

Undergraduate Honors Thesis

Influence of the Wall Heat Transfer on Flame Propagation

By

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Abstract

In the internal combustion engines, the flames interact with the walls of the cylinder, which affects the flame propagation characteristics and the engine performance. The flame tends to quench near the wall, which is due to wall heat fluxes. Also, wall heat transfer can play a significant role in undesirable auto-ignition of unburned mixture in the cylinder. The flame-unburned mixture-wall interactions can influence engine knock and affect the engine emission. The present research is aimed at understanding the effects of wall heat transfer on flame propagation. The flame propagation in the presence of the walls will be simulated and the effects of wall heat transfer on flame propagation properties will be investigated by changing wall temperature, pressures, channel widths, and equivalence ratios. By analyzing variations of those properties, we will be able to advance an understanding of flame-mixture ignition-wall heat transfer interactions, which will help reduce engine knock and emission for different types of engine structures. This research focuses mainly on the propagation of laminar premixed flames. The numerical method is used to solve the mass, momentum and energy conservation together with the combustion model. The first stage study focuses on using a single step chemistry reaction model to simulate flame propagating along one dimensional domain and two dimensional channels with adiabatic walls under different air fuel ratios, geometries, and injected flow velocity. This simulation is aimed to provide a reasonable distribution of temperature, flow velocity, pressure, fuel, oxidizer and products in the presence of the adiabatic walls. Based on the first stage, the second stage study focuses on adding heat transfer effects to the walls for two dimensional cases and analyze how wall heat transfer affects the distribution of the properties. For the first stage of research, the results from a single step chemistry model are compared with the experimental data. The results show that the single step chemistry model can accurately predict the flame consumption speed when air-fuel equivalence ratio ranges from 0.5 to 1. For the two dimensional channel with

adiabatic walls, the simulation shows that the presence of walls influences flame propagation through the flow velocity variation near the wall. In the second stage, wall heat transfer is included and the effects of wall heat transfer is analyzed in terms of flame quenching in the presence of walls. This research will lead to a better understanding of interactions of wall heat transfer and combustion in internal combustion engines, which can be a useful reference to analyze the engine knock and engine emissions.

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Chapter 1: Introduction

1.1 Background

The study of the fuel efficiency and exhaust emissions is of great importance in developing internal combustion (IC) engines. Many studies have shown that the walls in the IC engine will significantly affect the performance and the emission. The flame front tends to quench in the vicinity of the walls. The absence of the flame-wall interaction factor will influence the accuracy of the model that is used to predict the reaction rate, wall heat fluxes and temperature [1]. Furthermore, the presence of walls is the key factor that influences engine knock.

1.2 Flame Wall Interaction

The flame-wall interaction (FWI) was introduced in the 19th century by H. Davy [2]. The flame-wall interaction can be classified as Head-On Quenching (HOQ), Side-Wall Quenching (SWQ) and Quenching in Tube, as shown in Figure 1.

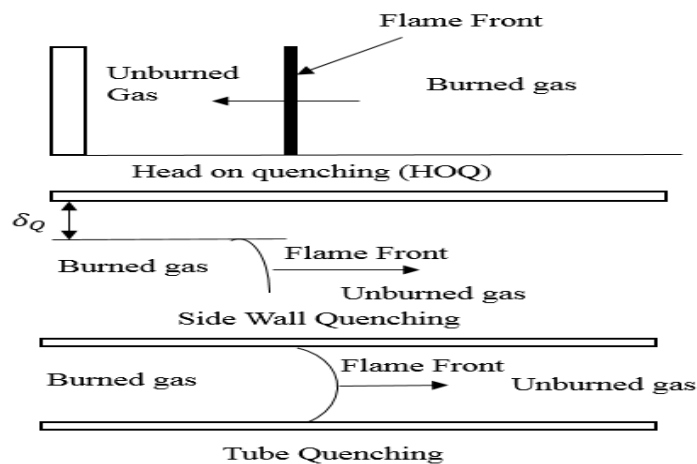


Figure 1: Illustration of Flame-Wall Interaction for Laminar Flame [1].

The Head-On Quenching is occurred when the flame propagates perpendicularly to a wall [2]. When the flame approaches a wall, the unburned gas ahead of the flame front, so called end-gas will be compressed, which leads to increase in the temperature and

pressure of the mixture. In this case, it can cause auto ignition of the end-gas before the flame reaches it, which is called “engine knock” [3]. The engine knock will generate high frequency pressure oscillations and damage the engine [3].

