where combustion can occur in its wake, must be eliminated. Therefore, the co-axial swirlers are designed to ensure that no recirculation bubbles form downstream.

Slow moving boundary layers on the premix tube walls must also be minimized to ensure that the flame will not creep upstream and cause flashback. The premixer tube converges to continually accelerate the flow and flatten any thick, developed boundary layers.

Chemical reaction can take place inside the premixer by autoignition after exposure to heated air. This spontaneous ignition can be avoided by ensuring that the residence time of the fuel/air mixture in the premixer is less than the autoignition delay time.

3.6.1 Injector Selection

Injector selection is a critical step in the premixer design. The nozzle(s) must provide a sufficiently fine mist of fuel droplets without requiring excessive fuel line pressure. Finer droplets require less time to evaporate and allow for shorter premixer tube lengths. A survey of pumps and existing fuel injection systems revealed that fuel pressures rarely exceed 1000 psi (6.9 MPa) (Lefebvre, 1999). Pressures above this value are beyond the capabilities of most small fuel pumps.

A major drawback of pressure atomizers is that the flow rate varies proportionally

Figure 3.9: Fuel schedule with multiple nozzles.

with the square root of the injection pressure differential across the nozzle, ΔP_n . Consequentially, good atomization only occurs over a narrow range of ΔP_n . Several individually addressable nozzles are used to improve the turndown ratio of the fuel injection system. They are activated/deactivated to maintain a relatively constant fuel line pressure while varying Q . This concept is illustrated in Figure 3.9. A disadvantage of the configuration is that the fuel system complexity and overall cost accumulate with the addition of nozzles.

Nozzle selection involves choosing a nozzle type to provide the required values of SMD and spray cone angle α_n at a specified ΔP_n . The number of nozzles is then selected to provide favourable fuel scheduling characteristics. Typically, nozzle manufacturers provide the mass flow, or volume flow rate, versus nozzle pressure relationship. For pressure nozzles

$$Q = K\sqrt{\Delta P_n} \quad (3.6.1)$$

where Q is the flow rate of fuel through the nozzle and K is a proportionality constant specific to the nozzle.

Another important parameter used in the premixer design is the ideal droplet

Figure 3.10: Premixer design flowchart.

discharge velocity

$$V_n = \sqrt{\frac{2\Delta P_n}{\rho_l}} \quad (3.6.2)$$

where ρ_l is the liquid density of fuel, equal to approximately 840 kg/m³ for diesel fuel (Gardner & Whyte, 1990).

The power required to drive the pump is helpful in validating the choice of nozzle and configuration. It is estimated by Equation 3.6.3.

$$W_{pump} = \frac{(\Delta P_n + P_3)(\dot{m}_f/\rho_l)}{\eta_{pump}} \quad (3.6.3)$$

where η_{pump} is the efficiency of the pump ($\eta_{pump} \approx 50\%$ for positive displacement pumps).

3.6.2 Swirler Design

The swirler design process is iterative, as the velocity in the premix tube is dependent on the unknown outer dimensions of the swirlers.

Figure 3.10 illustrates the swirler design process. The average velocity in the mixer tube, V_{avg} , is required to estimate the cross-sectional area of the mixer tube. It is defined by

$$V_{avg} = \frac{V_{in} + V_{out}}{2} \quad (3.6.4)$$

where V_{in} and V_{out} are the velocities at the inlet and outlet of the mixing tube,

respectively. They are based on the diameter of the mixing tube at each location

$$V_{in} = \frac{W_{premixer}\dot{m}_3}{\frac{\pi}{4}D_{mix,o}^2\rho_3} \quad (3.6.5)$$

$$V_{out} = \frac{W_{premixer}\dot{m}_3}{\frac{\pi}{4}D_{swirl}^2\rho_3} \quad (3.6.6)$$

The length of premix tube can then be defined once iteration is complete.

3.6.2.1 Axial Swirler Design Procedure

Three axial vane swirlers are used in the premixer. Design of these axial swirlers involves selecting the type of vanes used (flat or curved), the vane angle θ_v , vane thickness t_v , vane aspect ratio h/c , solidity σ , and inner diameter D_{hub} .

The solidity is a measure of the apparent blockage of the swirler or vane overlap. For a solidity less than unity, the vanes do not overlap and one can see through the swirler. A solidity of unity dictates that one cannot see through the swirler yet the vanes do not overlap. The vanes completely overlap for a solidity greater than unity.

From the parameters discussed above, the number of vanes n_v and the outer diameter D_{sw} required to provide the mass flow rate $\dot{m}_{sw}$ are computed. Figure 3.11 illustrates many of these parameters.

The mass flow rate of air through the swirler strongly depends on the pressure loss across this device. However, relating the two is difficult due to the lack of empirical correlations. Knight & Walker (1957) provided one correlation for an axial swirler admitting air into a combustion chamber. The relationship between mass flow and pressure loss is governed by

$$\dot{m}_{sw} = \left\{ \frac{2\rho_3\Delta P_{sw}}{K_{sw} \left[\left(\frac{\sec \theta_v}{A_{sw}} \right)^2 - \frac{1}{A_{liner}^2} \right]} \right\}^{0.5} \quad (3.6.7)$$

where ΔP_{sw} = the total pressure drop across the swirler
(approximately equal to that across the liner)

$\dot{m}_{sw}$ = the swirler mass flow

K_{sw} = 1.3 for flat vanes, 1.5 for curved vanes

Figure 3.11: Axial swirler dimensions.

A_{sw} = the frontal area of the swirler

Setting A_{liner} equal to the cross-sectional area of the dump downstream, A_{dump} , and rearranging for the swirler effective frontal area gives

$$A_{sw} = \sqrt{\frac{\sec^2 \theta_v}{\frac{2\rho_3 \Delta P_{mix}}{K_{sw} \dot{m}_{sw}^2} + \frac{1}{A_{dump}^2}}} \quad (3.6.8)$$

The swirler frontal area is defined from geometry as

$$A_{sw} = \frac{\pi}{4} (D_{sw}^2 - D_{hub}^2) + n_v \left(\frac{t_v}{\cos \theta_v} \right) \left(\frac{D_{sw} - D_{hub}}{2} \right) \quad (3.6.9)$$

The length of the vanes is determined from the aspect ratio

$$c_v = \left(\frac{D_{sw} - D_{hub}}{2(h/c)} \right) \quad (3.6.10)$$

and the number of vanes is

$$n_v = \sigma \frac{(\pi D)}{c_v \sin \theta_v} \quad (3.6.11)$$

where D is the diameter at the location the solidity is evaluated.

Simultaneous solution of Equations 3.6.8, 3.6.9, 3.6.10, and 3.6.11 yields both the number of vanes and the outer diameter of the swirler.

The swirl number S_n may be computed once the swirler geometry has been defined to ensure that the desired swirl strength is achieved. For axial swirlers (Lilley, 1977):

$$S_n = \frac{2}{3} \left[\frac{1 - (D_{hub}/D_{sw})^3}{1 - (D_{hub}/D_{sw})^2} \right] \tan \theta_v \quad (3.6.12)$$

3.6.2.2 Mixer Swirlers

The mixer swirlers are designed using the procedures discussed above. Flat vanes were chosen for both mixers due to their simplicity and manufacturability. Additionally, both swirlers are designed for a solidity of unity at the mean radius. This decision is based on the work of Martin (1988).

The division of mass flow between the swirlers and their vane angles were selected after consulting the work of Joshi *et al.* (1994). The author suggests a mass flow split of 4:1, and vane angles of 60° and 30° for the outer and inner swirlers, respectively. The inner flow with weaker swirl helps increase the axial velocity along the premixer centerline, hindering the formation of a recirculation vortex. Recirculation should be avoided inside the premixer as it acts as a potential flameholder.

The design parameters discussed above, as well as others, are tabulated in Table 3.1.

The downstream cross-sectional flow area of the mixer is approximated from the average velocity with

$$A_{premixer} = \frac{W_{premixer} \dot{m}_3}{\rho_3 V_{avg}} \quad (3.6.13)$$

3.6.2.3 Combustor Swirler

The combustor swirler is designed to utilize the remaining available pressure loss. For the combustor swirler

$$\Delta P_{swirl} = \Delta P_{3-4} - \Delta P_{diff} - \Delta P_{plenum} - \Delta P_{mix} \quad (3.6.14)$$

where ΔP_{mix} is the mixer swirler loss (1.0 percent) and ΔP_{plenum} is the expansion loss in the plenum region. The latter is determined from the corresponding losses for

Table 3.1: Design parameters for mixer swirlers.

	Inner	Outer
Rotation	counter-clockwise	clockwise
ΔP_{sw}	1.0%	1.0%
θ_v	30°	60°
$\dot{m}_{sw}$	$\frac{1}{1+4} W_{premixer} \dot{m}_3$	$\frac{4}{1+4} W_{premixer} \dot{m}_3$
K_{sw}	1.3 (flat vanes)	1.3 (flat vanes)
h/c	1	1
σ	1 at mean radius	1 at mean radius
t_v	specified by designer	specified by designer
D_{hub}	specified by designer	$D_{mix,i}$
A_{dump}	$A_{premixer}$	$A_{premixer}$

a sudden expansion

$$\Delta P_{plenum} = \left(1 - \frac{A_{an}}{A_{plenum}}\right)^2 \cdot q_{an} \quad (3.6.15)$$

The plenum area is approximately

$$A_{plenum} = \frac{\pi}{4} D_{ref,1}^2 - A_{premixer} \quad (3.6.16)$$

and the dynamic pressure in the annulus is

$$q_{an} = \frac{1}{2\rho_3} \left(\frac{W_{an} \dot{m}_3}{A_{an}} \right)^2 \quad (3.6.17)$$

The casing diameter in this region, $D_{ref,1}$, is defined in Section 3.7.

The design parameters for the combustor swirler are similar to those of the mixer swirlers. They are provided in Table 3.2. A vane angle of 60° is specified to provide a strong recirculation vortex in the primary combustion zone. This blade angle corresponds to a swirl number of approximately 1.15 for an axial swirler with a small hub (Equation 3.6.12 with $D_{hub}/D_{sw} \ll 1$). It is well in excess of the critical value of 0.6 required for recirculation. However, the high swirl number is chosen to ensure a strongly swirling flow entering the PZ. Additionally, a solidity of unity is specified at

Table 3.2: Design parameters for combustor swirler.

Rotation	counter-clockwise
ΔP_{sw}	from Equation 3.6.14
θ_v	60°
$\dot{m}_{sw}$	$W_{premixer}\dot{m}_3$
K_{sw}	1.3 (flat vanes)
h/c	1
σ	1 at outer radius
t_v	specified by designer
D_{hub}	specified by designer
A_{dump}	A_{liner}

the outer radius, instead of at the mean radius, so that any radiation from the PZ is blocked by removing the line of sight through the swirler.

3.6.3 Mixer Tube

The mixer is designed to ensure that complete evaporation and mixing occurs at all operating conditions without the occurrence of autoignition. For simplicity, it is assumed that mixing occurs instantaneously and the length is based solely on considerations for evaporation. This implies that evaporation occurs at a much slower rate than mixing, which is a reasonable assumption considering the low temperature and pressure inside the premixer.

3.6.3.1 Droplet Evaporation

The evaporation time for the liquid fuel is estimated by modeling the spray as a single droplet having the same SMD as that of the spray. The droplet, injected into the mixing tube, is tracked as it travels through the bulk flow and evaporates. The length necessary for complete evaporation is denoted by the axial distance where the droplet diameter is zero, as illustrated in Figure 3.12.

Figure 3.12: Injection of droplet into mixer tube.

The droplet evaporation model used is described in detail by Hallett (2003), Turns (2000), Kuo (1986), and Lefebvre (1989). It requires several simplifying assumptions:

1. the droplet is perfectly spherical,
2. quasi-steady state evaporation,
3. the transport properties remain constant spatially,
4. constant pressure,
5. pure fuel,
6. no chemical reaction,
7. uniform droplet temperature, and
8. unity Lewis number for the vapour/air mixture surrounding the droplet.

Quasi-steady evaporation means that any changes in droplet temperature or size cause the vapour field surrounding the droplet to adjust instantaneously. This allows the transient effects in the transport equations to be neglected.

With these assumptions, the evaporation of a droplet is described simply by

$$\frac{dD}{dt} = -\frac{2G}{\rho_l} \quad (3.6.18)$$

where D = the droplet diameter
 G = the mass flux of fuel vapour from the droplet surface
 ρ_l = the liquid fuel density

The mass flux G from the droplet surface, accounting for the effects of forced convection, is

$$G = Nu \frac{\lambda}{C_{pv} D} \ln(1 + B_T) \quad (3.6.19)$$

where Nu = the droplet Nusselt number
 λ = the mixture thermal conductivity
 C_{pv} = the fuel vapour specific heat
 B_T = the Spalding Transfer number

The Spalding transfer number,

$$B_T = \frac{Y_R - Y_\infty}{1 - Y_R} \quad (3.6.20)$$

where Y_R and Y_∞ denote the mass fraction of fuel vapour at the droplet surface and in the surrounding atmosphere, respectively.

The Nusselt number is estimated using the Ranz-Marshall correlation for the convective heat transfer to a sphere

$$Nu = 2 \left(1 + 0.3Re^{1/2} Pr^{1/3} \right) \quad (3.6.21)$$

where both the Prandtl number Pr and the Reynolds number Re are defined as

$$Pr = \frac{\eta C_p}{\lambda} \quad (3.6.22)$$

$$Re = \frac{\rho V_r D}{\eta} \quad (3.6.23)$$

where V_r is the relative velocity of the droplet. The mixture specific heat, viscosity, and density are denoted by C_p , η , and ρ , respectively.

A heat balance is performed on the droplet surface assuming that the droplet temperature is uniform throughout. The balance gives

$$q_R = Gh_{fg} + \frac{\rho_l C_{pl} D}{6} \frac{dT_l}{dt} \quad (3.6.24)$$

where q_R = the heat flux to the droplet surface
 h_{fg} = the latent heat of vaporization of the fuel
 C_{pl} = the specific heat of liquid fuel
 T_l = the droplet temperature

Rearranging gives

$$\frac{dT_l}{dt} = \frac{6}{\rho_l C_{p_l} D} (q_R - Gh_{fg}) \quad (3.6.25)$$

The heat transferred to the droplet surface is determined from the mass flux solution,

$$q_R = \frac{GC_{p_f} (T_\infty - T_l)}{\exp\left(\frac{GC_{p_f} D}{2\lambda}\right) - 1} \quad (3.6.26)$$

where T_∞ is the temperature of the ambient surroundings.

The heat and mass transfer solutions are coupled with the vapour-liquid equilibrium relationship at the liquid surface. Thus,

$$\frac{1}{Y_R} = 1 + \left(\frac{P_\infty}{P_v} - 1\right) \frac{MW_\infty}{MW_f} \quad (3.6.27)$$

where the vapour pressure P_v is given by the Antoine equation and P_∞ is the pressure of the surroundings. The molecular weight of the ambient surroundings (air) and the fuel are denoted by MW_∞ and MW_f , respectively.

Neglecting the variation of properties in the vapour field requires the use appropriate mean values. Hubbard *et al.* (1975) states that a 1/3 rule should be used.

$$\bar{T} = \frac{2}{3}T_l + \frac{1}{3}T_\infty \quad (3.6.28)$$

$$\bar{Y} = \frac{2}{3}Y_R + \frac{1}{3}Y_\infty \quad (3.6.29)$$

The properties for the mixture are then evaluated at the mean temperature $\bar{T}$ and fuel mass fraction $\bar{Y}$. However, evaluating the properties of mixtures is inconvenient for use in simple calculations. Hallett (2003) recommended evaluating the mixture ρ , λ , η , and C_p as for air at $\bar{T}$. In most cases $\bar{Y}$ is small enough that the mixture comprises mostly air.

The relevant properties of fuel (modeled as n-dodecane) along with those of air are listed in Tables A.3 and A.4.

3.6.3.2 Droplet Motion

The mean particle trajectory is modeled to determine the length required for complete evaporation inside the premixer. The motion of a particle in three dimensions is

Figure 3.13: Droplet trajectory.

described by Newton's second law of motion. In cylindrical coordinates (r, θ, x) , the axial, radial, and tangential components of particle velocity illustrated in Figure 3.13 are given by (Hallett, 2003)

$$\frac{du_p}{dt} = -\frac{1}{\tau} (u_p - u_b) \quad (3.6.30)$$

$$\frac{dv_p}{dt} = -\frac{1}{\tau} (v_p - v_b) - \frac{w_p^2}{r} \quad (3.6.31)$$

$$\frac{dw_p}{dt} = -\frac{1}{\tau} (w_p - w_b) - \frac{v_p w_p}{r} \quad (3.6.32)$$

where the effects of gravity and buoyancy have been neglected. The subscripts p and b denote the particle and bulk flow velocity components, respectively.

The quantity τ is the time constant for particle motion (Hallett, 2003)

$$\tau = \frac{4}{3} \frac{D}{C_D} \frac{\rho_l}{\rho} \frac{1}{V_r} \quad (3.6.33)$$

where the relative velocity is defined by the relation

$$V_r = \sqrt{(u_p - u_b)^2 + (v_p - v_b)^2 + (w_p - w_b)^2} \quad (3.6.34)$$

The drag coefficient is evaluated using the modified Stokes relation (Hallett, 2003).

$$C_D = \frac{24}{Re} (1 + 0.15Re^{0.687}) \frac{1}{1 + B_T} \quad (3.6.35)$$

Finally, the location of the particle may be tracked by integrating the partial differentials.

$$\frac{dx_p}{dt} = u_p \quad (3.6.36)$$

$$\frac{dr_p}{dt} = v_p \quad (3.6.37)$$

$$\frac{d\theta_p}{dt} = \frac{v_p}{r_p} \quad (3.6.38)$$

3.6.3.3 Effects of Turbulence

Turbulence in the free stream affects the particle trajectory and intensifies the rate of convective heat and mass transfer. Its effects on convection become important when the Reynolds number based on the turbulent fluctuations in the bulk flow is larger than the particle Reynolds number defined in Equation 3.6.23 (Gauthier, 2005).

Equation 3.6.21 is modified in this case by defining Re based on the turbulent fluctuations

$$Re' = \frac{\rho q D}{\eta} \quad (3.6.39)$$

The mean turbulent fluctuation of the bulk flow q is defined by White (1999) as

$$q = \sqrt{\frac{1}{3} (\overline{u'^2} + \overline{v'^2} + \overline{w'^2})} \quad (3.6.40)$$

In practice these fluctuations are commonly quoted as a percentage of the free stream velocity. This percentage is called the turbulence intensity TI ,

$$TI = \frac{q}{V_b} \quad (3.6.41)$$

where V_b is the bulk flow velocity. Taylor (1978) states that a turbulence intensity of 10-15 percent is typical in gas turbine combustors .

3.6.3.4 Solution

Equations 3.6.18, 3.6.25, 3.6.30, 3.6.31, 3.6.32, 3.6.36, 3.6.37, and 3.6.38 form a set of ordinary differential equations (ODEs). They may be integrated numerically using a computer with the known initial conditions. For the droplet, at time $t = 0$

$$\begin{aligned} T_{l_0} &= 300 \text{ K} \\ u_{p_0} &= V_n \cos \alpha_n / 2 \\ v_{p_0} &= V_n \sin \alpha_n / 2 \\ w_{p_0} &= 0 \\ x_{p_0} &= 0 \\ r_{p_0} &= 0 \\ D_0 &= SMD \end{aligned}$$

The bulk flow conditions at time $t = 0$

$$\begin{aligned} T_\infty &= T_3 \\ P_\infty &= P_3 \\ Y_\infty &= 0 \\ u_b &= V_{avg} \\ v_b &= 0 \\ w_b &= V_{avg} \tan \theta_v \end{aligned}$$

3.6.4 Autoignition

Autoignition is most likely to occur at full power when the temperature and pressure in the premixer are highest. It is avoided by ensuring that the mixture of fuel and air do not remain in the mixer tube for too long. A compromise between evaporation, mixedness, and autoignition must typically be made.

Wolfer (1938) provided a simple means of correlating the ignition delay time with temperature and pressure. The ignition delay time is thus

$$t_{ig} = 0.43P^{-1.19} \exp \left[\frac{4650}{T_u} \right] \quad (3.6.42)$$

where t_{ig} = the ignition delay time in milliseconds

P = the pressure in bars, $= P_3$

T_u = the initial mixture temperature in degrees Kelvin, $= T_3$

And the length until ignition occurs is

$$L_{ig} \approx V_{premixer} t_{ig} \quad (3.6.43)$$

The premixer length is then the smallest of the length required for evaporation and the distance at which autoignition occurs.

$$L_{premixer} = \min [L_{evap}, L_{ig}] \quad (3.6.44)$$

3.6.5 Flashback

According to Poeschl *et al.* (1994), flashback should not be a problem if the velocity in the premixer is higher than 20 m/s. Thus,

$$V_{avg} > 20 \text{ m/s} \quad (3.6.45)$$

3.6.6 Pilot

A pilot is required to ensure stable combustion during idle and other low power conditions. It is usually a small region in the combustor that operates stoichiometrically to supply hot gases and relight the primary zone in the event of a flameout. The same result is achieved by shortening the length of the premix tube to ensure that some fuel droplets escape the premixer into the combustion zone and create a small diffusion flame.

Figure 3.14: The dome region.

3.7 Dome Region

The dome, illustrated by Figure 3.14, is tailored to provide the desired flow pattern in the primary combustion zone. A steep wall angle of the dome, θ_{dome} , of 60° was chosen based on the recommendations of Sawyer (1985).

The dome length may then be computed from trigonometry

$$L_{dome} = \frac{D_{liner} - D_{swirl}}{2} \tan \theta_{dome} \quad (3.7.1)$$

The diameter of the casing near the dome region

$$D_{ref,1} = 0.6 D_{ref} \quad (3.7.2)$$

3.8 Combustor Length

The combustor overall length is the sum of the lengths of the primary and dilution zones, illustrated in Figure 3.15.

$$L_{tot} = L_{PZ} + L_{DZ} \quad (3.8.1)$$

Figure 3.15: Primary and dilution zone lengths.

