Theses and Dissertations

2013-06-12

Application of One Dimensional Turbulence (ODT) to Model Fire Spread Through Biomass Fuel Bed

Abinash Paudel

Brigham Young University - Provo

Follow this and additional works at: https://scholarsarchive.byu.edu/etd

Part of the Chemical Engineering Commons

BYU ScholarsArchive Citation

https://scholarsarchive.byu.edu/etd/3988

This Thesis is brought to you for free and open access by BYU ScholarsArchive. It has been accepted for inclusion in Theses and Dissertations by an authorized administrator of BYU ScholarsArchive. For more information, please contact scholarsarchive@byu.edu, ellen_amatangelo@byu.edu.
Application of One Dimensional Turbulence (ODT) to Model
Fire Spread Through Biomass Fuel Bed

Abinash Paudel

A thesis submitted to the faculty of
Brigham Young University
in partial fulfillment of the requirements for the degree of

Master of Science

David O. Lignell, Chair
Larry L. Baxter
Alonzo D. Cook

Department of Chemical Engineering
Brigham Young University
June 2013

Copyright © 2013 Abinash Paudel
All Rights Reserved
ABSTRACT

Application of One Dimensional Turbulence (ODT) to Model Fire Spread Through Biomass Fuel Bed

Abinash Paudel
Department of Chemical Engineering, BYU
Master of Science

Each year fires destroy millions of acres of woodland, lives, and property, and significantly contribute to air pollution. Increased knowledge of the physics and properties of the flame propagation is necessary to broaden the fundamental understanding and modeling capabilities of fires. Modeling flame propagation in fires is challenging because of the various modes of heat transfer with diverse fuels, multi-scale turbulence, and complex chemical kinetics.

Standard physical models of turbulence like RANS and LES have been used to understand the flame behavior, but these models are limited by computational cost and their inability to resolve sub-grid scales. Application of several other models and empirical studies in fire modeling are usually limited to fire spread rate only. In some fires, flame propagation often occurs through convective heating by direct flame contact as opposed to radiative preheating alone. Under these conditions, resolution of the flame front can provide the detailed physics and insights into the flame propagation.

The One Dimensional Turbulence (ODT) model is extended to turbulent flame propagation in biomass fuel beds representative of those in wildland fires. ODT is a stochastic model that is computationally affordable and can resolve both large and fine scales. ODT has been widely applied to many reacting and non-reacting flows like jet flames and pool fires. A detailed particle combustion model has been developed and implemented in the ODT model to investigate the fluctuating flame-fuel interface and to study flame propagation properties. The particle reaction is modeled as a single global decomposition reaction model. Radiative, convective, and internal particle conductive heat transfer are included. Gaseous combustion is modeled with a lookup table parameterized by mixture fraction and fractional heat loss using steady laminar flamelet solutions. Results are presented from simulations of flame propagation in buoyantly driven flows. Particle size and loading are varied to study their effects in flame spread. A timescale analysis is performed to compare radiative, convective, conductive, and reactive particle timescales to the turbulent fluctuations. The flame propagation in homogeneous turbulence is also studied which better represents the wildland fire. The timescales involved in the wildland fire are overlapped using LEM model to study their effects on the flame properties and flame spread.

Keywords: ODT, turbulence, combustion, biomass, fire, RANS, LES, spread rate, modeling, radiation, convection, flamelet, flame
ACKNOWLEDGMENTS

There are several people who have contributed to this project. Dr. David O. Lignell significantly and continually provided constructive techniques, principles, feedback and help. Dr. Larry L. Baxter, Dr. John Hedengren, and Dr. Alonzo Cook also gave inputs and insights. Credit also goes to Elizabeth E. Monson and Ryan Hintze for their contribution to the project. Without the financial support from USDA Forest Service Rocky Mountain Research Station, this project would not have been explored.

Lastly, I would like to thank my parents, my brother, and my wonderful wife, Sharmila Paudel, for their continuous support and encouragement.
# TABLE OF CONTENTS

**LIST OF TABLES** ................................................................. vi

**LIST OF FIGURES** .............................................................. vii

**NOMENCLATURE** ................................................................. ix

**Chapter 1**  **Introduction** ............................................... 1
  1.1 Summary ................................................................. 3

**Chapter 2**  **Literature Review** ........................................ 5
  2.1 Physical and Quasi-Physical Models .................................. 7
  2.2 Empirical, Quasi-Empirical and Statistical Models ............... 10
  2.3 Summary ................................................................. 10
  2.4 LEM ..................................................................... 11
  2.5 ODT ..................................................................... 12
  2.6 Biomass Combustion ................................................... 13

**Chapter 3**  **One Dimensional Turbulence** ........................... 15
  3.1 One-dimensional Eddy ................................................ 15
  3.2 Eddy Sampling ........................................................ 17
  3.3 Solution Procedure ................................................... 19
  3.4 Validation of ODT Model ............................................. 20
  3.5 Summary ................................................................. 21

**Chapter 4**  **Biomass Combustion** ....................................... 22
  4.1 Biomass Pyrolysis Model ............................................. 22
  4.2 Particle Heating ....................................................... 24
  4.3 Table Lookup and Flamelet Solution ................................ 27
  4.4 Radiation Model ....................................................... 29
  4.5 Soot Model ............................................................. 31
Chapter 5 Results ........................................ 38
  5.1 Laminar Test Cases .................................. 38
    5.1.1 Configuration .................................. 38
    5.1.2 Laminar Test Results .......................... 39
  5.2 Turbulent Simulations ................................. 46
    5.2.1 Buoyant Plume ................................ 46
    5.2.2 Homogeneous Turbulence ...................... 62

Chapter 6 Summary and Conclusion ....................... 73

Chapter 7 Future Work .................................. 75
  7.1 Inclusion of Moisture ............................... 75
  7.2 Inclusion of Char Oxidation ....................... 75
  7.3 Inclusion of Wind ................................ 75
  7.4 Inclusion of Slope ................................ 76
  7.5 Fuel Continuity .................................. 76

REFERENCES ............................................. 77

Appendix A Initial Burnout Profile for Homogeneous Turbulence ........ 84
LIST OF TABLES

4.1 Nunn’s devolatilization model parameters. ............................................. 23
4.2 Stream properties for pure fuel and pure air. .......................................... 29
5.1 Summary of reaction and heat transfer timescales. .................................. 45
5.2 Heat loss, initial, and final gas and particle temperatures for laminar test cases... 46
5.3 Summary of LEM model parameters. ....................................................... 63
1.1 Images comparing the calculated equivalence ratio contours (plot (a)) and temperature contours (plot (b)) using RANS and LES models against experimental data from [11]. Adapted from [10] with permission. ................................. 3

3.1 Representative image of an ODT domain. White line perpendicular to the flow direction is the ODT notional line of sight. ......................................................... 17

3.2 A representation of a 2D flow before and after an eddy along with ODT line represented as solid white line (a), and an illustration of a triplet map along the ODT line direction (b). ......................................................... 18

3.3 ODT of channel flow at Re=395. Plot (a) compares the scaled mean velocity profile with DNS; (b) shows the streamwise velocity profile (adapted from [69] with permission). ......................................................... 20

3.4 Comparison of ODT and DNS for a turbulent jet flame. Plot (a) shows the full width at half maximum of the mixture fraction profile. Plot (b) and plot (c) show the mean temperature as a function of mixture fraction (adapted from [69] with permission). ......................................................... 21

4.1 Experimental data and fitted curve for gas production from pyrolysis of sweet gum hardwood particles. Adapted from [59]. ......................................................... 23

4.2 Flamelet temperature profile (plot (a)) and density profile (plot (b)) as a function of mixture fraction at different heat loss values. ......................................................... 28

4.3 Acetylene concentration at various heat loss and mixture fraction; positive values indicate heat loss and and negative values indicate heat gain. ......................... 33

5.1 Representative image of orientation of a cylindrical particle in a grid cell. ...... 39

5.2 Initial gas mixture fraction, (ξ), and gas temperature profile. ......................... 40

5.3 Initial and final temperature profiles of gas and particles for various laminar test cases. Plot (a) shows profiles without radiation and convection, (b) shows profiles with convection but without radiation, (c) shows profiles with radiation but without convection, and (d) shows profiles with both radiation and convection. ................. 40

5.4 Final heat loss profiles for different laminar test cases corresponding to Fig. 5.3. . 42

5.5 Particle reaction timescale. ................................................................. 45

5.6 Initial mixture fraction profile (a), and the corresponding temperature profile (b). . 48

5.7 Buoyant plume baseline simulations at t=0.5, 1.0, 1.5, and 2.0 s. Plots in the left column show the gas and particle surface temperature profiles whereas plots in the right column show the mixture fraction and heat loss profiles. ......................... 50

5.8 Percent particle burnout for the baseline case at 0.5, 1.0, 1.5, and 2.0 s. Increasing width is directly correlated to increase in time. ................................. 51
5.9 Eddy size, time, and location map for the baseline simulation. ............... 53
5.10 Statistics from 120 realizations, showing mean gas (a) and particle temperatures (b), gas root mean square temperature fluctuations (c), mean mixture fraction profile (d), particle burnout width (e), and flame spread rate (f). ................. 55
5.11 Baseline mean velocity profile at 0.5 s, 1.0 s, 1.5 s and 2.0 s. Increasing width corresponds to increase in time. .................................................. 56
5.12 Mean gas temperature profiles, top row; mean particle temperature, middle row; mean mixture fraction, bottom row. The left, middle, and right columns are the d=0.7 mm, d=1.0 mm, and d=1.2 mm cases, respectively. ..................... 58
5.13 Mean gas temperature profiles, top row; mean particle temperature, middle row; mean mixture fraction, bottom row. The left, middle, and right columns are the φ=0.5, φ=1.0, and φ=2.0 cases, respectively. ......................... 60
5.14 Width of the particle burnout profile bounding 5% burnout (a), and rate of spread of this profile based on derivative of polynomial fits (b). ...................... 61
5.15 Initial temperature profile for homogeneous outflow case. .................. 64
5.16 Homogeneous turbulence baseline simulations at t=1.0, 2.0, 3.0, and 4.0 s. .... 65
5.17 Percent particle burnout for the baseline case at 1.0, 2.0, 3.0 and 4.0 s. Increasing width is directly correlated to increasing time. ......................... 68
5.18 Eddy size, time, and location map for the homogeneous outflow simulation. .... 68
5.19 Statistics from 120 realizations, showing mean gas (a) and particle temperatures (b), gas root mean square temperature fluctuations (c), mean mixture fraction profile (d), particle burnout width (e), and flame spread rate (f). ...................... 70
5.20 Width of the particle burnout profile bounding 5% burnout (a), and rate of spread of this profile based on derivative of polynomial fits (b). ...................... 72
NOMENCLATURE

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Definition</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>Particle surface area</td>
</tr>
<tr>
<td>α</td>
<td>Amplitude parameter in log normal distribution</td>
</tr>
<tr>
<td>α</td>
<td>Thermal diffusivity</td>
</tr>
<tr>
<td>β</td>
<td>ODT scaling parameter</td>
</tr>
<tr>
<td>C</td>
<td>ODT scaling parameter</td>
</tr>
<tr>
<td>Cc</td>
<td>Cunningham slip correction factor</td>
</tr>
<tr>
<td>Cp</td>
<td>Molar heat capacity</td>
</tr>
<tr>
<td>Cpp</td>
<td>Specific heat capacity of particles</td>
</tr>
<tr>
<td>D</td>
<td>Diffusivity</td>
</tr>
<tr>
<td>Dp</td>
<td>Diameter of a particle</td>
</tr>
<tr>
<td>DT</td>
<td>Turbulent diffusivity (LEM parameter)</td>
</tr>
<tr>
<td>Δt</td>
<td>Temporal or spatial advancement increment of an eddy</td>
</tr>
<tr>
<td>Δx</td>
<td>ODT cell size</td>
</tr>
<tr>
<td>Ea</td>
<td>Activation energy</td>
</tr>
<tr>
<td>Ekin</td>
<td>Eddy kinetic energy</td>
</tr>
<tr>
<td>Evp</td>
<td>Viscous penalty term</td>
</tr>
<tr>
<td>ε</td>
<td>Dissipation rate</td>
</tr>
<tr>
<td>εs</td>
<td>Solid emissivity</td>
</tr>
<tr>
<td>η</td>
<td>Arbitrary conservation property per unit mass / Kolmogorov length scale</td>
</tr>
<tr>
<td>f</td>
<td>Frequency</td>
</tr>
<tr>
<td>f(x)</td>
<td>Eddy triplet map as function</td>
</tr>
<tr>
<td>fv</td>
<td>Soot volume fraction</td>
</tr>
<tr>
<td>γ</td>
<td>Volume fraction of gas phase to total fraction (i.e. void fraction)</td>
</tr>
<tr>
<td>h</td>
<td>Heat transfer coefficient</td>
</tr>
<tr>
<td>h</td>
<td>Local enthalpy</td>
</tr>
<tr>
<td>ha</td>
<td>Adiabatic enthalpy</td>
</tr>
<tr>
<td>hs</td>
<td>Sensible enthalpy</td>
</tr>
<tr>
<td>hv</td>
<td>Enthalpy of the volatile</td>
</tr>
<tr>
<td>I</td>
<td>Radiative intensity</td>
</tr>
<tr>
<td>j</td>
<td>Diffusion flux</td>
</tr>
<tr>
<td>K</td>
<td>Plank mean absorption coefficient</td>
</tr>
<tr>
<td>k</td>
<td>Rate constant</td>
</tr>
<tr>
<td>k0</td>
<td>Pre-exponential factor</td>
</tr>
<tr>
<td>kb</td>
<td>Boltzmann constant</td>
</tr>
<tr>
<td>kg</td>
<td>Gas absorption coefficient</td>
</tr>
<tr>
<td>ks</td>
<td>Solid absorption coefficient</td>
</tr>
<tr>
<td>κ</td>
<td>Absorption coefficient</td>
</tr>
<tr>
<td>L</td>
<td>Domain length</td>
</tr>
<tr>
<td>LE</td>
<td>Eddy size</td>
</tr>
<tr>
<td>Lept</td>
<td>Optical thickness</td>
</tr>
<tr>
<td>Λ</td>
<td>Rate of all eddies</td>
</tr>
<tr>
<td>λ</td>
<td>Eddy frequency</td>
</tr>
<tr>
<td>λ</td>
<td>Thermal conductivity</td>
</tr>
<tr>
<td>Symbol</td>
<td>Description</td>
</tr>
<tr>
<td>--------</td>
<td>-------------</td>
</tr>
<tr>
<td>$m_p$</td>
<td>Mass per volume of grid cell</td>
</tr>
<tr>
<td>$\mu$</td>
<td>Dynamic viscosity</td>
</tr>
<tr>
<td>$M$</td>
<td>Molecular weight</td>
</tr>
<tr>
<td>$N_{odt}$</td>
<td>Number of ODT particles</td>
</tr>
<tr>
<td>$n_D$</td>
<td>Number density</td>
</tr>
<tr>
<td>$n_{real}$</td>
<td>Number of real particles</td>
</tr>
<tr>
<td>$n_{tot}$</td>
<td>Total number of particles</td>
</tr>
<tr>
<td>$Na$</td>
<td>Avogadro’s number</td>
</tr>
<tr>
<td>$Nu$</td>
<td>Nusselt number</td>
</tr>
<tr>
<td>$\nu$</td>
<td>Kinematic viscosity</td>
</tr>
<tr>
<td>$\Omega$</td>
<td>System being evolved by ODT</td>
</tr>
<tr>
<td>$\omega$</td>
<td>Reaction source term</td>
</tr>
<tr>
<td>$P$</td>
<td>Pressure</td>
</tr>
<tr>
<td>$P(x_0, l)$</td>
<td>Joint probability density function</td>
</tr>
<tr>
<td>$P_a$</td>
<td>Eddy probability</td>
</tr>
<tr>
<td>$\phi$</td>
<td>Rate factor</td>
</tr>
<tr>
<td>$\phi(x)$</td>
<td>Map of fluid system in the x domain</td>
</tr>
<tr>
<td>$\Pi$</td>
<td>Boundary of system being evolved by ODT</td>
</tr>
<tr>
<td>$Pr$</td>
<td>Prandlt number</td>
</tr>
<tr>
<td>$\dot{Q}$</td>
<td>Heat added to the ODT system</td>
</tr>
<tr>
<td>$q$</td>
<td>Heat flux</td>
</tr>
<tr>
<td>$q_{rad}$</td>
<td>Radiative heat flux</td>
</tr>
<tr>
<td>$Q_{rad}$</td>
<td>Volumetric radiative source</td>
</tr>
<tr>
<td>$Q_g$</td>
<td>Volumetric source term (gas)</td>
</tr>
<tr>
<td>$Q_s$</td>
<td>Volumetric source term (solid)</td>
</tr>
<tr>
<td>$R$</td>
<td>Universal gas constant</td>
</tr>
<tr>
<td>$Re$</td>
<td>Reynolds number</td>
</tr>
<tr>
<td>$\rho$</td>
<td>Density</td>
</tr>
<tr>
<td>$\rho_s$</td>
<td>Soot density</td>
</tr>
<tr>
<td>$S_{i,j}$</td>
<td>Rate of stress tensor</td>
</tr>
<tr>
<td>$S_{\Phi}$</td>
<td>Gas phase mass source term from devolatilization of wood particles</td>
</tr>
<tr>
<td>$\sigma$</td>
<td>Stephan-Boltzmann constant</td>
</tr>
<tr>
<td>$T$</td>
<td>Temperature</td>
</tr>
<tr>
<td>$t$</td>
<td>Time</td>
</tr>
<tr>
<td>$\theta$</td>
<td>Blowing factor in heat transfer to a particle</td>
</tr>
<tr>
<td>$\tau_i$</td>
<td>Particle time scale</td>
</tr>
<tr>
<td>$\tau_E$</td>
<td>Turbulent timescale</td>
</tr>
<tr>
<td>$\tau_{conv}$</td>
<td>Convective timescale</td>
</tr>
<tr>
<td>$\tau_o$</td>
<td>Optical thickness</td>
</tr>
<tr>
<td>$\tau_{rad}$</td>
<td>Radiative time scale</td>
</tr>
<tr>
<td>$\tau_{xy}$</td>
<td>Viscous stress tensor</td>
</tr>
<tr>
<td>$u$</td>
<td>Streamwise velocity (perpendicular to the ODT line)</td>
</tr>
<tr>
<td>$V$</td>
<td>Volume</td>
</tr>
<tr>
<td>$v^D$</td>
<td>Diffusion velocity</td>
</tr>
<tr>
<td>$\bar{v}_{cv}$</td>
<td>ODT control volume velocity</td>
</tr>
</tbody>
</table>
\( \vec{v}_{\text{sys}} \) ODT system velocity
\( \vec{v}_R \) Relative velocity of ODT system
\( X_i \) Mole fraction of chemical species
\( x \) ODT line direction
\( x_0 \) Eddie location
\( \xi \) Mixture fraction
\( Y \) Mass fraction of chemical species
\( Y_s \) Soot mass fraction
\( y \) Axes normal to the ODT line direction
\( Y_v \) Yield of pyrolysis gas
\( Y_{V,\infty} \) Ultimate pyrolysis yield
\( Z \) ODT scaling parameter