The Side-Wall Quenching will happen when the flame propagation is parallel to the wall [2]. Tube-Quenching will occur when the diameter of the tube is small enough [1]. The quenching distance is the smallest tube diameter for which the flame stops propagating.

1.3 Numerical Simulation

In this project, the flame-wall interaction will be investigated using numerical simulations. The discretized equations are solved by utilizing a second-order and conservative finite difference method [4]. The spatial derivatives are built and solved by using the second order centered finite difference for the velocity [4]. The third-order WENO scheme is used for convection term for scalars [4]. Also, the time integration is achieved by using a second-order semi-implicit projection method [4]. In this project, flows of interest are under laminar conditions, and all reactions and flow phenomena are resolved.

1.4 Motivation

Many studies indicate that flame wall quenching is one of the main sources that cause hydrogen carbon emissions in IC engines [6]. In order to effectively minimize the hydrocarbon emissions, this research analyzes which factors can contribute to decrease the quenching distance of a flame. The smaller the quenching distance is, the less hydrocarbon emissions it will generate. Compared with experiments, the use of numerical simulation can quickly extract detail information in a cost-effective and efficient way.

1.5 Objective

The objective of this research is to investigate flame-wall interactions under laminar flow conditions for the following specific cases.

1.5.1 Flame Propagation in One Dimensional Open Space

Simulate the flame propagation in one dimensional open space and study the relationship between the flame consumption speed and air-fuel equivalence ratio. For this problem, fuel considered is n-heptane. The initial temperature of unburned gas mixture is

298 K and the thermal dynamic pressure is set to be 1 atm. This is a validation study to set the chemical reaction rate parameters for use in flame-wall interaction studies.

1.52 Tube Quenching Simulation

Investigate how the heat transfer between flame and wall affects the flame propagation. The configuration is shown in figure 1. This specific study focuses on how the changes in air-fuel equivalence ratio and initial gas temperature affect and, wall temperature affect the quenching of flame. The air-fuel equivalence ratio varies from 0.5 to 4.5 and the initial wall temperature and unburned gas temperature range from 300 K to 1000 K.

Chapter 2: Initialization of Simulation

2.1 Initialization of Flow Condition

The initial conditions of flow include the geometry, mesh size and boundary conditions. Aside, we need to define the initial distribution of components' mass fraction, temperature, density and gas mixture velocity. The flame front position is set at the middle of simulation field.

2.1.1 Initialization of Geometry, Mesh Size and Boundary Condition

For one dimensional simulation, the flame propagates in one direction. Mesh points are built along the x direction. There is no wall surrounding the flame. For two dimension, flame propagates in the x and y directions. Mesh points are created along x and y direction. The flame can be constrained by top, bottom and front walls. The walls can be either adiabatic or conductive. Additionally, for each simulation, the inlet parameters including inlet flow velocity, temperature of injected gas mixture, and mass fraction of fuel, oxygen, nitrogen and products need to be specified.

2.1.2 Distribution of Species, Temperature and Fluid Velocity

The gas mixture components include fuel, oxygen, nitrogen and products. Both the distribution of species and initial flow velocity along the simulation domain follow the hyperbolic tangent function. This initialization enables flame to start propagating from the middle of simulation domain towards unburned region with unburned gas mixture continuously injected from inlet.