Dilution Zone The DZ provides the time for the hot PZ combustion gases to mix with the cooler annulus air. The length of the zone is determined based on jet mixing considerations since the DZ must provide the downstream turbine with a specific pattern factor Q . The pattern factor is defined as

$$Q = \frac{T_{max} - T_4}{T_4 - T_3} \quad (3.8.2)$$

where T_{max} is the maximum recorded temperature at the combustor exit. It is of little importance in this application because the combustor exhaust is dumped into a large plenum before entering the turbine. The combustion products will be given sufficient time to mix inside this plenum, which will provide the downstream turbine with a uniform mixture.

The DZ length is sized accordingly.

$$L_{DZ} = 0.2D_{liner} \quad (3.8.3)$$

Primary Zone Unlike the DZ, the length of the PZ is set to provide sufficient volume to operate stably. This value must typically be compromised to avoid an

Figure 3.16: Combustor reactor model.

excessively large combustor. A survey of existing designs reveals that this length is typically 90 percent of the liner diameter.

$$L_{PZ} = 0.9D_{liner} \quad (3.8.4)$$

This length is verified using reactor theory.

3.8.1 Reactor Modeling

Reactor theory allows the designer to describe the detailed evolution of the reaction within the combustor. The reader is directed to Turns (2000) for a complete mathematical description of the reactors used.

Here the combustor is modeled using a simplified network consisting of two reactors, a WSR and a PFR. The WSR simulates combustion in the PZ while the PFR simulates the dilution of the combustion products in the DZ. This configuration is shown in Figure 3.16.

The AEDsys program KINETX (Mattingly *et al.*, 2002) is used to model the combustion inside the liner. The program uses a 30-step reaction mechanism devised by Kollrack (1976) for modeling the combustion of "jet fuel", represented as $C_{12}H_{23}$, and includes the production of NO_x . Based on the air and fuel flow to each zone, and the dimensions of each zone (cross-sectional flow area and length), the program solves the mathematical model for the reactor network. It outputs the species concentrations and temperature in each reactor and computes relevant performance parameters such

as the combustor loading, the combustor loading at blowout, the residence time in each zone, and the combustion efficiency.

3.9 Dilution Hole Design

The liner dilution holes are designed to provide the desired penetration of the admitted jets. The hole diameter D_h and number of holes n_h are determined by the available pressure drop across the liner ΔP_{liner} , the total mass flow rate of the dilution air $\dot{m}_j$, and the mass flow rate of hot gas in the liner $\dot{m}_g$. The pressure drop across the liner is defined as

$$\Delta P_{liner} = \Delta P_{3-4} - P_{diff} \quad (3.9.1)$$

The Cranfield and the NASA design methods are both commonly used to perform the design of the dilution holes. This section details the use of the former.

3.9.1 Cranfield Design Method

The Cranfield design method, discussed in detail by Lefebvre (1999), provides an optimum hole size for a given momentum flux ratio J and selects the spacing between the adjacent holes to provide the desired $\dot{m}_j/\dot{m}_g$ ratio. This method requires a means for estimating both the penetration distance of the jets and the hole discharge coefficient.

3.9.1.1 Penetration Distance

Hole design begins with the computation of the jet diameter D_j necessary for a given penetration distance, which is the maximum depth reached by the jets. The jet diameter is depicted in Figure 3.17 along with other relevant parameters involved in hole design.

The maximum penetration distance of round air jets can be estimated by (Lefebvre, 1999)

$$Y_{max} = 1.25 D_j J^{0.5} \frac{\dot{m}_g}{\dot{m}_g + \dot{m}_j} \quad (3.9.2)$$

Figure 3.17: Jet penetration.

Lefebvre (1999) suggests designing for an optimum penetration distance of $0.33D_{liner}$ for tubular combustors.

The momentum flux ratio is defined as

$$J = \frac{\rho_j V_j^2}{\rho_g V_g^2} \quad (3.9.3)$$

where j denotes the dilution jet while g denotes the hot gas flow.

$$V_j = \sqrt{\frac{2\Delta P_{liner}}{\rho_3}} \quad (3.9.4)$$

$$V_g = \frac{\dot{m}_g}{\frac{\pi}{4} D_{liner}^2 \rho_g} \quad (3.9.5)$$

and

$$\rho_g = \frac{P_3}{RT_g} \quad (3.9.6)$$

The temperature at the end of the primary zone is roughly equal to the flame temperature T_{flame} .

The corresponding number of holes required is derived from mass conservation.

$$n_h = \frac{\dot{m}_j}{\frac{\pi}{4} D_j^2 \rho_3 V_j} \quad (3.9.7)$$

3.9.1.2 Discharge Coefficient

The discharge coefficient relates the actual hole area to the effective jet flow area as the jet contracts and accelerates through the opening. The equation for flow through a hole is

$$\dot{m}_h = C_D A_h [2\rho_3 (P_{an} - P_j)]^{0.5} \quad (3.9.8)$$

where P_{an} = the total pressure upstream of hole
 P_j = the static pressure downstream of hole
 A_h = the hole cross-sectional area
 $\dot{m}_h$ = the mass flow rate through the hole

Thus,

$$D_h = D_j / \sqrt{C_D} \quad (3.9.9)$$

A simple correlation for estimating the discharge coefficient for round sharp-edged holes is provided by Kaddah (1964). The author stated that

$$C_D = \frac{1.25 (K - 1)}{[4K^2 - K(2 - \beta)^2]^{0.5}} \quad (3.9.10)$$

where β = the ratio of hole mass flow rate to annulus mass flow rate ($\dot{m}_j/\dot{m}_{an}$)
 K = the ratio of the jet dynamic pressure to the annulus dynamic pressure upstream of holes

The coefficient K is approximated by the correlation

$$K = 1 + 0.64 \left\{ 2\mu^2 + \sqrt{4\mu^4 + 1.56\mu^2(4\beta - \beta^2)} \right\} \quad (3.9.11)$$

where α is the ratio of hole cross-sectional area to annulus cross-sectional area and $\mu = \beta/\alpha$.

Figure 3.18: Duct geometry.

3.9.2 Jet Trajectory

Modeling the cool jet as it penetrates and mixes with the hot combustion gases is useful for estimation of the pattern factor at the combustor exit. One empirical model for modeling crossflowing jets is described by Holdeman *et al.* (1987). The model provides a detailed prediction of jet penetration profile.

The author stated that the radial location of the jet centreline, Y , is described by

$$\frac{Y}{H_{eq}} = (a_1) (0.3575) (J)^{0.25} \left(\frac{S}{D_h}\right)^{0.14} \left(\frac{H_{eq}}{D_h}\right)^{-0.45} (C_D)^{0.155} \left(\frac{X}{H_{eq}}\right)^{0.17} [\exp(-b)] \quad (3.9.12)$$

where

$$\begin{aligned} H_{eq} &= \frac{H_0}{2} \\ a_1 &= \min \left[\left(1 + \frac{S}{H_{eq}} \right), 2 \right] \\ b &= (0.091) \left(\frac{X}{H_{eq}} \right)^2 \left[\left(\frac{S}{H_{eq}} \right) - \left(\frac{J^{0.5}}{3.5} \right) \right] \end{aligned}$$

The relevant geometrical parameters are illustrated in Figure 3.18, they include:

- H_0 = duct height
- S = circumferential spacing between orifice centres
- X = the axial distance downstream of orifice center

The jet spreading is characterized by the half width $W_{1/2}$, which is defined as the position where the jet temperature is equally between the peak temperature and the

Figure 3.19: Jet profile.

gas temperature (Figure 3.19). For the half width on the opposite side of the injection hole

$$\frac{W_{1/2}^+}{H_{eq}} = (0.1623) (J)^{0.18} \left(\frac{S}{D_h} \right)^{-0.25} \left(\frac{H_0}{H_{eq}} \right)^{0.5} (C_D)^{0.125} \left(\frac{X}{H_{eq}} \right)^{0.5} \quad (3.9.13)$$

On the injection side of the jet centreline,

$$\frac{W_{1/2}^-}{H_{eq}} = (0.2) (J)^{0.15} \left(\frac{S}{D_h} \right)^{0.27} \left(\frac{H_{eq}}{D_h} \right)^{-0.38} \left(\frac{H_0}{H_{eq}} \right)^{0.5} (C_D)^{0.055} \left(\frac{X}{H_{eq}} \right)^{0.12} \quad (3.9.14)$$

3.10 Igniter Sizing

The igniter chosen must be capable of providing the minimum amount of energy necessary to initiate combustion. It must also be located in a region where the fresh fuel and air mixture is expected.

3.10.1 Minimum Ignition Energy

For industrial combustors, ignition is most difficult during engine startup where the temperature and pressure inside the combustor are lowest. Since liquid fuel droplets do not have enough time to evaporate before entering the PZ, the igniter must initiate combustion in a heterogenous mixture of fuel droplets in air rather than a gaseous one. Heterogenous mixtures require more energy for ignition since additional energy must be supplied for evaporation. Thus, the minimum ignition energy for the combustor is determined based on that for a fully unevaporated spray at idle conditions.

Determining the minimum ignition energy first requires estimation of the quenching distance. Lefebvre (1999) stated that the quenching distance of slow-moving monodisperse sprays is

$$D_q = \left[\frac{\rho_l D^2}{\rho_a \phi \ln(1 + B_{st})} \right]^{0.5} \quad (3.10.1)$$

where B_{st} = stoichiometric fuel mass-transfer number

= 2.80 for Diesel fuel (Lefebvre, 1983)

ρ_l = the liquid fuel density

ρ_a = the air density

D = the droplet diameter, = SMD

ϕ = the equivalence ratio, = ϕ_{PZ}

The minimum ignition energy is related to the quenching distance by (Lefebvre, 1999)

$$E_{min} = C_{pa} \rho_a \Delta T_{st} \left(\frac{\pi}{6} \right) D_q^3 \quad (3.10.2)$$

where ΔT_{st} is the stoichiometric temperature rise found using the procedures outlined in Section 3.3.3.

3.10.2 Location

The location of the igniter is estimated using the concept of "magic circles" (Kretschmer, 2003). Magic circles, as depicted in Figure 3.20, attempt to capture the recirculation

Figure 3.20: Igniter location.

patterns inside the liner. These circles, half the diameter of the liner, are placed inside the liner touching the dome wall, the liner wall, and the liner centreline.

The igniter is placed at the location where the magic circles touch the liner wall, a region anticipated to be rich in fresh fuel/air mixture.

$$L_{ig} = \frac{D_{liner}}{4} \left[1 + \cot \left(\frac{\pi - \theta_{dome}}{2} \right) \right] + L_{dome} - \frac{D_{liner}}{8} \quad (3.10.3)$$

3.11 Liner Heat Transfer

The liner wall is fabricated from Hastelloy X. This material is suitable for long-term operation at temperatures up to 1150 K (Dodds & Bahr, 1990). Successful mechanical design requires adequate removal of heat to avoid damage of the liner since the hot gases contained within it are expected to greatly exceed this limit.

Designing the cooling scheme is made possible with estimates for the wall temperature. First, the uncooled wall temperature is computed so that the designer may gauge the cooling requirements and select an appropriate scheme. The scheme is then designed and the liner is analyzed once again to ensure sufficient heat removal.

3.11.1 Uncooled Wall Temperature

The uncooled wall temperature is computed using the heat transfer model developed by Lefebvre & Herbert (1960). This section outlines the gas temperature distribution assumed and the correlations used to perform the heat transfer analysis of the liner.

3.11.1.1 Gas Temperature Distribution

The first step in computing the wall temperature is estimating the temperature of the gas in the flame tube. Since combustion is premixed, it is reasonable to assume that the temperature of the hot gases everywhere in the primary zone is equal to the flame temperature. The assumption that the temperature rises linearly along the dome is also made to estimate the rate at which the fresh mixture reaches the peak flame temperature. Thus

$$T_{PZ} = \begin{cases} T_3 + (T_{flame} - T_3) \frac{x}{L_{dome}}, & x \leq L_{dome} \\ T_{flame}, & x > L_{dome} \end{cases} \quad (3.11.1)$$

where x denotes the axial position from the inlet of the liner.

The gas temperature drops in the dilution zone when jets of cool annulus air are admitted into the liner. The rate of mixing between the cool jets and the hot combustion products determines how quickly the temperature drops to T_4 . It is assumed that the circumferential hole spacing is small enough that mixing in the near wall regions occurs quickly. This warrants the assumption that

$$T_{DZ} = T_4 \quad (3.11.2)$$

everywhere in the dilution zone.

3.11.1.2 Heat Transfer Model

An energy balance on a section of liner gives

$$K_{1-2} = R_1 + C_1 = R_2 + C_2 \quad (3.11.3)$$

where the heat transfer along the liner walls in the axial direction has been neglected.

Internal Radiation The heat transferred by radiation from the flame to the liner wall is given by (Lefebvre & Herbert, 1960)

$$R_1 = \sigma \left(\frac{1 + \varepsilon_{liner}}{2} \right) \varepsilon_g T_g^{1.5} (T_g^{2.5} - T_{w_1}^{2.5}) \quad (3.11.4)$$

where σ = the Stefan-Boltzmann constant = $5.67 \times 10^{-8} W/(m^2 \cdot K^4)$
 T_g = the gas temperature in the liner
 T_{w_1} = the inner surface temperature of the liner wall
 ε_g = the gas emissivity at temperature T_g
 ε_{liner} = the liner wall emissivity

The emissivity for luminous gases can be estimated with

$$\varepsilon_g = 1 - \exp [-290PL(ql_b)0.5T_g^{-1.5}] \quad (3.11.5)$$

where P = the gas pressure in kPa, = P_3
 L = the flame luminosity factor
 l_b = the beam length of the liner in m, = $3.4(\text{volume})/(\text{surface area})$
 q = the fuel/air ratio by mass

Kretschmer & Odgers (1978) recommend the following for the luminosity factor

$$L = 0.0691(C/H - 1.82)^{2.71} \quad (3.11.6)$$

where C/H is the carbon to hydrogen ratio by mass.

Internal Convection The heat transferred by convection from the hot gas to the liner wall is given by (Lefebvre & Herbert, 1960)

$$C_1 = 0.020 \frac{\lambda_g}{D_{h,1}^{0.2}} \left(\frac{\dot{m}_g}{A_{liner} \eta_g} \right)^{0.8} (T_g - T_{w_1}) \quad (3.11.7)$$

where T_{an} = the annulus temperature, $\approx T_3$
 T_{w_1} = the inner surface temperature of the liner wall
 λ_g = the coefficient of thermal conductivity for the hot gases at T_g

- η_g = the dynamic viscosity for the hot gases at T_g
 $\dot{m}_g$ = the mass flow rate of gas through the liner
 A_{liner} = the liner cross-sectional flow area
 $D_{h,1}$ = the hydraulic diameter of the liner = D_{liner}

External Radiation The heat transferred by radiation from the liner wall to the casing (Lefebvre & Herbert, 1960)

$$R_2 = \sigma \frac{\varepsilon_{liner} \varepsilon_{casing}}{\varepsilon_{casing} + \varepsilon_{liner}(1 - \varepsilon_{casing})} (T_{w_2}^4 - T_{an}^4) \quad (3.11.8)$$

- where σ = the Stefan-Boltzmann constant = $5.67 \times 10^{-8} W/(m^2 \cdot K^4)$
 T_{an} = the annulus temperature, $\approx T_3$
 T_{w_2} = the outer surface temperature of the liner wall
 ε_{liner} = the liner emissivity
 ε_{casing} = the casing emissivity

External Convection The heat transferred by convection from the liner wall to the annulus air is given by (Lefebvre & Herbert, 1960)

$$C_2 = 0.020 \frac{\lambda_a}{D_{an}^{0.2}} \left(\frac{\dot{m}_{an}}{A_{an} \eta_a} \right)^{0.8} (T_{w_1} - T_{an}) \quad (3.11.9)$$

- where T_{an} = the annulus temperature, $\approx T_3$
 T_{w_2} = the outer surface temperature of the liner wall
 λ_a = the coefficient of thermal conductivity for air at T_{an}
 η_a = the dynamic viscosity for air at T_{an}
 $\dot{m}_{an}$ = the mass flow rate of air through the annulus
 A_{an} = the annulus cross-sectional flow area
 D_{an} = the annulus hydraulic diameter = $D_{ref} - D_{liner}$

Conduction The heat conducted through the liner wall

$$K_{1-2} = \frac{\lambda_{liner}}{t_{liner}} (T_{w_1} - T_{w_2}) \quad (3.11.10)$$

Figure 3.21: Trip strip dimensions.

- where σ = the Stefan-Boltzmann constant = $5.67 \times 10^{-8} \text{W}/(\text{m}^2 \cdot \text{K}^4)$
- T_{w1} = the inner surface temperature of the liner wall
- T_{w2} = the outer surface temperature of the liner wall
- t_{liner} = the liner wall thickness
- λ_{liner} = the liner wall coefficient of thermal conductivity

Equations 3.11.3 through 3.11.10 yield a set of 7 equations with 7 unknowns and was solved numerically using the iterative Secant method. When solved along the entire liner surface, a one dimensional temperature profile is achieved. The most important temperature is that of the inner wall, T_{w1} , since this value is greatest.

It should be emphasized that the wall temperatures computed are mean values only. Geometric discontinuities can be expected to produce hot streaks or peaks in the wall temperature that exceed the mean value computed above.

3.11.2 Trip Strips

The use of traverse ribs or trip strips is modeled by modifying the external convection term C_2 . Evans & Noble (1978) provided an analytical method for designing trip strips. These trip strips are illustrated by Figure 3.21.

For a duct of hydraulic diameter D_{an} and ribs of height e , the friction factor is given by

$$f_R = \frac{0.25}{\left[2 \log_{10} \frac{1}{(18e/D_{an})} + 1.74\right]} \quad (3.11.11)$$

where e/D_{an} is called the relative roughness.

The heat transfer coefficient is derived from the Nusselt number. It is determined using the following correlation:

$$Nu_{D,f} = 0.050(Re_{D,f})^{0.8}(Pr_f)^{0.4} \left(\frac{T_b}{T_f}\right)^{0.8} \quad (3.11.12)$$

where T_f and T_b denote the film and bulk temperature. Both the Prandtl and Reynolds number are evaluated at the film temperature, which is defined as

$$T_f = \frac{T_{an} + T_{w2}}{2} \quad (3.11.13)$$

The Reynolds number based on the hydraulic diameter

$$Re_{D,f} = \frac{\rho_f \left(\frac{\dot{m}_{an}}{\rho_a A_{an}}\right) D_{an}}{\eta_f} \quad (3.11.14)$$

where ρ_f and η_f are the density and viscosity of air evaluated at the film temperature.

The correlation for the Nusselt number, Equation 3.11.12, was generated with data for a relative roughness of 0.0275. To correct for other values of e/D_{an} , the following relationship is used:

$$Nu \propto f_R^{0.63} \quad (3.11.15)$$

The heat transfer coefficient h is thus

$$Nu = \frac{h D_{an}}{\lambda_f} \quad (3.11.16)$$

where the thermal conductivity for air, λ_f , is also evaluated at the film temperature.

Finally, the convective heat flux

$$C_2 = h(T_{w2} - T_{an}) \quad (3.11.17)$$

Figure 3.22: Thermal resistance induced by TBC.

3.11.3 Thermal Barrier Coating

As depicted by Figure 3.22, a TBC adds thermal resistance to the liner wall and effectively reduces the temperature "felt" by the liner. It is accounted for in the heat transfer model by modifying the conduction and internal radiation term.

Conduction Term With a TBC, the heat conducted through the liner wall becomes

$$K_{1-2} = U (T_{w_1} - T_{w_2}) \quad (3.11.18)$$

where the overall heat transfer coefficient is

$$U = \frac{1}{t_{liner}/\lambda_{liner} + t_{TBC}/\lambda_{TBC}} \quad (3.11.19)$$

Here the subscript 1 denotes the hot-side surface of the TBC and the subscript 2 denotes the cold-side surface of the liner wall.

The metal temperature at the interface between the liner wall and TBC, T_i , can be computed once the inner and outer surface temperatures are known using

$$K_{i-2} = \frac{\lambda_{liner}}{t_{liner}} (T_i - T_{w_2}) \quad (3.11.20)$$

or

$$K_{1-i} = \frac{\lambda_{TBC}}{t_{TBC}} (T_{w_1} - T_i) \quad (3.11.21)$$

Internal Radiation Term Equation 3.11.4 is still used to compute the internal radiation, only the emissivity of the wall is equal to that of the TBC material and not that of the liner.

$$\varepsilon_{liner} = \varepsilon_{TBC} \text{ for the inner wall} \quad (3.11.22)$$

3.12 Structural Considerations

The mechanical integrity of the combustor must be verified to ensure reliable operation. The two components subjected to the most severe loads are the casing and liner, which are essentially cylinders under high pressure. Of the two, the liner is the most critical as it is also subjected to a severe environment and high thermal load. The thermal loading creates steep temperature gradients that cause the liner to deform and expand.

3.12.1 Stress Estimate

A simple structural analysis is performed on both the casing and liner using the procedures discussed in Benham *et al.* (1996) for thin cylinders ($\frac{D}{t} \gg 10$). For this analysis, the liner is modeled with open ends since it has large holes cut out for the inlet and outlet. The casing, however, is modeled using closed ends as it is sealed from the surrounding atmosphere. The two cases are depicted in Figure 3.23.

For both open and closed thin cylinders, the hoop or circumferential component of stress resulting from the applied pressure differential may be estimated from

$$\sigma_{\theta} = \frac{\Delta P D}{2t} \quad (3.12.1)$$

where ΔP = the applied pressure differential, $= (P_i - P_o)$

D = the inner diameter

t = the cylinder wall thickness

The axial stress component due to the pressure differential varies between the two

Figure 3.23: Thin cylinder models used for structural analysis.

types.

$$\sigma_a = \begin{cases} \frac{\Delta P D}{4t}, & \text{closed} \\ 0, & \text{open} \end{cases} \quad (3.12.2)$$

Equations 3.12.1 and 3.12.2 are evaluated under the following assumptions: the static pressure in the combustor equals the total pressure, and the static pressure along the walls remains constant. Thus, $\Delta P \approx \Delta P_{liner}$ for the liner and $\Delta P \approx (P_3 - P_{amb})$ for the casing. These equations should also be evaluated where the diameter is largest (D_{ref} for the casing and D_{liner} for the liner) since the stress is highest in these locations.

