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**GAS TURBINE ENGINE DURABILITY IMPACTS OF HIGH FUEL-AIR RATIO COMBUSTORS
PART 1: POTENTIAL FOR SECONDARY COMBUSTION OF PARTIALLY REACTED FUEL**

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ABSTRACT

Demand for greater engine efficiency and thrust-to-weight ratio has driven the production of aircraft engines with higher core temperatures and pressures. Such engines operate at higher fuel-air ratios, resulting in the potential for significant heat release through the turbine if energetic species emitted from the combustor are further oxidized. This paper outlines the magnitude and potential for turbine heat release for current and future engines. The analysis indicates that in the future, high fuel-air ratio designs may have to consider changes to cooling strategies to accommodate heat release resulting from secondary combustion.

A characteristic time methodology is developed to evaluate the chemical and fluid mechanical conditions that lead to combustion within the turbine. The local concentration of energetic emissions partly determines the potential for energy release. An energy release parameter, here defined as a maximum increase in total temperature (ΔT_t), is used to specify an upper limit on the magnitude of impact. The likelihood of such impacts relies on the convective, mixing, and chemical processes that determine the fate and transport of energetic species through the turbine. Appropriately defined Damköhler numbers (Da)—the comparative ratio of a characteristic flow time (τ_{flow}) to a characteristic chemical time (τ_{chem})—are employed to capture the macroscopic physical features controlling the flow-chemistry interactions that lead to heat release in the turbine.

Keywords: turbine, durability, performance, emissions, design

NOMENCLATURE

ΔH_t	=	change in total enthalpy
ΔT_t	=	change in total temperature
η	=	film effectiveness
η_b	=	overall combustor/burner efficiency
η_{exit}	=	local efficiency at combustor exit
η_{local}	=	turbine flow combustion efficiency local to surface
ϕ	=	overall combustor equivalence ratio

ϕ_{exit}	=	local equivalence ratio at combustor exit
ϕ_{local}	=	turbine flow equivalence ratio local to surface
ϕ_{streak}	=	initial streak equivalence ratio
σ	=	standard deviation
τ_{chem}	=	characteristic chemical time scale
τ_{mix}	=	characteristic mixing time scale
τ_{flow}	=	characteristic flow time scale
τ_{ign}	=	ignition time
τ_{react}	=	reaction time
D	=	cooling hole diameter
Da	=	Damköhler number = τ_{flow}/τ_{chem}
EI	=	emission index
FAR	=	fuel-air ratio
H^*	=	enthalpy release ratio = $\Delta H_t/H_t$
H_t	=	total enthalpy
LHV	=	lower heating value
NGV	=	nozzle guide vane
P	=	pressure
S	=	mixedness parameter = σ/ϕ
T	=	temperature
TIT	=	turbine inlet temperature
X	=	mole fraction
Y	=	mass fraction

subscripts

∞	=	freestream
a	=	adiabatic
c	=	coolant
f	=	fuel
i	=	initial
o	=	oxidizer
$T3$	=	combustor inlet
$thresh$	=	threshold
w	=	wall

1. INTRODUCTION

Demand for greater engine efficiency and thrust-to-weight ratio has driven the production of aircraft engines with higher core temperatures and pressures. Such engines operate at higher fuel-air ratios, resulting in the potential for significant heat

release through the turbine if energetic species emitted from the combustor are further oxidized. Figure 1 illustrates the processes that lead to continuing heat release in the turbine. For fuel-air ratios approaching stoichiometric, increased combustor exit concentrations of CO, OH, H₂, and O are thermodynamically favored. Falling temperatures and pressures along the turbine flow path compel these energetic species to recombine. This can raise the effective temperature of the turbine flow, impacting performance [3,4] and adversely affecting durability through augmented heat transfer to internal surfaces. Perhaps more importantly, unmixedness and inadequate residence time will raise concentrations of energetic species such as CO and HC well beyond the equilibrium limit. If these energetic species are convected close to turbine surfaces, they may interact with oxidizer injected into the gas path in the form of cooling or purge flows, leading to elevated heat flux to cooled parts.

This paper outlines the magnitude and potential for turbine heat release for current and future engines. Particular emphasis is given to interactions with cooling and purge flows. Section 2 discusses the definition and use of fluid and chemical time scales to establish the basis for the analyses presented in this paper. Section 3 details the key thermodynamic, chemical, and fluid drivers of turbine heat release, including the roots of increased energetic emissions into the turbine. Turbine flow compositions representative of current and future engines are defined. In Section 4, time scales and example parameters typical of modern turbines are evaluated to determine the conditions and processes that are important to the mode and scope of turbine heat release. The analysis outlined in this paper demonstrates that turbine heat release may be a common occurrence, even in current engines, with the distinction that high fuel-air ratio combustion systems have a greatly increased potential for significant local temperature rise on the scale of a turbine blade row. Combined with experimental heat transfer results discussed in a companion paper [5], these results further indicate that higher fuel-air ratio designs may have to consider changes to cooling strategies to accommodate heat release resulting from secondary combustion.

2. IMPORTANT PHYSICAL PARAMETERS

The likelihood and impact of turbine heat release rely on the convective, mixing, and chemical processes that determine the fate and transport of energetic species in the turbine. Appropriately defined Damköhler numbers (Da)—the comparative ratio of a characteristic flow time (τ_{flow}) to a characteristic chemical time (τ_{chem})—are employed to capture the macroscopic, physical features controlling the associated flow-chemistry interactions. In this section, important flow, chemical, and thermodynamic parameters are discussed as the basis for specifying Da .

Since we are primarily interested in assessing the impacts on turbine durability, we define a convective flow time scale, τ_{conv} , related to a component length scale, such as a blade chord. Mixing time scales, τ_{mix} , associated with both turbulent diffusion in the freestream and decay of film cooling effectiveness, are also used. Mixing can play an important mitigating role, for example by diffusing streaks of energetic species enroute through the turbine and through limiting the rate at which these species combine with oxygen rich film cooling flows.

The relevant chemical time scales are the time required for a given mixture to ignite (τ_{ign}) and the time required to complete the reaction process (τ_{react}). To the limits of the chemical mechanism used [6], chemical time definitions are intended to provide an estimate of the fastest reaction rates possible for an estimated turbine flow mixture composition. Chemical times are determined using constant pressure, adiabatic, chemical kinetic calculations for a homogenous flow with no mixing [7]. Energy released during reaction is assumed to result in temperature rise with no contribution to alterations in the flow kinetic energy. These calculations are initiated at the highest local temperature encountered during the flow interaction hypothesized, therefore assuming that the hot-side temperature of a mixing layer drives ignition and reaction [8]. This means, for example, that in situations where the core flow encounters a colder flow, such as film cooling, we use the freestream temperature to initiate the calculation.