**Subscripts, superscripts, and other indicators**

- \( \dot{\text{[ ]}} \) indicates \( \text{[ ]} \) is a rate
- \( \hat{\text{[ ]}} \) indicates \( \text{[ ]} \) is an Eulerian property
- \( \text{[ ]} - \) indicates \( \text{[ ]} \) is in the direction of decreasing \( x \)
- \( \text{[ ]} + \) indicates \( \text{[ ]} \) is in the direction of increasing \( x \)
- \( \text{[ ]} 0 \) indicates \( \text{[ ]} \) is the initial value or property at \( x = 0 \) in the domain
- \( \text{[ ]} b \) indicates \( \text{[ ]} \) is a black body property
- \( \text{[ ]} E \) indicates \( \text{[ ]} \) is an eddy property
- \( \text{[ ]} e \) indicates \( \text{[ ]} \) is from the east face of a grid cell
- \( \text{[ ]} f \) indicates \( \text{[ ]} \) is a final value
- \( \text{[ ]} g \) indicates \( \text{[ ]} \) is a gas phase property
- \( \text{[ ]} i \) indicates \( \text{[ ]} \) is a property of chemical species \( i \) or a property of the \( i \)th grid cell
- \( \text{[ ]} \text{in} \) indicates \( \text{[ ]} \) is an inlet property at the wall
- \( \text{[ ]} \text{oo} \) indicates \( \text{[ ]} \) is a property of the surroundings
- \( \text{[ ]} L \) indicates \( \text{[ ]} \) is the property value at \( x = L \) in the domain
- \( \text{[ ]} m \) indicates \( \text{[ ]} \) is a mean property
- \( \text{[ ]} \text{max} \) indicates \( \text{[ ]} \) is the maximum value
- \( \text{[ ]} n \) indicates \( \text{[ ]} \) is the \( n \)th item in a list or series
- \( \text{[ ]} p \) indicates \( \text{[ ]} \) is a particle property
- \( \text{[ ]} \text{rad} \) indicates \( \text{[ ]} \) is a radiative property
- \( \text{[ ]} s \) indicates \( \text{[ ]} \) is a soot property
- \( \text{[ ]} w \) indicates \( \text{[ ]} \) is from the west face of a grid cell
- \( \text{[ ]} w \) indicates \( \text{[ ]} \) is a wall property
CHAPTER 1. INTRODUCTION

Each year in the US, fire destroys millions of acres of woodland and significantly contributes to pollution also destroying lives and property. Despite the federal government spending billions of dollars to suppress these fires, they are inevitable. Increased knowledge of the physics and properties of the flame propagation is necessary to broaden the understanding of these detrimental fires to help improve management techniques.

In wildland fires, flame propagation through unburnt fuel occurs by radiative preheating from the burnt fuel and by direct flame contact [1]. However, in fires with fine fuels, flame propagation occurs mainly by direct flame contact [2] as opposed to radiation preheating [3,4]. Thus, the resolution of flame brush can provide unique insights into the flame propagation. The results from the resolution of the flame brush can also be used as a sub grid modeling with other approaches for model development.

The computational models of flame propagation provide better understanding and a robust predictive technique that could save lives and property. Experiments are sometimes very expensive to perform and experimental analysis and observations do not provide insights of the detailed physics involved in these fires. The data from these computational models on the other hand, are cheaper to obtain, and can provide a rigorous, and robust indication of the result. The computational models are also applicable to various configurations that will help to improve management techniques.

A wide range of physical processes occurs in wildland flame propagation. The flame front at the interface between the flame and the unburnt fuel is typically turbulent where the flame is driven by buoyancy and wind effects [5–7]. These turbulent flames involve a wide range of length scales ranging from sub-millimeter thickness of an individual flame to the kilometer sized convective transport scales [8]. Wildland fires also involve many chemical species whose identity and reaction pathway may be unknown. Flames generally exist at the molecular level where fuel
and oxidizer mix together. Different modes of heat transfer and various kinds of fuels are involved in the wildland fires. The wide range of length scales and physical processes involved makes wildland fire modeling an exceptionally challenging problem.

To simulate flame propagation, all of the scales ranging from small individual flame scales to large flame scales, must be resolved. The only simulation model that can resolve flames in turbulent flows is direct numerical simulation (DNS). DNS numerically solves Cauchy equations of motions in a 3D domain without introducing any approximation or sub-grid models aside from continuum, Newtonian fluid media in incompressible flow. This is responsible for two drawbacks of DNS- its high computational cost, and limitation on the maximum Reynolds number [9]. Although DNS can resolve all scales, it is not feasible to use it to model turbulence and flame propagation in wildland fuels.

Reynolds-averaged Navier-Stokes (RANS) and large eddy simulation (LES) are computationally cheaper and are more feasible than DNS. However, these models cannot resolve individual flames. RANS solves the time averaged Navier-Stokes equations and LES solves the spatially-filtered Navier-Stokes equations. Since these models do not resolve individual flames, they require sub-grid models which generally are not coupled to evolving diffusive structures and require validation. Figure 1.1 shows images comparing the instantaneous equivalence ratio and temperature contours [10] calculated using RANS and LES models, respectively, against experimental data [11]. It can be inferred that RANS is able to capture only the large scales of turbulence while LES is able to capture unsteadiness, and a wide range of scales.

One dimensional turbulence (ODT) [12, 13], a relatively new 1D model, can resolve both large and fine scales. Since scales are resolved in 1D only, it is computationally cheaper than DNS, and LES. In ODT, reaction diffusion equations for mass, momentum, species, and energy are solved along a line-of-sight through a given flow. Chapter 3 describes the detailed description of ODT. This thesis describes the use of ODT to model turbulent flame propagation in biomass fuel beds representative of wildland fuel. Specifically, the objectives of this project were to:

1. develop a particle combustion model in ODT to understand the detailed description of the flame structure at the interface between flame and unburnt fuel.
2. Use ODT to simulate flame propagation in biomass fuel beds to quantify the particle properties on the spread.

3. Investigate particle heat-up and understand fuel propagation, effects of turbulence, feedback to the gaseous medium from the particle combustion, and important statistics that describe a flame front.

![Figure 1.1: Images comparing the calculated equivalence ratio contours (plot (a)) and temperature contours (plot (b)) using RANS and LES models against experimental data from [11]. Adapted from [10] with permission.](image)

1.1 Summary

Wildland fires are unavoidable and pose a threat to human lives and property. To suppress these fires, a better understanding of the physics in flame propagation is necessary. While DNS is computationally expensive and infeasible, RANS and LES cannot resolve individual scales of turbulent combustion. Hence, ODT, a one-dimensional stochastic turbulent model, which is computationally cheaper and can resolve both individual and large flame scales, is proposed to model flame propagation in representative wildland fuel.

The organization of this thesis is as follows. Chapter 1 provides the introduction and motivation for this project. Chapter 2 provides the background and reviews of the existing combustion...
model. Chapter 3 and chapter 4 provide the detailed description of the formulation of the ODT model and incorporation of biomass combustion in ODT respectively. The laminar test results and full turbulent simulation results are presented in Chapter 5. Finally, conclusions and the future work are provided in Chapter 6 and Chapter 7, respectively.
CHAPTER 2. LITERATURE REVIEW

Wildland fire modeling has been an active area of research since 1920 [14] after the largest and deadliest wildfire occurred in US in 1910 that destroyed 3 million acres of forest and caused millions of dollars in loss. Previous research like that of Albini [4] assumes that radiation is exclusively responsible for flame propagation, whereas Weber [15], and Finney [16] demonstrate the need of direct flame contact in addition to radiation for flame propagation in discontinuous fuel beds. Van Wagner [17] and Anderson [18] found that radiation provides less than 50% of the required heat transfer during crown fire experiments. The laboratory experiments performed by McCarter et al. [19] also showed that flame radiation from above the fuel bed contributes little to fuel preheating.

Anderson et al. [20] performed wind-tunnel fire studies to investigate convective heat transfer ahead of the fire moving across a porous fuel bed. In non-zero winds, the temperature decreased exponentially, whereas in zero winds, the temperature decreased very rapidly within the short distance of the flame front. Butler et al. [21] studied the radiative fluxes and temperature in a boreal forest crown fire. In the study, the peak temperature exceeded 1330 °C, and the average radiant flux from the flames across all the experiments was found to be about 200 kW/m². The data from the study also indicated that the radiative energy penetration in the forest canopy is significantly longer than it is in the lower levels of the forest sand. Butler and Cohen [22] mention that convection is a major energy transfer mechanism in wind-driven crown fires as the turbulent eddies migrate some distance ahead of the fire front. However, their theoretical model to predict safety zones for firefighters is solely based on the radiative heating. The model predicts that a firefighter standing 6 m away from a 21 m flame can feel a heat flux of up to 60 kW/m². Butler and Denison [23] measured radiant and total heat flux in a 0.8 m diameter cylindrical down-fired laboratory-scale reactor. A maximum radiant flux of 110 kW/m² was observed with less than 10% differences between radiative and total heat flux at all axial locations. Butler and Finney [24] developed a radiation-driven
numerical model to predict spread rate and intensity of forest crown fires. The model assumed that radiation was the dominant form of heat transfer between the fire and unignited fuel, while convection cooled the fuels ahead of the fire front. The results were compared with the measured spread rates. While the model overpredicted the magnitude of fire spread rates, it accurately predicted relative response of the fire spread rate. Catchpole et al. [25] conducted 357 experimental fires in an environmentally controlled large wind tunnel to study the effect of particle sizes, fuel bed depths, packing ratios, moisture contents, and windspeeds on the rate of spread of free-burning fires. It was found that spread rate decreases with moisture content, decreases as the square root of the packing ratio, and increases with the windspeed. Fuel diameters below 1 mm, and fuel bed depth did not have significant effect on the spread rate.

Frankman et al. [26] explored heat transfer behavior during the combustion of discontinuous fuel beds by measuring time-resolved radiation and convection heat fluxes in a series of experiments. They concluded that radiation flux heated the fuel, while convection heat transfer heated and cooled the fuel before the flame arrival. The convection could also heat or cool even after the flame arrival. In all experiments, convective heating pulses would occur before the combustion indicating that convective heating is a critical ignition source. Flame spread appeared to be governed by factors that affected the convective transport intensity. Rapid fluctuations were observed in both radiation and convection. Frankman and Webb et al. [27] measured time-resolved irradiance and convective heating and cooling of fast-response thermopile sensors in 13 natural and prescribed wildland fires. It was shown that convective heat transfer caused rapid fluctuations whereas irradiance increased monotonically with the approach of the flame and decreased largely with its passage. The radiation energy from the fire contributed 79% of the variance in fuel consumption whereas the convective heating at the sensor varied from 15% to fluxes exceeding that of the radiation. The results indicated that the contribution of convection and radiation in total energy release was dependent on the fuel and the environment.

Wildland fire is a complicated process that involves energy release in the form of heat due to chemical reactions and the transport of that energy to surrounding unburnt fuel [14]. Flame propagation via convective heating through the unburnt fuel is complex as the process is influenced by many factors. The flame front between the burnt and unburnt fuel is turbulent. The flow is guided by buoyant acceleration and the wind effects [5, 6]. The geometry and the dynamics of the
flame then determine whether the flame will contact with the adjacent unburned fuel resulting in fire spread [6].

Wildland fire occurs in a fuel composed of live and dead plant materials with primarily cellulose constituents which are polymers of glucose monomer [28]. The cellulose undergoes a thermal degradation reaction called pyrolysis when heated. In general, there are two competing degradation reactions: volatilization and char formation, which have different activation energies, heat releases, and products. Volatilization is an endothermic reaction that occurs in conditions of little or no moisture [29], whereas char formation is an exothermic reaction that occurs in the presence of moisture.

According to Balbi et al. [30], there are three different kinds of combustion modeling: statistical, empirical, and physical. Statistical models, which do not involve physical mechanisms, are the simplest. Empirical models (like the Rothermal model [31]), are based upon the conservation of energy, but they do not distinguish among the modes of heat transfer. Physical models differentiate the modes of heat transfer to predict the fire behavior [32].

This project illustrates the extension of the ODT model, a physical model in which the modes of heat transfer are differentiated to predict fire behavior. Hence, it is necessary to review the physical and quasi-physical models. Below is the brief description of different models since 1990 adapted from Sullivan [14,33,34] pertaining to physical and quasi-physical models to predict fire spread. A brief description of the empirical and statistical models is also provided to complete a discussion of the state of the art in wildland fire modeling.

2.1 Physical and Quasi-Physical Models

The physical and quasi-physical models fundamentally describe the physics and chemistry involved in fire behavior [34]. Weber’s model (1991) [35] was an early attempt to provide a framework necessary to provide a physical model of fire spread through wildland fuel. Weber provides a reaction-transport formulation of the conservation of energy equations and then formulates several components that contribute to the overall flux of energy. Heat is generated proportional to an Arrhenius chemical reaction. Weber treats the phase differences explicitly and the conservation of energy relations are coupled since the volatiles given off from the solid combust in the gas...
phase and the energy is released to the solid for further volatilization. The simple model compares favorably to the experimental result of Anderson et al. [36].

AIOLOS-F, developed in Greece, is a CFD model that utilizes the 3D form of the conservation laws to couple the combustion of the fuel layer with the atmosphere to model fire spread in wildland fuel. Fuel combustion is modeled through a 3D fuel mixture-fraction evolution equation and fuel consumption is modeled by an Arrhenius law. A thermal radiation heat transfer equation provides the radiant heat source term [14]. The set of equations are then solved iteratively for enthalpy, mixture fraction, and temperature. These values indicate the change in air density to account for the effect of buoyancy. The enthalpy, mixture fraction, and temperature are then updated with the new wind velocity. The process continues until the solution converges. This model runs in real time; however no comparison or performance data were published when it was first developed.

FIRETEC, developed at the Los Alamos National Laboratory [37] in 1997, is a coupled multiphase transport/wildland fire model. This model applies conservation of mass, momentum and energy equations. It is also a 3D model that is capable of representing the coupled interactions of the combustion, heat transfer and fluid mechanics involved in wildland fire. This model uses the finite volume method to solve its system of equations. The unique aspect of FIRETEC is that the variables in gas and solid phases are divided into mean and fluctuating components and ensemble averages of the equations which are similar to those used for turbulence modeling. Even though FIRETEC includes complex combustion reactions of wildland fire, it does not explicitly include the effects of radiation and consequently fails to propagate in zero wind situations or down slopes.

Forbes (1997) [38] developed a 2D model of fire spread utilizing radiative heat transfer, species consumption and flammable gas production. Forbes developed a set of differential equations to model the fire spread in eucalypt forest fires (the basis for his model are observations of eucalypt forests). The reaction rates are temperature dependent for both solid and gas phases. From the analysis of the 1D form of the equations, Forbes concludes that for most parameter values the fire does not sustain itself. Even though the performance data are not provided, he determines that the activation energies for each reaction, rate constants, and heat release coefficients govern the propagation of the fire. Moisture is ignored in this model.

Grishin [39] developed a physical model using the concepts and methods of reactive media mechanics based on the analysis of experimental data. Turbulent heat and mass transfer in the fire,
as well as heat and mass exchange between the near-ground layer of the atmosphere and the forest canopy are incorporated. The model takes into account the basic physiochemical processes like heating, drying and pyrolysis and utilizes the conservation of mass, momentum and energy in both solid and gas phases. Other equations along with the initial and boundary conditions are used to determine the flux, heat transfer, and mass loss rates based on the Arrhenius rate laws. Grishin uses an effective reaction rate whose mass rate is close to CO as CO is the most common pyrolytic product. Numerical analysis of the equations then enable the structure of the fire front and its development. The work of Grishin has long been recognized for its comprehensive and innovative approach to the problem of developing physical models of fire behavior [35].

The Wildland Fire Dynamics Simulator (WFDS), developed by the US National Institute of Safety Technology [40], is a 3D model that extends the Fire Dynamic Simulator (FDS). WFDS is based on the equations of motion for buoyant flow and was developed to predict the behavior of fires burning through wildlands. This 3D model assumes a two-stage endothermic thermal decomposition and uses the temperature-dependent mass loss rate expression to model the solid fuel degradation. WFDS assumes that combustion occurs because of fuel gas and oxygen mixing in stoichiometric proportion. Char oxidation is not accounted for and detailed chemical kinetics are not modeled. The model assumes that the time scale of the chemical reactions is much shorter than that of mixing. Radiation is incorporated in the model and soot production is an assumed fraction of the mass of fuel gas consumed. The model takes around 25 CPU hours for 100 seconds of simulated time in a 200 m × 200 m plot for a low intensity fire.