These approximations are crude since the flow in the annulus and liner is generally not stationary. Also, it is known that the velocity of the flow varies greatly as it flows through the combustor. A more accurate estimate of the static pressure distribution along the liner and casing walls would require the use of analytical techniques such as CFD.

3.12.2 Thermal Expansion

A material expands as it heats up with a change in length that is directly proportional to the temperature change. Of the combustor components, the growth of the liner is the most significant. The thermal expansion of the liner is estimated using the following relation for thermal strain:

$$\varepsilon_T = \alpha(T) (T - T_0) \quad (3.12.3)$$

where the strain in the radial and axial direction is

$$\varepsilon_r = \frac{\delta_r}{R}$$

$$\varepsilon_a = \frac{\delta_a}{L}$$

This expansion is not accompanied by a stress since the liner is free to move in either the radial or axial direction. Values of $\alpha(T)$ for various materials are tabulated in Appendix E.

Equation 3.12.3 assumes that the temperature distribution throughout the liner walls is constant. In practice, the temperature profile of the wall will vary greatly in both the axial and radial directions. The resulting localized temperature gradients which exist in the liner walls cause localized regions of the liner to expand and contract. Buckling of the liner walls will occur if the combination of thermal stresses, caused by these localized expansions and contractions, and the stresses due to the applied pressure become too severe.

Accurate estimation of the stress in the liner wall requires the inclusion of three-dimensional thermal effects. This can be performed with complex numerical models that combine FEM with CFD.

3.12.3 Failure Analysis

3.12.3.1 Yielding

Many materials become very ductile when subjected to the high temperatures typical of combustors. When loaded statically, these materials are prone to failure by distortion, or plastic deformation. The onset of this plastic deformation is called yielding. Both the liner and the casing must be designed to ensure that yielding does not occur under normal operating conditions.

The von Mises yield criterion is used to predict the failure of ductile materials. The criterion states that the onset of failure occurs when

$$\sigma_e = \sigma_Y \quad (3.12.4)$$

where σ_Y is the material yield strength. Values for the yield strength of various materials are given in Appendix E.

The equivalent stress, σ_e , in a two-dimensional system is

$$\sigma_e = \sqrt{\sigma_1^2 + \sigma_2^2 - \sigma_1\sigma_2} \quad (3.12.5)$$

where $\sigma_1 = \sigma_\theta$ and $\sigma_2 = \sigma_a$ for thin cylinders with no additional thermal stresses.

Figure 3.24: Creep deformation as a function of time.

3.12.3.2 Creep and Stress Rupture

One of the most common failure modes in the liner is creep distortion. Creep is the progressive deformation of a material at a constant stress, which occurs when the material is subjected to load under high temperatures (Figure 3.24). Over time, the reduction in cross-sectional area becomes significant enough that the rate of elongation begins to increase exponentially prior to failure. The time to failure by creep is called the stress rupture life.

The designer must ensure that the liner does not creep excessively and that failure does not occur by this mode. The data from creep and stress rupture tests for various metals can be found in Appendix E.

3.13 Summary

The procedures discussed in this chapter form a methodology for the preliminary design of a LPP combustor. Implementation of these procedures will be discussed in the following chapter.

Chapter 4

Preliminary Design

The preliminary design procedures outlined in the previous chapter were implemented in a FORTRAN program and applied to a marine gas turbine engine. The application necessitated many decisions made based on the engine specifications and requirements. The geometry and the results of the preliminary analysis output from the program are discussed in this chapter.

The engine is a 1-MW marine gas turbine that uses a twin-shaft configuration, as shown in Figure 4.1. It consists of a single-stage centrifugal compressor with a pressure ratio of 8 and two axial turbines. The first turbine drives the compressor on one shaft, called the gas generator, and the second powers the drive shaft. The temperature rise necessary for the first turbine is accomplished by a single can combustor operating on diesel fuel No. 1.

4.1 Engine Specifications

The engine specifications identify important aspects that govern the design of individual components. The specifications include:

1. low manufacturing cost
2. compact
3. safe
4. durable (2000 hours or 500 cycles between major overhaul)
5. maintainable

Figure 4.1: 1-MW marine gas turbine schematic.

6. operates on No. 1 diesel fuel
7. acceptable efficiency at full load (0.55 lbm / Hp·hr)
8. acceptable efficiency at part load
9. emissions meet applicable requirements
10. potential use in power generation

The engine design point occurs at full load or 100 percent throttle. It is fixed by Table 4.1.

4.2 Engine Cycle Calculations

Steady-state cycle calculations were performed to obtain the combustor inlet and outlet conditions at both full load and idle. These calculations were performed using GasTurb 9, which is a commercially available gas turbine cycle program for simulating engines.

The engine was modeled using GasTurb's two-spool turboshaft engine configuration. The design point was fixed with the inputs listed in Table 4.2 while the idle point required the use of GasTurb's off-design capabilities.

GasTurb's off-design model uses generic compressor and turbine maps to predict

Table 4.1: Engine design point.

Setting	Value
Ambient Temperature	288 K
Ambient Pressure	101.325 kPa
Inlet Flow Rate	5.5 kg/s
Pressure Loss Across Inlet	4 inches of H ₂ O
Compressor Polytropic Efficiency	85%
Compressor Pressure Ratio	8
Bleed From Compressor	1.5% of mass flow
Burner Efficiency	99%
Maximum Pressure Loss Across Burner	5% of burner inlet
Turbine Inlet Temperature (TIT)	1200 K
Turbine Polytropic Efficiency	88% each
Mechanical Efficiency	99%
Pressure Loss Across the Exhaust	8 inches of H ₂ O
Gearbox Efficiency	95%

Table 4.2: GasTurb Settings

Setting	Value
Total Temperature T1	288 K
Total Pressure P1	101.3 kPa
Relative Humidity	0 %
Inlet Corrected Flow Rate	5.5 kg/s
Intake Pressure Ratio	0.99
Pressure Ratio	8
Burner Exit Temperature	1200 K
Burner Design Efficiency	99.7 %
Burner Part-load Constant	1.6 (default value)
Fuel Heating Value	42.7392 MJ/kg
Water-to-fuel Ratio	0
Steam-to-fuel Ratio	0
Overboard Bleed	0.0825
Power Offtake	5 kW
HP Spool Mechanical Efficiency	94 %
Burner Pressure Ratio	0.95
Turbine Interduct Pressure Ratio	1
Turbine Exit Duct Pressure Ratio	1
Nozzle Pressure Ratio	1.0001
Nozzle Thrust Coefficient	0
LP Spool Mechanical Efficiency	99 %
Nominal PT Spool Speed	0 RPM
Compressor Polytropic Efficiency	85 %
HPT Polytropic Efficiency	88 %
PT Polytropic Efficiency	88 %

Table 4.3: Engine operating parameters.

Station	Full Load			Idle		
	P (kPa)	T (K)	$\dot{m}$ (kg/s)	P (kPa)	T (K)	$\dot{m}$ (kg/s)
1	101.3	288	5.5	101.3	288	2.2
2	100.3	288	5.5	100.3	288	2.2
3	802.3	574	5.4	260.4	414	2.1
4	762.2	1200	5.5	246.4	790	2.2
4.5	244.4	941	5.5	111.4	666	2.2
5	101.3	774	5.5	101.3	660	2.2
6	101.3	774	5.5	101.3	660	2.2

engine performance at low power settings. A particular operating point is found by setting limiters that restrict the maximum permissible power output and iterating over the cycle until these parameters reach their maximum value. In this case, idle was found by limiting the TIT until the net power output by the cycle was closest to zero.

The results from the GasTurb analysis are tabulated below. Table 4.3 displays the temperature and pressure at each location in the engine while Table 4.4 lists the fuel flow rate required by the engine.

4.3 Design Choices

Many decisions were made to perform the preliminary design. These decisions were based on the engine specifications, existing designs, and recommendations published in the literature.

Combustion Efficiency

It was reasonable to expect combustion efficiencies greater than 99 percent. Thus,

$$\eta_c = 99.7\% \quad (4.3.1)$$

Table 4.4: Required fuel flow rate.

	$\dot{m}_f$ (kg/s)
Design	0.09421
Idle	0.02131

Pressure Loss Coefficient

The combustor was designed to utilize the maximum permissible pressure loss,

$$\frac{\Delta P_{3-4}}{P_3} = 5\% \quad (4.3.2)$$

Pressure Loss Factor

A slightly higher pressure loss factor than typical aircraft can combustors was specified. The combustor flow pattern was expected to impart more resistance in the flow stream since the annulus air must make a complete 180° turn before it enters the pre-mixer.

$$\frac{\Delta P_{3-4}}{q_3} = 40 \quad (4.3.3)$$

Inlet and Outlet Area

The combustor inlet and outlet are sized to match the compressor exit and turbine inlet, respectively.

$$A_3 = 0.0093 \text{ m}^2 \quad (4.3.4)$$

$$A_4 = 0.0161 \text{ m}^2 \quad (4.3.5)$$

Fuel System

The injectors must be capable of providing extremely small droplets. Droplets any larger than 20 microns lead to excessive evaporation times and result in a lengthy pre-mixer. Thus,

$$SMD = 20 \mu \text{ m} \quad (4.3.6)$$

High fuel line pressures are required for nozzles to produce such small droplets. The high pressures will also reduce the number of pressure nozzles required to supply the desired mass flow rate. A pressure differential across the nozzle of 500 psi was chosen.

$$\Delta P_f = 3.4 \text{ MPa} \quad (4.3.7)$$

At high pressures, nozzles typically spray with wide cone angles.

$$\alpha_n = 90^\circ \quad (4.3.8)$$

Swirlers

The hub at the centre of the two co-axial swirlers was sized so that a fuel nozzle may be mounted inside.

$$D_{hub,1} = 20.0 \text{ mm} \quad (4.3.9)$$

A slightly larger value was used for the combustor swirler hub.

$$D_{hub,2} = 50.0 \text{ mm} \quad (4.3.10)$$

All swirlers are made up of flat vanes with a thickness of 1.5 mm.

$$t_v = 1.5 \text{ mm} \quad (4.3.11)$$

Liner to Casing Area Ratio

The liner-to-casing area ratio was selected based on consideration of the annulus flow and liner heat transfer.

$$k_{area} = \frac{A_{liner}}{A_{ref}} = 0.85 \quad (4.3.12)$$

Liner Wall

The liner is to be fabricated from the nickel-based alloy Hastelloy X. The emissivity (Omega, 1998) and conductivity: (Haynes International, 2005) for this material are:

$$\varepsilon_{liner} = 0.85 \quad (4.3.13)$$

$$\lambda_{liner} = 26 \text{ W/m K} \quad (4.3.14)$$

Dodds & Bahr (1990) recommend that the maximum operating temperature of this material not exceed 1150 K. Since the time between overhauls is only 2000 hours, a slightly higher limit of 1200 K was chosen.

The liner wall should be kept thin to minimize potential crack forming temperature gradients along the thickness direction. However, buckling will occur if the liner is too thin. Thus, the liner thickness was initially selected to be 3.0 mm.

$$t_{liner} = 3.0 \text{ mm} \quad (4.3.15)$$

Casing Wall

The casing was fabricated from stainless steel. The emissivity, (Holman, 1997)

$$\varepsilon_c = 0.6 \quad (4.3.16)$$

The thickness of the casing walls was chosen to ensure both formability and strength.

$$t_c = 3.175 \text{ mm} \quad (4.3.17)$$

TBC

The TBC selected was YSZ. Lefebvre (1999) states that the thermal conductivity of this ceramic is typically

$$\lambda_{TBC} = 2.6 \text{ W/m K} \quad (4.3.18)$$

The emissivity of ceramic zirconia is (Omega, 1998)

$$\lambda_{TBC} = 0.6 \quad (4.3.19)$$

Coating thicknesses range from 0.1-0.5 mm (Padture *et al.* , 2002). A thickness closer to the upper limit was selected to provide the maximum protection.

$$t_{TBC} = 0.5 \text{ mm} \quad (4.3.20)$$

Table 4.5: Airflow distribution.

Location	Mass Flow Fraction	Equivalence Ratio
Annulus	0.46	
Premixer	0.46	
Dilution Holes	0.54	
Primary Zone	0.46	0.54
Dilution Zone	1.00	0.25

Cooling

Trip strips were used to augment the rate at which the annulus air removes heat from the liner wall. Evans & Noble (1978) experimented with different configurations and stated that the optimum one had a spacing of $p/e=10$.

$$p = 26.0 \text{ mm} \quad (4.3.21)$$

$$e = 2.6 \text{ mm} \quad (4.3.22)$$

4.4 Results

The results from the preliminary design have been tabulated in Tables 4.5 and 4.6. The overall dimensions are documented in Figure 4.2, 4.3, and 4.4. Additionally, the preliminary design determined that a minimum ignition energy of 14.50 mJ and fuel pump power of 1 kW is required.

Preliminary design involved modeling combustion performance, liner heat transfer, droplet evaporation, and dilution jet mixing. The results from these models, along with those from the preliminary structural analysis, are presented here.

4.4.1 Combustion Performance

The AEDsys program KINETX (Mattingly *et al.*, 2002) was used to model the combustor with the reactor network of Figure 3.16. The program required specification of

Figure 4.2: Combustor dimensions.

Figure 4.3: Premixer dimensions.

SWIRLER	A	B	C	D	E	F
1	62	20	-30	1.5	21	12
2	155	62	+60	1.5	47	8
3	144	50	+60	1.5	47	10

Figure 4.4: Swirler dimensions.

Table 4.6: Pressure losses.

Location	Pressure Loss, $\frac{\Delta P}{P_3}$ (%)
Prediffuser	0.61
Sudden Expansion	0.84
Plenum	0.03
Mixer Swirlers	1.00
Combustor Swirler	2.52
Liner	3.55
Complete Combustor (Inlet to Outlet)	5.0

the area and length of each zone along with the mass flow rate of air and fuel. These values were obtained from the design specifications and the results of the preliminary design. The inputs used are depicted in Figure 4.5.

The results from KINETX are listed in Table 4.7. They indicate that the airflow distribution in the PZ and SZ have been chosen correctly. Both the PZ temperature, representing the flame temperature, and the DZ temperature, representing the TIT, match the design specifications above. They also indicate that the required combustion efficiency of 99.7 percent has been met, suggesting that sufficient residence time for combustion has been provided.

This adequate residence time is confirmed by the low emissions predicted: 3 ppmv CO, 12 ppmv NO, and 3 ppmv NO₂. It shows that the residence time was large enough to allow CO to almost completely react into CO₂ yet low enough to hinder the formation of excessive NO_x. The actual emissions are expected to be higher than predicted due to individual droplet burning and unmixedness. Nonetheless, these emissions are well below the 30 ppm of NO_x and 10 ppm of CO emitted by the OPRA OP16 gas turbine when operating on diesel fuel (Klein *et al.*, 2002).

The KINETX model was also used to analyze the primary zone stability. A stability loop, similar to that illustrated in Figure 2.14, was created with the program by varying the air/fuel ratio and plotting the equivalence ratio vs the combustor

Figure 4.5: Schematic of KINETX model.

Table 4.7: Results from KINETX.

Parameter	Value
T_{PZ}	1803
T_{DZ}	1203
η_c	99.71 %
CO	3 ppmv ¹
NO	12 ppmv ¹
NO ₂	3 ppmv ¹

¹Dry corrected to 15% O₂

Figure 4.6: Predicted combustor stability loop.

loading. The resulting plot is shown in Figure 4.6. The combustor loading at the design point is several orders of magnitude from the blowout value, confirming that the fuel/air mixture has ample residence time and that the combustor should operate stably at the design point.

Figure 4.6 also confirms that a pilot will be required to improve the off-design stability as reducing the engine output will lower the primary zone equivalence ratio. Since the design point is already close to the lean limit, lowering the equivalence ratio by a significant amount will result in operation outside the stable region of the loop. A pilot injector is not necessary, however, as the individual diffusion flames surrounding droplets that escaped the premixer un-evaporated are expected to provide a source of ignition.

4.4.2 Liner Wall Temperature

Figure 4.7 shows the temperature profile along the liner wall at full load predicted by the preliminary heat transfer analysis. A peak metal temperature in the liner wall of 1237 K is observed just upstream of the dilution holes. The figure also depicts a large

Figure 4.7: Predicted liner wall temperature profile at design.

discontinuity in the temperature profile across the liner admission holes, resulting from the step change in gas temperature between the PZ and DZ. It indicates that the conduction of heat along the liner wall cannot be neglected and must be included in the heat transfer model. Nonetheless, the analysis provides a good first estimate of the liner wall temperature.

4.4.3 Droplet Evaporation

The mean droplet trajectories and size distributions are summarized in Figure 4.8 and 4.9. It is evident that the operating conditions have a significant effect on the droplet lifetime and penetration.

The evaporation of a 20 micron droplet requires significantly more time at idle when compared to design. The mixer tube length is therefore based on the design point to avoid excessive combustor length. This allows unevaporated droplets to escape the premixer at low power settings and provide a pilot diffusion flame.

Figure 4.8: Predicted mean trajectory for 20 micron droplets.

Figure 4.9: Predicted droplet diameter distribution.

Figure 4.10: Predicted dilution jet trajectory.

4.4.4 Jet Mixing

The mixing of the dilution jets with the combustion products was investigated. Figure 4.10 illustrates the trajectory of the jet centerline Y_c and the location its half-widths, $Y_c - W_{1/2}^-$ and $Y_c + W_{1/2}^+$. It is evident that the jet penetrates steeply into the flow and spreads quickly.

4.4.5 Structural Analysis

Failure of the liner or casing during operation must not occur within the desired service life. Thus, a simple structural analysis was performed to help assess the probability of failure. Both the liner and casing were analyzed for possible yielding while the stress rupture life of the liner was estimated as well. Additionally, the thermal expansion of the liner was estimated to aid mechanical design.

The resulting stresses due to the internal pressure in the liner and casing are given in Table 4.8. The equivalent stresses in both components are well below the yield strength, suggesting that plastic deformation is not expected to occur.

Table 4.8: Estimates for stress in liner and casing.

	Liner	Casing
Hoop, MPa	-2.27	+ 57.87
Axial, MPa	+0.00	+ 28.94
Equivalent, MPa	2.27	50.12
Yield, MPa	55 ¹	214 ²

Tensile (+), Compressive (-)

¹Sheet at 1366 K

²Sheet at 589 K

SOURCE: Material properties from Appendix E

The stress rupture data for Hastelloy X is plotted in Figure 4.11 and compared to the liner life requirements. The liner must endure an applied stress of 2.27 MPa for 2000 hours at a maximum metal temperature of 1237 K. Based on the data presented in Figure 4.11, the liner is expected to have a life greater than 10,000 hours, exceeding the requirements.

Table 4.9 lists the thermal expansion estimates for the liner in both the radial and axial directions. The analysis predicts a change in length of 7.93 mm and 8.73 mm in the diameter and length, respectively. This growth must be considered during the mechanical design to ensure that the liner can grow unrestricted.

4.5 Summary

The results from the preliminary design indicate that the geometry should satisfy the mechanical, aerodynamic, and thermodynamic requirements. A detailed design, which includes verification with numerical techniques is performed prior to mechanical design. The topic of verification is covered in the ensuing chapter.

Figure 4.11: Stress rupture life for Hastelloy X. From Haynes International (2005)

Table 4.9: Estimates for liner thermal growth at 1237 K.

	δ mm
Diameter, D_{liner}	7.93
Length, L_{tot}	8.73

Chapter 5

Verification of Preliminary Design

To assess the success of the preliminary design procedure, the aerodynamic and structural performance of key components was verified using computational fluid dynamics (CFD) and finite element analysis (FEA), respectively. CFD refers to the application of any numerical technique to fluid flow whereas FEA is a specific numerical technique used for solving complex stress states.

The preliminary design procedure formed in Section 3 was verified with the following:

- CFD simulation of the combustor,
- CFD simulation of the premixer, and
- FEA of the liner.

5.1 Combustor CFD Analysis

CFX-5, a CFD software package, was used to analyze the combustor flowfield at the design point. The goal was to capture the heat released by the swirling flowfield inside the liner and the dilution of the hot combustion products.

The analysis was performed using the procedures outlined in the product documentation for CFX-5. The reader should consult this documentation for information on all models and settings that were used.

This section outlines the steps taken to perform the CFD simulation and discusses its results. The steps include:

- generating a modeling domain and grid,
- setting up the domain,
- specifying the boundary conditions,
- setting up the solver, and
- solving the domain.

5.1.1 Geometry and Grid Generation

A simplified three-dimensional solid model has been built and used to generate the computational grid.

Solid Model The internal flowpath of the combustor liner was modeled using several simplifications. These include:

- The flow inside the liner is uncoupled from the flow in the annulus. Only the inside of the liner was modeled due to the complexity of the combustor flowpath and limitations of computational resources.
- The main inlet is placed downstream of the combustor swirler. It is reasonable to assume that the flow exiting this swirler is uniform and that it follows the blades.
- The problem is axisymmetric. A 36° section was modeled using the periodic boundary condition to reduce the grid size and computation time. The section angle was chosen to ensure a whole number of both swirler vanes and dilution holes inside the domain.

The resulting solid model is shown in Figure 5.1. It was constructed using ANSYS DesignModeler from the dimensions shown in Figure 5.2. These dimensions were obtained from the results of the preliminary design.

Grid Generation An unstructured grid was generated with ANSYS CFX-MESH, illustrated in Figure 5.3, which consists of 150,000 nodes and 800,000 elements. The nodal density of the mesh was selected by studying its effects on the overall solution

Figure 5.1: Solid model of combustor flow domain.

Figure 5.2: Dimensions of combustor flow domain.

Figure 5.3: Combustor computational mesh.

and choosing one whose solution was grid independent. This study is discussed later in this section.

The density of the grid in the highlighted areas of Figure 5.3 was doubled to improve the resolution of the steep velocity and temperature gradients expected. Closeups of these areas are provided in Figure 5.4.

5.1.2 Combustor Domain

Setting up the combustor fluid domain required specification of the reaction, the fluid properties, and the fluid models. These are discussed below.

Reactions The air and fuel within the internal combustor flowfield was modeled as a reacting, variable composition mixture. The two-step reaction of Westbrook & Dryer (1981) for decane ($C_{10}H_{22}$) was used to compute the rate of fuel oxidation.