The local concentration of energetic emissions partly

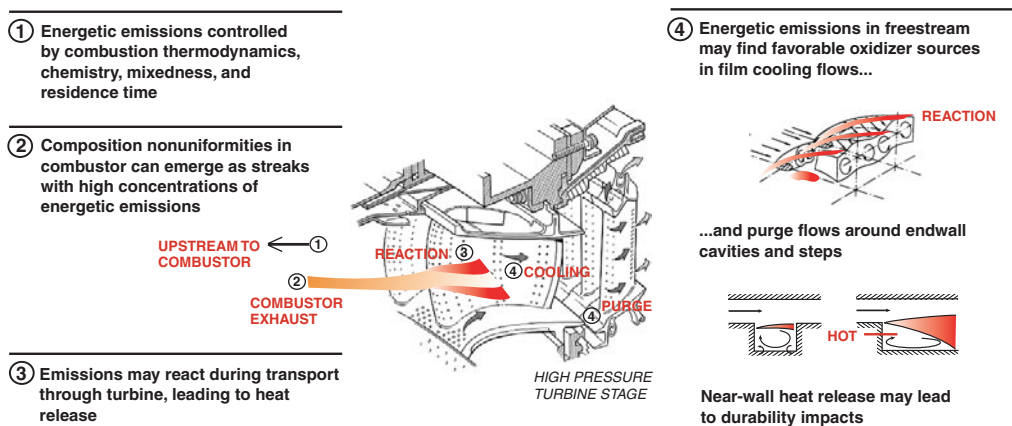


Figure 1. Phenomenology of secondary combustion in the turbine. (Original engine and film cooling graphics from [1] and [2].)

determines the potential for energy release. An energy release parameter, here defined as a maximum increase in total temperature (ΔT_t), is used to specify an upper limit on the magnitude of impact. The chemical time scales τ_{ign} and τ_{react} are specified explicitly in reference to ΔT_t . For this study, ignition is defined as the time to reach 5% of ΔT_t and the completion of reaction is defined as the time to reach 95% of ΔT_t . The value of ΔT_t , determined by a flame temperature relevant to a local, well-mixed, hypothetically isolated mixture, bounds the magnitude of heat release. As such, in this limit, we do not consider the mitigating impact of mixing on maximum temperature rise. By local we refer to a geometric scale that is much less than the one of interest. For instance, the local mixture could be that contained in a spherical volume of the scale of a cooling hole, the diameter of which is small relative to a turbine blade chord.

There are two important sets of inputs required for the definition of ΔT_t . The chemical composition of the turbine flow involved in a heat release event determines the available energy. This requires knowledge of energetic combustor emissions. So as to be pertinent to potential secondary combustion, the available energy is specified with reference to the lower heating value (LHV), again an upper limit. Thermodynamically, only part of this available energy will result in ΔT_t . Local temperature and pressures are used to specify the thermodynamic state of the mixture prior to combustion. Turbine operational point and aerodynamics determine the relevant temperatures and pressures.

The above parameters lend themselves directly to three definitions of Da . The magnitude of Da is indicative of the potential for and character of an event. In general, cases where $Da \ll 1$ are limited by reaction and cases where $Da \gg 1$ are limited by flow scales. The first definition of Da is the ratio of convective to ignition time, τ_{conv}/τ_{ign} , where τ_{conv} is calculated relative to a blade chord. If the ratio is much less than 1, the potential for any heat release to occur within the scale of the feature is minimal. In essence, the heat release is limited by chemical reaction and locally, the opportunity for surface damage is small. To understand whether heat release can occur over a physical scale relevant for turbine durability concerns, the ratio of convective to reaction time, τ_{conv}/τ_{react} , is important. If the ratio is approximately 1, heat release will occur in a distributed manner and the maximum local temperature increase may not be seen on the scale of the part. In any case, the impact on heat transfer (increased local heat flux) is related to the change in surface local temperature, but mixing scales have a key role in determining the extent of impact. The final definition of Da employed, the ratio of mixing to reaction time, τ_{mix}/τ_{react} , limits the distance to which energetic emissions can reach through the turbine by restricting the spatial scale of rich non-uniformities at the combustor exit that will not mix and burn before reaching a surface. Similarly, mixing interactions with cooling jets necessarily determine the location of the temperature rise relative to the surface.

3. ENERGETIC EMISSIONS INTO THE TURBINE

This section discusses the thermodynamic and practical drivers that determine concentrations of energetic emissions at

the combustor exit. Such emissions from the combustor are the necessary precursors to secondary combustion. Section 3.1 outlines the limits of combustor efficiency defined by thermodynamic considerations. Section 3.2 looks at the important role of combustor mixing in determining the level of energy entering the turbine. Finally, Section 3.3 outlines three turbine flow compositions that are representative of current and future engines. These mixture definitions are used in subsequent sections to determine the magnitude and potential for turbine heat release.

3.1. Thermodynamic Efficiency Limits for High Fuel-Air Ratio Combustors

As temperatures and pressures increase, higher concentrations of CO, OH, H₂, O, and other dissociated species are found at thermodynamic equilibrium. Figure 2 shows equilibrium concentrations of CO, OH, H₂, and O for aviation fuel, modeled as C₁₂H₂₃, combusted in air [9].