Albini [4,32] developed a 2D quasi-physical model of fire spread through a single homogeneous fuel layer. The fire front is an isothermal flame ignition interface between unburnt and burnt fuel expressed as an eigenvalue problem utilizing a 3-stage fuel heating model [14]. A modified version of the model, in which a thermally-inert zone is placed beneath the homogeneous fuel layer to simulate the fire propagation through the tree crowns, was compared with a series of field-based experimental crown fires conducted in immature Jack Pine. Only one experimental fire was used to obtain the model parameters and then the model parameters were applied to obtain flame geometry and radiative properties for the remaining fires. The model was found to perform reasonably well with an absolute error of 14%. Several other models have been developed using Albini’s model as a basis to improve the performance.
2.2 Empirical, Quasi-Empirical and Statistical Models

An empirical model is based upon observation and experiment rather than on theory. The primary use of empirical and quasi-empirical models is to estimate the likely spread in the direction of the wind. These models serve the primary purpose of suppression but are also used against the physical models to test the theory. Traditionally, the empirical models are 1D and are used to describe the behavior of the fire such as rate of forward spread (ROS), flame depth angle and height [33]. The straightforward implementation of these models, their computational efficiency, their direct relation to the behavior of real fires, and their development by forestry agencies for their own immediate use have made these models more popular among the wildland fire authorities [33]. A few of the empirical models are Canadian Forest Services (CFS), Conservation and Land Management (CALM), the Spinifex model developed by Western Australian Department of Conservation and Land Management, Canadian Forest Fire Behavior Prediction (CFBP) System, and CSIRO Grassland Fire Spread Meter developed in 1997. Some of the popular quasi-empirical methods developed so far are the United States Forest Service (USFS) model, and the trigonometric model developed by Nelson in 2002.

2.3 Summary

While some of the experiments and models conclude that radiation is the dominant heat transfer mechanism, few other studies and findings suggest that convection is a principal heat transfer mechanism necessary for pre-heating and pre-ignition during fire spread occurring in the fuel bed. Fire spread in wildland fuel beds has been an active area of research and physical, statistical and empirical models have been developed. Some of the models are computationally expensive; others cannot resolve the full range of scales of turbulent combustion. The statistical models do not provide insight and the empirical models do not recognize all the heat transfer modes. Physical models are developed mainly to study the behavior of fires under a variety of conditions in a wide range of fuels that is not obtained from experiments alone. The physical models such as WFDS ignore char oxidation, whereas a few models such as Forbes ignore moisture. The physical models do not resolve the flames and often need sub grid models. For example, Michael et al. [41] utilized FIRETEC with a sub-grid pocket model to simulate the local mixture fraction of sub-grid flame
sheets. Fire spread rates from the simulation were 25-100% higher than the previous FIRETEC simulations. The application of these physical models is currently limited to the fire spread rate only.

2.4 LEM

This study is based on the application of the ODT and the Linear Eddy Model (LEM) in biomass combustion.

LEM recognizes that the convection and diffusion which act on a scalar field are two distinctly different physical processes [42]. These two processes act and interact at all scales of the flow. A broad range of scales, from the integral scales that involve the largest coherent structures and typically are measured in many meters, to Kolmogorov scales that involve the smallest structures and shear layers and typically range in the millimeter to submillimeter range, is covered by the fluctuations in the velocity field. The unique feature of LEM is that it describes the scalar fields in high resolution and represents them in one dimension (1D). Using 1D allows LEM to resolve the full range of length and time scales, even for flows with high Reynolds (Re) and Schmidt (Sc) numbers [43, 44].

In LEM, convection, or stirring, is modeled in a stochastic manner. Times and sizes of the turbulent eddies are sampled randomly from a predefined eddy distribution function. The turbulent strain and rotational folding effects that cause mixing within a flow are represented through a mapping process called triplet mapping [43].

In LEM, molecular diffusion is treated as a Fickian diffusion process either as an effective or multicomponent diffusion coefficient. Flow field structures are an input to the model rather than the calculated as part of the model [44]. Hence, the LEM model allows one to overlap the timescales by manipulating parameters like the maximum allowable eddy size ($L_{\text{max}}$), the minimum eddy size ($L_{\text{min}}$) and the turbulent diffusivity parameter, $D_T$, which are all input parameters to the model. The turbulent diffusivity parameter dictates the rate of eddies in the model. According to Chakravarthy et al. [44], $D_T$ is estimated by

$$D_T(l_c) = 0.064 \nu_g \left[ \frac{L_c}{\eta} \right]^{4/3}$$  \hspace{1cm} (2.1)
where \( v_g \) is the kinematic viscosity of the gas, and \( l_c \) and \( \eta \) are the characteristic integral and Kolmogorov length scales, respectively. The eddy size distribution, \( f(l) \), and stirring frequency (eddy rate) per unit length of the spatial domain, \( \lambda \), is then given by

\[
f(l) = \frac{5}{3} \frac{l^{-8/3}}{\eta^{-5/3} - L^{-5/3}},
\]

\[
\lambda = 0.6912 \left[ \frac{L}{\eta} \right]^{4/3} \frac{\nu}{L^3} \left[ \frac{(L/\eta)^{5/3} - 1}{1 - (\eta/L)^{4/3}} \right],
\]

where \( l \) is the eddy size and other symbols have their usual meaning.

Even though LEM was first developed to investigate the qualitative mixing properties of the turbulent flows, it has been successfully used with other models to treat a variety of flow configurations including grid turbulence, planar mixing layers, and axis-symmetric jets [45–47]. Even though LEM has also been used to model reacting flows [48], it has not previously been applied to model wildland fires in fuel beds.

### 2.5 ODT

ODT was developed as an extension to the linear eddy model (LEM) [12, 43]. LEM and ODT solve two concurrent processes: (1) stochastic eddy events that model advection by local remapping processes with a given location, size, and rate; and (2) solution of unsteady reaction-diffusion equations for scalars on the 1D solution domain. In LEM, the eddy size distribution from which the eddies are sampled is fixed; however, in ODT, the eddy distribution advances in time and is calculated from the local, instantaneous velocity field. The LEM model assumes homogeneous turbulence, while in ODT model, turbulence is treated to be more general. The turbulent intensity variations with local shear profiles can be accounted in ODT but not in LEM. The distinctive feature of ODT, like LEM, is that the turbulent advection is represented by the stochastic process rather than the 3D evolution equations. A detailed description of the ODT model will be discussed later in Chapter 3.
Only a few studies have been done in turbulent reacting flows that use ODT as the base model because ODT is relatively a new model. Echekki et al. [49] used ODT to simulate the turbulent jet diffusion flames as a first application of ODT in combustion. Lignell et al. [50] simulated flame extinction in jet flames using ODT. ODT has also been applied in simulating the flame structure [51], hydrogen-fueled homogeneous charge compression ignition (HCCI) combustion [52], flame extinction and re-ignition in the planar ethylene jet flames [50], and in simulating autoignition of hydrogen/carbon monoxide fuel mixtures in a turbulent jet [53]. Application of ODT to non-reacting particle flows has also been studied [54]; however, biomass combustion using ODT has not been explored previously.

2.6 Biomass Combustion

There are a number of physical and empirical models of wood pyrolysis in literature. Some of the models (both simple and complex) reviewed by DiBalsi [55] are that of Thurner and Mann [56], Ward and Balshaw [57], Chan et al. [58], Nunn et al. [59], and Di Balsi and Branca et al. [60]. DiBalsi categorizes the pyrolysis models into two different categories: (1) one-component mechanism, and (2) multi-component mechanism of primary pyrolysis.

In one-component mechanisms, the majority of kinetic mechanisms consist of a single or three parallel reactions for the formation of main products. The virgin wood decomposes to char, tars, and gas via three parallel reactions. These three reactions are then represented by one global single reaction with one activation energy and rate constant. Some of the one-component models reviewed by DiBalsi are that of Thurner and Mann [56], Ward and Balswaw [57], and Chan et al. [58]. These models are studied in different feedstock and in different experimental systems. Most of the models are applicable to a very narrow range of temperatures. The pyrolysis model of Nunn et al. [59] is the only model that is applicable to temperatures up to 1400 K.

Multi-component mechanism consists of parallel and series reactions for the decomposition of the virgin wood material. Usually the multi-component mechanism consists of the primary and secondary reactions. In the primary reaction, the wood decomposes into gas, tar, and char via different parallel reactions with different activation energies and rate constants. In secondary reaction, tar further decomposes to gas and char [61]. Some of the multi-component models reviewed by DiBalsi are that of Font et al. [62], Graham et al. [63], Garcia et al. [63], and Morf et
al [64]. These models are applicable also to high temperature pyrolysis (usually above 1000 K but not above 1300 K).

The biomass pyrolysis model used in this work is that of Nunn et al. [59]. This form of model has been successfully used to model the devolatilization of both wood and other solid fuels [65–67] and has been useful in engineering calculations. A detailed description of this model is included in Chapter 4.
CHAPTER 3. ONE DIMENSIONAL TURBULENCE

As noted, ODT is an extension of LEM. ODT was first developed by Alan R. Kerstein and coworkers, with early codes in FORTRAN and later codes written in C++. The description of ODT in this thesis is adapted from [12, 13, 52, 68].

ODT is a numerical method to solve stochastic boundary-value problems on a 1D domain. The ODT code used here is written in a Lagrangian finite volume formulation in which the mass within the grid cells remains constant. ODT solves the diffusive-reactive equations for mass, momentum, species, and energy in one spatial dimension, and one evolution dimension.

The turbulent flows are driven by the eddies. The eddy motions in ODT are governed by a strain field that is characterized by the velocity field along the notional line of sight as shown in Fig. 3.1. In Fig. 3.1, the white line perpendicular to the flow direction represents the ODT notional line of sight. The production mechanisms, the induced motions and the viscous transport are distinctive quantities in ODT. Each scalar profile in ODT is transported by the molecular-diffusive transport based on the appropriate diffusive coefficient and by a mapping process, triplet mapping, which is representative of eddies in a 3D flow.

ODT is capable of solving two different types of flows: i) temporal flows (time is the evolution dimension) and ii) spatial flows (space is the evolution dimension) on a notional line of sight. In the temporal flows, the white line is stationary and the flow variables like temperature and velocity evolve temporally on the line of sight. In the spatial flows, the line of sight moves in the direction of the flow to capture the flow variables on the whole flow domain.

3.1 One-dimensional Eddy

Turbulent advection in the turbulent flows are caused by eddies. Each eddy event in ODT is an instantaneous event which is represented by a triplet map. Triplet mapping is an instantaneous mapping of a segment of the 1D domain onto itself. The triplet map is implemented by the physical
mapping of fluid at a given location, \( f(y) \) to other location, \( y \) as:

\[ \phi(y) \rightarrow \phi(f(y)). \]  \hspace{1cm} (3.1)

The map is defined as

\[
f(y) = \begin{cases} 
3(y - y_0) & \text{if } y_0 \leq y \leq y_0 + \frac{1}{3}l, \\
2l - 3(y - y_0) & \text{if } y_0 + \frac{1}{3}l \leq y \leq y_0 + \frac{2}{3}l, \\
3(y - y_0) - 2l & \text{if } y_0 + \frac{2}{3}l \leq y \leq y_0 + l, \\
y - y_0 & \text{otherwise.}
\end{cases}
\]  \hspace{1cm} (3.2)

Figure 3.2 illustrates the basic principle of triplet mapping. Figure 3.2 (a) shows a 2D flow before and after an eddy where the solid white line represents the ODT line orthogonal to the flow direction. Figure 3.2 (b) demonstrates the triplet map which is a representation of a 2D eddy in 1D. In the figure, the solid line represents the initial scalar profile (profile along the ODT line before an eddy) and the dashed line represents the profile after an application of a triplet map (profile along the ODT line after an eddy). The map takes a scalar profile on an eddy region and replaces it with three copies of the original, each compressed by a factor of three with the middle copy inverted. The triplet mapping modifies the profiles of velocity, density, temperature, and mixture fraction, simulating an eddy in 1D. The ODT model solves three velocity components \((u, v, w)\). The triplet mapping conserves all properties such as energy and velocity moments by implementing energy transfer among the velocity components [12, 13]. In addition to the triplet mapping, the velocity components are modified using functions \(J\) and \(K\) (functions \(J\), and \(K\) are described in [69]) to incorporate return to isotropy process associated with pressure-velocity coupling as follows:

\[ s(y) \longrightarrow s(f(y)) + c_s K(y), \]  \hspace{1cm} (3.3)

where \(s\) is \(u, v,\) or \(w\), coefficient \(c_s\) is chosen to conserve the momentum and total energy, and \(K(y)\) is a Kernel function defined as

\[ K(y) = y - f(y). \]  \hspace{1cm} (3.4)
Figure 3.1: Representative image of an ODT domain. White line perpendicular to the flow direction is the ODT notional line of sight.

3.2 Eddy Sampling

Turbulence is a chaotic process where properties of the flow change irregula-}

rily due to random occurrences of eddies. Turbulence causes the form-

ation of eddies of many different length scales. These irregular and wide range of eddies must be modeled in ODT to simulate the randomness of eddies occurrence in the real flows. The eddy events in real flow are characterized by the frequency of the eddies, the location, $x_0$ and the size, $l$, of the eddies.

In ODT, the local momentum and kinetic energy fields in the notional line of sight govern the eddy size, location and frequency. The frequency of eddy events is related to a given eddy $(x_0, l)$ and is denoted by $\lambda(x_0, l)$. The eddy rate is modeled using dimensional arguments and is given by

$$\lambda = \frac{1}{\tau l^2},$$  \hspace{1cm} (3.5)

where $\tau$ is the eddy time scale given by

$$\frac{1}{\tau} = C \sqrt{\frac{2}{\rho l^3} (E_{\text{kin}} - Z E_{\text{vp}})},$$  \hspace{1cm} (3.6)

where $C$ is the adjustable eddy rate parameter, and $Z$ is the viscous penalty parameter. The viscous penalty term, $E_{\text{vp}}$, avoids the artificially small eddies that are not energetically possible by
reducing the kinetic energy. \( E_{\text{kin}} \) is the measure of the kinetic energy within the eddy interval. The expressions for \( E_{\text{kin}} \) and \( E_{vp} \) are given in [70]. The rate of all eddies is given by

\[
\Lambda = \int \int \lambda(x_0, l) dx_0 dl,
\]

(3.7)
where \( \lambda(x_0, l) dx_0 dl \) is the rate of the eddies between locations \( x_0 \) and \( x_0 + dx_0 \) and between sizes \( l \) and \( l + dl \). A joint probability density function of eddy sizes and locations \( \hat{P}(x_0, l) \) is given by

\[
\hat{P}(x_0, l) = g(x_0)f(l)g(x_0)
\]  

(3.8)

where the expression for \( g(x_0) \) and \( f(l) \) are given in [69, 71].

Each eddy is accepted with the probability:

\[
P_a = \Delta t m \lambda / \hat{P}(x_0, l).
\]  

(3.9)

In ODT, the eddy sampling and advancement occurs as follows:

1. Randomly sample an eddy with location \( x_0 \) and size \( l \) from the probability distribution function \( P(x_0, l) \) (Eq. 3.8).

2. Accept the eddy with probability \( P_a \) (Eq. 3.9).

3. Advance the time from \( t \) to \( t + \Delta t \) even if the eddy was not accepted.

4. If the eddy was accepted, solve the unsteady diffusion reaction equations up to the current eddy implementation time.

5. Repeat from step 1.

### 3.3 Solution Procedure

The equation system solved is a coupled set of ordinary differential equations. For simplicity, the evolution procedure is shown for the temporal formulation using an explicit Euler method. The spatial formulation follows the similar procedure.

1. Initialize the case with the appropriate velocity, enthalpy and species profiles.

2. Compute mass, heat, and momentum fluxes along with the reaction source term at time \( t_n \).

3. Advance the momentum, species and energy equations to obtain the velocity, enthalpy and composition at time \( t_{n+1} \) in each grid cell.
4. Compute necessary dependent variables like temperature, mixture fraction, thermal conductivity, etc. from the velocity, enthalpy, and/or composition.

5. Update the density using ideal gas law \( \rho_{n+1} = \frac{M_{n+1} P_{n+1}}{RT_{n+1}} \).

6. Solve the continuity equation to calculate a new grid size for all grid cells \( \Delta x_{n+1} = \frac{(\rho \Delta x)_n}{\rho_{n+1}} \).

7. Repeat steps 2-8 until \( t_{n+1} \) equals the end time.

### 3.4 Validation of ODT Model

![Figure 3.3: ODT of channel flow at Re=395. Plot (a) compares the scaled mean velocity profile with DNS; (b) shows the streamwise velocity profile (adapted from [69] with permission).](image)

In order to validate the ODT code, it is necessary to compare it with either experimental data or some other verified model. Lignell et al. [50, 69] compared ODT with DNS for various flow configurations. Figure 3.3 shows the comparative results of ODT simulation and DNS of a channel flow configuration with a turbulent Reynolds number \( (Re) \) of 395. Plot (a) shows a scaled mean streamwise velocity profile with a very good agreement with that of DNS. ODT is capable of capturing the essential elements of the near-wall flow including viscous sublayer. Plot (b) shows the instantaneous streamwise velocity profile.

Lignell et al. [69] also validated ODT code against a reacting ethylene jet flame. A temporally evolving, planar, ethylene jet flame simulation was performed with periodic and boundary
conditions. The Reynolds number of the jet was 5,120, and a detailed chemical mechanism was implemented with 19 transported and 10 quasi-steady-state species. Figure 3.4 shows the comparison of ODT and DNS for a turbulent jet flame. Plot (a) compares the full width half maximum of the mixture fraction profile with that of DNS. Plot (b) and plot (c) show the conditional mean temperature as a function of mixture fraction of ODT and DNS simulation respectively of the same configuration. Plot (a) shows a very good agreement with DNS and that ODT compares relatively well with DNS.

Figure 3.4: Comparison of ODT and DNS for a turbulent jet flame. Plot (a) shows the full width at half maximum of the mixture fraction profile. Plot (b) and plot (c) show the mean temperature as a function of mixture fraction (adapted from [69] with permission).