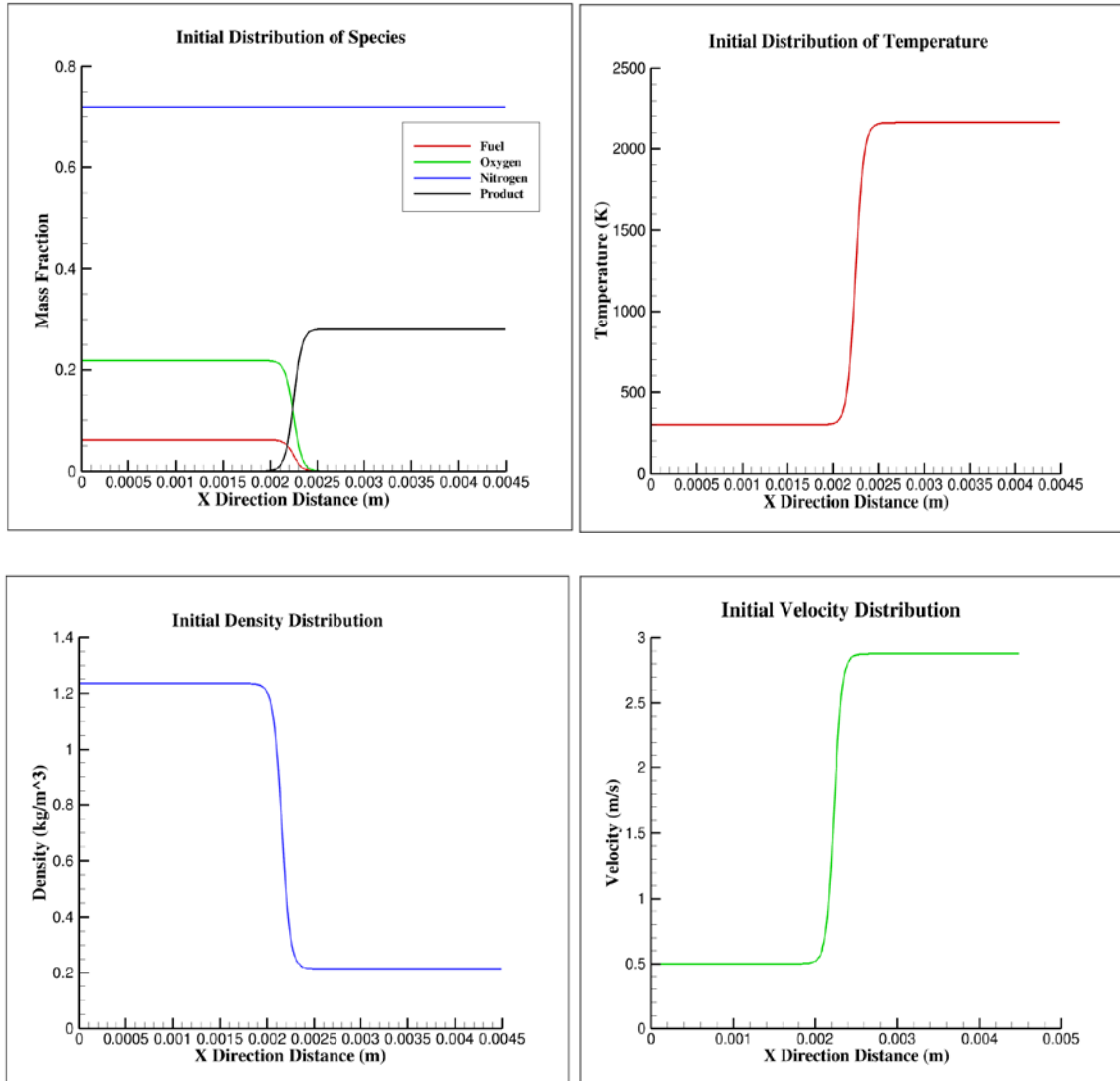


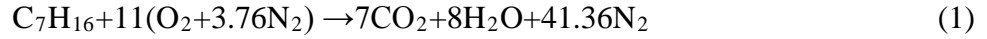
Figure 2: Initial Distribution of Species, Density and Fluid Velocity

The air-fuel equivalence ratio of unburned gas mixture is set to range from 0.5 to 4.5. The computer code can convert the air-fuel equivalence ratio to mass fraction of each species for simulation. For each simulation, the initial unburned gas mixture composition is set to be the same as the inlet mixture composition. Under different equivalence air-fuel ratio, the flame consumption speed is different.

2.2 Initialization of Combustion Model

2.2.1 Chemical Reaction Formula

The fuel that is used is n-heptane whose chemical formula is C_7H_{16} . The chemical reaction formula can be written as:



The molecular mass as well as the chemical reaction formula coefficient are put into the code for initialization. Those information combined with air-fuel equivalence are utilized to calculate the density and mass fraction of each species.