Figure 5.4: Closeup of refined areas in the combustor mesh.

The rate constants for each reaction are listed in Table 5.1.

The two-step Zeldovich mechanism was added to compute the mass fractions of thermal NO formed in the combustion process. CFX-5 solves these additional kinetic equations for NO without affecting the main combustion calculation. Thus, the effects of NO on the global flow and combustion are assumed negligible.

Fluid Properties A total of 7 species were modeled. These include: CO, CO₂, H₂O, O₂, NO, N₂, and diesel fuel vapour. Their properties were specified using CFX-5's predefined material library. Diesel fuel was modeled as Jet A since it was not included in the library.

Fluid Models Appropriate fluid models were chosen based on a review of published literature and recommendations in CFX-5's documentation. They are as follows:

- **Turbulence Model:** The $k-\epsilon$ turbulence model was chosen based on the work of Rizk & Mongia (1991). This two-equation model has a well established regime of predictive capability while compromising between accuracy and robustness (ANSYS Canada, 2004).
- **Combustion Model:** The reaction rates were computed using the combined Eddy Dissipation (ED) / Finite Rate Chemistry (FRC) model. It compares the reaction rate based on kinetics (FRC) and turbulent mixing (ED), selecting the

Table 5.1: Reaction rate constants. Units cm-sec-mole-kcal-Kelvins (Westbrook & Dryer, 1981)

$$\frac{d[A]}{dt} = k \exp(-E/RT) \times [A]^n [B]^m [C]^p$$

Reaction	[A]	[B]	[C]	k	E	n	m	p
5.1.1	C ₁₀ H ₂₂	O ₂	N/A	4.7×10^{11}	30.0	0.25	1.5	0
5.1.2	CO	H ₂ O	O ₂	$10^{14.6}$	40.0	1	0.5	0.25

minimum of the two. This model was chosen because combustion is expected to be largely kinetics controlled in the primary zone and mixing controlled in the dilution zone.

- Heat Transfer Model: The Total Energy heat transfer model was chosen to include the effects of kinetic energy.
- Radiation Model: Radiation was modeled using the Differential Approximation or P1. This relatively simple model was chosen because radiation in premixed combustion is not as significant when compared to diffusion combustion, where peak flame temperatures are much higher.

5.1.3 Boundary Conditions

Solution of the computational domain requires knowledge of the boundary conditions. The boundary conditions used were those corresponding to the engine design point and are provided below.

Inlet A specified mass flow rate boundary condition was used for both inlets. The total mass flow rate and the individual mass fractions of each species at design were estimated using the results obtained in Chapter 4. Table 5.2 summarizes the inlet boundary conditions.

Table 5.2: Inlet boundary conditions.

Condition	Value	
	Primary	Secondary
Flow Regime	Subsonic	Subsonic
Mass Flow Rate	0.25 kg/s	0.29 kg/s
Flow Direction	Cylindrical Components <ul style="list-style-type: none"> • r Component = 0 • Theta Component = 0.866 • Axial Component = 0.5 	Normal to Boundary
Turbulence	High (Intensity = 10%)	High (Intensity = 10%)
Total Temperature	575 K	575 K
Thermal Radiation	Local Temperature	Local Temperature
CO Mass Fraction	0.0	0.0
CO ₂ Mass Fraction	0.0	0.0
H ₂ O Mass Fraction	0.0	0.0
Fuel Mass Fraction	0.038	0.0
NO Mass Fraction	0.0	0.0
O ₂ Mass Fraction	0.223	0.232
N ₂ Mass Fraction	0.739	0.768

Table 5.3: Outlet boundary conditions.

Condition	Value
Flow Regime	Subsonic
Average Static Pressure	732 kPa
Thermal Radiation	Local Temperature

Outlet The average static pressure was set to match the inlet total pressure with that predicted by the preliminary design. The details of the outlet boundary condition are provided in Table 5.3.

Combustor Walls Wall boundary conditions were placed on both the swirler hub and the liner wall. The swirler hub was modeled as an adiabatic wall whereas the liner was modeled by specifying the overall heat transfer coefficient. Their details are listed in Tables 5.4 and 5.5.

The heat transfer coefficient allows one to model the thermal resistance outside the computational domain, as illustrated in Figure 5.5. In this case, it accounts for the effects of ABC cooling and the TBC. Thus,

$$U = \frac{1}{1/h + t_{liner}/\lambda_{liner} + t_{TBC}/\lambda_{TBC}} \quad (5.1.3)$$

where h is the heat transfer coefficient defined in Section 3.11.2.

Average values for U were applied to the liner wall by computing them at three locations using the procedures outlined in Chapter 3 and linearly blending them together. The values applied are shown in Figure 5.6.

5.1.4 Results

The combustor flow domain was solved using the high resolution advection scheme. This scheme blends between first and second order accuracy, providing a compromise between robustness and accuracy. The results from the analysis are discussed below.

Table 5.4: Wall boundary conditions for swirler hub.

Condition	Value
Wall Influence On Flow	No Slip
Wall Roughness	Smooth Wall
Heat Transfer	Adiabatic
Emissivity	0.8
Diffuse Fraction	1

Table 5.5: Boundary conditions for liner wall.

Condition	Value
Wall Influence On Flow	No Slip
Wall Roughness	Smooth Wall
Heat Transfer Coefficient	See Figure 5.6
Outside Temperature	575 K
Emissivity ¹	0.6
Diffuse Fraction	1

¹Wall emissivity for Zirconia ceramic on Inconel (Omega, 1998)

Figure 5.5: Definition of overall heat transfer coefficient.

Figure 5.6: Distribution of liner wall heat transfer coefficient.

Figure 5.7: Results from grid independence study for combustor CFD.

5.1.4.1 Grid Independence Study

The combustor was first analyzed using several grids of varying nodal density to ascertain the resolution required to achieve grid independence. This was accomplished by comparing the solution from four meshes.

The change in both the outlet NO and CO mass fractions with the mesh size are plotted in Figure 5.7. It is evident that a larger grid than the ones investigated is required. Despite the results of the study, the second largest mesh with 147,000 nodes was chosen to avoid excessively long simulations. The largest mesh, 314,000 nodes, required 48 hours of computation for only 400 iterations using two parallel processes on a dual XEON processor machine.

5.1.4.2 Convergence

Solution convergence was monitored using the residuals of the governing equations. The solution was said to have converged when all the residuals reached their minimum values and the solution appeared to no longer change with each iteration. The residuals achieved are listed in Table 5.6.

Table 5.6: RMS and MAX residuals for combustor CFD analysis.

Residual	RMS Value	MAX Value
Mass	1E-06	2E-04
Momentum	1E-06	2E-04
Energy	1E-05	2E-03
Radiation	3E-08	2E-06
Eddy Dissipation	1E-06	2E-04
Turbulent Kinetic Energy	2E-07	4E-05
Species Mass Fraction	1E-04	1E-02
Temperature Variance	1E-06	1E-04

5.1.4.3 Velocity Flowfield

The velocity inside the liner was plotted to verify that a strong swirling flow exists. Figure 5.8 depicts a large torroidal recirculation zone at the centre of the combustor. The type of recirculation zone observed is typical of strongly swirling flows where the induced axial pressure gradient is strong enough to force the flow to stick to the outer walls (Figure 5.9).

A second recirculation zone is observed in the aft portion of the liner, downstream of the dilution holes. This recirculation incurs additional pressure losses that are considered parasitic as they do not contribute to the overall combustion process. It is an unavoidable result of the sudden contraction at the exit of the liner.

Figure 5.8 also provides illustration of the mixing between the dilution jets and the main flow. They do not appear to reach the centreline as predicted in the preliminary design (Figure 4.10). This is because the predictions were based on flow through a constant area duct, neglecting the sudden contraction just downstream of the injection point. Nonetheless, the performance of the jets is satisfactory as the combustor outlet profile is not of importance in this application.

Figure 5.8: Velocity flowfield inside combustor.

Figure 5.9: Flow form observed for a strongly swirling air jet issuing from a divergent nozzle. From Beer & Chigier (1972).

5.1.4.4 Gas Temperature Distribution

Table 5.7 compares the maximum flame temperature and the TIT predicted in the preliminary design to that predicted in the CFD analysis. While good agreement between the predicted TITs was observed, CFD computed a much higher flame temperature.

Figure 5.10 depicts the temperature distribution inside the combustor. The figure shows the reaction progressing along the dome walls and reaching the peak flame temperature of 1885 K. As the flow recirculates, it entrains some of the dilution air and its temperature drops below the peak value. The consequent temperature distribution upstream of the dilution holes is one that is hotter near the liner walls and slightly cooler at the centreline.

Downstream of the dilution jets, the temperature of the flow drops almost immediately. This was expected since a large number of jets were used that create a blockage in the the hot combustion product flowpath. The blockage forces the flow to mix with the dilution jets, producing a large drop in gas temperature near the combustor walls.

The temperature distribution at the exit of the liner was plotted in Figure 5.11. A very large peak occurs at the centerline. This indicates that the length of the dilution zone is insufficient in providing enough time for the jets to mix with the combustion

Table 5.7: Comparison of predicted temperatures.

	Temperature (K)		
	Preliminary Design	CFD	Error (%)
Max Flame Temp.	1803	1885	4.4
TIT	1203	1190	1.1

Figure 5.10: Temperature distribution inside combustor.

Figure 5.11: Predicted combustor exit temperature profile.

products. However, it is not of concern as the combustor pattern factor is of little importance in this design.

5.1.4.5 Liner Wall Temperature Prediction

The liner wall temperature profile predicted by the CFD simulation was plotted in Figure 5.12 and compared to that predicted in the preliminary design (Figure 4.7). While both figures depict similar trends, the peak wall temperatures predicted and their respective locations differ.

These peak values are compared in Table 5.8. The preliminary design predicted a maximum temperature of 1237 K just upstream of the dilution holes whereas CFD concluded that a peak temperature of 1243 K occurred along the dome wall. This discrepancy is largely attributed to the crude approximation for the gas temperature distribution used in the preliminary design.

Contours of the temperature on the inside surface of the TBC are plotted in Figure 5.13. Two large temperature gradients are visible: one occurs along the dome where cold fuel and air react and the other occurs near the dilution holes. The first

Figure 5.12: Predicted liner wall temperature profile.

Table 5.8: Comparison of predicted maximum liner wall temperature.

Temperature (K)		
Preliminary Design	CFD	Error (%)
1237	1243	0.5

Figure 5.13: Contours of TBC surface temperature.

gradient is likely to cause buckling of the dome walls while the second is expected to induce cracking at the edge of the holes.

5.1.4.6 Emission Prediction

Both the formation of CO and NO were modeled in the CFD analysis. The simulation predicted very low CO and NO concentrations of 0.5 and 1 ppmv (dry basis corrected to 15 % O₂), respectively. These values should be regarded as crude approximations since emissions models are typically fairly inaccurate. Improving their accuracy requires the use of complex reaction schemes that are computationally intensive and difficult to obtain converged solutions.

The results from both the simplified reactor model and the CFD simulation are compared in Table 5.9. The large difference between the predicted values is attributed to the applied reaction schemes. The reactor model uses a more detailed 30-step reaction mechanism while the CFD simulation only models four reactions, two for fuel and CO oxidation as well as two for NO formation. Additionally, the NO model used in the CFD simulation only accounts for NO formed by the thermal mechanism.

Table 5.9: Comparison of predicted emissions.

Emissions (ppmv ¹)			
	Preliminary Design	CFD	Error (%)
CO	3	0.5	500
NO	12	1	1100
NO ₂	3	N/A	N/A

¹Dry corrected to 15% O₂

Since prompt NO_x is dominant in turbulent flames below 1800 K (Correa, 1991), its neglect underpredicts the total NO_x concentration.

Contours of the CO and NO mass fractions inside the liner have been plotted in Figure 5.14 and Figure 5.15, respectively. The former indicates that CO is formed near the flame where it has not yet had time to react into CO₂. NO concentrations, however, are highest in the slow-moving, reverse-flowing fluid near the combustor centerline. The fluid here has had sufficient time to allow N₂ to react with O₂ and form NO.

5.1.4.7 Concluding Remarks

The results of the CFD analysis indicate that the preliminary design adequately models the combustion process inside the liner. Reasonable first estimates for the wall temperature distribution, the gas temperature distribution inside the liner, and the pollutant concentrations in the exhaust were obtained. The results also suggest that the procedures of the previous chapter provide a stable combustor with an intense recirculation zone.

5.2 Premixer CFD Analysis

A CFD analysis was performed to measure the performance of the premixer at the design point with respect to mixing and evaporation. The analysis was performed

Figure 5.14: Contours of CO mass fraction inside liner.

Figure 5.15: Contours of NO mass fraction inside liner.

using ANSYS CFX-5 in a manner very similar to the combustor analysis outlined in Section 5.1.

This section outlines the steps taken to perform the CFD simulation and discusses the results. The steps include:

- generating a modeling domain and grid,
- setting up the domain,
- specifying the boundary conditions,
- setting up the solver, and
- solving the domain.

5.2.1 Geometry and Grid Generation

A three-dimensional solid model of the premixer flow domain was constructed and discretized using ANSYS DesignModeler and ANSYS CFX-MESH, respectively.

Solid Model The premixer flow domain was simplified to reduce the complexity of the problem. The simplifications include:

- No swirlers were included in the model. The size of the mesh was vastly reduced by placing the inlet to the domain downstream of the mixer swirlers. This required the assumption that the velocity profiles of the flow issuing from each mixer swirler are uniform and follows the blade.
- The fuel spray issuing from each nozzle is modeled as a single droplet with an initial diameter equal to the SMD.
- The problem is axisymmetric. A 90° section was modeled using the periodic boundary condition. The angle was chosen to ensure a whole number of mixer blades and fuel nozzles inside the domain.

The resulting solid model of the flow domain is shown in Figure 5.16. The dimensions used to create this model are provided in Figure 5.17, which were chosen based on the results of the preliminary design.

Figure 5.16: Solid model of premixer flow domain.

Figure 5.17: Dimensions of premixer flow domain.

Figure 5.18: Premixer computational grid.

Grid Generation The computational grid generated from the solid model is illustrated in Figure 5.18. The unstructured grid with 125,000 nodes and 690,000 elements was refined at the hub and thin wall to help capture the large velocity gradients expected.

5.2.2 Premixer Domain

The premixer domain consists of two phases: the continuous phase and the discrete phase. The continuous phase is the gaseous mixture of air and evaporated fuel vapour, while the discrete phase consists of the individual liquid droplets that travel through the domain. These droplets are tracked through the flow as they evaporate to form fuel vapour. Here, only the mean droplet path has been tracked.

Setting up the premixer flow domain requires specifying the fluid properties, selecting the fluid models for the continuous phase, selecting the fluid models for the discrete phase, defining the coupling between continuous and discrete phase, and creating injection points for the discrete phase.

Fluid Properties The continuous phase was modeled using a variable composition mixture of Jet A vapour and air whereas the discrete phase was modeled as liquid Jet A. Properties for these species were defined using CFX-5's material database.

Fluid Models The continuous phase fluid models were chosen as follows:

- Turbulence Model: *k- ϵ*
- Continuous Phase Heat Transfer Model: *Total Energy*
- Discrete Phase Heat Transfer Model: *Particle Temperature*
- Particle Coupling: *Fully Coupled*
- Drag Force: *Schiller Naumann*
- Turbulent Dispersion Force: *None*
- Heat Transfer: *Ranz Marshall*

The liquid evaporation model was used, the settings are as follows:

- Modification: *Light Oil*
- Latent Heat: *300000 J/kg*
- Reference Temperature: *25 °C*

Particle Injection Regions The domain consists of two particle injection points: one along the premixer axis, and another on the tube walls just downstream of the swirlers. The first point injects fuel from the centre of the hub, parallel with the flow, while the second injects radially downwards from the tube wall, just downstream of the mixer swirlers.

The settings used were derived from the preliminary design implemented in Chapter 4 and are listed in Table 5.10. Slightly larger droplets were specified because of problems tracking ones any smaller.

5.2.3 Boundary Conditions

Inlet There are two inlets: a primary (outer) and a secondary (inner) inlet. They were modeled as total pressure inlets with uniform swirl. Boundary condition settings were obtained from Chapter 4 and are given in Table 5.11.

Outlet The outlet is modeled using the specified mass flow condition. The settings are provided in Table 5.12.

Table 5.10: Particle injection regions.

Parameter	Value	
	Main	Secondary
Injection Method	Cone	Cone
Injection Center (X,Y,Z)	(0,0,0)	(0.01 m, 0.0737 m, 0 m)
Cone Angle ($^{\circ}$)	45	45
Injection Velocity (m/s)	89	89
Number Positions	1	1
Cartesian Components	(X,Y,Z)=(1,0,0)	(0,-1,0)
Temperature (K)	300	300
Particle Diameter (μm)	38	38
Particle Mass Flow Rate (kg/s)	0.0047	0.0188

Table 5.11: Inlet boundary conditions.

Condition	Value	
	Inner	Outer
Flow Regime	Subsonic	Subsonic
Total Pressure	780 kPa	780 kPa
Flow Direction	Cylindrical Components	Cylindrical Components
	r Component = 0	0
	Theta Component = -0.5	0.866
	Axial Component = 0.866	0.5
Turbulence	High (Intensity = 10%)	High (Intensity = 10%)
Total Temperature	575 K	575 K
Fuel Mass Fraction	0.0	0.0

Table 5.12: Outlet boundary conditions.

Condition	Value
Flow Regime	Subsonic
Mass Flow Rate	0.645 [kg/s]

Premixer Walls All remaining surfaces were modeled as walls, except for two periodic surfaces. Table 5.13 gives the settings used.

5.2.4 Results

This section discusses the results from the analysis of the premixer, which was performed using the high resolution advection scheme.

5.2.4.1 Grid Independence Study

The grid independence study for the premixer model was performed in the same manner as for the combustor model, discussed in Section 5.1. The results from this study, plotted in Figure 5.19, indicate that a mesh size greater than 310,000 nodes is necessary to obtain sufficient resolution of the swirling flow. Despite the results, the second largest mesh with 125,000 nodes was selected to avoid excessive computation time.

5.2.4.2 Convergence

The residuals achieved are listed in Table 5.14.

5.2.4.3 Velocity Flowfield

The velocity flowfield is depicted in Figure 5.20. A small undesirable central recirculation bubble is observed behind the hub as a result of the strong swirling flow. The fuel concentration in the bubble, however, is too low for combustion to occur here. Nonetheless, the recirculation zone suggests that the axial velocity of the flow

Table 5.13: Wall boundary conditions.

Condition	Value
Wall Influence On Flow	No Slip
Wall Roughness	Smooth Wall
Heat Transfer	Adiabatic
Perpendicular Restitution Coefficient	0
Parallel Coefficient	1.0

Figure 5.19: Results from premixer grid independence study.

Table 5.14: RMS and MAX residuals for premixer CFD analysis.

Residual	RMS Value	MAX Value
Mass	3E-08	5E-07
Momentum	1E-06	5E-05
Energy	1E-06	1E-04
Eddy Dissipation	2E-07	2E-05
Turbulent Kinetic Energy	2E-07	1E-05
Species Mass Fraction	1E-06	1E-04
Particle Sources	N/A	1E-05

issuing from the inner mixer swirler is not high enough to hinder reverse flow along the centrebody.

The velocity flow angle of the air inside the premixer at various axial locations, labeled in Figure 5.20, was studied to determine the strength of swirl exiting the premixer. Figure 5.21 illustrates that the tangential velocity of the flow degrades due to the shear layer formed between the two counter-rotating streams. The figure supports the assumption that a third swirler is necessary at the exit of the premixer to impart sufficient swirl for PZ recirculation.

5.2.4.4 Particle Trajectories

The mean trajectories of the individual droplets were plotted in Figure 5.22. The figure shows two different sets of droplets: those leaving the main injection point and those leaving the secondary injection point. The droplets leaving from the main injector completely evaporate in the time provided without impacting any walls. The droplets leaving the secondary injector, however, impact the walls downstream of their injection point. This unfavourable collision forces evaporation to occur near the walls, outside of the intense shear layer.

Figure 5.20: Velocity flowfield inside premixer.

Figure 5.21: Premixer outlet flow angle.

Figure 5.22: Droplet mean trajectories and evaporation history.

A comparison between the evaporation distance predicted in the preliminary design and that predicted using CFD is given in Table 5.15. For the purposes of verification, the droplet evaporation model used in the preliminary design was applied a second time using the larger initial drop size specified in the CFD analysis. The droplet model of Section 3.6.3 predicted a much larger evaporation distance than calculated using CFD. This was attributed to differences between the fluid model used in the preliminary design and CFD. Both thermodynamic and transport properties in the vapour phase surrounding the droplet have a significant effect on droplet evaporation rates (Hallett, 2003).

5.2.4.5 Mixedness

Mixedness quantifies the degree to which one species mixes into another. It is defined as

$$\text{Mixedness} = \frac{\text{local average mole fraction } i}{\text{maximum inlet mole fraction } i} \quad (5.2.1)$$

The mixedness of the fuel vapour was plotted along the premixer tube cross-section

Table 5.15: Comparison of predicted evaporation distance for one droplet injected with an initial diameter of $38\text{ }\mu\text{m}$.

Distance (mm)		
Preliminary Design	CFD	Error (%)
517	190	172

in Figure 5.23. Three small pockets of fuel are observed that surround the individual evaporating fuel droplets. As they progress downstream, the pockets become less concentrated but never fully mix with the free stream.

This lack of mixedness is confirmed by the contours at the outlet, illustrated in Figure 5.24. Several pockets of concentrated fuel vapour exist that are surrounded by fresh air. The inclusion of the effects of droplet size distributions, more realistic of actual sprays, is expected to yield more favourable results.

5.2.4.6 Concluding Remarks

The results from the CFD analysis of the premixer illustrate the difficulties involved when modeling evaporating fuel droplets. The assumptions used to model the droplet, and the methods used to estimate the properties of the vapour phase surrounding the droplet, greatly affect the evaporation rate. Further efforts should be focused on integrating a more complex and realistic droplet evaporation model into the preliminary design and CFD.