3.5 Summary

Computationally affordable resolution of the viscous scales in fully developed turbulence is a key feature of ODT [12]. ODT resolves all length scales in 1D and hence various statistical quantities can be abstracted, unlike conventional closure methods. ODT requires minimal empiricism and is able to incorporate complex molecular processes without introducing additional approximations [12]. It is applicable to many flows of fundamental and practical interest which are homogeneous in at least one spatial coordinate [12, 68]. ODT has been successfully applied to a variety of canonical flows [54, 68, 72] and many reacting and non-reacting flows including jet flames and pool fires.
CHAPTER 4. BIOMASS COMBUSTION

The ODT model is used as a base model for the turbulent biomass combustion. Wildland fires involve complex chemistry, and various modes of heat transfers in a wide range of fuels. Thus, the chemistry, heat transfer modes, and the biomass pyrolysis model must be incorporated in the ODT model. This chapter provides a detailed description of the sub-models added into ODT to extend its capability to model the biomass combustion. It also provides a brief description on the coupling of the gas and particles, interaction between them, and a short description on the mechanism involved in the particle heating.

4.1 Biomass Pyrolysis Model

The biomass pyrolysis model used in conjunction with ODT model is that of Nunn et al [59]. Nunn et al. studied the pyrolysis of dry sweet gum hardwood particles of size 45-88 µm. Lu et al. [73] notes that moisture in biomass occurs in two forms: free water and bound water. Free water exists in liquid forms in pores and cells, whereas bound water exists as chemically bound to surface sites or hydrated species. The Nunn et al. model does not have free water content as the fuel studied is assumed dry. The rate of formation of products assuming isothermal particles is given by the expression

\[
\frac{dY_v}{dt} = k(Y_{v\infty} - Y_v),
\]

where \(Y_{v\infty} = 0.93\) is the ultimate attainable yield, i.e., the yield at high temperature and long residence times, \(k\) is the reaction rate constant given by

\[
k = k_0 \exp \left( -\frac{E_a}{RT} \right)
\]

where \(k\) is the rate constant, \(k_0\) is the kinetic pre-exponential factor with a value of 33,884 1/s, \(E_a\) is the activation energy with a value of 16.5 kcal/mol, \(T\) is the temperature of the particle and \(R\) is the
Table 4.1: Nunn’s devolatilization model parameters.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Symbol</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Ultimate Yield</td>
<td>$Y_{v,\infty}$</td>
<td>92.97</td>
</tr>
<tr>
<td>Pre-exponential factor</td>
<td>$k_0\left(\frac{1}{s}\right)$</td>
<td>33,884</td>
</tr>
<tr>
<td>Activation energy</td>
<td>$E_a\left(\frac{kcal}{mol}\right)$</td>
<td>16.5</td>
</tr>
</tbody>
</table>

universal gas constant. Table 4.1 summarizes the Nunn’s model parameters. This pyrolysis model includes the formation of chemical (nonmoisture) water and the secondary cracking of biomass tars to form light gases.

Nunn’s pyrolysis model is a single reaction first order decomposition model which assumes that virgin material decomposes through a single reaction, i.e. from wood to volatiles. The model parameters were derived by nonlinear least-squares regression fitting procedure where the sum of the squared errors, between the calculated and observed yields, is minimized. Figure 4.1 shows the experimental data and model-generated curve for the gas production resulting from the pyrolysis of sweet gum hardwood particles. The fitted curve shows a statistically good agreement with the experimental values.

Figure 4.1: Experimental data and fitted curve for gas production from pyrolysis of sweet gum hardwood particles. Adapted from [59].
Assuming a constant volume (as biomass pyrolyzes, the biomass changes its density but the volume remains constant), volatile yield, $Y_v$, is given by

$$Y_v = 1 - \frac{\rho_0}{\rho}, \quad (4.3)$$

where $\rho_0$ is the initial density of the particle and $\rho$ is the density of the particle at time $t$. The spatial variation of temperature or composition within the particles is not considered. However, this model is applicable to a wide range of temperatures, whereas other single reaction models reviewed by Di Blasi et al. [55] are not able to predict the volatilization rate above 900 K.

The rate of change in mass of the particle is then given by

$$\frac{\partial m_p}{\partial t} = -\dot{Y}_v V_p \rho_p, \quad (4.4)$$

where $\rho_p$ is the density and $V_p$ is the volume of the particle. Mass leaving the particle is the gas fuel source term which is given as:

$$S_\phi = -\frac{\partial m_p}{\partial t}. \quad (4.5)$$

Nunn’s model is a relatively simple model that is applicable to temperatures up to 1400 K. Since the model is a single reaction model, it is computationally faster than the complex multi component models and is relatively easier to implement. This model has been successfully used to model the devolatilization of both wood and other solid fuels. Other models of devolatilization were also considered; however, Nunn’s model was chosen to be the suitable, and sufficient model that accomplishes the essence of the project which is to resolve the flame, and understand the detailed physics involved in the fire propagation rather than to quantify the gas and particle properties in the process.

4.2 Particle Heating

In wildland fires, biomass heat-up occurs mainly via radiation and convective heat transfer from the hot combustion gases to the surface of the biomass. The biomass then conducts the surface heat to the particle core via conduction. Pyrolysis then occurs as the biomass reaches its ignition temperature.
The biomass can be of different composition with varying size (typical aspect ratios between 2 and 15 [74]) and shape (cylindrical or spherical). The biomass particles considered in this project are cylinders and are that of sweet gum hardwood with size ranging from 0.5 mm to 1.2 mm. To neglect internal temperature gradients, the Biot number should be less than 0.1 [75]. However, the analysis on these particles revealed a Biot number (Bi) of about 0.5. This suggests that lumped capacitance method cannot be used and internal temperature gradient must be modeled.

The internal particle temperature gradients along with internally varying devolatalization rates based on those gradients are computed by solving the unsteady particle heat and mass equations. The mass and energy balances on the particle at cell $i$ results:

$$\frac{\partial m_i h_i}{\partial t} = q_w A_w - q_e A_e - h_v \dot{m}_i,$$  \hspace{1cm} (4.6)

$$\frac{\partial m_i}{\partial t} = -\dot{m}_i.$$  \hspace{1cm} (4.7)

The subscript $i$ denotes cell $i$, and $A_e, A_w$ are the east and west face areas, $q$ is the heat flux given by $q = -k(T) \frac{\partial T}{\partial r}$. The temperature dependent thermal conductivity $k(T)$ is taken from [73]. The boundary conditions to solve the above equations are

$$q|_{r=0} = 0,$$  \hspace{1cm} (4.8)

$$q|_{r=R} = -hA(\theta(T_o - T_p)) - (q^+ + q^-),$$  \hspace{1cm} (4.9)

where $q^+$ and $q^-$ are the radiative fluxes in opposite directions.

The solution of Eq. 4.6 and Eq. 4.7 using the above mentioned boundary conditions result in enthalpy and mass at each particle grid cell $i$. The temperature of the particles is then computed from the non-linear equation

$$f(T) = (\Delta h_f - h_i) + \int_{T_{ref}}^{T} C_p(T) dT = 0.$$  \hspace{1cm} (4.10)

Here, $\Delta h_f$ is the heat of formation of a particle, $h_i$ is the enthalpy of the particle, $T_{ref}$ is the reference temperature, and $C_p(T)$ is the temperature dependent heat capacity approximated as:
\[ C_p(T) = \left( a \left[ \frac{T}{K} \right]^3 + b \left[ \frac{T}{K} \right]^2 + c \left[ \frac{T}{K} \right] + d \right) \frac{J}{kg \cdot K}, \quad (4.11) \]

where \( T \) is the temperature of a particle in units of K, \( a = 7.6658 \times 10^{-7} \), \( b = -3.6353 \times 10^{-3} \), \( c = 5.9279 \), and \( d = -3.26.96 \) are the polynomial constants. For \( T > 2000 \text{K} \), Eq. 4.11 is evaluated at \( T = 2000 \text{K} \). Equations 4.10 and 4.11 are then solved to find the temperature, \( T \), of a particle by using Newton’s method where \( T \) at time \( t \) is given by

\[ T_i^{k+1} = T_i^k - \frac{f(T_i^k)}{f'(T_i^k)}, \quad (4.12) \]

where \( k \) is an iteration number and \( f'(T_i^k) \) is estimated as

\[ f'(T_i^k) = \frac{f(T_i^k + dT) - f(T_i^k)}{dT}, \quad (4.13) \]

where \( dT \) is a step size. Equation 4.12 converges if \( \text{abs}(T_i^{k+1} - T_i^k) \leq T_{tol} \), where \( T_{tol} \) is the tolerance level which is currently set to a value of \( 1 \times 10^{-3} \text{K} \).

Each particle in the ODT model is a pseudo particle. Each pseudo particle represents a number of other particles with identical properties. Doing this reduces the computational cost as the transport equation needs to be solved only for one pseudo particle rather than for all the particles with identical properties. The number of real particles that a pseudo particle represents is given by:

\[ n_{real} = \frac{(1 - \gamma)V}{V_p N_{pseudo}}, \quad (4.14) \]

where \( V_p \) is the volume of the particle, \( N_{pseudo} \) is the total number of pseudo particles in the given cell, \( \gamma \) is the void fraction, and \( V \) is the volume of the cell defined as \( A\Delta x \), where \( A = 1m^2 \) is the perpendicular area. The void fraction, \( \gamma \) is defined as \( (V - V_p)/V \), which reduces to

\[ \gamma = 1 - \left( \frac{\xi}{1 - \xi} \right) \left( \frac{\rho_{air}}{\rho_p} \right), \quad (4.15) \]

assuming \( V_{air} \gg V_p \).
4.3 Table Lookup and Flamelet Solution

The forest fuels involve a number of compounds and a complex combustion chemistry. Implementing detailed chemistry is difficult and modeling chemical kinetics can be computationally expensive. However, there are evidences that the forest fires are large fires and are limited by mixing instead of the complex chemical kinetics involved. This serves as a motivation to use lookup tables or global mechanisms. The global mechanism approach involves solving transport equations for chemical species which makes it computationally more expensive than the lookup table approach. Hence, lookup table approach is chosen as a representation of the combustion process as lookup tables are easier to implement as well.

The lookup table to model solid biomass combustion is pre-generated using a flamelet formulation process assuming a unity Lewis number in which a solution is obtained as a function of the mixture fraction using 102 uniformly spaced grid points. The solution is obtained for 19 different heat loss points uniformly spaced between -0.5 and 0.4. The mixture fraction and enthalpy are the only two transported variables since the products from the pyrolysis are not modeled. This reduces the number of scalars being transported. All other properties like the temperature, products, and other gas-phase properties like temperature, density, viscosity, soot formation etc. are functions of the mixture fraction and heat loss. Some of the reasonable assumptions inherent using table lookup approach include:

1. All biomass offgases have the same elemental composition, and heat of formation

2. All offgases species have the same diffusivity

3. All kinetic reaction rates go instantly to equilibrium, limited only by mixing off gases with bulk gas.

Figure 4.2 shows the temperature and density profile as a function of heat loss and mixture fraction. In the figure, positive values denote heat loss, and negative values indicate heat gain.
Figure 4.2: Flamelet temperature profile (plot (a)) and density profile (plot (b)) as a function of mixture fraction at different heat loss values.

Heat loss, $\gamma$, is calculated as

$$\gamma = \frac{h_a - h}{h_{sens}},$$

(4.16)
Table 4.2: Stream properties for pure fuel and pure air.

<table>
<thead>
<tr>
<th>Compound</th>
<th>Mole Fraction</th>
<th>Compound</th>
<th>Mole Fraction</th>
</tr>
</thead>
<tbody>
<tr>
<td>$CH_4$</td>
<td>0.35975</td>
<td>$CH_4$</td>
<td>0.0</td>
</tr>
<tr>
<td>$CO_2$</td>
<td>0.625</td>
<td>$CO_2$</td>
<td>0.0</td>
</tr>
<tr>
<td>$CO$</td>
<td>0.015625</td>
<td>$CO$</td>
<td>0.0</td>
</tr>
<tr>
<td>$O_2$</td>
<td>0.0</td>
<td>$O_2$</td>
<td>0.21</td>
</tr>
<tr>
<td>$N_2$</td>
<td>0.0</td>
<td>$N_2$</td>
<td>0.79</td>
</tr>
</tbody>
</table>

$T = 469.0 \text{ K}$  

$T = 298.0 \text{ K}$

where $h_a$ is the adiabatic enthalpy, $h$ is the local enthalpy and $h_{sens}$ is the local adiabatic sensible enthalpy. The adiabatic enthalpy, and adiabatic sensible enthalpy are functions of mixture fraction. The pyrolyzed gases are burned at the given enthalpy according to the flamelet solution.

Since the pyrolysis products are unknown, a surrogate gas composition and temperature were found that matches the elemental composition and enthalpy of formation of the pyrolysis products. The biomass is taken as a sweet gum dry hardwood with chemical formula $C_{32}H_{36}O_{21}$ [59]. The surrogate pyrolysis products were taken to be $CH_4$, $CO_2$, and $CO$ at 469 K with mole fractions of 0.35938, 0.01563, 0.625, respectively. The streams properties corresponding to mixture fraction values of $\xi = 0$ (pure air) and $\xi = 1$ (pure fuel) are summarized in Table 4.2.

4.4 Radiation Model

Wildland fires involve high temperature combustion processes. At such high temperatures, radiation heat transfer appears as an important heat transfer mechanism [76]. Hence, to better predict the temperature profile of the gas and the biomass, radiation must be accounted for and modeled.

Flames emit radiation from gases and soot. Infrared active species such as $CO$, $CO_2$, and $H_2O$ are present in the products of combustion that are responsible for the non-luminous radiation in gases [76]. Soot is a major contributor of thermal radiation. Radiation causes heat loss which is generally in the range of 10-20%. Even though radiation is an important heat transfer mechanism in combustion systems, the inclusion of radiative heat transfer into the analysis of the combustion system is very difficult.
There are various radiation models of which the optically thin model [77, 78] and the two-flux model [50, 79] are implemented in the ODT code. The optically thin model is the simplest of the two and accounts for absorption and emission along a line. This model allows for direct radiation exchange with the surroundings. The volumetric radiative source for the optically thin model is given by

\[ Q_{\text{rad}} = -4\sigma K(T^4 - T_{\infty}^4), \]

where \( \sigma \) is the Stephan-Boltzman constant, \( T \) is the gas temperature, \( T_{\infty} \) is the surrounding gas temperature, and \( K \) is the Plank mean absorption coefficient which is calculated as

\[ K = \sum_i x_i PK_i, \]

where \( i \) is the species, \( x \) is the mole fraction, \( P \) is the pressure and \( K_i \) is the temperature dependent species absorption coefficient adapted from [79]. Since compounds like \( CO_2 \), \( H_2O \) and soot contribute to the thermal radiation, \( K_i \) is taken as a combination of the gas and soot values as:

\[ K = K_g + 1220f_vT, \]

where \( f_v \) is the soot volume fraction and \( T \) is the temperature of the gas. The drawback of the optically thin model is that it over-predicts radiative losses from the \( CH_4 \) flames due to neglect of strong reabsorption by the 4.3-micron band of \( CO_2 \) [80].

Radiation of the gas phase including the particles is modeled using the two-flux model. This model is a more accurate model than the optically thin model and is a common approximation in 1D configurations. The two-flux model computes the two radiative heat flux, \( q^+ \) and \( q^- \) which are given by

\[ \frac{dq^+}{dx} = 2k_g\sigma T_g^4 + 2k_s\sigma T_s^4 - 2k_gq^+ - 2k_sq^+, \]

\[ \frac{dq^-}{dx} = -2k_g\sigma T_g^4 - 2k_s\sigma T_s^4 + 2k_gq^- + 2k_sq^- . \]

In Eq. 4.20 and 4.21, \( k_g \) and \( k_s \) are the gas and solid absorption coefficients, respectively, \( \sigma \) is the Stefan-Boltzman constant, \( T_s \) is the surface temperature of the particles, \( T_g \) is the gas temperature,
and \( x \) is the ODT line direction. The solid absorption coefficient is given by \( k_s = \varepsilon_s \frac{\pi}{4} d^2 n_d \) [79], where \( n_d \) is the particle number density, and \( \varepsilon_s \) is the solid emissivity.

The volumetric radiation source terms as a function of radiative heat flux (Eq. 4.20 and 4.21) for the gas and particles respectively are then given by

\[
Q_g = 4k_g \sigma T_g^4 - 2k_g (q^+ + q^-),
\]

\[
Q_s = 4k_s \sigma T_s^4 - 2k_s (q^+ + q^-).
\]

4.5 Soot Model

Soot is a collection of impure carbon particles that are produced due to incomplete combustion of the hydrocarbons in fuel. As noted above, soot accounts for the bulk of the thermal radiation in flames and fires. Thus, it is necessary to approximate the soot formation and composition in flames. Soot formation is a complex process. Even though soot contributes to the flame temperature, due to its complexity, soot modeling is subject to large uncertainties.

The model of Leung et al. [81], a semi-empirical model, is adapted and implemented in the ODT code. This model reasonably predicts the soot formation and is a computationally inexpensive model. According to this model, soot is formed in four steps: i) nucleation, ii) growth, iii) oxidation and, iv) coagulation. Soot is described using the number density and the mass fraction, and is assumed to have a monodispersed size distribution. The soot volume fraction, \( f_v \), is given as

\[
f_v = \rho Y_s / \rho_s
\]

where \( Y_s \) is the soot mass fraction, \( \rho \) is the gas density, and \( \rho_s \) is the soot density.