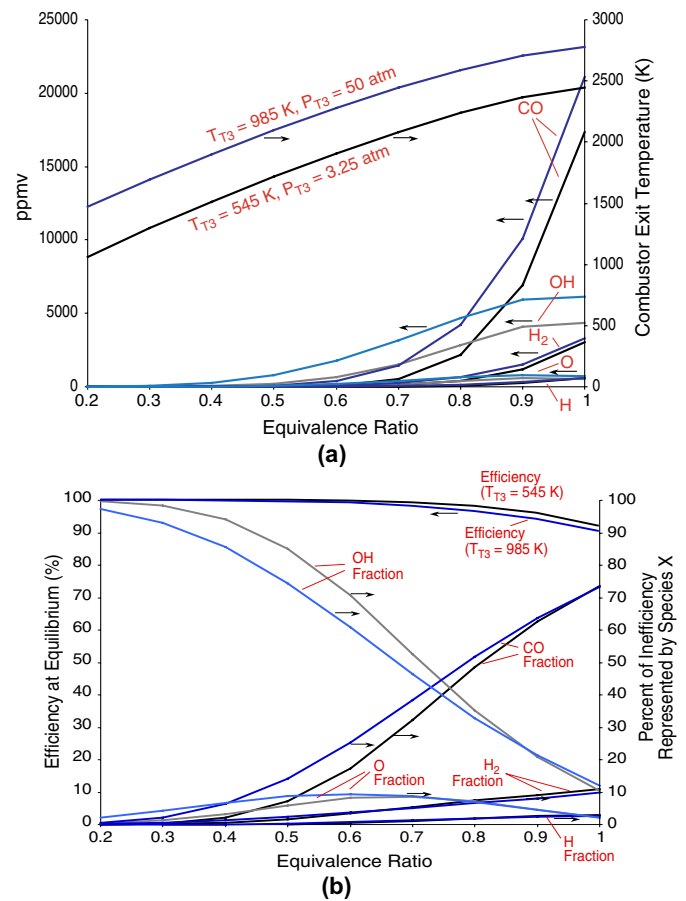


Figure 2. X_{CO} , X_{H_2} , X_{OH} , X_O , and efficiency (η_b) at constant enthalpy and pressure equilibrium for lean and stoichiometric aviation fuel-air mixtures.

Two conditions are shown in Figure 2, one characteristic of a 1970's era commercial aircraft engine at the cruise condition ($T_{T3} = 545 \text{ K}$, $P_{T3} = 3.25 \text{ atm}$) and another that might be characteristic of a 2010's era military aircraft engine at take-off

($T_{T3} = 985$ K, $P_{T3} = 50$ atm). Combustor exit temperatures resulting from a range of fuel-air ratio specifications are also shown. A lean overall equivalence ratio (ϕ) of 0.3 is typical of current era commercial combustors. Future military applications may demand operation at ϕ near 1. Also shown in Figure 2 is a measure of combustor efficiency (η_b), defined with reference to lower heating values (LHV). The relative contributions of energetic species that represent the inefficiency ($1-\eta_b$) are indicated.

Importantly, where idle has historically been the most inefficient condition, future engines will also contend with inefficiency at high power conditions. As temperature increases for a constant pressure, η_b decreases and a greater proportion of the fuel energy enters the turbine gas path. Note that not all of the equilibrium inefficiency exhibited at higher temperatures is contained in the measurable emissions of CO. Other dissociated species, particularly OH, H₂, and O, are important components of inefficiency at high temperatures. For $\phi = 1$, a significant fraction of the fuel energy, a maximum of $\sim 10\%$ for the cases shown, may be available for heat release in the turbine, more if combustion is not complete. Reducing P_{T3} from the values shown in Figure 2 would lead to a further decline in efficiency resulting from increases in equilibrium levels of dissociated species for a given turbine inlet temperature (TIT). At equilibrium, as equivalence ratios increase, CO becomes the primary source of inefficiency, replacing OH. However, as the next section will discuss, unmixedness in practical devices promotes species where oxidation is kinetically limited, such as CO and HC, to levels much greater than equilibrium such that they can become the dominant carriers of energy into the turbine.

3.2. Impacts of Combustor Mixedness and Residence Time

Historical emissions measurements indicate that combustor efficiencies for current commercial and military aircraft engines at high power conditions are typically above 99.9% [10-13]. The levels of CO and HC concentrations associated with these efficiencies are several times greater than equilibrium and inefficiencies can be higher for low power conditions such as idle. Deviations from equilibrium are due to the coupled effects of inadequate combustor residence time, less than perfectly mixed fuel and air, requirements for liner cooling (manifest in the profile factor), and tailoring of the TIT for turbine life. Current engines exhibit higher efficiency at higher power conditions and there is a general trend towards lower fractional energy in HC as power level (and thus engine temperature and pressure) is increased [10,11], primarily a result of better mixing.

Given the larger equilibrium levels of energetic emissions, higher fuel-air ratio designs will place a premium on improved mixedness to avoid turbine durability and performance problems. While there is a proportional decrease in mixedness with an increase in primary zone equivalence ratio for the same geometry [14,15], history suggests that better dilution hole placement, fuel injection, air or fuel staging, and improved liner cooling have made it possible to increase engine temperatures while maintaining combustor efficiency. But,

generally, as ϕ approaches stoichiometric levels, near perfect mixing (and thus an end to temperature profile tailoring) is required to maintain combustor efficiency near the equilibrium limit.

One important characterization of mixedness is found in the distribution of fuel-air ratio at the combustor exit. As fuel-air ratios increase and efficiencies decrease, there is a greater probability of finding a rich mixture in the turbine in addition to greater overall concentrations of energetic species. Such rich mixtures are more likely to flow further through the turbine than a well-stirred, leaner mixture because of the addition of a mixing constraint beyond the chemistry required for oxidation. This increases the likelihood that heat release may impact blade surfaces. The impact of reductions in combustor mixedness is illustrated by Figure 3 where the distribution of equivalence ratio at the combustor exit (ϕ_{exit}) is contrasted for engines with various mixedness levels and overall fuel-air ratios.

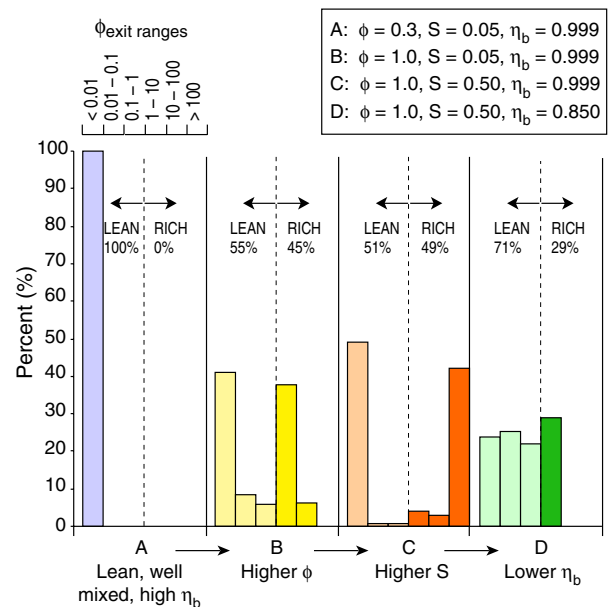


Figure 3. Impact of mixedness on fuel emissions.