2.2.2 Single Step Chemical Reaction Parameters

In order to generate the combustion and flame propagation, the chemistry model needs to be utilized. Recent research has proved the importance of detail chemical kinetics in modeling the structure of flames. On the other hand, there is a continuing need for simple and reliable chemistry model to generate experimental flame propagation. [5] To be specific, the simulation of flame propagation in 2D and 3D geometry needs to consume a large amount of computational capacity. The use of detailed kinetic chemistry model would further the computational cost. In this case, the single step chemistry reaction model is utilized. The simplified combustion model sets the temperature, activation energy, concentration of fuel and oxidizer as the factors that influence the reaction rate. The equation is defined as:

$$K_{ov} = AT^n \exp(-E_a/RT) [Fuel]^a [Oxidizer]^b \quad [5] \quad (2)$$

where

$$K_{ov} = \text{Reaction Rate (mole/s)}$$

$$E_a = \text{Activation Energy (J)}$$

$$R = \text{Ideal Gas Constant (8.314 J * mole}^{-1}\text{K}^{-1}\text{)}$$

$$T = \text{Temperature (K)}$$

$$a, b = \text{Rate Parameters}$$

$$[Fuel], [Oxidizer] = \text{Concentration (mole/m}^3\text{)}$$

$$AT^n = \text{Pre Exponential Factor (m}^3\text{/(mole.S))}$$

Refer to equation 2, several parameters need to be set for each specific fuel. Since this simulation use n-heptane as fuel, according to recent research, the parameters are set as following.

Pre-exponential Factor AT^n ($m^3/(mole.S)$)	5.1×10^{11}
Activation Energy E_a (J)	125520
Rate Parameter a	0.25
Rate Parameter b	1.5
Flammability Limits	0.5 to 4.5

Table 1: Single Step Reaction Rate Parameters [5]

Aside, the change of temperature is based on the reaction rate, heat of combustion of fuel and specific heat of gas mixture. Heat of combustion can be calculated by taking the difference of enthalpy of formation of reactants and products. For heptane, the heat of combustion is 4.58×10^6 J/mole under complete combustion condition. Furthermore, based on detail chemistry simulation, the temperature ranges from 298 K to 2200 K for equivalence air-fuel ratio being set to be 1. The temperature of flame front is around 1500K.

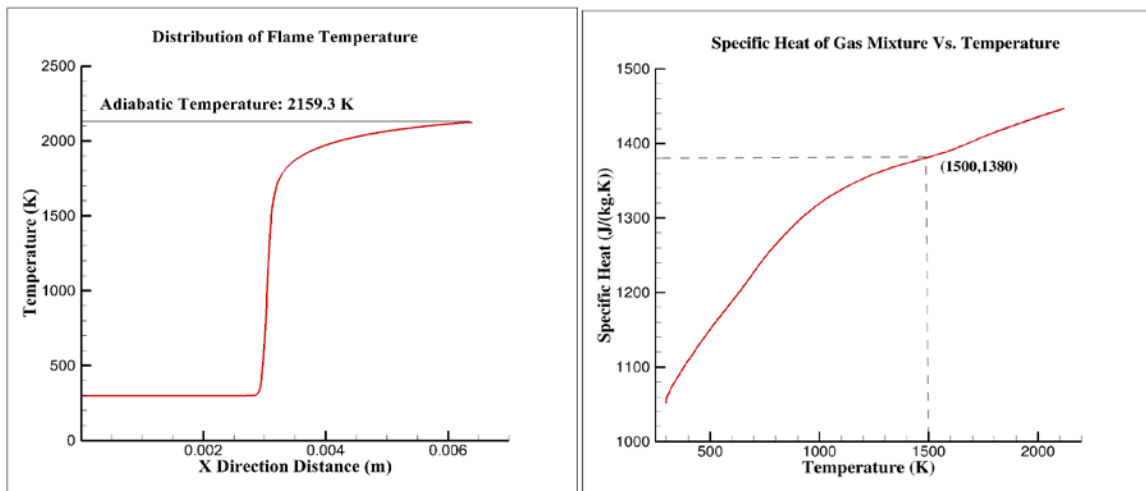


Figure 3: Adiabatic Flame Temperature (Stoichiometric n-heptane Air-Fuel Equivalence Ratio) and Specific Heat Distribution