Soot in this model nucleates and grows via acetylene mass fraction whereas oxidation is based on the oxygen mass fraction. The soot formation can be approximated using acetylene as an available species. The mixture fraction and enthalpy are transported during the ODT simulation. Based on the the instantaneous heat loss and mixture fraction, acetylene concentration can be approximated using lookup table. This acetylene concentration is then used to calculate the soot formation rates [82] given as:
\[ r_1 \left[ \text{kmol/m}^3\text{s} \right] = 0.1 \times 10^5 \exp \left( -\frac{21100}{T} \right) [C_2H_2], \]  

\[ r_2 \left[ \text{kmol/m}^3\text{s} \right] = 0.6 \times 10^4 \exp \left( -\frac{12100}{T} \right) [C_2H_2] \sqrt{A_s}, \]  

\[ r_3 \left[ \text{kmol/m}^3\text{s} \right] = 0.1 \times 10^5 \sqrt{T} \exp \left( -\frac{19680}{T} \right) [O_2] A_s, \]  

\[ r_4 \left[ \text{1/m}^3\text{s} \right] = 2C_a \left( \frac{6MW_C}{\pi \rho_s} \right)^{1/6} \left( \frac{6k_b T}{\rho_s} \right)^{1/2} \left( \frac{\rho Y_s}{MW_C} \right)^{1/6} n^{11/6}, \]  

where MW\(_C\) is the mean molecular weight of carbon, [C\(_2\)H\(_2\)] and [O\(_2\)] denote the concentrations of acetylene and oxygen, \( n \) is the number density, \( Y_s \) is the soot mass fraction, \( \rho_s \) is the soot density, and \( A_s \) is the particle surface area per unit volume given as:

\[ A_s = \pi \left( \frac{6\rho Y_s}{\pi \rho_s} \right)^{2/3} n^{1/3}. \]  

These rates are then used to calculate the number density and soot mass fraction source terms [82, 83] which are given as:

\[ S_n = \frac{2N_A r_1}{C_{min}} - r_4, \]  

\[ S_{Y_s} = MW_C \cdot (2r_1 + 2r_2 - r_3), \]

where \( N_a \) is the Avogadro’s number, and \( C_{min} = 100 \) is the number of carbons present in a particle resulted from the nucleation of particles with acetylene. Figure 4.3 shows the acetylene concentration as a function of mixture fraction at several heat loss values. In the figure, positive values denote heat loss, and negative values indicate heat gain.
4.6 Gas and Particle Interaction

Wildland fires evolve and spread due to interaction between the hot gases and pyrolyzing fuel. The combustible hot gases are evolved due to the pyrolysis of the biomass. The hot gases continually combust and transfer heat to the fuel via conduction, convection, and radiation. The heat from the gas pyrolyzes the biomass and evolves the combustible hot gases. This cycle of pyrolysis, combustion, and heat transfer spreads and sustains the flame. To extend the capability of ODT to model combustion, ODT must be fused with the reacting particles.

A fully Lagrangian particle model [54, 84] is implemented into the ODT model. The Lagrangian particles interact with both events, the diffusive process and the eddy events. Hence, these Lagrangian particles flow with the gas and gain velocity through turbulent advection. However, for the biomass combustion, the fuel is primarily fixed such that the particle velocity is zero. Since the particles in the model are stationary, they act as a momentum sink for the gas and create drag.

Without the biomass reaction, there is no source term for the gas mass, and the continuity equation in cell $i$ reduces to $\rho_i\Delta x_i = C$ which implies that as the density $\rho$ changes, the grid spacing, $\Delta x$, in grid cell $i$ also changes accordingly. With the biomass combustion, a gas mass source term
is introduced. Solving the Reynolds transport theorem

\[
\frac{d}{dt} \int_{\Omega(t)} \rho \eta dV = \frac{\partial}{\partial t} \int_{\Omega} \rho \eta dV + \int_{\Pi} \rho \eta \vec{v}_{sys} \cdot \vec{n} dA,
\]

where \( \eta \) is a variable representing some quantity per mass, \( \rho \) is the density, \( \vec{v}_{sys} \) is the system velocity, \( \Omega(t) \) is the Lagrangian system, \( \hat{\Omega} \) is the Eulerian control volume and \( \hat{\Pi} \) is the boundary,

the continuity equation for any given cell becomes:

\[
\frac{d(\rho \gamma \Delta x)}{dt} = S_{\phi} \gamma \Delta x,
\]

where \( \gamma \) is the cell void fraction calculated as \( \gamma_i = (V_i - V_p)/(V_i) \) where \( V_i \) is the volume of the cell \( i \) and \( V_p \) is the total volume of particles in that cell. With the biomass combustion, the set of unsteady differential equations in the ODT code are also changed accordingly. For example, a given scalar \( \eta \) representing an intensive property (e.g., \( u, v, w, h \), etc.) has the following transport equation without a source term from the biomass combustion.

\[
\frac{d\eta}{dt} = -j_e - j_w \rho \Delta x + S_{\eta} \rho.
\]

(4.33)

With the incorporation of the biomass combustion, the above equation modifies to,

\[
\frac{d\eta}{dt} = -\frac{j_e - j_w}{\rho \gamma \Delta x} + \frac{S_{\eta}}{\rho} - \frac{\eta S_{\phi}}{\rho}.
\]

(4.34)

Equation 4.34 differs from Eq. 4.33 by the addition of \(-\eta S_{\phi}/\rho\) term on the right hand side and division of the second term by void fraction, \( \gamma \). Equations (4.35-4.40) summarize the gas-phase system including the source term from the wood pyrolysis derived using the Reynolds transport equation.
\[
\frac{dm}{dt} = S_\phi \gamma \Delta x, \tag{4.35}
\]

\[
\frac{du}{dt} = -\frac{\tau_{u,e} - \tau_{u,w}}{\rho \gamma \Delta x} + \frac{\sum_i (u_{p,i} - u_{g}) m_{p,i} f / \tau_i}{\rho \gamma \Delta x} + \frac{(\rho_\infty - \rho) g}{\rho} - \frac{S_\phi}{\rho}, \tag{4.36}
\]

\[
\frac{dv}{dt} = -\frac{\tau_{v,e} - \tau_{v,w}}{\rho \gamma \Delta x} + \frac{\sum_i (v_{p,i} - v_{g}) m_{p,i} f / \tau_i}{\rho \gamma \Delta x} - \frac{S_\phi}{\rho}, \tag{4.37}
\]

\[
\frac{dw}{dt} = -\frac{\tau_{w,e} - \tau_{w,w}}{\rho \gamma \Delta x} + \frac{\sum_i (w_{p,i} - w_{g}) m_{p,i} f / \tau_i}{\rho \gamma \Delta x} - \frac{S_\phi}{\rho}, \tag{4.38}
\]

\[
\frac{dh}{dt} = -\frac{q_e - q_w}{\rho \gamma \Delta x} - \frac{\sum_i h_{i} \Delta p_i (T - T_{p,i})}{\rho \gamma \Delta x} + \frac{S_{\text{rad}}}{\rho} + \frac{h_{\nu} S_\phi}{\rho} \frac{S_\phi}{\rho}, \tag{4.39}
\]

\[
\frac{d\xi}{dt} = -\frac{\dot{j}_{\xi,e} - \dot{j}_{\xi,w}}{\rho \gamma \Delta x} + \frac{S_\phi}{\rho \gamma} - \frac{\xi}{\rho} \frac{S_\phi}{\rho}. \tag{4.40}
\]

In the above equations, \(\tau\) represents the stress tensor, \(e\) and \(w\) represent the east and west cell faces, \(u\) is the streamwise velocity, \(h\) is the enthalpy, \(\xi\) is the mixture fraction, \(S\) represents the source term, \(\nu\) is the kinematic viscosity of the gas, \(g\) is the gravitational acceleration that appears in the buoyant term of the streamwise (\(u\)) momentum equation, \(\phi\) represents the biomass combustion, \(f\) is the friction factor (third term in the momentum equations) resulted due to stationary particles, and subscript \(p\) and \(g\) denote the particle and gas properties respectively.

The relations for stress tensor, \(\tau\), and species and heat flux, \(j\) and \(q\) are given as:

\[
\tau_{xx} = -\mu \frac{du}{dx}, \tag{4.41}
\]

\[
j_i = -\frac{\rho Y_i D_i \, dx_i}{X_i} = \rho D_i \frac{dY_i}{dx} - \frac{\rho D_j Y_j \, dM}{M \, dx}, \tag{4.42}
\]

\[
q = \lambda \frac{dT}{dx} + \sum_i h_{ij_i}, \tag{4.43}
\]
where \( M \) is the mean molecular weight, \( D_i \) is the species diffusion coefficient, \( X_i \) is the mole fraction of species \( i \), \( \lambda \) is the gas thermal conductivity, \( T \) is the gas temperature, and \( h_i \) is the enthalpy of species \( i \).

The enthalpy equation (Eq. 4.39) involves the convection between the particles and the gas. The convection heat transfer is characterized by the heat transfer coefficient \( \tilde{h} \). The heat transfer coefficient between the particle and the gas is computed as \( \tilde{h} = k Nu / D \), where \( k \) is the gas thermal conductivity, \( D \) is the diameter of the particle, and \( Nu \) is the Nusselt number computed using correlations adapted from [75] as shown in Eq. 4.44 and 4.45.

\[
Nu_{sphere} = 2 + 0.6Re^{1/3}Pr^{1/3},
\]

\[
Nu_{cylinder} = 0.3 + \frac{0.6Re^{1/2}Pr^{1/3}}{1 + \left( \frac{0.4}{Pr} \right)^{3/4}} \left[ 1 + \left( \frac{Re}{282000} \right)^{3/4} \right]^{4/3},
\]

(4.44)

(4.45)

Here \( Re \) is the Reynolds number and \( Pr \) is the Prandtl number.

The wildland fires involve high temperature combustion resulting in high heat and mass transfer rates. For mass heat transfer rates, the heat transfer coefficient is modified by including a blowing factor [85, 86] given as:

\[
\theta = \frac{\phi}{e^\phi - 1},
\]

(4.46)

where \( \phi \) is the rate factor given as:

\[
\phi = \frac{N_{A0} \bar{C}_{pA} + N_{B0} \bar{C}_{pB}}{h_{loc}},
\]

(4.47)

where \( N \) is the flux of species \( A \) or \( B \), \( \bar{C}_p \) molar heat capacity, and \( h_{loc} \) is the local heat transfer coefficient.

### 4.7 Summary

ODT is an extension to the LEM model. ODT, like LEM, is a stochastic model to compute statistical properties of the velocity and scalar fields on a 1D domain. ODT is applicable to many reacting and non-reacting flows including jet flames and pool fires. ODT is extended to model
turbulent flame propagation in biomass fuel beds representative of wildland fires. A two-flux model is used to model radiation as it is reasonably accurate and is a common approximation in 1D configurations. A semi-empirical soot model of Leung et al. [81] is adapted and implemented in the ODT code.

Wildland fires involve complex combustion chemistry. The lookup table approach is implemented in the ODT code as it is easier to implement and is computationally faster. The Lagrangian particles are implemented in ODT model and the Eulerian transport equations are modified by including the source terms from biomass combustion. The biomass pyrolysis model is adapted from Nunn et al. [59], which has been successfully used to model both wood and other solid fuels. Particle heating mainly occurs via radiation, convection, and conduction. The particle internal temperature distribution is considered. The particles are assumed to be dry and char oxidation is neglected.
CHAPTER 5. RESULTS

5.1 Laminar Test Cases

5.1.1 Configuration

Before the simulation of turbulent biomass combustion, a few temporal laminar test cases were performed in buoyant environment to understand the heat loss, and basic behavior of heat transfer via convection, internal particle conduction, and radiation from gas to the particles. In these laminar test cases, the eddies (triplet maps in ODT) are turned off and ODT variables (like mixture fraction and enthalpy) transport only by diffusion. These tests were performed on a 0.2 m domain with a simulation time of 1 s. Initially the domain is divided into 300 cells with one ODT particle in each cell.

The loading of the biomass is 0.0985 kg throughout the domain which is based on the stoichiometric void fraction of 0.999717. The void fraction is an input parameter to the model. Equation 4.15 is evaluated at the desired mixture fraction value to compute the void fraction. Then Eq. 4.14 is evaluated to compute the number of real particles each pseudo particle represents. Each ODT particle (pseudo particle) is a sweet gum hardwood particle with an initial density of 700 kg/m$^3$ and a diameter of 1 mm. Each particle is a cylindrical particle with a constant length of 1 m. The length of the particles is chosen to be significantly greater than the diameter for simplicity. The particles are stationary and as they burn, their density changes but the volume essentially remains constant. The 1 m long particle is coiled inside the grid cell on which it resides. The effective intertwining of the particle inside the grid cell is in such a way that there are $L/\Delta x$ number of length segments which do not cross each other. The representative image of a particle residing in a grid cell $i$ is shown in Fig. 5.1. Such orientation of the particle is chosen for the ease of the involved calculations.
Ignition is started on the domain with a predefined mixture fraction profile. The hyperbolic tangent mixture fraction profile with the corresponding temperature profile is shown in Fig. 5.2. The width of the mixture fraction profile is 0.04 m (1/5th of the domain) with a transition width of 0.02 m on each side. As the simulation progresses, only 2 variables, mixture fraction and enthalpy, get transported and other scalars are computed using the lookup table. The two-flux model is used for radiation and buoyancy is incorporated. It is assumed that the radiation heat exchange exists between the particle external surface and surrounding environment. Pyrolysis products and the solid particles are assumed to have the same temperature. The gas phase and particle phase both are reactive. The dependent scalars are then calculated using the lookup table using the following steps:

1. Calculate heat loss based on the enthalpy and mixture fraction at cell \(i\) using Eq. 4.16.

2. Find the mixture fraction, \(\xi\), and heat loss, \(\gamma\) that bounds the cell mixture fraction and heat loss.

3. Bilinearly interpolate to obtain dependent scalars like temperature, density, viscosity, etc.

### 5.1.2 Laminar Test Results

Four different cases with four modes of heat transfer were run as the test cases. The cases included A) case with no radiation and convection, B) case with convection but no radiation, C) case with radiation but no convection, and D) case with both convection and radiation.

Figure 5.3 shows the initial and final temperature profiles of gas and particles and Fig. 5.4 shows the corresponding heat loss profiles for these test cases. In both figures, plot (a) corresponds to case A and so on for other cases.
Figure 5.2: Initial gas mixture fraction, ($\xi$), and gas temperature profile.

Figure 5.3: Initial and final temperature profiles of gas and particles for various laminar test cases. Plot (a) shows profiles without radiation and convection, (b) shows profiles with convection but without radiation, (c) shows profiles with radiation but without convection, and (d) shows profiles with both radiation and convection.
Case A (Without Radiation and Convection)

Without any modes of heat transfer, the initial gas temperature peaks broaden and the width of the profile increases uniformly on both sides of the profile. The peak temperature of 2100 K stays at 2100 K because of mixing and ongoing combustion. The conduction from the gas to the particles is neglected, hence the particle temperature does not change due to absence of convection and radiation heat transfer. The corresponding heat loss profile (Fig. 5.4 (a)) indicates that there is no heat loss in the system.

Case B (With Convection but Without Radiation)

When convection is added, the temperature profile broadens and increases in width but not as much as in case A. The peak temperature falls from 2100 K to about 2000 K at the end of 1 s. The corresponding heat loss profile (Fig. 5.4 (b)) indicates that the drop in temperature is due to heat loss from the flame zone. The overall heat loss is around 10%. Particle temperature rises from an initial temperature of 298.15 K to a peak temperature of 500 K. The particle temperature profile is similar to the shape of the gas temperature profile but there is a noticeable time lag between these two profiles. Particle surface temperature is always below the gas temperature.

Case C (With Radiation but Without Convection)

When the radiation is added, the peak gas temperature of 2100 K falls to a final temperature of about 1000 K, a drop of 1100 K in 1 s. The deep concave up profile in the middle of the gas temperature profile reduces to a shallow flat profile, but the outside profile remains the same. It is indicative from the temperature profile that there is a significant amount of heat loss from the flame. The corresponding overall heat loss profile (Fig. 5.4 (c)) confirms that there is as much as 70% of heat loss in the flame zone. In turbulent flames, the significant decrease in temperature is not apparent as the decrease in temperature is compensated by the continual combustion of gases resulted from pyrolysis.
Unlike case A, the change in particle temperature is small. The corresponding heat loss profile indicates that there should be more heat loss in the flame zone than on the sides of the flame. The numerical values of the particles’ final temperature value confirm that the particle temperature rises to 340 K in the flame zone and decreases to 335 K on the sides of the flame.

The significant heat loss strongly impacts the particle heating, particle reaction rate, soot precursor concentrations, and soot formation rate. There is a dual competing behavior of high temperature producing soot that contributes to high radiation, and heat loss due to radiation that suppresses the formation of soot which reduces the radiation rates.
Case D (With Convection and Radiation)

When convection and radiation are both turned on, the radiation causes the gas temperature to decrease more significantly than the convection. The final gas temperature profile and the corresponding heat loss profile resemble case C. The final gas temperature has the combined effect of both convection and radiation. The peak particle temperature is 442 K as compared to 500 K in case B (with convection only) and 335 K in case C (with radiation only).

Time Scales

Insight into the behavior and simplified mechanism of the system can be gained by analyzing timescales. Hence, it is important to compare the various timescales involved to see how they are overlapped with each other or separated from each other. The timescales analyzed for this system are those of conduction, convection, radiation, and biomass pyrolysis (reaction). The intrinsic timescale for any process can be defined as \( \Delta T_{\text{max}} / (dT/dt)_{\text{max}} \). This definition gives the time taken for a system to reach a maximum temperature, \( T_{\text{max}} \), from a minimum system temperature, \( T_{\text{min}} \) with a maximum rate of change in temperature, \( dT/dt|_{\text{max}} \).