5.3 FEA of the Liner

A FEA analysis was performed on the liner using ANSYS to ascertain the operating life. This section outlines the steps taken to perform the analysis and discusses the results. The section includes:

- generating a solid model and a grid,
- specifying material properties,

Figure 5.23: Fuel vapour mixedness along axial direction in premixer.

Figure 5.24: Fuel vapour mixedness at outlet of premixer.

- setting the boundary conditions, and
- solving the problem.

5.3.1 Geometry

Solid Model The problem was simplified using several assumptions. These include:

- The problem is axisymmetric. A 36° section was modeled by taking advantage of periodicity. The angle was chosen to ensure a whole number of dilution holes was included in the domain.
- The liner temperature is uniform throughout. This is not the case since the results from the combustor CFD simulation (Section 5.1) prove otherwise. However, an accurate estimation of the liner wall temperature requires a complex conjugate heat transfer analysis.
- The applied static pressure along the liner wall is constant and equal to the total pressure. Specification of the static pressure distribution on both sides of the liner would require CFD analysis of the entire combustor flow domain. Estimates for the total pressure inside the annulus and liner are used instead.
- The front face of the liner, connected to the premixer, is assumed fixed.

The resulting solid model is illustrated in Figure 5.25. It was constructed using the geometry determined in the preliminary design.

Grid Generation The unstructured computational grid is shown in Figure 5.26. It consists of 110,000 nodes which was chosen as a result of a grid independent study. The results of this study are discussed later.

5.3.2 Material Properties

The properties of Hastelloy X were specified using the datasheet provided in Appendix E.

Figure 5.25: Liner solid model.

Figure 5.26: Liner computational grid.

5.3.3 Boundary Conditions

The boundary conditions were as follows:

- the section is modeled as a "Cyclic Sector",
- the front of the liner is fixed in the axial direction (as shown in Figure 5.25),
- the applied pressure on the inner surface of the liner is 760 MPa,
- the applied pressure on the outer surface of the liner is 800 MPa,
- the liner is at a uniform temperature of 1200 K, and
- the reference temperature is 298 K.

5.3.4 Results

This section discusses the results of the liner FEA. It includes discussion of the grid independent study, liner stress distribution, liner thermal expansion, and concluding remarks.

5.3.4.1 Grid Independence Study

Grid independence was checked by performing the analysis using a grid with half the number of nodes. Since the solution did not change by more than 1 percent, grid independence was achieved.

5.3.4.2 Liner Stress Distribution

The stress distribution along the liner wall is plotted in Figure 5.27. A maximum stress of approximately 16.3 MPa occurs at the edge of the dilution holes due to stress concentrations induced by the geometric discontinuity. This value is significantly higher than the 2.3 MPa estimated in the preliminary design, which did not take into account these stress concentrations. Table 5.16 compares the maximum stresses predicted.

At point A on Figure 5.27, away from the dilution holes, FEA predicts a stress of only 2.8 MPa, illustrating the stress raising effect that holes produce. This corresponds to only an 18 percent error.

Table 5.16: Comparison between predictions for maximum stress.

Von Mises Stress (MPa)		
Preliminary Design	FEA	Error (%)
2.3	16.3	86

Figure 5.27: Von Mises stress distribution on liner.

Table 5.17: Comparison between predictions for maximum displacement.

Direction	Length (mm)		
	Preliminary Design	FEA	Error (%)
Radial	3.9	2.8	39
Axial	8.7	11.1	22

5.3.4.3 Thermal Expansion

Contours of the radial and axial displacement of the liner are plotted in Figures 5.28 and 5.29, respectively. Both depict uniform thermal expansion throughout the liner as a result of the constant wall temperature specified.

Table 5.17 compares the maximum displacement in each direction predicted using the preliminary design and FEA. The two sets of values differ largely due to the wall temperature estimate used to form the prediction. Additionally, the liner in the detailed design is slightly longer.

5.3.4.4 Life Prediction Based on Stress Rupture

The stress rupture life of the liner predicted using FEA and that predicted in the preliminary design are plotted in Figure 5.30 for comparison purposes. The preliminary design predicted a life greater than 10,000 hours whereas FEA revealed that it was only 570 hours. This discrepancy is due to the high operating temperature and stresses predicted using numerical analysis in combination with the exponential relationship between the applied stress and the inverse of the time to rupture.

Figure 5.30 also reveals the strong effect of temperature on the stress rupture life. For example, at an applied stress of 16.3 MPa, increasing the temperature from 1255 to 1366 K results in a reduction in rupture life from 570 to 11 hours. This corresponds to a 9 percent increase in temperature and a 98 percent reduction in life.

Figure 5.28: Radial displacement of liner.

Figure 5.29: Axial displacement of liner.

Figure 5.30: Comparison between predictions for liner stress rupture life.

5.3.4.5 Concluding Remarks

FEA provided a detailed solution of the complex stress state in the liner walls, however, its ability to accurately model the liner was compromised by the simplifying assumptions made. The assumption that the temperature of the liner wall was uniform throughout neglected the large thermal gradients that were observed in Figure 5.13. These gradients are accompanied by thermal stresses which can potentially cause failure, either by buckling of the liner wall or cracking near the dilution holes.

5.3.5 Summary

CFD and FEA were used to verify the preliminary design procedures. The results indicated that more complex numerical models and experimental validation are required. Nonetheless, the preceding analysis suggested that the combustor designed should meet most of the requirements.

The results from the preliminary design were used to perform a complete mechanical design of the combustor. This is discussed in the next chapter.

Chapter 6

Mechanical Design

This section provides an overview of the mechanical design for the combustor. It documents the design of the overall combustor assembly, illustrated in Figure 6.1 and Figure 6.2, which consists of the following components:

1. the liner assembly,
2. the premixer assembly,
3. the casing assembly,
4. the injector assembly, and
5. the igniter.

The design included the creation of a three-dimensional solid model and a set of detailed engineering drawings using PRO/ENGINEER. These drawings are provided in Appendix F.

6.1 Liner

6.1.1 Liner Assembly

The main features of the liner are displayed in Figure 6.3. The liner has circumferential ribs on the outer surface and a through-hole to allow the igniter to penetrate into the flow. It is fastened to the premixer using threads, eliminating the need for welding. Lastly, a stopper prevents the liner from sliding off the premixer in the event that differences in thermal expansion separate the two.

Figure 6.1: Combustor cross-section.

Figure 6.2: Combustor cross-section.

Figure 6.3: Liner assembly.

Figure 6.4: Illustration of liner axial growth.

Igniter Hole The igniter is placed slightly further upstream than specified in Figure 4.2. This design change was made to reduce the chance of interference between the igniter and the liner. As the liner expands during startup, the igniter hole will travel downstream and contact the igniter. Placing the liner hole further upstream reduces the amount of travel experienced by the igniter hole and lowers the chance of interference between the two components. This is shown in Figure 6.4.

Even at its new location, the hole still travels downstream by a small amount. Clearance is provided by cutting an oblong hole (Figure 6.5) with the major dimension aligned with the combustor axis.

6.1.2 Fabrication

The liner consists of two parts butt welded together using TIG or plasma welding, as shown in Figure 6.6. Each part is manufactured by casting Hastelloy X to form a rough shape and then machining to specification. The rough shape has a wall thickness of 1/4 inch with the inner diameter cast to specification, eliminating the

Figure 6.5: Igniter hole.

Figure 6.6: Liner assembly, exploded view.

Figure 6.7: Premixer assembly, exploded view.

costly process of machining the inner surface. Instead, the surface is grit blasted to reduce the roughness.

The aft section requires one additional process to drill the dilution holes once it has been machined to specification.

6.2 Premixer

6.2.1 Fabrication

The premixer consists of three swirlers and a premix tube, as illustrated in Figure 6.7. The entire assembly is fabricated using 300-series stainless steel; the swirlers are machined individually whereas the tube is spun from thin sheet 3.175 mm (1/8 inch) thick. Six 60 mm wide square holes placed circumferentially around the premix tube are machined to allow air to pass into the premixer. The whole premixer assembly is tack welded together at the locations illustrated in Figure 6.8.

Figure 6.8: Premixer assembly, cross-sectional view.

6.2.2 Fuel Injection

Fuel is injected into the premix tube through holes in the premixer using five nozzles. They are suspended at various locations by the injector assembly, illustrated in Figure 6.9. Lying on the premixer centerline, the main nozzle protrudes through a hole in the inner mixer swirler hub. The other four nozzles are suspended into equally spaced holes placed circumferentially around the premix tube. This configuration allows fuel to be distributed at various radii, maximizing the mixing potential.

6.3 Casing

The main purpose of the casing is to act as a pressure vessel. It must contain the high pressure compressor discharge without any bleed loss while allowing the delivery of fuel and air through the walls. The casing also provides structural support for the components within.

As illustrated in Figure 6.10, the complete assembly consists of two parts bolted together. The design includes several features: mounting surfaces, two flanges, and

Figure 6.9: Close-up of fuel injection assembly.

an inlet.

6.3.1 Overall Fabrication

The casing was split into two parts to allow assembly and disassembly with ease. Both parts are fabricated from welded sections of 300-series stainless steel sheet 3.175 mm (1/8 inch) thick that are spun into shape. The flanges and mounts are machined from separate pieces of similar grade stainless steel prior to being welded onto the casing. The inlet duct, also welded to the casing, is constructed from both stainless tube and sheet.

6.3.2 Mounting Surfaces

The mounting surfaces allow the injector and igniter assemblies to be fastened to the casing. Thus, bolt holes must be drilled and tapped as specified for each mount.

Five injector mounts are required: one on the end of the casing and four placed circumferentially. A hole is cut in the casing for each mount to allow the injector to penetrate into the combustor.

The igniter mount is similar to the injector mounts except its shape differs to match the igniter.

Figure 6.10: Casing assembly.

6.3.3 Flanges

The flanges, as shown in Figure 6.11, allow the two parts of the casing to be bolted together, providing a seal between the high pressure air and surrounding ambient atmosphere. Both are roughly 10 mm thick to rigidly fasten the two parts of the casing. To improve the seal effectiveness and the ease of alignment, a lip is machined into one flange and a groove into the other. Assembly/dissassembly is also improved by threading one side of the flange and eliminating the need for nuts.

Flange design included the selection of the type and number of bolts. 1/4 inch - 28 UNF bolts were chosen. The geometric properties for this type of bolt is provided in Table 6.1. Selecting smaller bolts would require a larger number of them to sustain total applied the load and increase the assembly/dissassembly time. Larger bolts, however, need more clearance surrounding the head and cannot be spaced as closely together. This hinders the clamping effectiveness of the flange as the large unsupported region between the bolts has a lower clamping force.

Figure 6.11: Casing flanges.

Table 6.1: Basic dimensions of 1/4-28 UNF bolt.

Property	Value
Major Diameter, D_r (in.)	0.2500
Minor Diameter, D (in.)	0.2062
Tensile Stress Area, A_t (in. ²)	0.0364

The number of bolts was computed using the procedures outlined by Juvinall & Marshek (2000) to ensure that they do not become overloaded. This requires the estimation of the axial separating force F_{tot} between the two sections of the casing, which is caused by the applied internal pressure P acting on the casing end walls. To provide a conservative estimate, the axial separating force is based on an estimated effective end wall area, A_{eff} , of 500 mm in diameter. Thus, the total separating force:

$$\begin{aligned} F_{tot} &= PA_{eff} \\ &\approx (8 \text{ bar}) \left[\frac{\pi}{4} (500 \text{ mm})^2 \right] \\ &\approx 160 \text{ kN or } 35 \text{ ksi} \end{aligned} \quad (6.3.1)$$

The individual bolt force F_{bolt} is related to the number of bolts n_{bolt} by:

$$F_{bolt} = \frac{F_{tot}}{n_{bolt}} \quad (6.3.2)$$

This can then be related to the bolt tensile stress σ_t with

$$\sigma_t = \frac{F_{bolt}}{A_t} = \frac{F_{tot}}{n_{bolt}A_t} \quad (6.3.3)$$

where A_t is the cross-sectional area of the bolt subjected to tensile load.

The minimum number of bolts was computed to provide a tensile stress in each individual bolt equal to the proof stress, σ_p . The proof stress corresponds to the axially applied load that the bolt can withstand without any permanent deformation. Using SAE Grade 2 bolts with $\sigma_p = 55 \text{ ksi}$ (380 kPa) (Juvinall & Marshek, 2000), the minimum number of bolts is

$$\sigma_p = \sigma_t \quad (6.3.4)$$

$$\sigma_p = \frac{F_{tot}}{n_{bolt}A_t} \quad (6.3.5)$$

$$n_{bolt} = \frac{F_{tot}}{A_t \sigma_p} \quad (6.3.6)$$

$$n_{bolt} \approx 18 \text{ bolts}$$

Twenty bolts are specified to provide a safety factor.

Figure 6.12: Clearance between liner outlet and casing end-wall.

6.3.4 Liner Support

The casing provides axial and radial support for both the premixer and liner assemblies while allowing for thermal expansion of the liner. Thus, design was performed using the estimates for thermal growth from the liner FEA.

The assembled liner and premixer is fastened to the casing wall using five bolts. The other end of this assembly, the outlet of the liner, protrudes through a hole cut in the casing end wall. The liner is able to slide through this hole as it lengthens since a small clearance is left between the outer surface of the liner and the inner surface of the hole. Additional clearance is provided between the liner and hole to allow for radial growth. The stopper on the liner, discussed in Section 6.1, is sized larger than this hole so that the casing wall will prevent excessive movement. Figure 6.12 illustrates the clearance for the liner outlet.

Radial support of the liner is given by 5 flexible struts placed in the annulus. These struts, shown in Figure 6.13, are thin rectangular sheets of 300-series stainless steel 1.59 mm thick. They are fillet welded to the casing at an angle so that they act as springs, displacing when the liner grows radially. This configuration ensures that radial support occurs at all times. Care was taken to ensure that the struts were not too stiff as to crush the liner during operation.

Figure 6.13: Liner radial supports.

6.3.5 Inlet

The inlet duct (Figure 6.14) is comprised of 300-series stainless tubing 1.59 mm thick welded to the casing. Its exact configuration is tailored to mate with the compressor and turbine while conforming to the engine layout.

The inlet duct consists of a mandrel bent tube that is butt welded to a spun cone, the prediffuser. Guide vanes were added so that the pressure losses incurred when the flow travels through the bend are minimized. These guide vanes are bent from stainless steel sheet 1.59 mm thick and plug welded into the tubing. In order to plug weld, holes were drilled in the tube along the mating surface.

6.4 Injector Assembly

The injector assembly delivers the fuel from outside the casing to inside the premixer. Five of these assemblies are bolted to the casing in the locations shown in Figure 6.15. Each injector assembly consists of a plate, two weld fittings, a tube, and a nozzle, as

Figure 6.14: Inlet sub-assembly.

illustrated in Figure 6.16. While the plate is machined from thick 300-series stainless sheet, the remaining parts are off-the-shelf products.

The tube and fittings are stainless steel parts from SWAGELOK while the nozzle is the BETE PJ24 fogger with a 1/4 inch male pipe thread. This impingement type nozzle sprays a cone-shaped fog with a spray angle of 90°. The datasheet for the nozzle is provided in Appendix E.

6.4.1 Fabrication

The fabrication procedure is simple. A weld fitting with a 1/4 inch tube socket on one end and a 1/4 inch NPT male connector on the other is fillet welded to the plate. The threaded male connector fastens to the main fuel line and a 1/4 inch OD tube is fillet welded to the tube socket end of the fitting. The tube delivers fuel to the nozzle, which is fillet welded to another weld fitting. This fitting has a tube socket on one end as well and a 1/4 inch NPT female connector on the other end. The nozzle has a male 1/4 inch NPT that threads into this female connector. All parts are as

Figure 6.15: Injector locations.

Figure 6.16: Injector assembly, exploded view.

Figure 6.17: Allison igniter.

specified in the drawings provided in Appendix F.

6.5 Igniter

A Champion spark igniter (p/n CH34168) is specified to initiate combustion. The igniter selected is also used on the Allison 250, an engine similar in output to the one currently being designed. An example of the igniter is shown in Figure 6.17.

6.6 Summary

The complete design is documented in Appendix E, which includes a detailed set of engineering drawings.

Chapter 7

Conclusions and Recommendations

7.1 Conclusion

Motivated by new lowering governmental emissions limits, a methodology for the preliminary design of a modern LPP gas turbine combustor was successfully developed and applied to a 1-MW marine gas turbine. The design was verified using numerical analysis tools. Reasonable agreement between predictions from the preliminary design and numerical analysis was achieved which indicated that the design procedures have been developed successfully. Additionally, a detailed mechanical design was performed to provide working drawings for component fabrication. The sequences for manufacture and assembly are also provided in the preceding sections.

The preliminary design procedures provided a combustor geometry that was verified to operate stably with high combustion efficiency, and emitted ultra-low concentrations of pollutants. The liner life, however, was not able to meet the design criteria based on current analyzes. Nonetheless, the procedures quickly provided a detailed definition of the geometry and an assessment of its performance. The initial step before complex numerical analysis with CFD and FEA, the methodology developed greatly simplified the transition from preliminary to detailed design.

Verification was performed by analyzing the reacting flow inside the liner, the flow inside the premixer, and computing the stress state in the liner. The results from numerical analysis agreed well with the predicted values for TIT, peak flame temperature, and peak wall temperature. Significant discrepancies, however, existed between

the estimated liner wall temperature profiles, emissions concentrations, droplet evaporation rates, peak stresses, thermal growths, and liner operating lives. Some error is attributed to the simplified assumptions made to reduce the complexity of the numerical models. More realistic models, in addition to experimentation, are required to improve the assessment of the preliminary design.

It should be emphasized that, despite these large discrepancies, numerical analysis confirmed that the preliminary design was successful. Since further improvements are made at the detailed design phase, the preliminary design is only required to provide a geometry with a reasonable degree of conformance. The combustor designed met most of the specifications and requirements and is therefore acceptable for prototype manufacturing.

7.2 Recommendations

The recommendations for future work include:

- Full scale experimental testing of the combustor. This should be performed to validate the results from both the preliminary and detailed design.
- Sensitivity analysis of the preliminary design methodology. This would provide the designer with valuable insight into parameters that greatly affect the resulting design.
- CFD simulation of the entire combustor flow domain in combination with a more detailed reaction scheme. This is necessary to improve the accuracy of the detailed design phase. It would provide estimates for the static pressure distribution along the liner wall, the airflow distribution throughout the combustor, and the overall total pressure loss. The analysis would also include the effects of annulus flow on dilution jet performance. Additionally, it would reveal any asymmetry in the annulus flow induced by the combustor inlet configuration.
- Investigation of the flow through the inlet and into the annulus. A non-uniform flow in the annulus will affect the transfer of heat from the liner to the annulus

air and consequentially the thermal stress on the liner. The performance of the dilution jets will be affected as well.

- More complex FEA of the liner. The life of this component cannot be accurately assessed without including the effects of thermal gradients and a reasonable estimate for the static pressure distribution along the liner walls. This would require a conjugate heat transfer analysis of the liner and a CFD simulation of the complete combustor flow domain.
- Life prediction based on environmental effects and cyclic fatigue. Accurate liner life prediction requires the inclusion of the detrimental effects of high temperature oxidation and corrosion. Additionally, the effects of frequent start-ups/shutdowns and changes in engine output during operation should be included.
- Investigation of the combustor off-design performance. This is necessary to fully gauge the performance and feasibility of the combustor.
- Implementation of a more complex droplet evaporation model into the preliminary design. Particularly, a multicomponent model is necessary to accurately simulate the evaporation of complex fuels such as Diesel.
- Modification of the preliminary heat transfer analysis. The effects of heat conducted axially along the liner wall and a more realistic gas temperature distribution should be included in the model.
- Modification of the liner nozzle to remove the second recirculation zone created by the sudden contraction at the combustor exit. This sudden contraction can be removed by tapering the liner walls.
- Update correlation for ignition delay. The literature should be reviewed to find a more recent correlation.
- Include the effects of turbulent fluctuations on the particle trajectory in the premixer CFD model.