In this case, for simplification, the value of specific heat is set to be 1380 J/ (kg.K), which is corresponding to 1500 K temperature. Given specific heat of gas mixture and heat of combustion, the temperature change can be calculate as following:

$$\Delta T = \frac{QK_{ov}}{c_p\rho} \quad (3)$$

where

$$\nabla T = \text{Change of temperature} \left(\frac{K}{s} \right)$$

$$Q = \text{Heat of Combustion (J/mole)}$$

Based on the governing equation, the change of temperature along the flame propagation can be evaluated as:

$$\Delta T = \frac{Q \times K_{ov}}{c_p \times \rho} \quad (4)$$

where

$$Q = \text{Heat of Combustion (J/kg)}$$

$$c_p = \text{Specific heat of mixture gas} \left(\frac{J}{kg \times K} \right)$$

$$\rho = \text{Density of mixture gas} \left(\frac{kg}{m^3} \right)$$

In return, the temperature change would affect the reaction rate. Basically, the chemical reaction would release heat, which increase the temperature. The increase of temperature would intensify the chemical reaction. Both temperature and reaction rate will reach a steady states after the flame is fully developed. Aside, the consumption of fuel and oxidizer can be calculated as:

$$St = \frac{nu * W * K_{ov}}{\rho} \quad (5)$$

where

$$St =$$

Consumption rate of fuel and oxidizer, Producing rate of product ($\frac{1}{s}$)

ν = Chemical formula coefficient

W = Molecular Weight ($\frac{kg}{mole}$)

According to the temperature change and consumption and producing rate equations, we can map the distribution of temperature, fuel, oxidizer and product along the flame propagation range.

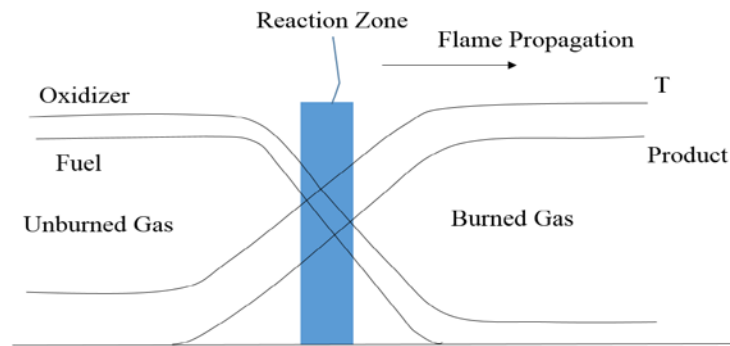


Figure 4: Distribution of Temperature, Product, Fuel and Oxidizer.

Chapter 3: Flame Propagation, Heat Transfer, and Quenching

3.1 Flame Propagation

There are several parameters being used to monitor the flame propagation. One of them is the flame consumption speed. It is evaluated by integrating along the simulation domain:

$$S_L = -\frac{1}{\rho Y_f} \int_{-\infty}^{+\infty} K_{ov} dx \quad (6)$$

where

Y_f = Mass Fraction of Fuel in Unburned Mixture

ρ = Density of Unburned Mixture kg/m^3

S_L = Consumption Speed $\left(\frac{m}{s}\right)$

Consumption speed of flame highly depends on chemical reaction rate, initial pressure, and unburned gas temperature. Chemical reaction rate is determined by the Equivalent air-fuel ratio, temperature and fuel type. In this case, under the same initial conditions, the adjustment of Equivalent air-fuel ratio and fuel type would change the consumption speed.

The Consumption speed is a reference parameter to set the inlet flow velocity. At the steady state, if the inlet flow velocity is smaller than the consumption speed, flame front will move towards to inlet. If the inlet flow velocity is larger than consumption speed, the flame front will be pushed to the end of simulation domain. In order to have a stable and fully developed flame, the inlet flow velocity should be equal to the flame consumption speed.

Another important factor is the flame thickness. There are several definitions of flame thickness. One definition that is used in this research is based on the temperature change.

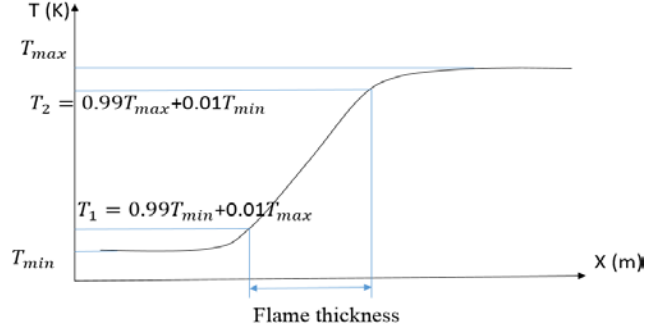


Figure 5: Definition of Flame Thickness

Flame thickness is mainly determined by the diffusivity of the gas mixture and chemical reaction rate. Diffusivity changes with temperature, which is shown as following. Aside, the correlation of viscosity has the same form as diffusivity.