From the application of the definition of the timescale, the intrinsic time scale for the gaseous radiation is given by

\[
\tau_{\text{rad}} = \frac{T - T_{\infty}}{4\sigma k_g \rho c_p (T^4 - T_{\infty}^4)}. \tag{5.1}
\]

The intrinsic convection time scale is given as:

\[
\tau_{\text{conv}} = \frac{m_p C_{p, p}}{h_{\text{conv}} A_p} = \frac{R^2 L^2 \rho C_{p, p}}{2k_g Nu R (R + L)}, \tag{5.2}
\]

where \( k_g \) is the gas thermal conductivity, and \( Nu \) is the Nusselt number. With gas velocity \( v=0 \) m/s the peak gas temperature and initial particle temperature of 300 K, the convection timescale is \( \tau_{\text{conv}}=3.9 \) s, for cylindrical, 1 m length particles with diameter \( D=1 \) mm. This means that via convection, for particles to reach a temperature of 1,000 K, at least 3.9 s of simulation time is required. This result is consistent with the change in the particle temperature through the simulation in case B.
The internal particle conduction timescale is given as

\[ \tau_{\text{cond}} = \frac{R_p^2}{\alpha} = \frac{D^2 \rho C_{p,p}}{4k_p}, \]  

(5.3)

With D=1 mm, the conduction timescale, \( \tau_{\text{cond}} = 1.5 \) s.

The timescale for radiative transport to the particles is estimated using \( m_p C_{p,p} dT_p/dt = Q_s/n_d \) along with \( \tau_{\text{rad},p} = (T_g - T_p)/(dT_p/dt) \), where \( Q_s \) is given by Eq. 4.23, and \( m_p \) is the mass of a single particle.

\[ \tau_{\text{rad}} = \frac{m_p C_{p,p} n_d (T_g - T_p)}{Q_s}. \]  

(5.4)

The particle radiative timescale is \( \tau_{\text{rad},p} = 25 \) s at the initial conditions for cylindrical particles. This value is consistent with the particle temperature rise in case C.

The ignition temperature, the temperature after which significant pyrolysis reaction starts to occur, for the particles is around 850 K. In all laminar test cases, the temperature of the particles is well below their ignition temperature which results in almost zero pyrolysis reaction. However, the reaction timescale may be estimated as the time to react the particle at a given particle reaction rate. The timescale gives the approximate time for a particle of any size to react at the given temperature. If the reaction timescale is less than the heat transfer timescale, it is indicative that particle heating takes longer than the reaction and the process of pyrolysis becomes heat transfer limited. Conversely, if the heat transfer timescale is less than the reaction timescale, the biomass pyrolysis will be reaction limited. The particle reaction rate is temperature dependent, hence the timescales are also temperature dependent. Figure 5.5 shows the reaction timescale as a function of particle temperature. The graph is based on \( \tau_{\text{rxn}} = m_p/(dm_p/dt)_{\text{max}} \) and Eq. 4.2. The reaction timescale (y-axis) is shown on a log scale. As the temperature of the particles increase, the reaction timescale decreases exponentially. Table 5.1 summarizes the reaction and heat transfer timescales for gas and cylindrical particles.
Table 5.1: Summary of reaction and heat transfer timescales.

<table>
<thead>
<tr>
<th>Properties</th>
<th>Definition</th>
<th>Symbol</th>
<th>Value (s)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Gas Rad</td>
<td>$(T - T_{\infty})\rho C_p/4\sigma k_a (T^4 - T_{\infty}^4)$</td>
<td>$\tau_{g,\text{rad}}$</td>
<td>0.14</td>
</tr>
<tr>
<td>Part. Rad</td>
<td>$m_p C_p p d (T_g - T_p) / Q_s$</td>
<td>$\tau_{p,\text{rad}}$</td>
<td>25</td>
</tr>
<tr>
<td>Part. Conv ($v = 0 \text{ m/s}$)</td>
<td>$R^2 L^2 \rho C_p p / 2k g R (R + L)$</td>
<td>$\tau_{\text{conv}}$</td>
<td>4.7</td>
</tr>
<tr>
<td>Part. Conv ($v = 5 \text{ m/s}$)</td>
<td>&quot;</td>
<td>&quot;</td>
<td>0.7</td>
</tr>
<tr>
<td>Part. Cond</td>
<td>$D^2 \rho C_p p / 4kp$</td>
<td>$\tau_{\text{cond}}$</td>
<td>1.5</td>
</tr>
<tr>
<td>Reaction (800 K)</td>
<td>&quot;</td>
<td>$\tau_{\text{rxn}}$</td>
<td>1</td>
</tr>
<tr>
<td>Reaction (1,030 K)</td>
<td>&quot;</td>
<td>&quot;</td>
<td>0.1</td>
</tr>
<tr>
<td>Reaction (1,440 K)</td>
<td>&quot;</td>
<td>&quot;</td>
<td>0.001</td>
</tr>
</tbody>
</table>

Figure 5.5: Particle reaction timescale.

Summary

These laminar test cases, indicate that the heat transfer from the hot gases to the particles happens mainly due to convection. There is a significant amount of heat loss due to radiation, and it is due to the radiation that the gas temperature falls significantly in 1 s (case D). The heat transfer is high in the flame zone where the gas temperature is high.

Based on the laminar test cases, the radiation time scale for particle heating was found to be 25 s, the convection time scale to be 3.9 s, the conduction time scale to be 1.55 s, and reaction time scale at 800 K to be 1 s, and at 1,000 K to be 0.1 s. From these time scales analyses, it is evident that the combustion process is reaction limited until the particles reach the temperature of 800 K after which the process becomes heat transfer limited.
Table 5.2 summarizes the effect of radiation and convection on gas and particle temperatures. It also shows the peak heat loss in different cases at the end of 1 s.

Table 5.2: Heat loss, initial, and final gas and particle temperatures for laminar test cases.

<table>
<thead>
<tr>
<th>Cases</th>
<th>A</th>
<th>B</th>
<th>C</th>
<th>D</th>
</tr>
</thead>
<tbody>
<tr>
<td>$T_{0,\text{gas}}$</td>
<td>2,100 K</td>
<td>2,100 K</td>
<td>2,100 K</td>
<td>2,100 K</td>
</tr>
<tr>
<td>$T_{\text{final,,gas}}$</td>
<td>2,100 K</td>
<td>2,000 K</td>
<td>1,000 K</td>
<td>1,000 K</td>
</tr>
<tr>
<td>$T_{0,\text{part}}$</td>
<td>300 K</td>
<td>300 K</td>
<td>300 K</td>
<td>300 K</td>
</tr>
<tr>
<td>$T_{\text{final,,part}}$</td>
<td>300 K</td>
<td>500 K</td>
<td>345 K</td>
<td>442 K</td>
</tr>
<tr>
<td>$\gamma$</td>
<td>0%</td>
<td>10%</td>
<td>70%</td>
<td>70%</td>
</tr>
</tbody>
</table>

5.2 Turbulent Simulations

Combustion requires that the fuel and air be mixed at the molecular level as combustion happens at the molecular level. Turbulence causes mixing because of the range of different size eddies that cause strain and shear at the interface between the eddies that increase mixing. Turbulence also increases the rate of heat transfer to the particles (via convection) such that conduction and reaction become the limiting process. To understand the effect of turbulence in biomass combustion, and flame spread, a series of ODT simulations were performed.

Two different configurations of the simulation were performed. The first configuration involves a temporal simulation of a buoyant plume with ODT as a base model. The second configuration is a homogeneous turbulent flow with LEM as a base model. Both set of simulations consist of baseline cases with variations on particle diameter and biomass loading.

5.2.1 Buoyant Plume

Configuration

The buoyant plume simulations were run in a 10 m domain with a simulation time of 2 s. The domain is intentionally set significantly larger than the particle size to attain a steady state spread rate. Initially, the domain is divided into 15,000 equally spaced cells. Each grid cell contains one ODT particle and the void fraction in each cell is 0.999717. The domain is loaded
with a stoichiometric loading value of 1.97 kg of biomass. Each ODT particle is a sweet gum hardwood particle with an initial density of 700 kg/m$^3$ and a diameter of 1 mm. Each particle is a cylindrical particle with a constant length of 1 m.

To ignite the particles, the simulation is started with a hyperbolic tangent mixture fraction profile as in laminar test cases. However, the width of the fuel zone is 0.06 m ($3/500^{th}$ of the domain) with a hyperbolic tangent transition from air to fuel on either side of width 0.03 m. The mixture fraction at the center of the domain is one (pure fuel) and decreases to zero (pure air) on both sides. The corresponding temperature profile is then initialized using the lookup table with adiabatic conditions (initial heat loss of zero throughout the domain). This initial mixture fraction profile is not the pyrolyzed gas from the particles as particles throughout the domain are unburned at 300 K initially with zero reaction rate. The initial profile adds enough pyrolyzed gas to the domain to increase the particle temperature to the ignition temperature, thus sustaining and spreading the flame in the domain.

Figure 5.6 shows the initial mixture fraction profile along with the corresponding gas temperature profile. The initial velocity on the line is zero. The 2-flux radiation model is used along with Leung soot model. Enthalpy and mixture fraction are the only two transported variables. All other scalars are computed using the lookup table. All simulations were performed with $C=10$ (eddy rate parameter), $Z=700$ (viscous penalty parameter), and $\beta=0.7$ (elapsed time large-eddy suppression parameter) as the ODT parameters. The void fraction is nearly unity throughout the domain, so particle volume is ignored in the transport equations. Outflow boundary conditions are used and the boundaries are assumed to be at 298.15 K and at 1 atm pressure. The emissivity of the particles is taken as $\varepsilon=1$.

Buoyancy drives the turbulent energy cascade. As the simulation progresses, gas velocity increases with time due to buoyancy.
The increased velocity then creates shear stress which then results in an eddy cascade. These eddies induce more eddies which drives the mixing and increases the heat transfer from the gas to the particles. Increased heat transfer causes biomass to pyrolyze, which then combusts, causing a decrease in gas density in the flame zone. Sheared velocity is created in the flame zone which then enduces mixing. This cycle of eddy cascade and combustion continues throughout the simulation and the flame spreads on both sides of the domain.

**Single Realization Result**

Figure 5.7 shows the results of the baseline case for a typical buoyant plume realization at four times: 0.5, 1.0, 1.5, and 2.0 s. The plots on the left show the particle surface temperature and the gas temperature profile whereas the plots on the right show the heat loss and mixture fraction profiles at different times. On the left side of the plot, the highly fluctuating solid black line represents the gas temperature profile and the dotted blue line represents the particle surface temperature. The solid line on the right side of the plot represents the mixture fraction whereas the dotted line represents the heat loss. Plots on the right side also show two horizontal dotted lines at $y=0$ and $y=0.145$ to indicate the zero heat loss line and stoichiometric mixture fraction line.

At $t=0.0$ s, the particles are cold, whereas the gas at the middle of the domain is as hot as 2100 K. There is no pyrolysis as the particles are cold, and there is only radiative heat transfer between the gas and the particles. As the simulation progresses, the hot gas then diffuses and mixes
due to turbulent advection which causes the mixture fraction profile to spread and gas to combust. Gas combustion results in heat transfer from the gas to the particles which results in pyrolysis. The pyrolysis in turn results in combustion and thus sustained flame spread.

At $t=0.5$ s, the mixture fraction profile decreases from an initial peak value of 1 to a peak value of about 0.3 but increases its width from 0.06 m to about 1 m. As it diffuses, the fuel combines with the entering air resulting in a mixture fraction closer to the stoichiometric point at which combustion occurs. The peak gas temperature significantly reduces from 2100 K to 1800 K but increases its width from about 0.012 m to about 1 m. The reduction in temperature is due to convection and radiation loss from the gas to the particles. The particles increase their temperature from 300 K to a peak value of 800 K.

By $t=1.0$ s, convection and radiation are increased due to mixing and the particles reach a temperature just above 800 K at which point pyrolysis just starts to occur. The gas decreases in temperature to 1500 K but continues to diffuse and increase the width. Significant pyrolysis has not yet occurred. The particle temperature outside of the main flame zone gradually increases but stays well below the ignition temperature. This indicates that the effect of radiation on the particles is relatively low as the radiation timescale is high (25 s).

By $t=1.5$ s, particles in the middle of the domain breach the ignition temperature and begin to pyrolyze as the reaction at 1.5 s is much faster than at 1.0 s. The mixture fraction increases from a peak value of 0.25 to 0.35. The pyrolyzed gas combusts and the gas temperature also increases. At the same time, turbulent advection and diffusion cause the temperature peak to increase its width and cause heat loss. This heat loss on both sides of the flame cause the particles to heat up and then pyrolyze as they reach the ignition temperature. This way the flame spreads on both sides of the domain.

At $t=2.0$ s, the spread of the flame is significant. The particle temperature reaches an equilibrium temperature with the gas temperature at the middle of the domain where reactions have ceased and particle pyrolysis is complete.
Figure 5.7: Buoyant plume baseline simulations at t=0.5, 1.0, 1.5, and 2.0 s. Plots in the left column show the gas and particle surface temperature profiles whereas plots in the right column show the mixture fraction and heat loss profiles.
The gas temperature has a peak value at 3 m and 7 m of the domain whereas the peak values of the mixture fraction is at 5 m and 6 m. The temperature of the gas increases from a peak value of around 1600 K at \( t = 1.5 \) s to a peak value of around 1800 K at \( t = 2.0 \) s. The particle temperature reaches as high as 1300 K at which point the particles burn to 93% (Fig. 5.8) of their original mass.

Figure 5.8 shows the percentage burnout of the particles at the four different times. At 0.5 s very little reaction has occurred. By 1 s the particles at the center of the domain have reached 50% burnout, while the burnout is complete (93%) by 2 s at the center of the domain. The remaining particle mass consists of char and ash. Char oxidation and ash are not currently implemented in the model.

Discussion

It is noted that, at any time, the heat loss is higher where the mixture fraction is higher. Particles at that point in space are high in temperature which causes more pyrolysis resulting in a higher mixture fraction at that point. The temperature of the gas is high at \( t=0 \) s, then falls to the lowest peak value of 1500 K at \( t=1.0 \) s, and increases again to about 1800 K by the end of the simulation. This results in a necking effect in the temperature profile of the gas which is more evident in the mean gas temperature profile as shown in Fig. 5.10 (a).
At all times, the fluctuations of the gas temperature is significantly higher than that of the particle temperature. The timescales of the turbulent fluctuation are shorter than the convective and internal conductive timescales. This difference in timescales filters the fluctuation level on the surface temperature of the particles. This suggests that molecular flame structures do not need to be resolved and simplified sub-grid models for RANS and LES may be accurate. Towards the end of the simulation, the time between two consecutive eddies, calculated as the ratio of run time to the number of eddies, is approximately 0.0002 s whereas the average convective and internal conductive timescale are 0.47 s and 0.5 s respectively. Even though the turbulent fluctuation occurs quickly, it takes more time for the heat to get transferred to the surface of the particles. This separation of heat transfer timescale from the turbulent fluctuation time creates a filtering effect on particle surface temperature.

The separation of conduction and convection timescale from the integral turbulent scale causes the particle surface temperature to lag behind the gas temperature. As the flame spreads outwards from the domain, the particle temperature and gas temperature reach an equilibrium after which the particle temperature decreases continuously.

**Eddy Information**

Figure 5.9 shows the time and location of the eddies during the simulation. It also shows the size of the eddies (the widths of the horizontal line segments) and the frequency of the eddies. On a typical simulation, the largest eddy size is 2.5 m and the minimum eddy size is 0.0169 m. The standard deviation of the eddy distribution is 0.0933 m and the mean is 0.0654 m. The increase in the width of the plume is directly correlated with the number of eddies. As time progresses, the velocity of the gas increases due to buoyancy effects. An increase in the velocity profile creates strain and shear which increases the number of eddies. An increase in the eddies induces more shear which result in more eddies. Hence as the width increases, the number of eddies also increases.

At around 1.5 s, the eddy map profile widens significantly. At this time, particle temperatures are just reaching the average temperature of 1,050 K which is above the particles ignition temperature at which point vigorous reaction occurs. Strong pyrolysis causes increased combustion reaction; thus increasing the heat transfer, reaction, eddy cascade, and mixing. This causes the
broadening of the profile beyond 1.5 s. As noted above, the reaction timescale drops an order of magnitude (1.0 s-0.1 s) from particle temperatures of 800 K to 1,030 K.

Figure 5.9: Eddy size, time, and location map for the baseline simulation.

Summary

The initial profile sustains and spreads the flame. Radiation has very little effect on particle surface temperature as the radiation time scale is significantly higher than the run time. The pyrolysis occurs vigorously as the temperature breaches the ignition temperature.

There is a competing effect between the combustion and turbulent advection. The combustion of the pyrolyzed gas tends to increase the temperature but it is offset by the turbulent advection as it diffuses and decreases the temperature. The gas temperature is highest near the stoichiometric point where the mixture fraction profile crosses the stoichiometric line at $\xi=0.145$. The highest particle temperature occurs at the middle of the domain. The decrease in initial mixture fraction is offset by the combustion after 1.5 s, after which vigorous reaction occurs.
Base Case Multiple Realizations

Eddy location and sizes in ODT are randomly selected. Hence, a single realization does not represent the true profile of any scalars. To obtain statistics and true mean profiles of the transported scalars, 120 realizations were performed with the same configuration but with random eddy locations, frequencies, and sizes. Then the scalar profiles were averaged and the mean and root mean square profiles were obtained which better represented the variables that characterize the turbulent combustion.

Fig. 5.10 shows the mean profile of temperatures and mixture fraction along with the width of the flame. The plots are at 50 different times linearly spaced between 0 s and 2 s. The average gas and particle temperatures are shown in Fig. 5.10 (a) and 5.10 (b). Since these profiles represent the average of a scalar, the highly fluctuating behavior as seen in a single realization, was eliminated.

The mean gas and particle temperatures peak around 1,400 K, unlike the single realization due to averaging. Figure 5.10 summarizes the flame behavior.