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Appendix A

Properties of Gases and Liquids

Table A.1: Curvefit coefficients for the enthalpy of select gases at standard reference state (298.15 K, 1 atm).

$$\frac{\bar{h}}{R_u T} = a_1 + \frac{a_2}{2}T + \frac{a_3}{3}T^2 + \frac{a_4}{4}T^3 + \frac{a_5}{5}T^4 + \frac{a_6}{T}$$

Species	<i>T</i> (K)	<i>a</i> ₁	<i>a</i> ₂	<i>a</i> ₃	<i>a</i> ₄	<i>a</i> ₅	<i>a</i> ₆
<i>CO</i> ₂	1000 – 5000	+0.04453623 <i>E</i> + 02	+0.03140168 <i>E</i> – 01	–0.12784105 <i>E</i> – 05	+0.02393996 <i>E</i> – 08	–0.16690333 <i>E</i> – 13	–0.04896696 <i>E</i> + 06
	300 – 1000	+0.02275724 <i>E</i> + 02	+0.09922072 <i>E</i> – 01	–0.10409113 <i>E</i> – 04	+0.06866686 <i>E</i> – 07	–0.02117280 <i>E</i> – 10	–0.04837314 <i>E</i> + 06
<i>H</i> ₂ <i>O</i>	1000 – 5000	+0.02672145 <i>E</i> + 02	+0.03056293 <i>E</i> – 01	–0.08730260 <i>E</i> – 05	+0.12009964 <i>E</i> – 09	–0.06391618 <i>E</i> – 13	–0.02989921 <i>E</i> + 06
	300 – 1000	+0.03386842 <i>E</i> + 02	+0.03474982 <i>E</i> – 01	–0.06354696 <i>E</i> – 04	+0.06968581 <i>E</i> – 07	–0.02506588 <i>E</i> – 10	–0.03020811 <i>E</i> + 06
<i>N</i> ₂	1000 – 5000	+0.02926640 <i>E</i> + 02	+0.14879768 <i>E</i> – 02	–0.05684760 <i>E</i> – 05	+0.10097038 <i>E</i> – 09	–0.06753351 <i>E</i> – 13	–0.09227977 <i>E</i> + 04
	300 – 1000	+0.03298677 <i>E</i> + 02	+0.14082404 <i>E</i> – 02	–0.03963222 <i>E</i> – 04	+0.05641515 <i>E</i> – 07	–0.02444854 <i>E</i> – 10	–0.10208999 <i>E</i> + 04
<i>O</i> ₂	1000 – 5000	+0.03697578 <i>E</i> + 02	+0.06135197 <i>E</i> – 02	–0.12588420 <i>E</i> – 06	+0.01775281 <i>E</i> – 09	–0.11364354 <i>E</i> – 14	–0.12339301 <i>E</i> + 04
	300 – 1000	+0.03212936 <i>E</i> + 02	+0.11274864 <i>E</i> – 02	–0.05756150 <i>E</i> – 05	+0.13138773 <i>E</i> – 08	–0.08768554 <i>E</i> – 11	–0.10052490 <i>E</i> + 04

SOURCE: From Turns (2000)

Table A.2: Curvefit coefficients for enthalpy of select fuels at standard reference state (298.15 K, 1 atm).

$$\bar{h} \text{ (kJ/kmol)} = 4184 \left(a_1\theta + \frac{a_2}{2}\theta^2 + \frac{a_3}{3}\theta^3 + \frac{a_4}{4}\theta^4 - \frac{a_5}{\theta} + a_6 \right)$$

where $\theta \equiv T(\text{K})/1000$

Formula	Fuel	MW	a_1	a_2	a_3	a_4	a_5	a_6	a_8^1
CH_4	Methane	16.043	-0.29149	26.327	-10.610	1.5656	0.16573	-18.331	4.300
C_3H_8	Propane	44.096	-1.4867	74.339	-39.065	8.0543	0.01219	-27.313	8.852
C_6H_{14}	Hexane	86.177	-20.777	210.48	-164.125	52.832	0.56635	-39.836	15.611
C_8H_{18}	Isooctane	114.230	-0.55313	181.62	-97.787	20.402	-0.03095	-60.751	20.232
CH_3OH	Methanol	32.240	-2.7059	44.168	-27.501	7.2193	0.20299	-48.288	5.3375
C_2H_5OH	Ethanol	46.07	6.990	39.741	-11.926	0	0	-60.214	7.6135
$C_{8.26}H_{15.5}$	Gasoline	114.8	-24.078	256.63	-201.68	64.750	0.5808	-27.562	17.792
$C_{7.76}H_{13.1}$	Gasoline	106.4	-22.501	227.99	-177.26	56.048	0.4845	-17.578	15.232
$C_{10.8}H_{18.7}$	Diesel	148.6	-9.1063	246.97	-143.74	32.329	0.0518	-50.128	23.514

SOURCE: From Turns (2000)

¹ To obtain 0 K reference state for enthalpy, add a_8 to a_6 .

Table A.3: Properties of n-dodecane ($C_{12}H_{26}$)

Molecular Weight ¹ , MW (kg/kmol)	170.340
Boiling Temperature at 1 atm ¹ , T_{bp} (K)	489.5
Liquid Density at 293 K ¹ , ρ_l (kg/m ³)	749
Latent Heat of Vaporization at 1 atm ² , h_{fg} (kJ/kg)	256
Liquid Specific Heat at 293 K ³ , C_{pl} (kJ/kg·K)	2.173
Critical Temperature ¹ , T_c (K)	658.2
Critical Volume ¹ , V_c (cm ³ /mole)	713
Acentric factor ¹ , ω	0.575
Dipole Moment ¹ , μ (debyes)	0
Polar Correction Factor ¹ , κ	0
Vapour Specific Heat ¹ , C_p	Equation B.1.1
	• $A = -9.328E + 03$
	• $B = +1.149E + 03$
	• $C = -6.347E - 01$
	• $D = -1.359E - 04$
Vapour Viscosity, η	Equation B.2.5
Vapour Thermal Conductivity, λ	Equation B.3.2
Vapour Pressure ¹ , P_v	Equation B.4.1
	• $A = 16.1134$
	• $B = 3774.56$
	• $C = -91.31$

SOURCES:

¹From Reid *et al.* (1987).²From Turns (2000).³From Hallett (2003).

Table A.4: Properties of air

Molecular Weight ¹ , MW (kg/kmol)	28.97
Molecular Diameter ² , σ Å	3.711
Characteristic Energy Constant ² , ϵ/k (K)	78.6
Specific Heat ¹ , C_p	Equation B.1.1
	<ul style="list-style-type: none"> • $A = +0.2745E + 05$ • $B = +0.6184E + 04$ • $C = -0.8993E + 03$ • $D = 0$
Viscosity, μ	Equation B.2.1
Thermal Conductivity, λ	Equation B.3.2

SOURCES:

¹From Rolle (2005).²From Reid *et al.* (1987).

Appendix B

Thermodynamic and Transport Properties of Ideal Gases

Many researchers have developed entirely empirical and semi-empirical methods for determining both thermodynamic and transport properties of ideal gases. Reid *et al.* (1987) provided a detailed review of these correlations applicable to ideal gases as well as liquids.

B.1 Specific Heat

The specific heat for many gases is determined using curvefits of the form

$$C_p = \frac{1}{MW} (A + B \cdot T + C \cdot T^2 + D \cdot T^3) \quad (\text{B.1.1})$$

where C_p = the specific heat in J/kg·K

MW = the molecular weight in kg/kmol

T = the temperature in K

The constants A , B , C , and D are determined experimentally.

B.2 Viscosity

Correlating viscosity with the temperature of a fluid is made possible using the Chapman-Enskog equation (Reid *et al.* , 1987), which is defined as

$$\eta = \frac{(2.669E - 06)(MW \cdot T)^{1/2}}{\sigma^2 \Omega_v} \quad (\text{B.2.1})$$

where η = the viscosity in N·s/m²
 MW = the molecular weight in kg/kmol
 T = the temperature in K
 σ = the molecular diameter in Å
 Ω_v = the collision integral

The collision integral is determined from the empirical relation

$$\Omega_v = [A(T^*)^{-B}] + C[\exp(-DT^*)] + E[\exp(-FT^*)] \quad (\text{B.2.2})$$

where T^* is the dimensionless temperature, $A = 1.16145$, $B = 0.14874$, $C = 0.52487$, $D = 0.77320$, $E = 2.16178$, and $F = 2.43787$. Equation B.2.2 is only applicable for $0.3 \leq T^* \leq 100$.

The dimensionless temperature is computed using

$$T^* = \frac{kT}{\epsilon} \quad (\text{B.2.3})$$

where k is Boltzmann's constant and ϵ is the characteristic energy of the medium.

The value of k/ϵ together with that of σ effectively determine the ability of Equation B.2.1 to accurately predict the viscosity over a wide range of temperatures. Obtaining meaningful values for these parameters is difficult as there are infinite sets that will yield acceptable results. Many researchers have specified rules to characterize k/ϵ and σ . One such method, discussed by Reid *et al.* (1987), is outlined below.

Equation B.2.1 is employed with

$$\frac{\epsilon}{k} = \frac{T_c}{1.2593} \sigma = 0.809 V_c^{1/3} \quad (\text{B.2.4})$$

The final result is

$$\eta = (4.0785E - 06) \frac{F_c (MT)^{1/2}}{V_c^{2/3} \Omega_v} \quad (\text{B.2.5})$$

where the critical volume V_c is in cm³/mole. The factor F_c is defined as

$$F_c = 1 - 0.2756\omega + 0.059035\mu_r^4 + \kappa \quad (\text{B.2.6})$$

where ω = acentric factor
 μ_r = dimensionless dipole moment
 κ = polar correction factor

The dimensionless dipole moment,

$$\mu_r = 131.3 \frac{\mu}{(V_c T_c)^{1/2}} \quad (\text{B.2.7})$$

where the dipole moment μ is in debyes.

B.3 Thermal Conductivity

The Eucken correlation provides a means of estimating the thermal conductivity for polyatomic gases (Reid *et al.*, 1987). The Eucken equation is defined by

$$\frac{\lambda}{\eta C_v} = 1 + \frac{9R}{4C_v} \quad (\text{B.3.1})$$

where λ = the thermal conductivity in W/m·K
 η = the viscosity in N·s/m²
 C_v = the constant volume specific heat in J/kg·K

Substituting the ideal-gas relation ($C_p - C_v = R$) and rearranging yields

$$\lambda = \eta \left(C_p + \frac{5}{4} R \right) \quad (\text{B.3.2})$$

where R is the gas constant and C_p is the constant pressure specific heat for the gas, both in J/kg·K.

B.4 Vapour Pressure

The vapour pressure of liquids may be calculated using the Antoine equation (Reid *et al.*, 1987)

$$P_v = \exp \left[A - \frac{B}{T + C} \right] \quad (\text{B.4.1})$$

where T is in K and P_v is in Torr (760 Torr = 1 atm = 101.325 kPa).

Appendix C

Verification

Figure C.1: Verification of curvefits for enthalpy of select gases.

Figure C.2: Verification of curvefit for enthalpy of diesel vapour.

Figure C.3: Verification of curvefits for properties of air.

Figure C.4: Verification of curvefits for properties of $\text{C}_{12}\text{H}_{26}$ vapour.

Figure C.5: Verification of curvefits for vapour pressure of $C_{12}H_{26}$.

Appendix D

Preliminary Design

Figure D.1: Overall preliminary design flowchart.

Figure D.2: Flowchart for determination of airflow distribution.

Figure D.3: Flowchart for resizing for premixed combustion.

Figure D.4: Flowchart for sizing diffuser.

Figure D.5: Flowchart for designing premixer.

Table D.1: Geometry inputs for preliminary design.

Parameter	Description
$AR_{prediff}$	Prediffuser area ratio.
D_3	Inlet diameter.
D_4	outlet diameter.
$D_{hub,1}$	Mixer swirler hub diameter.
$D_{hub,2}$	Combustor swirler hub diameter.
e	Trip strip height.
(h/c)	Swirler vane aspect ratio (for each swirler).
k_{area}	Liner-to-casing area ratio.
K_{sw}	Swirler vane parameter (for each swirler).
n_n	Number of nozzles.
p	Trip strip spacing.
SMD	Nozzle spray SMD.
t_{casing}	Wall thickness of casing.
t_{liner}	Liner thickness.
t_v	Swirler vane thickness (for each swirler).
ΔP_{mixer}	Mixer swirler pressure loss.
ΔP_n	Fuel nozzle pressure differential.
α_n	Spray cone angle.
σ	Swirler vane solidity (for each swirler).
θ_{cont}	Contraction wall angle.
θ_{dome}	Dome wall angle.
$\theta_{prediff}$	Prediffuser wall angle.
θ_v	Swirler vane angle (for each swirler).

Figure D.6: Flowchart for dilution hole sizing.

Table D.2: Operating condition inputs for preliminary design.

Parameter	Description
$\dot{m}_3$	Mass flow rate through combustor.
P_3	Combustor inlet total pressure.
T_3	Total temperature at combustor inlet.
T_4	Total temperature at combustor outlet.
T_{flame}	Flame temperature
TI	Combustor turbulence intensity.
T_{l_0}	Injection temperature of liquid fuel.
$\frac{\Delta P_{3-4}}{P_3}$	Pressure loss coefficient.
$\frac{\Delta P_{3-4}}{q_{ref}}$	Pressure loss factor.
η_{pump}	Pump isentropic efficiency.
ϕ_{PZconv}	PZ equivalence ratio for and equivalent conventional combustor.

Figure D.7: Flowchart for checking structural integrity.

Table D.3: Fluid properties input in preliminary design.

Parameter	Description
B_{st}	Stoichiometric fuel mass-transfer number.
(C/H)	Fuel carbon-to-hydrogen mass ratio.
C_{pa}	Specific heat of air.
C_{pv}	Specific heat of fuel vapour.
E	Activation energy for Semenov equation.
Fuel	The type and chemical formula of the fuel burned.
$\bar{h}_i$	Enthalpy of gases (CO_2 , H_2O , N_2 , O_2 , fuel vapour).
h_{fg}	Fuel latent heat of vaporization.
MW_a	Molecular weight of air.
MW_f	Molecular weight of fuel.
P_v	Fuel vapour pressure.
R_a	Ideal gas constant for air.
R_u	Universal gas constant.
$S_{L_{ref}}$	Reference flame speed for Semenov equation.
η_a	Viscosity of air.
η_v	Viscosity of fuel vapour.
λ_a	Thermal conductivity of air.
λ_v	Thermal conductivity of fuel vapour.
ρ_l	Density of liquid fuel.

Table D.4: Material properties input in preliminary design.

Parameter	Description
α	Coefficient of thermal expansion for liner wall.
ε_{casing}	Emissivity of casing material.
ε_{liner}	Emissivity of liner wall material.
ε_{TBC}	Emissivity of TBC.
λ_{liner}	Thermal conductivity of liner wall material.
λ_{TBC}	Thermal conductivity of TBC.
σ	Stefan-Boltzmann constant.
σ_Y	Yield stress for liner and casing material.

Table D.5: Outputs from preliminary design for operating conditions.

Parameter	Description
AR_{plenum}	Plenum sudden expansion area ratio.
E_{min}	Minimum ignition energy.
S_n	Swirl number for combustor swirler.
t_{ig}	Autoignition time.
T_{flame}	Flame Temperature.
V_{ref}	Reference velocity.
V_n	Droplet injection velocity.
W_{an}	Fraction of mass flow through annulus.
$W_{dilution}$	Fraction of mass flow through dilution holes.
W_{DZ}	Fraction of mass flow in DZ.
$W_{premixer}$	Fraction of mass flow through premixer.
W_{PZ}	Fraction of mass flow in PZ.
W_{pump}	Required pump power.
ΔP_{dump}	Dump total pressure loss.
ΔP_{liner}	Total pressure drop across the liner.
ΔP_{plenum}	Plenum total pressure loss.
$\Delta P_{prediff}$	Prediffuser total pressure loss.
ΔP_{swirl}	Combustor swirler total pressure loss.
ϕ_{DZ}	DZ equivalence ratio.
ϕ_{PZ}	PZ equivalence ratio.

Table D.6: Geometry outputs of preliminary design.

Parameter	Description
A_{an}	Annulus flow area.
C_D	Dilution hole discharge coefficient.
c_v	Swirler vane length (for each swirler).
D_h	Dilution hole diameter.
D_{liner}	Liner diameter.
$D_{liner,2}$	Liner diameter after contraction.
$D_{mix,i}$	Inner mixer swirler tip diameter.
$D_{mix,o}$	Outer mixer swirler tip diameter.
$D_{prediff}$	Prediffuser outlet diameter.
D_{ref}	Casing diameter.
$D_{ref,1}$	Casing diameter in plenum region.
$D_{ref,2}$	Casing diameter after contraction.
L_2	Contraction length.
L_{dome}	Dome length.
L_{DZ}	DZ length.
L_{ig}	Igniter hole location.
$L_{prediff}$	Prediffuser length.
$L_{premixer}$	Premixer tube length.
L_{PZ}	PZ length.
n_h	Number dilution holes.
n_v	Number of swirler vanes (for each swirler).

Table D.7: Output from heat transfer analysis.

Parameter	Description
T_g	Gas temperature distribution.
T_i	Interface temperature distribution.
T_{w1}	Inner wall temperature distribution.
T_{w2}	Outer wall temperature distribution.

Table D.8: Output from droplet evaporation model.

Parameter	Description
x_p	Droplet axial location history.
r_p	Droplet radial location history.
D	Droplet diameter history.
T_l	Droplet temperature history.

Table D.9: Output from dilution jet model.

Parameter	Description
X_c	Jet centerline axial location history.
Y_c	Jet centerline radial location history.
$W_{1/2}^-$	Jet half-width on injection side of centerline.
$W_{1/2}^+$	Jet half-width on opposite side of centerline.

Table D.10: Output from reactor model.

Parameter	Description
$Loading$	Combustor loading.
T_{flame}	Flame temperature.
T_4	Combustor outlet temperature.
Y_i	Species molar concentration.
η_c	Combustion efficiency.
χ_i	Species molar concentration.

Appendix E

Datasheets

HASTELLOY[®] X alloy

UNS No. N06002

47Ni^a-22Cr-18Fe-9Mo-1.5Co-0.6W-0.10C-1Mn*1Si-0.008B*

^aAs Balance *Maximum

Fabrication Guidelines	A nickel-base alloy with an exceptional combination of oxidation resistance, fabricability and high-temperature strength. <u>Principal Features</u> <u>Creep and Stress-Rupture Strength</u> <u>Tensile Properties</u> <u>Physical Properties</u> <u>Oxidation</u> <u>Carburization</u> <u>Health and Safety</u> <u>Sales Office Addresses</u> H-3009A
Technical Papers	
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MSDS	
Request for Literature	
Applicable Specifications	
<u>Heat-Resistant Alloys For Industrial Applications</u>	

For more information send e-mail to: Alloy_Information@haynesintl.com

HAYNES HOME

PRINCIPAL FEATURES

Strength and Oxidation Resistance

HASTELLOY® X alloy is a nickel- chromium-iron-molybdenum alloy that possesses an exceptional combination of oxidation resistance, fabricability and high-temperature strength. It has also been found to be exceptionally resistant to stress-corrosion cracking in petrochemical applications. X alloy exhibits good ductility after prolonged exposure at temperatures of 1200, 1400, 1600F (650, 760 and 870C) for 16,000 hours.

Haynes International Inc X alloy

Easily Fabricated

HASTELLOY X alloy has excellent forming and welding characteristics. It can be forged and, because of its good ductility, can be cold worked. X alloy can be welded by both manual and automatic methods including shielded metal arc (covered electrodes), gas tungsten arc (GTAW) and gas metal arc (GMAW). The alloy can also be resistance welded.

Haynes International Inc X alloy

ASME Boiler and Pressure Vessel Code

HASTELLOY X alloy plate, sheet, strip, bar, tubing and pipe are covered by ASME specifications SB-435, SB-572, SB-619, SB-622 and SB-626 under UNS Number N06002.

Haynes International Inc X alloy

Heat-Treatment

Wrought forms of HASTELLOY X alloy are furnished in the solution heat-treated condition unless otherwise specified. X alloy is solution heat-treated at 2150F (1177C) and rapid cooled. Bright annealed products are cooled in hydrogen.

Haynes International Inc X alloy

Available in Convenient Forms

HASTELLOY X alloy is available in the form of plate, sheet, strip, billet, bar, wire, covered electrodes, pipe and tubing.

Haynes International Inc X alloy

Useful for Aircraft, Furnace and Chemical Process Components

X alloy has wide use in gas turbine engines for combustion zone components such as transition ducts, combustor cans, spray bars and flame holders as well as in afterburners, tailpipes and cabin heaters. It is recommended for use in industrial furnace applications because it has unusual resistant to oxidizing, reducing and neutral atmospheres. Furnace rolls of this alloy were still in good condition after operating for 8,700 hours at 2150F (1177C). HASTELLOY X alloy is also used in the chemical process industry for retorts, muffles, catalyst support grids, furnace baffles, tubing for pyrolysis operations and flash drier components.

Haynes International Inc X alloy

Specifications

For information on specifications to which this alloy can be ordered, please contact one of the locations shown on the Haynes Home Page and ask for bulletin H-1034 Applicable Specifications.

Nominal Chemical Composition, Weight Percent

Ni	Cr	Fe	Mo	Co	W	C	Mn	Si	B
47*	22	18	9	1.5	0.6	0.10	1*	1*	0.008*

⁴As balance

*Maximum

[Back to HASTELLOY X alloy Home](#)
[Forward to Creep and Stress-Rupture Strength](#)

HAYNES HOME

Test Temperature		Approximate Rupture Life Strength for Time Indicated, Ksi (MPa)			
°F	°C	10 Hours	100 Hours	1,000 Hours	10,000 Hours
1200	649	67.0 (462)	48.0 (331)	34.0 (170)	24.9 (170)
1400	760	32.0 (221)	22.5 (155)	15.8 (77)	11.1 (77)
1600	871	17.0 (117)	10.6 (73)	6.5 (28)	4.0 (28)
1800	982	6.5 (45)	3.7 (26)	2.1 (8)	1.2 (8)
2000	1093	2.4 (17)	1.2 (8)	0.6 (4)	-

* Solution heat-treated. Based on over 100 tests.

Average Rupture Data, Plate and Bar*

Test Temperature		Approximate Rupture Life Strength for Time Indicated, Ksi (MPa)			
°F	°C	10 Hours	100 Hours	1,000 Hours	10,000 Hours
1200	649	72.0 (496)	47.9 (330)	34.0 (234)	24.0 (165)
1350	732	36.0 (248)	25.0 (172)	18.0 (124)	12.5 (86)
1500	818	21.0 (145)	14.0 (96)	10.0 (69)	6.8 (47)
1650	899	12.0 (83)	7.5 (52)	4.7 (32)	3.0 (21)
1800	982	7.0 (48)	4.2 (29)	2.4 (17)	1.4 (10)
1950	1066	3.7 (26)	1.9 (13)	1.0** (7)	-
2100	1149	1.7 (12)	0.7 (5)	0.3** (2)	-

* Solution heat-treated. Based on over 100 tests for sheet and over 150 tests for plate and bar.

** Extrapolated

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HAYNES HOME

TYPICAL TENSILE PROPERTIES

Average Room Temperature Tensile Data*

Form	Condition	Ultimate Tensile Strength		Yield Strength at 0.2% Offset		Elongation in 2 in. (50.8 mm)
		Ksi	MPa	Ksi	MPa	%
Sheet, 0.091 to 0.312 in. (2.3 to 7.9mm) thick	Heat-treated at 2150°F (1177°C), Rapid Cooled	109.5	755	55.9	385	45
Plate, 3/8 to 2 in. (9.5 to 50.8mm) thick	Heat-treated at 2150°F (1177°C), Rapid Cooled	107.7	743	49.1	339	51

* Based on 58-69 tests.

Average Tensile Data*

Form	Condition	Test Temperature		Ultimate Tensile Strength		Yield Strength at 0.2% Offset		Elongation in 2 in. (50.8 mm)
		°F	°C	Ksi	MPa	Ksi	MPa	%
								44
Sheet 0.045-0.060 in. (1.1-1.5 mm) thick	Bright Annealed at 2150°F (1177°C), Hydrogen Cooled	Room		111.2	767	54.9	379	
		1000	538	89.0	614	35.6	245	
		1200	649	84.2	581	35.4	244	
		1400	760	67.1	463	34.4	237	
		1600	871	44.9	310	28.1	194	
		1800	982	25.6	177	13.2	91	
		2000	1093	14.0	97	6.3	43	60

* Strain rate was 0.005-inch/inch/minute to 0.2% yield point and 0.5-inch/inch/minute to failure.