In Figure 3, it is assumed that the distribution of equivalence ratios in the combustor primary zone is Gaussian in form and described by the parameter $S (= \sigma/\phi)$ [16, see also 17]. The fuel is combusted to the indicated efficiency and it is assumed for convenience that all energetic emissions emerge from the combustor as CO. Each bar in Figure 3 represents the percentage of the combustor exit area that can be characterized by ϕ_{exit} in the indicated logarithmic range. For combustors with high efficiencies, the distribution of ϕ_{exit} is narrow around a small number. This is represented by case A in Figure 3. Temperatures are lower, there is enough air to combust the injected fuel at all points in the combustor, and almost all locations at the exit are characterized by a low fuel-air ratio. The temperature profile at the combustor exit is thus a manifestation of liner cooling requirements and TIT tailoring.

As ϕ_{exit} increases to one, represented by the comparison of cases A and B in Figure 3, temperature variations at the

combustor exit would likely be a manifestation of both cooling requirements and inefficiencies. Some fluid, in this case about 45%, enters the turbine with rich equivalence ratios. As mixedness decreases, comparing case B to case C, mixtures are more likely to emerge with very high equivalence ratios—essentially slugs of partially reacted fuel. Further decreasing the efficiency, progressing from case C to case D in Figure 3, spreads the distribution of equivalence ratio more evenly across the spectrum since some of the fluid in lean regions would not be fully reacted.

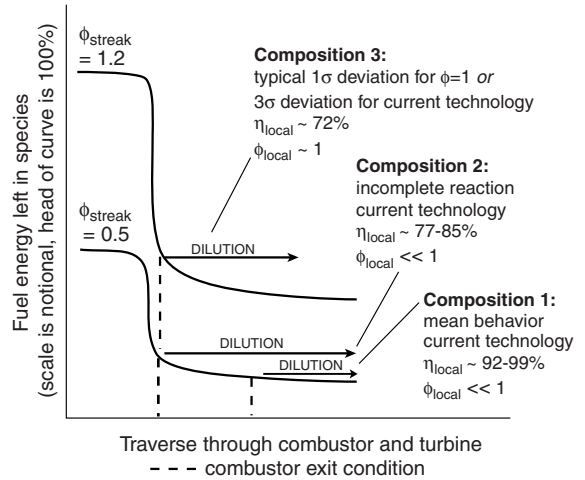
There are several spatial distributions of equivalence ratio at the combustor exit that could satisfy cases A through D of Figure 3. High equivalence ratio non-uniformities emitted into the turbine can be regarded as streaks, and the combustor exit may be characterized by several small-scale or fewer large-scale non-uniformities. In addition to spatial expressions of inefficiency, the combustion process can be intermittently inefficient due to the inherent temporal unsteadiness of the combustion process. Godin *et al.* [4] estimate that a 200 K standard deviation, about a TIT of 1544 K, is equivalent to a 0.08% drop in η_b , and sensitivity to unsteadiness increases with temperature. In contrast, an 80 K standard deviation around a 2300 K TIT equates to a 0.30% drop in η_b [4].

3.3. Representative Turbine Flow Compositions

The thermodynamic and fluid impacts of mixedness and residence time considered in the previous sections indicate that a wide range of turbine flow compositions is possible. In this section, we develop three representative cases where local speciation in the turbine is estimated for the purpose of determining the likelihood and potential magnitude of secondary combustion. For the estimate, we assume that the combustor exit composition is a good approximation for mixtures that might be found through the turbine, with the exception that local mixing with cooling or purge air may perturb the mixture composition. The appendix discusses the validity of this assumption through an analysis of the mixing of rich streaks emitted by the combustor. It is found that over the length of a turbine stage, reactive streaks larger than ~20% of span do not have time to fully mix with the surrounding flow.

Turbine flow compositions are constructed by tracking the fate of a streak of fluid, with an assumed fuel type and initial equivalence ratio (ϕ_{streak}) that reflects the state of mixedness in the combustor. To estimate the post-combustor composition, equilibrium is calculated for a specified temperature and pressure relevant to the local site, followed by a compositional perturbation of CO, HC, and H₂ to enforce variable partitions among kinetically limited species. Thus, compositions of OH, O, and other energetic species are assumed to be at their equilibrium levels. The actual mole or mass fractions of these species are therefore by-products of these specifications. Because it is less likely that high carbon number species would emerge from the combustor, all HC was assumed to be CH₄, chosen also for its inhibitory impact on CO and H₂ oxidation and thus to specify a lower bound for kinetic rates. Because we are primarily concerned with durability impacts, we additionally perturb the mixture to account for mixing with

oxidizer associated with local cooling or purge flows to provide a worst case near-surface equivalence ratio (ϕ_{local}). This process and the range of turbine flow conditions that results are illustrated in Figure 4. Since the total available energy defines the maximum local temperature rise, CO, OH, H₂, O, and HC emissions are summed into a CO-equivalent molar quantity by energy content in Figure 4.



Note: Portion of fuel energy represented by CO/HC/H₂ specified at 99%, 85%, and 73% for C1 through C3. η_{local} further reduced with changes in local T and P .

Composition	CO-equivalent (ppmv)
1	1400 - 12000
2	21000 - 32000
3	77000 - 81500

Figure 4. Representative current and future turbine flow compositions.

The fuel-lean mixture, Composition 1, with $\phi_{streak} = 0.5$ and $\eta_{local} \sim 92-99\%$, is chosen to represent the mean performance of a well-mixed, current era combustor of high efficiency. The range of efficiency reflects the influence of different combustor exit temperatures and pressures on equilibrium concentrations of energetic species other than CO, HC, and H₂. Composition 2 reflects an increase in CO-equivalent emissions for the same current era combustor as described for Composition 1, resulting in $\eta_{local} \sim 77-85\%$. This decline in efficiency could, for example, be due to lack of adequate residence time for complete combustion. The fuel-rich mixture, Composition 3, with $\phi_{streak} = 1.2$ and $\eta_{local} \sim 72\%$, represents the additional dimension of unmixedness. This case selects, for example, the 1 σ deviation case along the upper tail of a future, stoichiometric combustor with a $S = 0.2$ exit distribution, a level of unmixedness typical of current combustors, or a greater than 3 σ deviation along the upper tail of the Composition 2, current era combustor with inadequate residence time. Thus, Composition 3 is more likely for future turbine flows as compared to current designs. The resulting mole fraction of energetic emissions for Composition 3 is in excess of 75000 ppmv equivalent CO by energy, nearly three times the stoichiometric equilibrium value. Six fractional

distributions of CO, HC, and H₂ were employed for each composition specified. For cases that include hydrogen, a 10% fractional distribution was assumed. The HC/CO fraction was varied from 0.5 to 1.0 for each case where both were included in the mixture.