From the figure, it is clear that the particle temperature continually rises. Since the flame starts at the center and spreads on both sides of the domain, particle temperature is high at the center and low towards the sides. Gas temperature first increases, then decreases, and increases again producing a necking effect. This necking effect is due to the initial mixture fraction profile mixed out in the absence of an external fuel source. The temperature rises as initial rich mixtures are brought to the stoichiometric point, then decreases as these stoichiometric mixtures are diluted with air, and as convective and radiative losses occur. The temperature rises again as particles evolve pyrolysis products and combust. This behavior is also seen in the single realization. Plot (d) shows the mean mixture fraction profile, which reflects this behavior. There is a clear increase in the mixture fraction beginning at 1.0 s, but rising rapidly around 1.5 s.
Figure 5.10: Statistics from 120 realizations, showing mean gas (a) and particle temperatures (b), gas root mean square temperature fluctuations (c), mean mixture fraction profile (d), particle burnout width (e), and flame spread rate (f).
In all simulations, the width of the flame is measured as the width of the domain that is bounded on both sides by the 5% particle burnout. Figure 5.10 (e) shows the width of the particle burnout profile. There are 3 different somewhat linear regions in this profile. Region from 0 s to 0.6 s where the width burnout profile is essentially zero. The width of the burnout profile in the region from 0.6 s to 1.5 s slowly increases as the particles reach their ignition temperature and reaction starts to occur. The width in the region from 1.5 s and beyond increases rapidly as the particles reach far beyond their ignition temperature, and vigorous pyrolysis reaction occurs. This vigorous reaction increases the rate of heat transfer and combustion, thus increasing the burnout profile.

The width of the burnout profile at different times is shown as symbols. A solid line is then fitted to the data to obtain the spread rate. The solid line in the figure is a 6th order polynomial fit that is then differentiated to yield plot (f) which represents the profile of the flame spread rate. The spread rate depends on the polynomial fit which does not truly represent the actual width values but increase in the width of the burnout profile is evident from the increasing rate. Figure 5.11 shows the mean velocity profile at four different times.

![Image](image_url)

(a)

Figure 5.11: Baseline mean velocity profile at 0.5 s, 1.0 s, 1.5 s and 2.0 s. Increasing width corresponds to increase in time.
Summary

From the mean profiles, it is clear that the particle temperature rises monotonically. The gas temperature increases at first, then decreases, and then increases again producing a necking effect. The RMS profile of the gas temperature increases in width as flame spreads across the domain. The width of the flame is essentially zero until 0.5 s, increases slowly from 0.5 s to 1.5 s, then increases rapidly from 1.5 s to simulation end time of 2.0 s.

Parametric Simulations

To see the effect of biomass loading and particle sizes on the flame spread rate, parametric simulations were performed. Particle sizes of 0.7 mm and 1.2 mm were chosen to halve and double the internal conduction timescale. A total of 120 realizations were performed and the simulation configuration is similar to the baseline case. Figure 5.12 shows the comparison of mean gas and particle temperature profiles along with the gas mixture fraction profile. In the figure, the top row is the mean gas temperature profile, the middle row is the mean particle surface temperature profile, and the bottom row is the mean gas mixture fraction profile. The left column is the mean profiles for d=0.7 mm, the middle column for d=1.0 mm (baseline case), and the right column for d=2.0 mm particles.

The particles reach beyond the ignition temperature at about 1.0 s for 0.7 mm, at around 1.3 s for the 1 mm particles, and at around 1.8 s for the 1.2 mm particles. The temperature profiles and mixture fraction profile of the smaller particles are wider than the larger particles. There are three high gas temperature wings for particles that are the size of 0.7 mm whereas for larger particles, the gas temperature is high in the middle only. For all three cases, the gas and particle temperature profiles rise monotonically.
The general trend associated is that the pyrolysis rate and the spread rate are faster with decreasing particle size. The conduction time scale is directly proportional to the square of the diameter according to Eq. 5.3. Hence, as the particle size decreases by 0.3 mm, the conduction time decreases by about a half. This means, it takes less time for the smaller particles to heat up to a certain temperature, or the smaller particles reach a higher temperature if exposed to the same amount of time. Thus, for a simulation time of 2 s, smaller particles heat up quickly and reach a temperature higher than the larger particles. This causes faster pyrolysis, faster combustion, faster
heat transfer rate, and more spread. This also implies that for the same particle loading, the flame on smaller particles does not last for a long time, and flames on larger particles will last for a longer period of time but it may take time for particles to reach their ignition temperature. Bigger particles have a larger conduction timescale. This larger conduction timescale may cause the flame to quench if the initial flame is not sufficiently wide to heat up the particles.

It is clear from the figure that there is no necking effect on the smaller particles, whereas for the larger particles, the necking effect is distinct and is seen for longer periods of time. The larger particles heat up slowly because of an increase in conduction timescale that causes the necking effect. By the time the initial mixture fraction mixes, particles gain enough heat to pyrolyze and the necking behavior disappears in the smaller particles.

The equivalence ratios were varied from 1.0 to 0.5 and 2.0, halving (lean) and doubling (rich) respectively the biomass loading. A total of 120 realizations were performed. The simulation configuration is similar to the baseline case. Figure 5.10 (c) shows the comparison of mean gas and particle temperature profiles along with the gas mixture fraction profile. In the figure, the top row is the mean gas temperature profile, the middle row is the mean particle surface temperature profile, and the bottom row is the mean gas mixture fraction profile. The left column is the mean profiles for $\phi=0.7$, the middle column for $\phi=1.0$ (baseline case), and the right column for $\phi=2.0$.

The width of the temperature profiles for both the rich and lean cases are lower than the baseline case. The width of the mean particle surface temperature also has a similar trend. The lower temperature width is directly associated with the low pyrolyzed gas. The necking effect is prominent in all 3 loadings.

Lean loading results in less heat loss from the gas which results in less amount of evolved volatiles. This combined effect results in the necking effect in regions after 0.5 s. It is clear from the mixture fraction profile that the lean case evolves less fuel than the other two cases. However, the heat loss in the lean case is lower than the rich case which results in somewhat higher gas temperature than the rich case. The rich case shows much higher rate of pyrolysis as indicated by the mean mixture fraction profile. However, the heat loss for the rich case is higher than the other two cases, which results in lower gas temperature. The spread of mixture fraction in the rich case is higher than the lean case as there is more fuel to spread on in contrast to less fuel in lean case.
Figure 5.13: Mean gas temperature profiles, top row; mean particle temperature, middle row; mean mixture fraction, bottom row. The left, middle, and right columns are the $\phi=0.5$, $\phi=1.0$, and $\phi=2.0$ cases, respectively.

Similar to the effect of large particles, high particle loading may also cause extinction of the flame. Higher particle loading means more heat loss from the gas to the particles and more heat transfer area. Rich loading is affected by both high heat transfer area and more heat loss. The combined effect reduces the temperature of the gas, quenches the flame, and decreases the spread rate.

Figure 5.14 (a) shows the comparison of the width of the 5% burnout profile and spread rate corresponding to the fitted polynomial for five different cases, 2 size variations, 2 loading
variations, and the baseline case. The width and spread rate for the particle size of 0.7 mm at stoichiometric loading has the highest spread rate, whereas the particle size of 1.2 mm at stoichiometric loading has the lowest spread rate. The trend is that the smaller particles spread earlier and faster than the larger particles. The spread width and flame rate for 1 mm particles with half the stoichiometric loading is about the same as for a 1 mm particle with double the stoichiometric loading. For all the cases, the width and the spread rate trends are similar in that as the particles reach beyond the ignition temperature, the rate increases rapidly.

![Graphs showing width of particle burnout profile and flame spread rate](a) and (b)

Figure 5.14: Width of the particle burnout profile bounding 5% burnout (a), and rate of spread of this profile based on derivative of polynomial fits (b).

**Summary**

A decrease in the particle size decreases the conduction timescale, whereas an increase in the particle size increases the conduction timescale. A decrease in timescale causes particles to gain heat quickly from the gas. Thus particles with smaller size pyrolyze and burnout much faster than the particles with larger size which take more time to heat up but burn for a longer time. If there is more mass of the fuel, then the flame will quench before the particles reach their ignition temperature. Likewise, if there is less fuel than the amount of air, the flame will spread but will not sustain itself for longer period of time as the particles burn out quickly.

If the initial ignition source is not sufficiently wide, the gas temperature will decrease faster than the particles can be heated. When the particles do not attain a temperature enough to produce
reaction sufficient to sustain the flame, the flame will extinguish. This condition occurs for a narrow initial ignition zone, for large particles with low heating rates, or for a high particle loading having a higher heat capacity.

5.2.2 Homogeneous Turbulence

Turbulent buoyant plume simulations gave insights on how the flame spreads on different sizes and loading of fuels. In the turbulent buoyant plume simulation, it was noted that implicit filtering occurs in particle surface temperature due to separation of heat transfer timescales from the turbulent fluctuation timescale. Towards the end of the simulation, the process was convection limited. This inspires us to overlap the convection timescale with that of the turbulent fluctuation timescale to investigate the response of the particle surface temperature with the fluctuating turbulence. This overlapping of timescales are done in a homogeneous turbulence environment.

A turbulent flow is homogeneous if all statistics are invariants under a shift in position in the flow domain [87]. To overlap the convection timescale with the integral time scale in a homogeneous environment, the LEM model (described in Chapter 3) is used as it was first developed to investigate the qualitative mixing properties of turbulent flows assuming homogeneous behavior.

Reasonable parameters are set by specifying particle size, eddy size, and velocity fluctuation. A velocity fluctuation is estimated using 10% of a 10 m/s velocity. Assuming average gas temperature of 1,000 K, the convection time scale is estimated, using Eq. 5.2, to be 0.5 s. To overlap the convection timescale with the conduction timescale and the integral timescale, the input parameters are manipulated. The diameter of the particles is chosen to be 0.5 mm and the integral length scale to be 0.5 m. These parameters result in $D_T=0.032 \text{ m}^2/\text{s}$, Kolmogorov length scale, $\eta=0.0097614 \text{ m}$, particle Reynolds number, $\text{Re}_p=20.492$, and turbulent Reynolds number, $\text{Re}_T=4.098$. Table 5.3 summarizes the LEM model parameters derived to overlap different timescales.
Table 5.3: Summary of LEM model parameters.

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Symbol</th>
<th>Definition</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Gas velocity</td>
<td>( v_\infty \text{(m/s)} )</td>
<td>-</td>
<td>10</td>
</tr>
<tr>
<td>Gas temperature</td>
<td>( T_\infty \text{(K)} )</td>
<td>-</td>
<td>1,000</td>
</tr>
<tr>
<td>Integral length scale</td>
<td>( L_1 \text{(m)} )</td>
<td>-</td>
<td>0.5</td>
</tr>
<tr>
<td>Particle radius</td>
<td>( r_p \text{(mm)} )</td>
<td>-</td>
<td>0.25</td>
</tr>
<tr>
<td>Gas velocity fluctuation</td>
<td>( v' \text{(m/s)} )</td>
<td>( 0.1v_\infty )</td>
<td>1</td>
</tr>
<tr>
<td>Reynolds number</td>
<td>( \text{Re}_p )</td>
<td>( 2v_\infty r_p/v_g )</td>
<td>20.492</td>
</tr>
<tr>
<td>Turbulent Reynolds number</td>
<td>( \text{Re}_T )</td>
<td>( u'L_1/v_g )</td>
<td>4,098</td>
</tr>
<tr>
<td>Kolmogorov length scale</td>
<td>( \eta \text{(m)} )</td>
<td>( L_1/\text{Re}_T^{3/4} )</td>
<td>0.000976</td>
</tr>
<tr>
<td>Conduction timescale</td>
<td>( \tau_{\text{cond}} \text{(s)} )</td>
<td>( r_p^2/\alpha )</td>
<td>0.525</td>
</tr>
<tr>
<td>Convection timescale</td>
<td>( \tau_{\text{conv}} \text{(s)} )</td>
<td>( 2mpr_pC_p,p/Nuk_gA_p )</td>
<td>0.468</td>
</tr>
<tr>
<td>Integral timescale</td>
<td>( \tau_{L} \text{(s)} )</td>
<td>( L_1/v'_\infty )</td>
<td>0.5</td>
</tr>
<tr>
<td>Turbulent diffusivity</td>
<td>( D_T \text{(m}^2/\text{s)} )</td>
<td>( 0.064v_g \left( \frac{L_1}{\eta} \right)^{4/3} )</td>
<td>0.032</td>
</tr>
</tbody>
</table>

**Configuration**

To match the eddy turnover time, and attain a steady state flame spread, the simulation was run for 4 s on a 2 m domain with 2,000 initial grid points and 2,000 initial particles, one on each grid cell. The diameter of the particles are chosen to be 0.5 mm to reduce the conduction timescale such that it matches with that of the convective and integral timescale. The particles are divided into 5 control volume instead of 10 in the buoyant simulation. All other properties of the particles are the same as in the buoyant plume simulation. In the buoyant plume simulation, the initial velocity of the gas is zero and rises up to 20 m/s because of buoyancy. However, in the homogeneous flow simulation, velocity in the domain through out the simulation is based on the eddy velocity and the velocity associated with expansion/contraction of the cells. The Radiation model, soot model, and boundary conditions are unchanged from the buoyant plume simulation, and the buoyancy is not incorporated.
The simulation is started with the initial temperature profile as shown in Fig. 5.15 which was obtained by an in-house code (refer Appendix A). The initial profile is developed such that the first half of the domain (zone “A”) consists of unburnt fuel and the second half (zone “B”) consists of fully burnt fuel (93% of the original mass). The initial gas temperature in zone “B” is assumed to be the same as that of particles with an initial heat loss of zero throughout the domain. The particles in zone “A” are fresh particles with stoichiometric loading of 0.12 kg, void fraction of 0.999717, and a zero gas mixture fraction whereas the fuel loading in zone “B” is 0 kg (contains only char and ashes) with a stoichiometric mixture fraction of 0.145. The hot gas in zone “B” mixes due to turbulent advection and drives the flame spread through zone “A”.

**Single Realization Result**

Figure 5.16 shows the results of the baseline case for a typical homogeneous outflow realization at four times: 1.0, 2.0, 3.0, and 4.0 s. The plots on the left show the particle surface temperature and gas temperature profile, whereas the plots on the right show the heat loss and mixture fraction profiles at different times. The bottom plot shows the initial gas temperature and initial mixture fraction, and two vertical lines to distinguish zone “A” from zone “B”. On the left side of the plot, the highly fluctuating solid line represents the gas temperature profile and the dotted line represents the particle surface temperature profile. The solid line on the right side of the plot represents the mixture fraction profile whereas, the dotted line represents the heat loss profile.
Plots on the right side also show two horizontal fixed solid lines at $y=0$ and $y=0.145$ to indicate the zero heat loss line and the stoichiometric mixture fraction line.

Figure 5.16: Homogeneous turbulence baseline simulations at $t=1.0$, 2.0, 3.0, and 4.0 s.
Initially the particles in zone “A” are cold (300 K) and there is no pyrolysis. The gas and the particles in zone “B” are as hot as 1,300 K. The particles in zone B are fully burnt. As the simulation progresses, the hot gas front from zone “B” of the domain heats the unburnt particles in zone “A”.

By t=1.0 s, due to turbulent advection, the mixture fraction mixes and spreads to the zone “A” region. Mixture fraction decreases in zone “B”, whereas it increases in zone “A”. Due to the pyrolysis of hot particles in the flame zone, the gas rises to a temperature of about 1,400 K. The gas mixture fraction is close to the stoichiometric point at the end of zone “B”. The heat loss profile indicates that even though the mixture fraction profile is close to stoichiometric point at the end of zone B, the temperature is lower than at the flame zone due to more heat loss. The particles in the flame front zone are at a temperature of about 800 K - 1,000 K, whereas particles outside of the flame front zone in zone “A” are at 390 K (due to radiation). The particles in zone “B” have decreased in temperature due to mixing and heat loss. The particles and gas temperature in zone “B” fall from 1,300 K to about 1,050 K.

By t=2.0 s, the gas temperature in zone “A” reaches a peak value of 1,500 K as the particles reach a temperature of 1,000 K, well above the ignition temperature, causing vigorous pyrolysis reaction. The flame front shifts its position from 0.9 m to 0.8 m. The temperature of particles and gas at the end of zone “B” continues to decrease due to heat loss and mixing. However, due to the vigorous pyrolysis reaction, the mixture fraction between 0.8 m and 1.1 m (the position of flame front at t = 1.0 s) increases beyond the stoichiometric point, and spreads towards the beginning region of zone “B”. The particles in this region continue to pyrolyze as the flame front moves towards the left. The spread of mixture fraction due to turbulent advection causes the gas and particle temperature to increase at the beginning of zone “B”, creating a hump effect.

By t=3.0 s, the flame front shifts its position from 0.8 m to 0.5 m where the peak temperature is around 1,500 K. The width of the hump (between 0.75 m and 1.45 m) increases significantly due to mixing and radiative readsorption. The particles within the hump reach a peak temperature of 1,100 K where the particles have completely burnt (burnout percentage is 93%). The particles in the flame region reach a temperature of about 900 K. The zone “A” mixture fraction in the flame front zone is at the stoichiometric point, whereas the mixture fraction on the right side of the flame front continues to increase above the stoichiometric point due to the time lag between the gas and
particle surface temperature. Heat loss in zone “A” is highest at the flame zone which is as high as 58%. The highest heat loss in the domain is at the end of zone “B” which causes the temperature profile to tail off.

By t=4.0 s, the flame reaches a position of about 0.25 m from 0.5 m where the peak temperature stays around 1,500 K. The particles at the right side of the flame have reached a temperature of about 1,000 K, whereas the particles in the flame zone are at around 850 K. The width of the hump continues to increase due to spread, mixing, and re-adsorption creating a flat temperature profile between 0.5 m and 1.6 m. The mixture fraction in zone “A” at the right side of the flame front is far above the stoichiometric point but does not have a gradient, which indicates that particles at the right side of the flame continue to burn which is also evident from the burnout profile as shown in 5.17. Heat loss in zone “A” is highest at the points where the mixture fraction is at stoichiometric, but the heat loss difference compared to the region at the right side of the flame front is very little. The particles at the end zone “B” have a tailing temperature profile due to significant amount of heat loss.