$$D = D_0 \left(\frac{T}{T_0} \right)^{0.75} \quad (7)$$

where

$$D = \text{Diffusivity at Temperature } T \left(\frac{m^2}{s} \right)$$

$$D_0 = \text{Diffusivity at } T_0 = 298K \left(\frac{m^2}{s} \right)$$

The diffusivity is in direct relation to the flame thickness. To be specific, the larger the diffusivity, the quicker the unburned gas mixture is feed into the flame. Under the same chemical reaction rate, this amount of fuel needs to be consumed over a longer distance. The chemical reaction rate is in indirect relation to flame thickness. Under the same diffusivity, the larger the chemical reaction rate, the quicker the fuel gets burned. In this case, the fuel would be burned in a shorter distance.

The last parameter to evaluate the flame propagation is the maximum flame temperature. The flame temperature is determined by the chemical reaction rate, specific heat of gas mixture and heat of combustion. The simulation temperature is often used to compare with the experimental results for validation.

3.2 Walls Heat Transfer

The presence of conduction enables the energy to transfer from the flame across walls. In order to account for the heat transfer, each grid point except for the walls has an energy balance governing equation based on the first law of thermodynamics. In terms of differential equation, the energy balance equation can be rewritten as:

$$\rho C_p \frac{DT}{Dt} = \dot{\omega}'_T + \nabla \cdot (\lambda \nabla T) \quad (9)$$

where

$$\dot{\omega}'_T = - \sum_{k=1}^N h_k \dot{\omega}_k$$

$h_k = \text{Enthalpy of species } k$

The wall heat transfer flux is evaluated as

$$q_w = -\lambda \frac{\partial T}{\partial n} \quad (10)$$

where

$n = \text{wall normal direction.}$

Additionally, the temperature of walls is set to be constant. Given this boundary condition and energy balance equation, the heat transfer occurs between the fluid and the walls.

3.3 Quenching

When the flame propagates through a small diameter tube whose diameter ranges 1mm to 10 mm, the heat transfer of the walls can cause the quenching of flame. The quenching distance is refer to the minimum diameter of tube for flame propagation

through. In order to derive the equation. There are two criteria that are ignition criteria and quenching criteria. The ignition criteria mentions that the ignition will happen only if enough heat is transferred to the unburned gas that is as thick as steady laminar flame thickness to adiabatic flame temperature. The quenching criteria defines that the quenching will occur when the heat from chemical reaction of the flame front is less than the heat transfer out of the system.

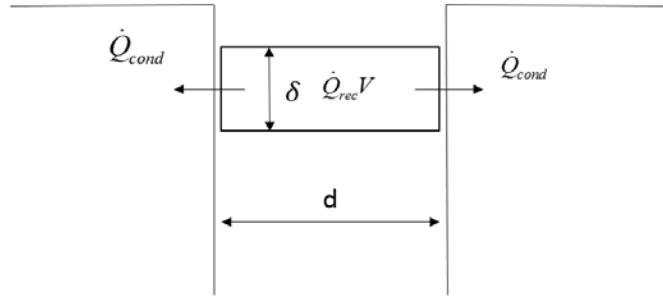


Figure 6: Illustration of Flame Quenching Model [8]

The energy balance equation is listed as following:

$$\dot{Q}_{rec} V = \dot{Q}_{cond,tot} \quad (10)$$

where

$V = \text{Volume of Flame Front Region}$

$\dot{Q}_{rec} = \text{Heat Release From Chemical Reaction}$

$\dot{Q}_{cond,tot} = \text{Heat Loss from Conduction}$

The volumetric heat release rate \dot{Q}_{rec} can be calculated as following:

$$\dot{Q}_{rec} = -\dot{m}_F \Delta h_c \quad (11)$$

where

$\dot{m}_F^m = \text{Volumetric Mass Production Rate of Fuel (kg/s - m}^3)$

$\Delta h_c = \text{Change of Enthalpy } \left(\frac{J}{Kg}\right)$

The heat conduction across the walls can be calculated as following:

$$\dot{Q}_{cond} = -kA \frac{dT}{dx} \Big|_{T_w} \quad (12)$$

where

$A = 2\delta L$ (Area between the Flame Front and Walls (m^2))