4. OXIDATION OF EMISSIONS IN THE TURBINE

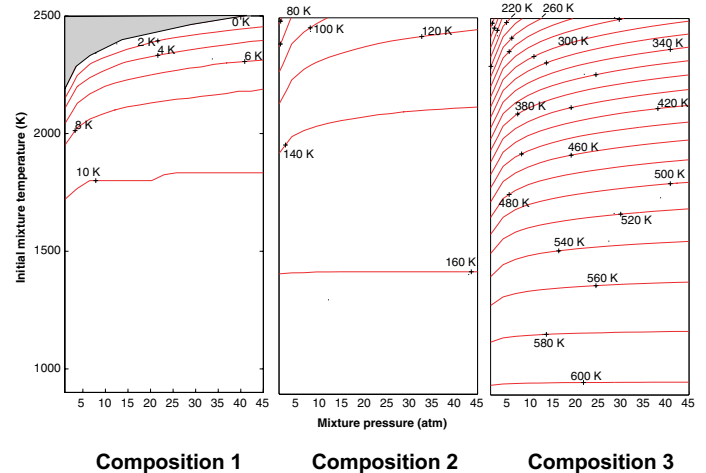
In the previous section, we proposed a reasonable range of estimates for the quantity of energetic emissions entering the turbine for current and future engines. In the high-pressure turbine (HPT), oxidative chemical interactions occur among air constituents, primary combustion products, and trace species (*e.g.* CO, H₂, NO_y, SO_x, HO_x, and O) [18,19]. These oxidative processes are quenched by rapid temperature changes through the HPT as well as local conditions. Thus, whether these species actually react within the turbine is a function of the time evolution of temperature and pressure through the turbine, local fluid mechanical effects, as well as the constituency of such emissions.

Section 4.1 places bounds on the magnitude of energy release that can occur as a result of such emissions by estimating ΔT_T for the compositional scenarios described in the previous section. However, only part of the energy resident in emissions from the combustor is recovered through the turbine. In Section 4.2, we consider the potential heat release by estimating Da . In the course of this discussion, we also discuss the importance of various factors that govern chemical time scales. For illustrative purposes, we investigate ΔT_T and Da bounds over a temperature and pressure range of 900-2500 K and 0.1-45 atm, respectively, representing the local temperatures and pressures that might be experienced within the engine over a range of operating conditions. A subset of temperatures and pressures, between approximately 1500-2100 K and approximately 10-25 atm, is employed as characteristic of the nozzle guide vane (NGV) flow through a current era advanced engine turbine.

The impact of secondary heat release on durability is dependent on the magnitude of the energy release and subsequently, the scale of augmentation in heat flux that might be experienced. Mixing processes within the energetic freestream and its interaction with oxidizer-rich cooling or purge can impact the potential, magnitude, and heat flux impact of heat release. Section 4.3 looks at mixing of freestream and local oxidizer flows to evaluate potential limitations on surface heat transfer.

4.1. Bounds on the Magnitude of Energy Release

The adiabatic flame temperature is a thermodynamic limit to the magnitude of heat released and subsequent temperature rise resulting from emissions of energetic species in the turbine. Figure 5, shows results for ΔT_T calculated in this manner for the representative turbine conditions described in Figure 4. Thermodynamic data are taken from [6]. Also shown in Figure 5 are ranges for H^* , the ratio of enthalpy change to initial enthalpy ($\Delta H/H_i$). This is an alternative interpretation of the temperature rise.



Composition	CO-equivalent (ppmv)	H^*
1	1400 - 12000	0.05% - 0.5%
2	21000 - 32000	6% - 11%
3	77000 - 81500	18% - 35%

Figure 5. Potential local temperature rise.
(Grayed area indicates thermodynamically incompatible T, P, X combinations.)

The primary driver of ΔT_T is the total local available energy or concentration of local energetic emissions. Compositions marked by higher total energetic emissions have a greater potential for larger ΔT_T . Thermodynamically, there is a generally greater dependence on initial local temperature as opposed to pressure, but pressure dependencies are important up to $P \sim 10$ atm. Although there are some compositional effects due to differences in the assumed mixing between freestream and oxidizer, such details are not as important. The local temperature rise, ΔT_T , is greater for low initial temperature and high-pressure conditions. Sensitivity to local conditions increases with greater amounts of energetic emissions because equilibrium values of CO and H₂ vary more widely over the temperature and pressure space.

Composition 3 (C3) exemplifies an upper bound for a high equivalence ratio, future engine of thermodynamically limited efficiency, and unmixedness typical of current era combustors. Over a range of temperatures and pressures characteristic of the NGV, local temperature rise can exceed 500 K, or an H^* of $\sim 35\%$. This is much higher than the approximately 10 K rise ($H^* \sim 0.45\%$) of Composition 1 (C1) for the same temperature and pressure space, which exemplifies a current, highly-efficient, well-mixed engine. Increasing unmixedness for current engines, as shown in C2, increases the potential local temperature rise to 150 K ($H^* \sim 11\%$). Generally, a combustor efficiency of less than 98.5% is required to realize a temperature rise (ΔT_T) greater than 50 K for C3 where an efficiency of less than 95% is required for C1.

4.2. Actual Energy Release Depends on Fluid and Chemical Time Scales

Faster reaction rates are associated with a greater risk that heat release will occur locally. To determine the important factors governing chemical time scales, knowledge of the speciation at the combustor exit is required. Because empirical characterizations across the range of potential combustor exit conditions are limited, the turbine flow composition cases discussed previously were used to estimate characteristic chemical time scales. Chemical times— τ_{ign} and τ_{reac} defined by the time to reach 5% and 95% of ΔT_T , respectively—were estimated using a constant pressure, adiabatic, chemical kinetic model for homogenous flow with no mixing using the GRIMech2.1 reaction set [6]. The method of determining characteristic chemical times was chosen to provide the shortest scales and thus a worst case estimate for reactions progressing in the direction of the flow. Figure 6 plots results for τ_{ign} and τ_{reac} . The height of each solid bar in Figure 6 indicates the median chemical time for a temperature and pressure range of approximately 1500-2100 K and approximately 10-25 atm characteristic of the NGV flow through a current era advanced engine turbine. The bracketed lines indicate the full range of chemical times for this temperature and pressure range.