Figure 5.17 shows the percentage burnout of the particles at 4 different times: 1.0, 2.0, 3.0 and 4.0 s. At 1 s, very little reaction has occurred as only few particles reach beyond the ignition temperature and the particles are transitioning from preheating to the ignition temperature. As the simulation progresses, the gap between the subsequent burnout profile increases as the flame spreads across zone “A”. As the flame spreads, particles in the flame front continue to increase in temperature, whereas the particles at the right side of the flame front continue to pyrolyze.

**Eddy Information**

Figure 5.18 shows the time and location of eddies during part of the homogeneous turbulent simulation. It also shows the size of the eddies and the frequency of the eddies. Due to large number of eddies, only the first 2,000 eddies are shown. On a typical simulation, the largest eddy size is 0.5 m, whereas the minimum eddy size is 0.0010 m. The standard deviation of the eddy distribution is 0.0053 m and the mean of the eddies is 0.0024 m.
Figure 5.17: Percent particle burnout for the baseline case at 1.0, 2.0, 3.0 and 4.0 s. Increasing width is directly correlated to increasing time.

Figure 5.18: Eddy size, time, and location map for the homogeneous outflow simulation.

**Discussion and Summary**

It is clear from the figure that the flame propagates to the left and the flame front temperature stays at around 1,500 K throughout the simulation time. Like the buoyant plume simulation, the gas temperature fluctuates more than the particle temperature at all times even though the conduction and convection time scales are overlapped with that of turbulent fluctuation. This suggests that implicit fluctuation occurs in the particle surface temperature in almost all wildland fires. Like
the buoyant plume simulation, the gas temperature in the flame front is always higher than that of the particles despite the overlapping of heat transfer timescales to that of integral scale.

Heat loss and gas temperature in zone “A” are highest at the flame zone. The gas temperature always leads the particle surface temperature as in the buoyant plume simulation. Unlike the buoyant plume simulation, the gas temperature increases monotonically eliminating the necking effect. The fully burnt particles at the right side of the domain first decrease in temperature due to mixing. However, after 2 s, the burnt particles increase in temperature due to a heat gain from the burning particles and due to the mixing. The mixture fraction at the end of the domain remains near stoichiometric where the temperature of the gas is expected to be as hot as in the flame zone. However, heat loss at the end of the domain is high at all times, which causes the gas temperature to decrease creating a tailed temperature profile.

**Homogeneous Case Multiple Realizations**

Figure 5.19 shows the mean profile of the temperatures and mixture fraction along with the width of the flame. The plots are at 50 different times linearly spaced between 0 s and 4 s. The average gas and particle temperatures are shown in plot (a) and plot (b), respectively. Plot (c) shows the gas temperature root mean square (RMS) fluctuating profile.

From the figure, it is clear that the gas and particle temperatures in zone “A” rise monotonically, whereas the gas and particle temperature in zone “B” first decrease and then increase again. The increase in the gas and particle temperature in zone “A” is due to heat transfer from zone “B” and combustion in zone “A”. The decrease in the particle temperature in zone “B” is due to mixing and heat loss, whereas the increase in temperature after about 1.5 s in zone “B” is due to mixing and radiative re-adsorption.

Plot (d) shows the mean mixture fraction profile. The mixture fraction increases monotonically in zone “A” as fresh mass starts to pyrolyze and as the zone “B” gases are mixed. However, in zone “B”, the mixture fraction decreases monotonically until about 1.5 s. After 1.5 s, the turbulent mixing causes the mixture fraction in zone “B” close to zone “A” to increase. The mixture fraction at the right end of zone “B” decreases monotonically from the initial time to the ending simulation time.
Figure 5.19: Statistics from 120 realizations, showing mean gas (a) and particle temperatures (b), gas root mean square temperature fluctuations (c), mean mixture fraction profile (d), particle burnout width (e), and flame spread rate (f).
Plot (e) shows the width of the particle burnout profile. The width is essentially zero up to the particle ignition time of about 1.0 s. The width then increases rapidly from the ignition time to the simulation end time. A vigorous reaction occurs after about 1.0 s and the heat transfer to and from the gas increases. The heat transfer process shifts the flame front from the middle of the domain towards zone “A”.

Plot (f) shows the derivative of the fitted curve in plot (e) which represents the flame spread rate. The flame spread rate continues to rise from the initial time up to 3 s. After 3 s, the flame front propagates up to the beginning of the domain after which the flame spread rate decreases and the spread reaches an equilibrium. After this point, the particles pyrolyze and sustain the flame until they convert to chars and ashes.

**Parametric Simulations**

To see the effect of biomass loading and particle sizes on the flame spread in a homogeneous environment, parametric simulations were performed. Particle sizes of 0.34 mm and 0.7 mm were chosen to halve and double the internal conduction timescale, and the equivalence ratios were varied from 1.0 to 0.5 and 2.0, halving and doubling respectively the biomass loading. A total of 120 realizations were performed. The simulation configuration is similar to the baseline case.

Figure 5.20 (a) shows the comparison of the width of the 5% burnout profile and Fig. 5.20 (b) shows the comparison of the spread rate corresponding to the fitted polynomial for four different cases, 1 size variation, 2 loading variations, and the baseline case. The profile for 0.7 mm particles is not shown as the flame quenches before it spreads.

The flame in the 0.34 mm particles starts to spread almost from the start time, whereas the flame in the 0.5 mm particles starts to spread after about 1.5 s. However, the spread rate for 0.5 mm particles is faster than the 0.34 mm particles. The pre-heating, pyrolysis, and spread occurs in smaller period of time in smaller particles, whereas pre-heating occurs for longer period of time before the flame spreads in larger particles. Since the smaller particles take a shorter period of time to preheat and pyrolyze, the flame spread rate is slower than in larger particles where pre-heating occurs for a longer amount of time after which the flame spreads rapidly.
When the biomass loading is rich, the flame spreads more rapidly than in the lean condition. When the conduction timescale is overlapped with the turbulent fluctuation scale, the combustion of the biomass becomes reaction limited. As the biomass loading is rich, there is a larger amount of fuel which causes faster spread in the flame.
CHAPTER 6. SUMMARY AND CONCLUSION

Wildland fires are hazardous and pose a threat to human lives and property. To suppress these fires, flame propagation, flame behavior, flame rate, and underlying physics must be understood. Several turbulent models have been applied to model biomass combustion. However, these combustion models are currently limited to fire spread rate only. Some models do not resolve all the scales at the flame fuel interface while other are computationally expensive.

ODT, a relatively simple 1D model, can resolve both individual and large scales. Since scales are resolved in 1D only, it is computationally cheaper. ODT is an extension of the LEM model in which scalars are transported by 2 processes: diffusion and turbulent advection. Turbulent advection in ODT is modeled by a mapping process called triplet mapping. ODT is a numerical method to solve stochastic boundary-value problems on a 1D domain. Computationally affordable resolution of viscous scales in fully developed turbulence is a key feature of ODT. ODT has been widely applied to many non-reacting and reacting flows including jet flames and pool fires.

The ODT model’s capability was extended by implementing the biomass combustion model. Forest fuels involve a number of compounds and complex combustion chemistry. To reduce the computational cost, a lookup table is used to represent the combustion process for the intended use. Lagrangian particles in the ODT model are fixed in space. Particles interact with the gas phase through eddy events and the diffusive process. Particles create a drag and act as a momentum sink. The diffusive transport equations (i.e., mass, momentum, energy, and mixture fraction) are modified to couple the biomass combustion.

The biomass considered were dry (does not include free moisture content) with high volatile yields so that the key processes are particle heat-up and pyrolysis only. Evaporation and char oxidation were ignored. The biomass pyrolysis model used in this work was that of Nunn et al. [59]. This model is a single reaction first order decomposition model which assumes a virgin material decomposes through a single reaction, from wood to volatiles.
The internal temperature gradients in the particles were solved by dividing the particles into 5-10 control volumes as the particles are thermally thick. Wood can be of either spherical, cylindrical, or flat plate in shape; however, only 1 m cylindrical particles with density of 700 kg/m³ were studied here. Heat transfer from the gas medium to the particles (and vice versa) occurs via convection, radiation, and conduction.

To investigate the basic behavior of various heat transfer processes and establish timescales, 4 different laminar test cases were performed. Laminar test cases revealed that heat loss mainly occurs via radiation, and heat transfer occurs mainly via convection. Internal conduction becomes the limiting processes after the particles reach the ignition temperature.

A particle size of 1 mm was chosen for the base configuration as representative of a fine fuel. Turbulent simulations were performed in a buoyantly driven flow and in a homogeneous flow. The buoyantly driven flow consists of a baseline case where the basic flame behavior was established. The effect of different particle sizes and loading on the flame spread rate were studied. The buoyantly driven flow simulations revealed that a decrease in particle size decreases the conduction timescale which causes particles to gain heat quickly and increase the pyrolysis rate. Low loading decreases the amount of volatiles which decreases the rate of flame spread. Likewise, high loading increases the heat loss which decreases the flame spread rate. Spread rate is higher in smaller size particles but the flame does not sustain itself for a long time. Larger particles take time to heat but the flame is sustained for a longer period of time. Too large of particles (conduction time scale > \( \tau_{mix} \)) cause the flame to quench.

Inherent separation of particle heating timescales from the corresponding turbulent fluctuations occurs in the buoyantly driven plume. This suggests that molecular flame structures do not need to be resolved and simplified sub-grid models for RANS and LES may be accurate.

Heat transfer occurs mainly via convection. To see the effect of high turbulent fluctuations on the particle surface temperature and to represent the simulation close to the real fires, homogeneous turbulence simulation was performed using LEM. The homogeneous turbulence simulation revealed that even though the heat transfer timescales overlapped with turbulent integral length scales, implicit filtration occurs between the gas temperature and particle surface temperature.
CHAPTER 7. FUTURE WORK

7.1 Inclusion of Moisture

Water vapor production from biomass burning is important in large open fires as any water content in the fuel must be driven off before the first stage of combustion can occur. Driving off the water content requires energy which may reduce the combustion temperature. This study is based on the sweet gum hardwood particles which are considered relatively dry. Inclusion of moisture in the model will be a better representation of the fuel. Treating the fuel with some moisture content will also better predict the combustion temperature. Lu et al. [73], developed a 1D particle model that simulates the drying, rapid pyrolysis, gasification, and char oxidation processes of particles. A simulation study with varying moisture content will be insightful.

7.2 Inclusion of Char Oxidation

The one component mechanism of primary wood pyrolysis involves 3 parallel reactions in which virgin wood decomposes to char, tars, and volatiles [28]. Char then oxidizes further resulting in tars and gases. This project involves a single first order reaction where virgin material is assumed to directly convert to volatiles during pyrolysis and where char oxidation is ignored. A study with a complex chemistry model might provide more accurate devolatilization rates and gas compositions; however, the general trends and behaviors are not expected to be different than those studied here.

7.3 Inclusion of Wind

Wind is one of the primary factors that greatly affects the flame propagation in wildland fire [88]. It causes high turbulent fluctuations and increases the heat transfer rates. The direction of the flame rate is greatly affected by wind and the entrainment of more air in the flame zone changes
the behavior of flame evolution. Including the wind in the ODT biomass combustion model would provide better prediction capabilities for the flame spread, flame behavior, flame structure, and statistics of the flame front. Wind can be incorporated in the ODT model by moving the ODT domain with the same velocity but in the opposite direction of the cross flow, or by adding a cross flow velocity to the ODT velocity components. Alternatively, wind can be incorporated using 3D version of LEM, or ODTLES as ODT is limited to the kinds of configurations allowed.

7.4 Inclusion of Slope

In steep slope, flame is closer to fuel and hence wildfires progress far more rapidly uphill than they do downhill. On a slope, fuel uphill is pre-heated by radiation and convection from the flame downhill which causes the flame to spread more rapidly. The steeper the slope, the higher the spread rate. Understanding the effect of slope on the intensity, rate, and directions of a fire is important to improve management techniques.

7.5 Fuel Continuity

Uniform fuels that are distributed continuously over the given area provides an unbroken path for a fire whereas fuel distributed unevenly, provides a broken path for a fire. The fuel continuity profoundly affects the eventual outcome of a fire.
REFERENCES


79


APPENDIX A. INITIAL BURNOUT PROFILE FOR HOMOGENEOUS TURBULENCE

clc;
clear;
global Tref heatLossVec mixfVec hSensVec rhoTable enthTable tempTable ...
    nInPseudoPart Hf_part figureOn initDx initRho_gas initMass_fakePart ...
    initTemp_part mixfStoic
Tref=298.15;
figureOn=0;
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%----Set these variables%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
up=1; % uprime
Li=1 % Integral length scale
nu=1.5E-5;
kine=1/2*up^2;
eps=kine^(3/2)/Li;
eta=(nu-3/eps)^0.25 % Kolmogorov length
cL_p = 5.9787; % initial guesses for params
ce_p = 0.4036;
%--Gas properties
mixfStoic = 0.145;
mixfMax=0.06;
odtParamFile = fopen('../odtParam.inp');
while(~feof(odtParamFile))
    ln=fgetl(odtParamFile);

    found = regexp(ln,'(domainLength)');
if length(found)~=0
    domLength = [sscanf(ln,'%f')];
end

found = regexp(ln,'(ngrd_0)');
if length(found)~=0
    ngrd = [sscanf(ln,'%f')];
    break;
end
end
fclose(odtParamFile);
ngrd
domLength
L=domLength;
%ngrd=150 --Particle properties
particleFile = fopen('..\particle.inp');
while(~feof(particleFile))
    ln=fgetl(particleFile);
    found = regexp(ln,'(part_Radius)');
    if length(found)~=0
        part_rad = [sscanf(ln,'%f')];
    end
    found = regexp(ln,'(part_Length)');
    if length(found)~=0
        part_length = [sscanf(ln,'%f')];
    end
    found = regexp(ln,'(part_Shape)');
    if length(found)~=0
        part_Shape = [sscanf(ln,'%f')];
    end
end
found = regexp(ln,'(part_Density)');
if length(found)~=0
    rhop = [sscanf(ln,'%f')];
end
found = regexp(ln,'(pTempInitial)');
if length(found)~=0
    initTemp_part = [sscanf(ln,'%f')];
end
end

%initVoidFrac=0.999717; initVoidFrac=0.999965;
initRho_part= rhop
part_Shape
part_rad
part_length
initTemp_part
Hf_part=-5344000;
%------------------------------------------------------------------------
%/%/%/%/%/%/%/%--Calculate cell face and cell position--%/%/%/%/%/%/%/%/%/%/%/%/
ngrdf=ngrd+1;
Xf=linSpace(0,domLength,ngrdf);
for x=1:ngrd
    X(x)=(Xf(x+1)+Xf(x))/2;
end
initDx=Xf(2)-Xf(1);
%------------------------------------------------------------------------
%/%/%/%/%/%/%/%/%/%/%/%--Mixture fraction profile--%/%/%/%/%/%/%/%/%/%/%/%/
tranWidth=0.1;
locOfTrans=domLength/2;

86
eta0 = 0.5 * mixfMax*(1+tanh(2/tranWidth*(X-locOfTrans))); 
%eta0 = fliprl(eta0); 
%------------------------------------------------------------------------------------------------------------------- 
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%---Read Table---%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
readTable; 
%------------------------------------------------------------------------------------------------------------------- 
%---Create table for different variables for bilinear interpolation---
firstTable = Table.get(0); 
mixfVec=firstTable(:,mixfIndex); 
hSensVec=firstTable(:,hSensIndex); 
for i=1:Table.size() 
    tableForAllVars=Table.get(i-1); 
    rhoTable(:,i)=tableForAllVars(:,rhoIndex); 
    enthTable(:,i)=tableForAllVars(:,enthIndex); 
    tempTable(:,i)=tableForAllVars(:,tempIndex); 
    cpTable(:,i)=tableForAllVars(:,cpIndex); 
    heatLossVec(i)=heatLossLL.get(i-1); 
end 
%------------------------------------------------------------------------------------------------------------------- 
heatLoss_zeroTable = Table.get( min(find(heatLossVec >=0)) -1 ); 
initRho_gas = interp1(heatLoss_zeroTable(:,mixfIndex), ... 
                      heatLoss_zeroTable(:,rhoIndex),0) 

initVoidFrac = 1-(mixfStoic/(1-mixfStoic))*(initRho_gas/rhop); 
%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%---Initial mass of the gas and particle---%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
if part_Shape == 2
    initVol_part=4.0/3.0*pi*part_rad^3;
elseif part_Shape == 1
    initVol_part=pi*part_rad^2*part_length;
else

87
error('s', 'Unrecognized particle shape-->', 'part_shape);
end

initMass_fakePart=initVol_part*initRho_part;
nInPseudoPart=(1-initVoidFrac)*initDx/initVol_part;
limitingMass_fakePart = initMass_fakePart * 0.1;

%---------------------------------------------------
%%/%%%----Solve for heatLoss based on initial mixf profile----%/%%%%%
guessHL=0.0;
initMass_part = initMass_fakePart * nInPseudoPart;
for i=1:ngrd
    myMixf=eta0(i);
    heatLossInLine(i)=fzero(@(hl) solnForHL(hl,myMixf,initMass_part),
                        guessHL,optimset('Display','off'));
    guessHL=heatLossInLine(i);
    numMessage = strcat(num2str(i/ngrd*100),'

end
T  = interp2Extrap( heatLossInLine, eta0, tempTable );
rho = interp2Extrap( heatLossInLine, eta0, rhoTable );

T = partTemp(eta0,T);

Tadia = interp2(heatLossVec,mixfVec,tempTable,heatLossInLine(11),eta0);

mpart = initMass_part - initRho_g*initDx.*eta0;
mFakePart = mpart./nInPseudoPart;

for i=1:ngrd
    hFakePart(i) = Hf_part + integral(@(t) heatCap_part(t), ...
                                      initTemp_part,T(i));
end
for i=1:ngrd
    eta1(i)=interp2Extrap(heatLossInLine(i),eta0(i),enthTable);
end

%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%%
% Save the useful data to load in ODT--%\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\\