Average Effect of Cryogenic Temperatures on Tensile Properties

Form	Condition	Test Temperature		Ultimate Tensile Strength		Yield Strength at 0.2% Offset		Elongation in 2 in. (50.8 mm)
		°F	°C	Ksi	MPa	Ksi	MPa	%
								46
Plate	Heat-treated at 2150°F (1177°C), Rapid Cooled	-321	-196	150.2	1036	-	-	
		-108	-78	118.8	819	-	-	
		72	22	104.5	721	47.0	324	

Average Short-Time Tensile Data, Cold-Reduced and Welded 0.109 in. (2.8mm) Sheet

Condition	Form	Ultimate Tensile Strength		Yield Strength at 0.2% Offset	
		Ksi	MPa	Ksi	MPa
As Cold-Reduced	Reduced 5 percent	123.0	848	82.0	565
	Reduced 15 percent	137.0	945	106.0	731
	Reduced 30 percent	161.0	1110	137.0	945
Cold-Reduced and Welded, As-Welded	Reduced 5 percent	114.9	792	68.0	469
	Reduced 15 percent	113.1	780	72.1	497
	Reduced 30 percent	112.9	778	69.9	482

NOTE All cold-reduced sheet and the various weld samples were produced from material which had been solution heat-treated prior to cold reduction or welding.

All data were obtained at room temperature and are the result of a limited number of tests.

Average Short-Time Tensile Data, Weldments

Condition	Weld Method	Material	Ultimate Tensile Strength		Yield Strength at 0.2% Offset		Elongation in 2 in. (50.8 mm)
			Ksi	MPa	Ksi	MPa	%
As-Welded	Shielded Metal Arc (covered electrodes)	Sheet, 0.125 in. (3.2mm)	110.2	760	55.2	381	26
		Plate, 0.250 in. (6.4mm)	109.8	757	56.7	391	26
		Plate, 0.375 in. (9.5mm)	110.2	760	55.4	382	26
As-Welded	Gas Tungsten Arc (GTAW)	Sheet, 0.125 in. (3.2mm)	110.2	759	59.1	407	26
		Plate, 0.250 in. (6.4mm)	107.1	738	53.1	365	25
		Plate, 0.375 in. (9.5mm)	107.6	742	54.9	379	22
As-Welded	Gas Metal Arc (GMAW)	Sheet, 0.125 in. (3.2mm)	103.7	715	53.1	366	22
		Plate, 0.250 in. (6.4mm)	110.8	764	55.0	379	33
		Plate, 0.375 in. (9.5mm)	106.4	734	57.0	393	24

Average Aged Tensile Data, Room Temperature*

Form	Aging Temperature		Aging Time, Hours	Ultimate Tensile Strength		Yield Strength at 0.2% Offset		Elongation in 2 in. (50.8 mm)
	°F	°C		Ksi	MPa	Ksi	MPa	%
Sheet, 0.125 in. (3.2mm) thick	SHT		-	114.3	788	55.2	381	57
	1200	649	1000	125.0	862	61.0	421	35
			4000	143.8	991	76.2	525	19
			8000	147.9	1020	78.6	542	19
			16000	148.0	1020	78.1	538	15
	1400	760	1000	137.0	945	65.3	450	23
			4000	134.6	928	64.3	443	18
			8000	131.0	903	61.3	423	19
			16000	126.1	869	59.3	409	17
	1600	871	1000	123.0	848	53.2	369	26
			4000	117.9	813	49.3	340	29
			8000	115.0	793	48.2	332	30
			16000	111.1	766	46.1	318	29
Plate, 1/2 in. (12.7mm) thick	SHT		-	109.9	758	49.5	341	47
	1200	649	1000	121.4	837	56.5	390	33
			4000	142.5	983	73.4	506	18
			8000	143.6	990	73.0	503	18
	1400	760	1000	129.4	892	56.9	392	23
			4000	129.9	896	56.9	392	21
			8000	129.2	891	56.3	388	20
	1600	871	1000	119.0	820	47.6	328	31
			4000	116.7	805	44.9	310	28
			8000	113.7	784	43.9	303	26
			16000	109.0	752	42.7	294	26

* Test data for each form are from a single heat.
SHT= Solution heat-treated (not aged)

Average Welded and Aged Tensile Data, Room Temperature*

Form	Aging Temperature		Aging Time, Hours	Ultimate Tensile Strength		Yield Strength at 0.2% Offset		Elongation in 2 in. (50.8 mm)	Reduction of area
	°F	°C		Ksi	MPa	Ksi	MPa		
Plate, 1/2 in. (12.7mm) thick	1600	871	8000	109.0	752	47.9	330	-	22
Gas Tungsten Arc Welded Plate, 1/2 in. (12.7mm) thick	1200	649	1000	126.9	875	66.0	455	-	33
			4000	150.1	1035	86.5	596	-	19
			8000	145.5	1003	82.9	572	-	18
	1400	760	1000	128.2	884	58.2	401	-	19
			4000	127.4	878	62.3	430	-	18
			8000	125.2	863	62.3	430	-	15
	1600	871	4000	105.3	726	49.7	343	-	15
			8000	98.0	676	46.9	323	-	16
All Weld Metal**	1200	649	1000	123.0	848	87.5	603	4	8
			4000	139.3	960	86.0	593	7	8
			8000	131.8	909	86.8	598	6	9
	1400	760	1000	113.5	783	62.7	432	10	12
			4000	110.5	762	60.6	418	7	6
			8000	97.7	674	59.8	412	4	7
	1600	871	1000	92.8	640	48.3	330	8	9
			8000	92.7	639	46.3	319	13	11

* Test data for each form are from a single heat.

** Single test data. Gas tungsten arc welded.

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TYPICAL PHYSICAL PROPERTIES

	Temperature, °F	British Units	Temperature, °C	Metric Units
Density	72	0.297 lb/in. ³	22	8.22 g/cm ³
Melting Range	2300-2470		1260-1355	
Electrical Resistivity	72	46.6 microhm-in. (712 ohms per cir. mil-ft)	22	1.18 microhm-cm
Thermal Conductivity	70	63 Btu-in./ft. ² hr.-°F	21	9.1 W/m-K
	200	76 Btu-in./ft. ² hr.-°F	93	11.0 W/m-K
	500	98 Btu-in./ft. ² hr.-°F	200	14.1 W/m-K
	1100	144 Btu-in./ft. ² hr.-°F	593	20.8 W/m-K
	1300	159 Btu-in./ft. ² hr.-°F	704	22.9 W/m-K
	1500	174 Btu-in./ft. ² hr.-°F	816	25.1 W/m-K
	1700	189 Btu-in./ft. ² hr.-°F	927	27.2 W/m-K
Specific Heat	Room	0.116 Btu/lb.-°F	Room	486 J/K g-K
	200	0.117 Btu/lb.-°F	93	490 J/K g-K
	400	0.118 Btu/lb.-°F	204	494 J/K g-K
	600	0.119 Btu/lb.-°F	316	498 J/K g-K
	800	0.123 Btu/lb.-°F	427	515 J/K g-K
	1000	0.130 Btu/lb.-°F	538	544 J/K g-K
	1200	0.139 Btu/lb.-°F	649	582 J/K g-K
	1400	0.151 Btu/lb.-°F	760	632 J/K g-K
	1600	0.167 Btu/lb.-°F	871	699 J/K g-K
	1800	0.186 Btu/lb.-°F	982	799 J/K g-K
	2000	0.205 Btu/lb.-°F	1093	858 J/K g-K
Mean Coefficient of	79-200	7.7 microinches/in.-°F	26-93	13.9 10 ⁻⁴ m/m-°C
Thermal Expansion	79-1000	8.4 microinches/in.-°F	26-538	15.1 10 ⁻⁴ m/m-°C
	79-1200	8.6 microinches/in.-°F	26-649	15.5 10 ⁻⁴ m/m-°C
	79-1350	8.8 microinches/in.-°F	26-732	15.8 10 ⁻⁴ m/m-°C
	79-1500	8.9 microinches/in.-°F	26-816	16.0 10 ⁻⁴ m/m-°C
	79-1650	9.1 microinches/in.-°F	26-899	16.4 10 ⁻⁴ m/m-°C
	79-1800	9.2 microinches/in.-°F	26-982	16.6 10 ⁻⁴ m/m-°C
Poisson's	-108	0.328	-78	0.328
Ratio	72	0.320	22	0.320
Magnetic Permeability	Room	1.002 at 200 oersteds (15,900 A/m)		

Dynamic Modulus of Elasticity, Plate

Condition	Temperature, °F	Dynamic Modulus of Elasticity, 10 ⁶ psi	Temperature, °C	Dynamic Modulus of Elasticity, GPa
Heat-treated at 2150°F (1177°C) Rapid Cooled	Room	29.8	Room	205
	200	29.4	93	203
	400	28.6	204	197
	600	27.8	316	192
	800	26.7	427	184
	1000	25.8	538	178
	1200	24.7	649	170
	1400	23.3	760	161
	1600	22.2	871	153
	1800	20.4	982	141

Average Hardness, Room Temperature

Form	Condition	Average Hardness Rockwell B
Sheet, 0.012 to 0.090 in. (0.30 to 2.3mm) thick	Heat Treated at 2150°F (1177°C), Rapid Cooled	87*
Sheet, 0.091 to 0.312 in. (2.3 to 7.8mm) thick	Heat-treated at 2150°F (1177°C), Rapid Cooled	92*
Plate, 3/8 to 2 in. (9.5 to 50.8mm) thick	Heat-treated at 2150°F (1177°C), Rapid Cooled	89*

* Based on 58-65 tests

Average Hardness, Welded Sheet

Weld Method	Test Area	Average Hardness Rockwell B
Shielded Metal Arc (covered electrodes)	Weld area	92
	Heat-affected zone	93
	Base metal	91
Gas Tungsten Arc (GTA W)	Weld area	89
	Heat-affected zone	93
	Base metal	91
Gas Metal Arc (GMA W)	Weld area	90
	Heat-affected zone	93
	Base metal	91

Note: Sheet was solution heat-treated prior to welding. Hardness was determined at room temperature in the as-welded condition.

Average Hardness, Room Temperature*

Form	Aging Temperature		Aging Time, Hours	Hardness, Rockwell A
	°F	°C		
Sheet	SHT		-	54
	1200	649	1000	56
			4000	62
			8000	63
	1400	760	1000	62
			4000	61
			8000	60
	1600	871	1000	61
			4000	58
			8000	55
Plate	SHT		-	54
	1200	649	1000	57
			4000	62
			8000	63
	1400	760	1000	60
			4000	59
			8000	58
	1600	871	1000	56
			4000	56
			8000	54
All Weld Metal**	1200	649	1000	64
			4000	65
			8000	63
	1400	760	1000	62
			4000	60
			8000	60
	1600	871	1000	56
			4000	55
			8000	54

SHT=Solution heat-treated (not aged)

* Single tests from a single heat for each form

** Gas tungsten arc welded

Formability, Sheet

Condition	Typical Olsen cup depth,	
	in.	mm
Heat Treated at 2150°F (1177°C),	0.48	12.3
Rapid Cooled		

Average Impact Strength, Plate

Condition	Test Temperature		Average Charpy V-Notch Impact Strength	
	°F	°C	Ft. lb.	Joules
Heat-treated at 2150°F (1177°C) Rapid Cooled	-321	-196	37	50
	-216	-157	44	60
	-108	-78	51	69
	-20	-29	56	76
	Room		54	73
	1500	815	58	79

Average Impact Strength, Aged Plate*

Aging Temperature		Aging Time, Hours	Average Charpy V-Notch Impact Strength	
°F	°C		ft.lb.	Joules
SHT		-	95	129
1200	649	1000	24	33
		4000	12	16
		8000	15	20
1400	760	1000	10	14
		4000	10	14
		8000	8	11
1600	871	1000	15	20
		4000	12	16
		8000	15	20
		16000	12	16

SHT=Solution heat-treated (not aged)

* Average of four tests on 1/2-in. (12.7mm) plate from a single heat.

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OXIDATION RESISTANCE

Comparative Static Oxidation Data in Flowing Air for 1008 Hours*

Alloy	1800°F (980°C)				2000°F (1095°C)			
	Metal Loss/side,		Metal Loss +		Metal Loss/side,		Metal Loss +	
	Mils	µm	Mils	µm	Mils	µm	Mils	µm
HASTELLOY® X alloy	0.29	0.007	0.74	0.019	1.5	0.038	2.7	0.069
INCONEL® alloy 600	0.32	0.008	0.90	0.023	1.1	0.028	1.6	0.041
INCONEL alloy 601	0.53	0.013	1.3	0.033	1.2	0.031	2.6	0.06
Alloy 625	0.32	0.008	0.72	0.018	3.3	0.083	4.8	0.12
Alloy 800H	0.94	0.024	1.8	0.046	5.4	0.137	7.4	0.19

* Cycled to room temperature once a week

** CIP=Continuous Internal Penetration

INCONEL is a trademark of Inco Family of Companies

Schematic Representation of Metallographic Technique used for Evaluating Oxidation Tests

1. Metal Loss = $(A-B)/2$
2. Average Internal Penetration = C
3. Maximum Internal Penetration = D
4. Average Metal Affected = $\{(A-B)/2\} + C$
5. Maximum Metal Affected = $\{(A-B)/2\} + D$

Comparative Average Hot Corrosion Resistance*

Test Temperature		Test Period, Hours	Total Metal Affected/side, Mils (mm)					
			HASTELLOY® X alloy		HASTELLOY S alloy		HAYNES® 188 alloy	
°F	°C		Mils	mm	Mils	mm	Mils	mm
1650	900	200	3.0	0.8	2.7	0.07	2.1	0.05
1650	900	1000	6.8	0.17	7.5	0.19	3.7	0.09

* All tests performed by exposure to the combustion products of No. 2 fuel oil (0.4 percent sulfur) and 5 ppr of sea salt. Gas velocity over samples was 13 ft./sec. (4m/s). Thermal shock frequency was one/hour.

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CARBURIZATION RESISTANCE

Tests were performed in a carburizing environment with an inlet gas mixture (volume %) of 5.0% H₂, 5.0% CO, 5.0% CH₄ and the balance argon. The calculated oxygen potential and carbon activity at 1800F (980C) were 9×10^{-22} atm. and 1.0, respectively.

The results are presented in terms of the mass of carbon pickup per unit area, which was obtained from the equation $M = C (W/A)$ where M = the mass of carbon pickup per unit area (mg/cm²), C = difference in carbon (weight fraction) before and after exposure, W = weight of the unexposed specimen (mg) and A = surface area of the specimen exposed to the test environment (cm²).

Comparative Carburization Resistance at 1800F (980C) for 55 Hours

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HEALTH AND SAFETY INFORMATION

Welding can be a safe occupation. Those in the welding industry, however, should be aware of the potential hazards associated with welding fumes, gases, radiation, electric shock, heat, eye injuries, burns, etc. Also, local, municipal, state, and federal regulations (such as those issued by OSHA) relative to welding and cutting processes should be considered.

Nickel-, cobalt, and iron-base alloy products may contain, in varying concentrations, the following elemental constituents; aluminum, cobalt, chromium, copper, iron, manganese, molybdenum, nickel and tungsten. For specific concentrations of these and other elements present, refer to the Material Safety Data Sheets (MSDS) available from Haynes International, Inc.

Inhalation of metal dust or fumes generated from welding, cutting, grinding, melting, or dross handling of these alloys may cause adverse health effects such as reduced lung function, nasal and mucous membrane irritation. Exposure to dust or fumes which may be generated in working with these alloys may also cause eye irritation, skin rash and effects on other organ systems. The operation and maintenance of welding and cutting equipment should conform to the provisions of American National Standard (ANSI/AWS Z49.1, "Safety in Welding and Cutting". Attention is especially called to Section 4 (Protection of Personnel) and 5 (Health Protection and Ventilation) of ANSI/AWS Z49.1. Mechanical ventilation is advisable and, under certain conditions such as a very confined space, is necessary during welding or cutting operations, or both, to prevent possible exposure to hazardous fumes, gases, or dust that may occur.

Properties Data:

The data and information in this publication are based on work conducted principally by Haynes International, Inc. and occasionally supplemented by information from the open literature, and are believed to be reliable. However, we do not make any warranty or assume any legal liability or responsibility for its accuracy, completeness or usefulness, nor do we represent that its use would not infringe upon private rights. Any suggestions as to uses and applications for specific alloys are opinions only and Haynes International, Inc. makes no warranty of results to be obtained in any particular situation. For specific concentrations of elements present in a particular product and a discussion of the potential health effects thereof, refer to the Material Safety Data Sheet supplied by Haynes International, Inc.

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316/316L

UNS S31600 AND UNS S31603

Type 316 is an austenitic chromium-nickel stainless steel containing molybdenum. This addition increases general corrosion resistance, improves resistance to pitting from chloride ion solutions, and provides increased strength at elevated temperatures. Properties are similar to those of Type 304 except that this alloy is somewhat stronger at elevated temperatures. Corrosion resistance is improved, particularly against sulfuric, hydrochloric, acetic, formic and tartaric acids; acid sulfates and alkaline chlorides.

Type 316L is an extra-low carbon version of Type 316 that minimizes harmful carbide precipitation due to welding.

Typical uses include exhaust manifolds, furnace parts, heat exchangers, jet engine parts, pharmaceutical and photographic equipment, valve and pump trim, chemical equipment, digesters, tanks, evaporators, pulp, paper and textile processing equipment, parts exposed to marine atmospheres and tubing. Type 316L is used extensively for weldments where its immunity to carbide precipitation due to welding assures optimum corrosion resistance.

COMPOSITION

	Type 316 %	Type 316L %
Carbon	0.08 max.	0.03 max.
Manganese	2.00 max.	2.00 max.
Phosphorus	0.045 max.	0.045 max.
Sulfur	0.030 max.	0.03 max.
Silicon	0.75 max.	0.75 max.
Chromium	16.00 - 18.00	16.00 - 18.00
Nickel	10.00 - 14.00	10.00 - 14.00
Molybdenum	2.00 - 3.00	2.00 - 3.00
Nitrogen	0.10 max.	0.10 max.
Iron	Balance	Balance

AVAILABLE FORMS

AK Steel produces Types 316 and 316L Stainless Steels in thicknesses from 0.01" to 0.25" (0.25 to 6.35 mm) max. and widths up to 48" (1219 mm). For other thicknesses and widths, inquire.

MECHANICAL PROPERTIES

Typical Room Temperature Properties

	UTS ksi (MPa)	0.2% YS ksi (MPa)	Elongation % in 2" (50.8 mm)	Hardness Rockwell
Type 316	84 (579)	42 (290)	50	B79
Type 316L	81 (558)	42 (290)	50	B79

316/316L-S-11-01-99

AK STEEL 316/316L STAINLESS STEEL DATA SHEET

SPECIFICATIONS

Types 316 and 316L Stainless Steel sheet and strip are covered by the following specifications:

Type 316	Type 316L
AMS 5524	AMS 5507
ASTM A 240	ASTM A 240
ASTM A 666	ASTM A 666

PHYSICAL PROPERTIES

Density, 0.29 lbs/in³
7.99 g/cm³

Electrical Resistivity, microhm-in
(microhm-cm) 68°F (20°C) – 29.4 (74)

Specific Heat, BTU/lb/°F (kJ/kg • K)
32 - 212°F (0-100°C) – 0.12 (0.50)

Thermal Conductivity, BTU/hr/ft²/ft/°F
(W/m • K)
at 212°F (100°C) – 9.4 (16.2)
at 932°F (500°C) – 12.4 (21.4)

Modulus of Elasticity, ksi (MPa)
28.0 x 10³ (193 x 10³) in tension
11.2 x 10³ (77 x 10³) in torsion

Mean Coefficient of Thermal Expansion,
in/in/°F (µm/m • K)

32 - 212°F (0 - 100°C) – 8.9 x 10⁻⁶ (16.0)
32 - 600°F (0 - 315°C) – 9.0 x 10⁻⁶ (16.2)
32 - 1000°F (0 - 538°C) – 9.7 x 10⁻⁶ (17.5)
32 - 1200°F (0 - 649°C) – 10.3 x 10⁻⁶ (18.5)
32 - 1500°F (0 - 871°C) – 11.1 x 10⁻⁶ (19.9)

Magnetic Permeability, H = 200
Oersteds, Annealed – 1.02 max.

Melting Range, °F (°C) – 2500 - 2550
(1371 - 1399)

CORROSION RESISTANCE

Types 316 and 316L Stainless Steels exhibit better corrosion resistance than Type 304. They provide excellent pitting resistance and good resistance to most chemicals involved in the paper, textile and photographic industries.

HEAT TREATMENTS

Types 316 and 316L are non-hardenable by heat treatment.

Annealing: Heat to 1900 - 2100°F
(1038 - 1149°C), then rapidly quench.

FORMABILITY

Types 316 and 316L can be readily formed and drawn.

WELDABILITY

The austenitic class of stainless steels is generally considered to be weldable by the common fusion and resistance techniques. Special consideration is required to avoid weld "hot cracking" by assuring formation of ferrite in the weld deposit. These particular alloys are generally considered to have poorer weldability than Types 304 and 304L. A major difference is the higher nickel con-

tent for these alloys which requires slower arc welding speed and more care to avoid hot cracking. When a weld filler is needed, AWS E/ER 316L and 16-8-2 are most often specified. Types 316 and its low-carbon "L" version are well known in reference literature and more information can be obtained in this way.

METRIC CONVERSION

Data in this publication are presented in U.S. customary units. Approximate metric equivalents may be obtained by performing the following calculations:

Length (inches to millimeters) –
Multiply by 25.4

Strength (ksi to megapascals or
meganeutons per square meter) –
Multiply by 6.8948

The information and data in this product data sheet are accurate to the best of our knowledge and belief, but are intended for general information only. Applications suggested for the materials are described only to help readers make their own evaluations and decisions, and are neither guarantees nor to be construed as express or implied warranties of suitability for these or other applications.

Data referring to mechanical properties and chemical analyses are the result of tests performed on specimens obtained from specific locations with prescribed sampling procedures; any warranty thereof is limited to the values obtained at such locations and by such procedures. There is no warranty with respect to values of the materials at other locations.