In the absence of mixing effects (e.g. strain, turbulence), τ_{ign} is typically on the order of 0.01 ms (0.005 to 0.1 ms range). The reaction time, τ_{reac} , is typically on the order of 0.1 ms (0.01 to 0.5 ms range). Since the temperatures are high in advanced engines, τ_{ign} is predominantly the time required for requisite radical concentrations to be generated. For the initial composition of the mixtures used for the analysis, radical concentrations are set to the levels found at chemical equilibrium at the given temperature and pressure. For higher equivalence ratio engines, radical concentrations emitted from

the combustor may be greater than those assumed and subsequently, τ_{ign} may be lower.

Chemical times are overwhelmingly driven by local temperature, but for any given temperature and pressure, the range of τ_{reac} is less than an order of magnitude for the cases explored, despite the large range in turbine flow composition. For chemistry in the turbine, local reactions occur under significantly more dilute conditions than in the combustor, which may dampen the impact of concentration changes on reaction rate as compared to temperature. In Figure 6, the impact of composition is smaller than that of temperature with the most consistent perturbation related to the presence of H_2 in the mixture, which affects τ_{ign} most prominently. As equivalence ratios increase, if mixing cannot be improved, HC emissions will increase which can inhibit CO and H_2 oxidation. The inhibitory impact of CH_4 on chemical times is only evident where the fuel-air ratio is consistently set to a constant factor, which is the case for the high fuel-air ratio Composition 3. For the other composition cases, there is an increase in chemical rates with increased HC fraction, which arises because of the substitution of CH_4 for CO effectively increases the equivalence ratio of the mixture.

The nonlinear balance between the total mass of energetic emissions and how fast they are consumed through oxidation contributes to the decline and then increase in characteristic chemical times as the total fraction of energetic emissions increases. Generally, second-order reactions are involved which have rates that are geometric with species concentrations so that for a given ϕ_{streak} , chemical times will decrease with increasing fuel content. However, there is more fuel to consume and a larger relative radical concentration to develop prior to oxidation, trends that act to increase chemical times.

In Figure 7, a Damköhler number (Da) comparing a

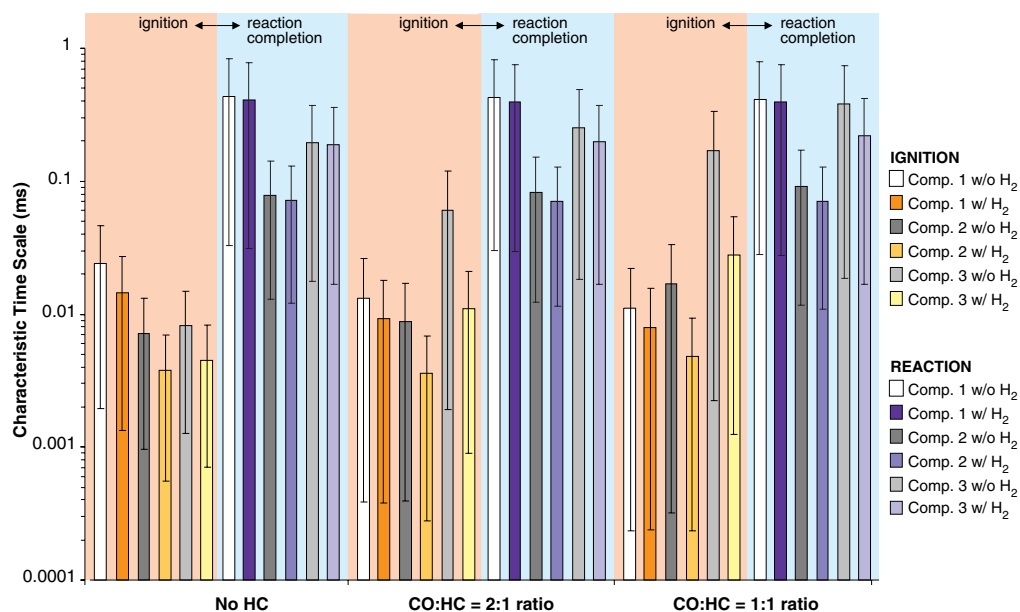


Figure 6. Characteristic chemical time scales for turbine heat release for representative flow compositions.
(Solid bars represent median values and bracketed lines indicate ranges for a typical NGV flow space for a current era advanced engine.)

convective time of 0.5 ms, typical of a turbine blade row, and the ignition time is shown for the mixture compositions outlined in Figure 3. The results are overlapped upon ΔT_T results from Figure 5. Note that this specification of Da essentially decouples the chemical and fluid scales (*e.g.* there is no dependence on temperature change in τ_{flow}). For the temperatures shown, CO kinetics are explosive over a greater range of pressures as temperatures increase, but lower pressures exhibit a significant change in rates as the explosive regime is exited, shown by the knee in the Da curves at lower temperatures. The addition of H_2 reduces this effect.

Moving from the lower left to the upper right of each temperature and pressure space, the current direction of technology development, conditions for secondary combustion become more favorable because temperatures and pressures (and thus concentrations) are increasing. Since temperatures are highest at high power conditions, take-off, climb-out, or other accelerations are the most probable time for oxidation and heat release within the turbine. For a current era engine, in which the NGV flow may be described a temperature and pressure space of 1500-2100 K and 10-25 atm, Da based on a typical blade row convective time of 0.5 ms is 10-500 for ignition and 1-25 for reaction for all cases. For situations in which residence times would be much greater, for recirculation zones for example, the total temperature and pressure space would consist of high Da for the well-mixed, adiabatic limit.

Given the magnitude of Da over the NGV temperature and pressure space, oxidation of energetic emissions due to interactions with cooling air can be fairly common even in current era turbines, provided fluid mechanical mechanisms of extinction are not in evidence. Within an individual design, the potential for secondary combustion becomes weaker as the high

and low turbines are traversed and temperatures and pressures decline. Following a streamline through the NGV, Da for reaction declines to order one, indicating that the HPT would be the section of most concern for heat release impacts. Furthermore, the magnitude of Da indicates that reactions, if begun in the NGV, may lead to the highest temperatures towards the end of the NGV or beginning of the first rotor of the HPT. For Composition 3, which represents an unmixed slug emitted into the turbine of a stoichiometric engine, realizing the full extent of reaction can have a significant impact on durability since maximum temperature increases may exceed 500 K.

4.3. Mixing Time Scales Mitigate Wall Impacts

Reactions and heat release important to durability will occur within mixing layers between freestream and coolant flows. The rapidity with which near-wall thermal layers change subsequent to coolant injection is a relevant mixing scale that has been well-documented [*e.g.* 20]. In application to the reacting coolant layer durability problem, we calculate a characteristic mixing time scale based upon the decay of adiabatic film effectiveness, η , as a surrogate for species mixing. As a time scale, this analysis essentially provides an estimate of when the cooling oxidizer would be exhausted and a maximum surface impact can be expected. The freestream and coolant are assumed to be initially separated (freestream equivalence ratio $\sim \infty$), subsequently mixing turbulently with $Le=Pr=Sc=1.0$. Since the cooling effectiveness maps the near-wall mixing state and is a non-dimensionalized scalar quantity, we replace temperatures with mass fractions (Equation 1). For the analysis, we assume that the oxidizer stream is air and that

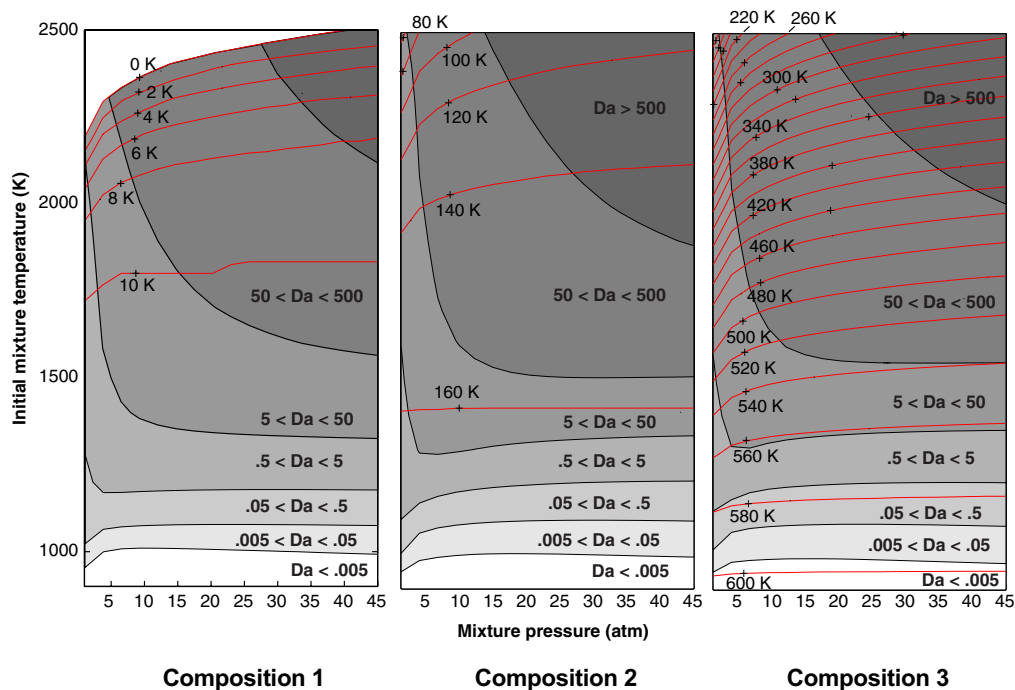


Figure 7. ΔT_T and Da comparing ignition and blade row traverse times for representative flow compositions. (Blank area indicates thermodynamically incompatible T, P, X combinations.)

the freestream is a diluted fuel where the O₂ has been exhausted.

$$\eta = \frac{(T_{ow} - T_{\infty})}{(T_c - T_{\infty})} \quad (1)$$

$$\eta = \frac{(Y_{fw} - Y_{f\infty})}{(Y_{fw,i} - Y_{f\infty})} \text{ where } Y_{fw,i} = 0 \text{ and } Y_{f\infty} = 1$$

At the wall, since a bimodal component composition has been assumed, the oxidizer and freestream fractions are directly related. The value of the effectiveness, which indicates a particular mixture of freestream and coolant at the wall, is related to a fuel-air ratio at the wall (FAR_w) which is a function of composition and concentration (Equation 2).

$$Y_{ow} = 1 - Y_{fw}$$

$$FAR_w = Y_{fw} / Y_{ow} = Y_{f\infty} (1 - \eta) / (1 - Y_{f\infty} (1 - \eta))$$

$$\eta = 1 / (FAR_w + 1) \quad (2)$$

$$\text{For threshold } \eta, \text{ set } FAR_w = FAR_{thresh} \left(Y_{O_2|o} / Y_{(HC+CO)|f} \right)$$

Here, FAR_w is set to specify a stoichiometric wall value to determine when all of the oxidizer would be mixed with the freestream.

For a set FAR_w, the threshold value of η is smaller for increased HC content, giving a longer mixing time scale. This is because the required O₂ for a combustible mixture changes as the fuel composition changes and the relative mass of fuel in freestream increases with CO content. For low concentrations of freestream energetic emissions such as C1 in Figure 4, the threshold value of η is ~ 0.01 , increasing to ~ 0.15 for C3, with a higher freestream concentration of energetic emissions. This indicates a significant mixing limitation on the depth into a cooling layer the maximum temperature rise can penetrate. For a typical cooling hole with an attached jet [e.g. 20], the threshold value of η is not reached along the centerline for distances downstream of greater than 40 cooling hole diameters (D), a scale equivalent to a turbine blade chord. Thermal perturbation of the near-wall flow would be effectively delayed, even in cases where reaction times are fast. In contrast, a lifted coolant jet could realize threshold η values at less than 10*D which indicates that maximum temperatures could be experienced at more vulnerable locations between cooling holes. Furthermore, cooling flows are three-dimensional and for typical lateral hole spacings (typically 3*D), there will be regions where the maximum temperature is reached nearer to the surface than would be supposed by centerline or average effectiveness profiles. However, in reference to Figure 7, because non-reactive, cooler air is diffused into the reacting zone, the realized temperature rise would be smaller than suggested by the local adiabatic flame temperature. If the resulting mixture temperature is low, coolant flows could quite possibly quench reactions.

5. CONCLUSION

In this paper, a characteristic time methodology was developed to evaluate the chemical and fluid mechanical conditions that can lead to secondary combustion within aircraft

turbines. Representative turbine flow compositions for current and future engines were estimated and used to determine the maximum increase in total temperature (ΔT) and therefore specify an upper limit on the magnitude of impact. These compositions were further employed to evaluate the likelihood of such impacts through the definition of Damköhler numbers (Da) appropriate to the macroscopic, physical features controlling the flow-chemistry interactions that lead to heat release in the turbine.

As temperatures and pressures increase, higher concentrations of CO, OH, H₂, O, and other dissociated species are found at thermodynamic equilibrium. As a result, 10% of the fuel energy may be available for secondary combustion, more as unmixedness in the combustor increases or if the geometry of the combustor is such that there is inadequate residence time in which to complete combustion. Thus, where idle has historically been the most inefficient condition, future aircraft engines will also contend with inefficiency at high power conditions.