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Smallest Physical Size

DESIGN FEATURES

- High energy efficiency
- One-piece, compact construction
- No whirl vanes or internal parts
- 1/8" or 1/4" male connection
- 100-mesh screen or 10 micron paper filter optional

SPRAY CHARACTERISTICS

- Finest fog of any direct pressure nozzle
- Produces high percentage of droplets under 50 microns

Spray pattern: Cone-shaped Fog

Spray angle: 90°. For best 90° pattern, operate nozzle at or above 4 bar

Flow rates: 0.043 to 5.34 l/min

Metal

Fog

Male

Fog Pattern

PJ Flow Rates and Dimensions

Impingement, 90° Spray Angle, 1/8" or 1/4" Pipe Sizes, BSP or NPT

Male Pipe Size	Nozzle Number	K Factor	LITERS PER MINUTE @ BAR								Approx. Orifice Dia. (mm)	Approx. Cov. D (mm)	Approx. Spray Height H (mm)	Pipe Size	Dim. (mm)		Wt. (g) Metal
			2 bar	3 bar	5 bar	10 bar	20 bar	30 bar	50 bar	70 bar					A	B	
1/8	PJ6	0.0137				0.043	0.061	0.075	0.097	0.114	0.152	203	103	1/8	19.1	11.1	7
	PJ8	0.0259			0.058	0.082	0.116	0.142	0.183	0.217	0.203	254	127				
	PJ10	0.0387		0.067	0.087	0.123	0.173	0.212	0.274	0.324	0.254	254	127				
	PJ12	0.0524		0.091	0.117	0.166	0.234	0.287	0.371	0.439	0.305	254	127				
	PJ15	0.0843	0.119	0.146	0.189	0.267	0.377	0.462	0.596	0.705	0.381	254	127				
OR	PJ20	0.153	0.216	0.264	0.341	0.483	0.683	0.836	1.08	1.28	0.508	310	155	1/4	24.6	14.2	
	PJ24	0.228	0.322	0.395	0.510	0.721	1.02	1.25	1.61	1.91	0.610	400	200				
	PJ28	0.296	0.419	0.513	0.662	0.937	1.32	1.62	2.09	2.48	0.711	460	230				
1/4	PJ32	0.410	0.580	0.710	0.917	1.297	1.83	2.25	2.90	3.43	0.813	560	280	1/4	24.6	14.2	
	PJ40	0.638	0.902	1.11	1.43	2.02	2.85	3.49	4.51	5.34	1.02	610	305				

$$\text{Flow Rate (l/min)} = K \sqrt{\text{bar}}$$

Standard Materials: Brass, 303 Stainless Steel and 316 Stainless Steel. See chart on page 21 for complete list.

Appendix F

Engineering Drawings

BILL OF MATERIALS		
ITEM	DESCRIPTION	QTY
GT1340-F-001-001-01	COMBUSTOR LINER MAIN	1
GT1340-F-001-002-01	COMBUSTOR LINER NOZZLE	1
GT1340-F-002-003-01	MIXER INNER SWIRLER	1
GT1340-F-002-004-01	MIXER OUTER SWIRLER	1
GT1340-F-002-005-01	MIXER TUBE	1
GT1340-F-002-006-01	COMBUSTOR SWIRLER	1
GT1340-F-003-007-01	CASING PART 1	1
GT1340-F-003-022-01	INJECTOR MAIN BASE	1
GT1340-F-003-023-01	INJECTOR SEC BASE	4
GT1340-F-004-009-01	CHAMPION IGNITER P/N. CH34168	1
GT1340-F-005-010-01	INJECTOR MOUNT PLATE	1
GT1340-F-005-011-01	SWAGELOK SS-4-TSW-1-4	1
GT1340-F-005-012-01	SWAGELOK SS-T4-S-049-20	1
GT1340-F-005-013-01	SWAGELOK SS-4-TSW-7-4	1
GT1340-F-005-014-01	BETE PJ24, 1/4IN NPT STAINLESS	1
GT1340-F-006-015-01	INJECTOR MOUNT PLATE	4
GT1340-F-006-016-01	SWAGELOK SS-4-TSW-1-4	4
GT1340-F-006-017-01	SWAGELOK SS-T4-S-049-20	4
GT1340-F-006-018-01	SWAGELOK SS-4-TSW-7-4	4
GT1340-F-006-019-01	BETE PJ24, 1/4IN NPT STAINLESS	4
GT1340-F-007-020-01	HEX SOCKET HEAD CAP SCREW, 1/4"-28 L3/8"	25
GT1340-F-007-021-01	HEX SOCKET HEAD CAP SCREW, 1/4"-28 L7/8"	20
GT1340-F-008-008-01	CASING PART 2	1
GT1340-F-008-024-01	CASING INLET BEND	1
GT1340-F-008-025-01	INLET DIFFUSER	1
GT1340-F-008-026-01	INLET RING	1
GT1340-F-008-027-01	IGNITER MOUNT BASE	1
GT1340-F-008-028-01	LINER SUPPORT	5
GT1340-F-008-029-01	INNER BEND GUIDE VANE	1
GT1340-F-008-030-01	OUTER BEND GUIDE VANE	1

REV	DESCRIPTION	DATE	INITIAL
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UNLESS OTHERWISE STATED ALL DIMENSIONS IN MILLIMETERS TOLERANCES LINEAR: ANGULAR:		BOM	
		CARLETON UNIVERSITY DEPT. OF MECH AND AERO ENGINEERING	
MATERIAL	SIZE B	ASSEM	PART NO GT1340-F-000-000-01
FINISH			SHEET 1 of 2

ASSEMBLY BOM				
ITEM		TYPE	DESCRIPTION	QTY
1	GT1340-F-001-000-01	ASSEMBLY	LINER ASSEMBLY	1
2	GT1340-F-002-000-01	ASSEMBLY	PREMIXER ASSEMBLY	1
3	GT1340-F-003-000-01	ASSEMBLY	CASING ASSEMBLY PART 1	1
4	GT1340-F-008-000-01	ASSEMBLY	CASING PART 2 ASSEMBLY	1
5	GT1340-F-004-009-01	PART	CHAMPION IGNITER P/N. CH34168	1
6	GT1340-F-005-000-01	ASSEMBLY	INJECTOR ASSEMBLY	1
7	GT1340-F-006-000-01	ASSEMBLY	INJECTOR ASSEMBLY	4
8	GT1340-F-007-020-01	PART	HEX SOCKET HEAD CAP SCREW, 1/4"-28 L3/8"	25
9	GT1340-F-007-021-01	PART	HEX SOCKET HEAD CAP SCREW, 1/4"-28 L7/8"	20

REV	DESCRIPTION	DATE	INITIAL
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UNLESS OTHERWISE STATED ALL DIMENSIONS IN MILLIMETERS TOLERANCES LINEAR: ANGULAR:	COMBUSTOR OVERALL ASSEMBLY		
	CARLETON UNIVERSITY DEPT. OF MECH AND AERO ENGINEERING		
MATERIAL	SIZE B	ASSEM	PART NO GT1340-F-000-000-01
FINISH			SHEET 2 of 2

SCALE 0.100

ASSEMBLY BOM				
	ITEM	TYPE	DESCRIPTION	QTY
1	GT1340-F-001-001-01	PART	COMBUSTOR LINER MAIN	1
2	GT1340-F-001-002-01	PART	COMBUSTOR LINER NOZZLE	1

REV	DESCRIPTION	DATE	INITIAL
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YSZ TBC 0.5 MM THICK APPLIED TO INNER SURFACE

BUTT WELD

SECTION A-A
SCALE 0.200

SCALE 0.200

UNLESS OTHERWISE STATED ALL DIMENSIONS IN MILLIMETERS TOLERANCES LINEAR: ANGULAR:	LINER ASSEMBLY		
	CARLETON UNIVERSITY DEPT. OF MECH AND AERO ENGINEERING		
MATERIAL	SIZE B	ASSEM	PART NO GT1340-F-001-000-01
FINISH			SHEET 1 of 1

MATERIAL	HASTELLOY X	SIZE	B	PART	PART NO	GT1340-F-001-001-01
FINISH	MACHINED					SHEET 1 of 2

ALL DIM AS NORMAL

UNLESS OTHERWISE STATED ALL DIMENSIONS IN MILLIMETERS TOLERANCES LINEAR: ANGULAR:		COMBUSTOR LINER NOZZLE		
		CARLETON UNIVERSITY DEPT. OF MECH AND AERO ENGINEERING		
MATERIAL	HASTELLOY X	SIZE	PART	PART NO
FINISH	MACHINED	B		GT1340-F-001-002-01
				SHEET 1 of 1

ASSEMBLY BOM				
	ITEM	TYPE	DESCRIPTION	QTY
1	GT1340-F-002-003-01	PART	MIXER INNER SWIRLER	1
2	GT1340-F-002-004-01	PART	MIXER OUTER SWIRLER	1
3	GT1340-F-002-005-01	PART	MIXER TUBE	1
4	GT1340-F-002-006-01	PART	COMBUSTOR SWIRLER	1

UNLESS OTHERWISE STATED ALL DIMENSIONS IN MILLIMETERS TOLERANCES LINEAR: ANGULAR:	PREMIXER ASSEMBLY			
	CARLETON UNIVERSITY DEPT. OF MECH AND AERO ENGINEERING			
MATERIAL	SIZE B	ASSEM	PART NO	GT1340-F-002-000-01
FINISH				SHEET 1 of 2

REV	DESCRIPTION	DATE	INITIAL
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SECTION A-A
SCALE 0.500

ALL DIM AS NORMAL

UNLESS OTHERWISE STATED ALL DIMENSIONS IN MILLIMETERS TOLERANCES LINEAR: ANGULAR:	PREMIXER ASSEMBLY		
	CARLETON UNIVERSITY DEPT. OF MECH AND AERO ENGINEERING		
MATERIAL	SIZE B	ASSEM	PART NO GT1340-F-002-000-01
FINISH			SHEET 2 of 2

REV	DESCRIPTION	DATE	INITIAL

SCALE 1.500

ALL DIM ± 0.025 MM UNLESS OTHERWISE SPECIFIED

UNLESS OTHERWISE STATED ALL DIMENSIONS IN MILLIMETERS		MIXER INNER SWIRLER	
TOLERANCES		CARLETON UNIVERSITY	
LINEAR:		DEPT. OF MECH AND AERO ENGINEERING	
ANGULAR:		SIZE B	PART NO
MATERIAL		STAINLESS	GT1340-F-002-003-01
FINISH		MACHINED	SCALE 1.500
		SHEET 1 of 1

SECTION A-A

8 BLADES EQUALLY SPACED

REV	DESCRIPTION	DATE	INITIAL
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SCALE 0.750

ALL DIM ± 0.025 MM UNLESS OTHERWISE SPECIFIED

UNLESS OTHERWISE STATED ALL DIMENSIONS IN MILLIMETERS TOLERANCES LINEAR: ANGULAR:	MIXER OUTER SWIRLER			
	CARLETON UNIVERSITY DEPT. OF MECH AND AERO ENGINEERING			
MATERIAL STAINLESS	SIZE B	PART	PART NO GT1340-F-002-004-01	
FINISH MACHINED		SCALE 0.750	SHEET 1 of 1

Ø 18
4 HOLES
EQUALLY SPACED

SCALE 0.200

ALL DIM AS NORMAL

UNLESS OTHERWISE STATED ALL DIMENSIONS IN MILLIMETERS TOLERANCES LINEAR: ANGULAR:		MIXER TUBE			
		CARLETON UNIVERSITY DEPT. OF MECH AND AERO ENGINEERING			
MATERIAL	STAINLESS	SIZE	PART	PART NO	GT1340-F-002-005-01
FINISH	MACHINED	B			
					SHEET 1 of 1

REV	DESCRIPTION	DATE	INITIAL
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SCALE 0.750

ALL DIM ± 0.025 MM UNLESS OTHERWISE SPECIFIED

UNLESS OTHERWISE STATED ALL DIMENSIONS IN MILLIMETERS TOLERANCES LINEAR: ANGULAR:		COMBUSTOR SWIRLER	
		CARLETON UNIVERSITY DEPT. OF MECH AND AERO ENGINEERING	
MATERIAL	STAINLESS	SIZE	PART
FINISH	MACHINED	B	NO
		PART NO	GT1340-F-002-006-01
		SCALE	0.750
		SHEET 1 of 1	

ASSEMBLY BOM			
ITEM	TYPE	DESCRIPTION	QTY
1	GT1340-F-003-007-01	PART CASING PART 1	1
2	GT1340-F-003-022-01	PART INJECTOR MAIN BASE	1
3	GT1340-F-003-023-01	PART INJECTOR SEC BASE	4

REV	DESCRIPTION	DATE	INITIAL
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FILLET WELD

SCALE 0.200

UNLESS OTHERWISE STATED ALL DIMENSIONS IN MILLIMETERS		CASING ASSEMBLY PART 1	
TOLERANCES		CARLETON UNIVERSITY	
ANGULAR:		DEPT. OF MECH AND AERO ENGINEERING	
MATERIAL		SIZE	PART
FINISH		B	NO
		ASSEM	GT1340-F-003-000-01
		SHEET 1 of 1

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SECTION C-C
SCALE 0.300

1/4-28 UNF - 2B CLEAR
F DRILL (6.53) THRU - (20) HOLE

20 HOLES EQUALLY SPACED

REV	DESCRIPTION	DATE	INITIAL
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UNLESS OTHERWISE STATED ALL DIMENSIONS IN MILLIMETERS TOLERANCES LINEAR: ANGULAR:			
CARLETON UNIVERSITY DEPT. OF MECH AND AERO ENGINEERING			
MATERIAL	SIZE B	PART	PART NO GT1340-F-003-007-01
FINISH			SHEET 2 of 2

REV	DESCRIPTION	DATE	INITIAL
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UNLESS OTHERWISE
STATED ALL DIMENSIONS
IN MILLIMETERS
TOLERANCES
LINEAR:
ANGULAR:

MATERIAL STAINLESS
FINISH MACHINED

INJECTOR MAIN BASE

CARLETON UNIVERSITY
DEPT. OF MECH AND AERO ENGINEERING

SIZE B PART PART NO GT1340-F-003-022-01

SHEET 1 of 1

SCALE 1.000

SECTION B-B

ALL DIM ± 0.13 MM UNLESS OTHERWISE SPECIFIED

UNLESS OTHERWISE STATED ALL DIMENSIONS IN MILLIMETERS TOLERANCES LINEAR: ANGULAR:	INJECTOR SEC BASE			
	CARLETON UNIVERSITY DEPT. OF MECH AND AERO ENGINEERING			
MATERIAL	STAINLESS	SIZE	PART	PART NO GT1340-F-003-023-01
FINISH	MACHINED	B		
				SHEET 1 of 1

ASSEMBLY BOM				
	ITEM	TYPE	DESCRIPTION	QTY
1	GT1340-F-005-010-01	PART	INJECTOR MOUNT PLATE	1
2	GT1340-F-005-011-01	PART	SWAGELOK SS-4-TSW-1-4	1
3	GT1340-F-005-012-01	PART	SWAGELOK SS-T4-S-049-20	1
4	GT1340-F-005-013-01	PART	SWAGELOK SS-4-TSW-7-4	1
5	GT1340-F-005-014-01	PART	BETE PJ24, 1/4IN NPT STAINLESS	1

REV	DESCRIPTION	DATE	INITIAL
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UNLESS OTHERWISE STATED ALL DIMENSIONS IN MILLIMETERS TOLERANCES LINEAR: ANGULAR:				INJECTOR ASSEMBLY	
				CARLETON UNIVERSITY DEPT. OF MECH AND AERO ENGINEERING	
MATERIAL	SIZE	ASSEM	PART NO	GT1340-F-005-000-01	
FINISH	B			SHEET 1 of 1

REV	DESCRIPTION	DATE	INITIAL
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SCALE 1.500

UNLESS OTHERWISE STATED ALL DIMENSIONS IN MILLIMETERS TOLERANCES LINEAR: ANGULAR:				INJECTOR MOUNT PLATE			
MATERIAL STAINLESS				CARLETON UNIVERSITY DEPT. OF MECH AND AERO ENGINEERING			
FINISH MACHINED		SIZE B	PART	PART NO	GT1340-F-005-010-01		
				SCALE	1.500	SHEET 1 of 1	

REV	DESCRIPTION	DATE	INITIAL
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SECTION A-A
SCALE 3.000

SCALE 2.000

ALL DIMENSIONS IN INCHES

ALL DIM AS NORMAL

UNLESS OTHERWISE STATED ALL DIMENSIONS IN MILLIMETERS TOLERANCES LINEAR: ANGULAR:	SWAGELOK SS-T4-S-049-20		
	CARLETON UNIVERSITY DEPT. OF MECH AND AERO ENGINEERING		
MATERIAL FINISH	STAINLESS	SIZE B	PART NO
		GT1340-F-005-012-01	
			SHEET 1 of 1

REV	DESCRIPTION	DATE	INITIAL
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ASSEMBLY BOM

	ITEM	TYPE	DESCRIPTION	QTY
1	GT1340-F-006-015-01	PART	INJECTOR MOUNT PLATE	1
2	GT1340-F-006-016-01	PART	SWAGELOK SS-4-TSW-1-4	1
3	GT1340-F-006-017-01	PART	SWAGELOK SS-T4-S-049-20	1
4	GT1340-F-006-018-01	PART	SWAGELOK SS-4-TSW-7-4	1
5	GT1340-F-006-019-01	PART	BETE PJ24, 1/4IN NPT STAINLESS	1

SECTION A-A
SCALE 1.500

SCALE 1.000

UNLESS OTHERWISE STATED ALL DIMENSIONS IN MILLIMETERS TOLERANCES LINEAR: ANGULAR:	INJECTOR ASSEMBLY		
	CARLETON UNIVERSITY DEPT. OF MECH AND AERO ENGINEERING		
MATERIAL	SIZE B	ASSEM	PART NO GT1340-F-006-000-01
FINISH			SHEET 1 of 1

REV	DESCRIPTION	DATE	INITIAL
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SCALE 1.500

UNLESS OTHERWISE STATED ALL DIMENSIONS IN MILLIMETERS TOLERANCES LINEAR: ANGULAR:				INJECTOR MOUNT PLATE	
MATERIAL STAINLESS				CARLETON UNIVERSITY DEPT. OF MECH AND AERO ENGINEERING	
FINISH MACHINED		SIZE B	PART	PART NO	GT1340-F-006-015-01
		SCALE 1.500		SHEET 1 of 1	

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REV	DESCRIPTION	DATE	INITIAL
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SECTION A-A
SCALE 3.000

SCALE 2.000

ALL DIMENSIONS IN INCHES

ALL DIM AS NORMAL

UNLESS OTHERWISE STATED ALL DIMENSIONS IN MILLIMETERS TOLERANCES LINEAR: ANGULAR:	SWAGELOK SS-T4-S-049-20		
	CARLETON UNIVERSITY DEPT. OF MECH AND AERO ENGINEERING		
MATERIAL STAINLESS	SIZE B	PART	PART NO GT1340-F-006-017-01
FINISH			SHEET 1 of 1

ASSEMBLY BOM				
ITEM		TYPE	DESCRIPTION	QTY
1	GT1340-F-008-008-01	PART	CASING PART 2	1
2	GT1340-F-008-024-01	PART	CASING INLET BEND	1
3	GT1340-F-008-025-01	PART	INLET DIFFUSER	1
4	GT1340-F-008-026-01	PART	INLET RING	1
5	GT1340-F-008-027-01	PART	IGNITER MOUNT BASE	1
6	GT1340-F-008-028-01	PART	LINER SUPPORT	5
7	GT1340-F-008-029-01	PART	INNER BEND GUIDE VANE	1
8	GT1340-F-008-030-01	PART	OUTER BEND GUIDE VANE	1

UNLESS OTHERWISE STATED ALL DIMENSIONS IN MILLIMETERS TOLERANCES LINEAR: ANGULAR:	CASING PART 2 ASSEMBLY			
	CARLETON UNIVERSITY DEPT. OF MECH AND AERO ENGINEERING			
MATERIAL	SIZE B	ASSEM	PART NO	GT1340-F-008-000-01
FINISH				SHEET 1 of 2

5 PIECES EQUALLY SPACED

ALL DIM AS NORMAL

REV	DESCRIPTION	DATE	INITIAL
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75

SECTION A-A
SCALE 0.250

UNLESS OTHERWISE
STATED ALL DIMENSIONS
IN MILLIMETERS
TOLERANCES
LINEAR:
ANGULAR:

MATERIAL

FINISH

CARLETON UNIVERSITY
DEPT. OF MECH AND AERO ENGINEERING

SIZE
B

ASSEM

PART
NO

GT1340-F-008-000-01

SHEET 2 of 2

REV	DESCRIPTION	DATE	INITIAL

SECTION A-A

SCALE 0.750

ALL DIM AS NORMAL

UNLESS OTHERWISE STATED ALL DIMENSIONS IN MILLIMETERS TOLERANCES LINEAR: ANGULAR:		INLET RING			
MATERIAL - STAINLESS		SIZE B		PART NO	PART GT1340-F-008-026-01
RING					SHEET 1 of 1

REV	DESCRIPTION	DATE	INITIAL
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R266.305

ALL DIM AS NORMAL

UNLESS OTHERWISE STATED ALL DIMENSIONS IN MILLIMETERS TOLERANCES LINEAR: ANGULAR:	IGNITER MOUNT BASE			
	CARLETON UNIVERSITY DEPT. OF MECH AND AERO ENGINEERING			
MATERIAL STAINLESS	SIZE B	PART	PART NO	GT1340-F-008-027-01
FINISH MACHINED				SHEET 1 of 1

REV	DESCRIPTION	DATE	INITIAL
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ALL DIM AS NORMAL

UNLESS OTHERWISE STATED ALL DIMENSIONS IN MILLIMETERS TOLERANCES LINEAR: ANGULAR:	OUTER BEND GUIDE VANE			
	CARLETON UNIVERSITY DEPT. OF MECH AND AERO ENGINEERING			
MATERIAL	STAINLESS	SIZE B	PART	PART NO GT1340-F-008-030-01
FINISH	MACHINED			SHEET 1 of 1