Compared to a typical blade row convective time of 0.5 ms, Da based on ignition time is on the order of 10-500 for NGV flow conditions applicable to current era engines over a wide range of species conditions. Comparatively, Da based on reaction time is on the order of 1-25, and reactions, if begun at the end of the blade row, may continue into the next blade row. Thus, provided fluid mechanical mechanisms of extinction are not in evidence, oxidation of energetic emissions due to interactions with cooling air can be fairly common even in current era combustors. Since temperatures are highest at high power conditions, take-off, climb-out, or other accelerations are the most probable time for oxidation and heat release within the turbine. Mixing processes work to reduce the extent of reactive, fuel-rich streaks at the combustor exit. However, over the length of a turbine stage, reactive streaks larger than $\sim 20\%$ of span do not have time to fully mix with the surrounding flow. This may be a significant mode in which energetic emissions can survive well downstream of the combustor. Heat release impacts on turbine heat flux are limited by relatively slow mixing with coolant, particularly for attached flows.

Mixedness improvements and other changes applied to bring the combustor exit flow closer to an equilibrium condition are mitigating measures that cannot overcome the fundamental thermodynamic determinants that govern the emission of energetic species into the turbine. In the near-term, turbine durability impacts will constrain the introduction of higher temperature and pressure engines. A companion paper [5] details experiments which measure up to 70% of the temperature rise suggested by ΔT_T for a single row of reacting film cooling jets. As fuel-air ratios increase further, an unprecedented merging of combustor and turbine designs will be required. If future aircraft engines are to continue increases in peak temperature and pressure, combustor and turbine design procedures will need to be altered to accommodate secondary combustion.

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APPENDIX: MIXING OF FUEL RICH STREAKS ENROUTE TO DOWNSTREAM SURFACES

Low mixedness is manifest at the combustor exit by spatial non-uniformities in fuel-air ratio. As a result of low mixedness, and in contrast to the emission of CO or HC resulting from inadequate residence time, some portion of the flow will be characterized by a greater than stoichiometric composition. In time, these spatial non-uniformities are emitted into the gas path as streaks. These streaks can transport energetic emissions to downstream of the combustor where, mixing with oxygen-rich film cooling or purge flows, stoichiometric combustion can occur somewhere in the interaction. Streaks that carry lean mixtures will be further diluted at the site and are relatively less consequential, both because of the reduced opportunity for reaction and the diminished impact of the subsequent heat release. It can be expected that upon emission, a streak would abut other regions of the exit flow that are lean. The fate of the streak—whether a portion of the streak survives the transit—depends on the scale of the non-uniformity as well as the relative mixing, convective, and chemical time scales.

We determine the importance of a particular non-uniformity by the definition of a minimum scale, representing the smallest streak that survives a single, representative HPT stage. Two simplified mixing models were employed to measure the extent of mixing in transit, a diffusive transport analysis governed by turbulence levels, and the development of a shear layer due to a velocity differential. The resulting estimates were used to define

τ_{mix}/τ_{conv} . If τ_{mix}/τ_{conv} is greater than one, some portion of the streak will arrive unperturbed. If oxidation reactions are slow, such that Da , defined by the reaction time and appropriate convective scale, τ_{conv}/τ_{reac} , is small, some of the partially reacted fuel in the mixed-out region may also survive.

Turbulent diffusion is one of the primary mechanisms through which a streak is mixed with adjacent flow. The temperature ratio of the streak to adjacent flow is taken to be unity and thus the density and velocity ratios are also taken to be unity. Under the assumption of isotropic turbulence with no cross velocities, no shear stresses form and thus, no shear layer develops. Additionally, no superimposed molecular diffusion is employed as it is assumed that the turbulence is dominant. In the manner shown in Figure 8, the diffusion, or mixing time, τ_{mix} , is defined as the time that it takes for a parcel of fluid at the outside of the reactive streak to diffuse to the centerline of the streak.

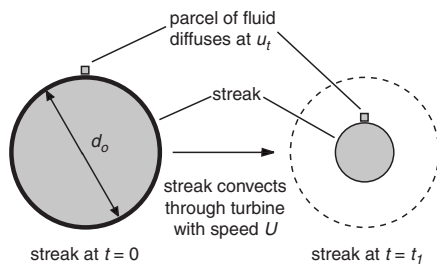


Figure 8. Turbulent diffusive mixing

An estimate for τ_{mix} is dependent on the diffusion velocity (V_d) which is related to the turbulent diffusivity (D_t) by analogy to the equation for Fickian diffusion. For turbulent diffusion, V_d is approximately the turbulent velocity, u_t . The turbulent velocity is estimated from typical levels of turbulent intensity ($u'/U = 0.1-0.15$) at the combustor exit. Using the assumption of homogenous isotropic turbulence, the turbulent velocity is estimated from the turbulent intensity as approximately u' . This gives a conservative definition for the mixing time, $\tau_{mix} = d_o / (2u_t)$.

Streak parameters such as temperature, pressure, position, and time were taken from streamlines calculated for a current era advanced engine. Turbulent intensities were varied from 5-50% of the freestream velocity and a range of streak diameters from 5-50% of the combustor exit span was considered. Results indicated that streaks larger than $\sim 10\%$ of span will not have sufficient time to mix out over the length scale of a turbine stage. The same approach can be used on other downstream components.

In addition to the turbulent diffusive mixing, the development of a shear layer between the streak and the adjacent flow also provides a potential mixing mechanism. Different velocities will result from different stream temperatures. Analytical distributions of velocity and pressure for a 2-D, turbulent shear layer between two co-flowing streams of different velocities and temperatures were employed for the analysis. Temperature differences were derived from typical combustor exit profiles. The temperature difference leads to the velocity non-uniformity between the cold and hot streams,

either of which may carry energetic emissions. The definition of τ_{mix} in this case is conservatively defined as the time required for the velocity at the centerline of the streak to slow down by 1%. This is essentially a criterion for the penetration of the shear layer to the center of the streak.

The shear mechanism provides a more vigorous mechanism for mixing than diffusive turbulence. Over the length of a turbine stage, and for a range of velocity ranges, it was found that reactive streaks larger than $\sim 20\%$ of span do not have time to fully mix with the surrounding flow. Thus, in the limit of fast chemistry, $\tau_{conv}/\tau_{reac} \gg 1$, some amount of energetic emissions can survive through the HPT unmixed.