$K = \text{Heat Transfer Coefficient (W/ (m}^2K))$

$T_w = \text{Wall Temperature (K)}$

Based on above relation, the energy balance Equation can be derived as following:

$$(-\bar{\dot{m}}_F^m \Delta h_c)(\delta dL) = k (2\delta L) \frac{T_b - T_w}{d/b} \quad (13)$$

$$\bar{\dot{m}}_F^m = \frac{1}{T_b - T_u} \int_{T_u}^{T_b} \dot{m}_F^m dT \quad (14)$$

where

$\bar{\dot{m}}_F^m = \text{Average Volumetric Mass Production Rate of Fuel (kg/s-m}^3)$

$T_b = \text{Burned Region Temperature (K)}$

$b = \text{Parameter that is greater than 2}$

$\frac{T_b - T_w}{d/b} = \text{Temperature Gradient from Centerline to Walls}$

Based on above relations, the quenching distance d can be correlated with flame thickness δ .

$$d = \sqrt{b\delta} \quad (15)$$

The theoretical result indicates that the quenching distance is greater than flame thickness for various fuels.

Chapter 4: Simulation of Flame Propagation

4.1 Flame Propagates in One dimensional Open Space

The one dimensional open space simulation is used for validating the overall simulation model by comparing the results from single step simulation with experimental data.

4.1.1 Initialization of Model

For open space simulation, there is no walls defined around the flame. The grid points are used to mesh x direction only. The left set of domain is set to be inlet and the right side is set to be outlet. Fluid is free to flow in either direction in open space.

X Distance (m)	Y Distance (m)	X Direction Mesh	Y Direction Mesh	Thermal Dynamic Pressure (Pa)	Initial Unburned Gas Temperature (K)
0.0045	0.0020	750	1	101325	298

Table 2: Initial and Boundary Conditions of One Dimensional Open Space Flame Propagation

4.1.2 Model Validation

After flame stabilizes, the temperature profile is shown as following. There is no temperature gradient along vertical direction since the walls are absent. The simulation domain is divided by the flame front to the unburned and burned region.

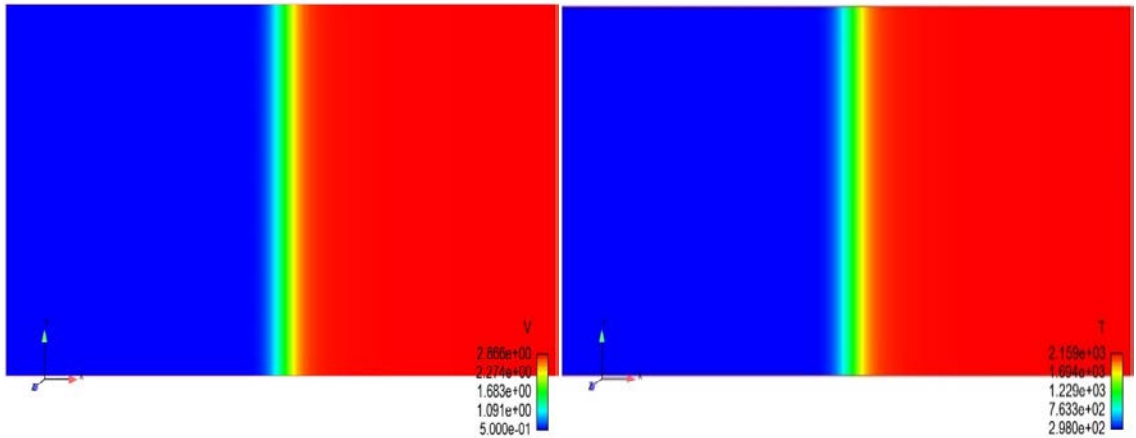


Figure 7: Distribution of Fluid Velocity and Temperature (Unit:m/s,K, Stoichiometric *n*-heptane Air-Fuel Equivalence Ratio, $T_{Unburned} = 298K, P_{Initial} = 1atm, Simulation Time: 5.7595E-03 s$)

For comparison, the consumption speed and adiabatic flame temperature are extracted from single step chemistry simulation to compare with the experimental data [7].

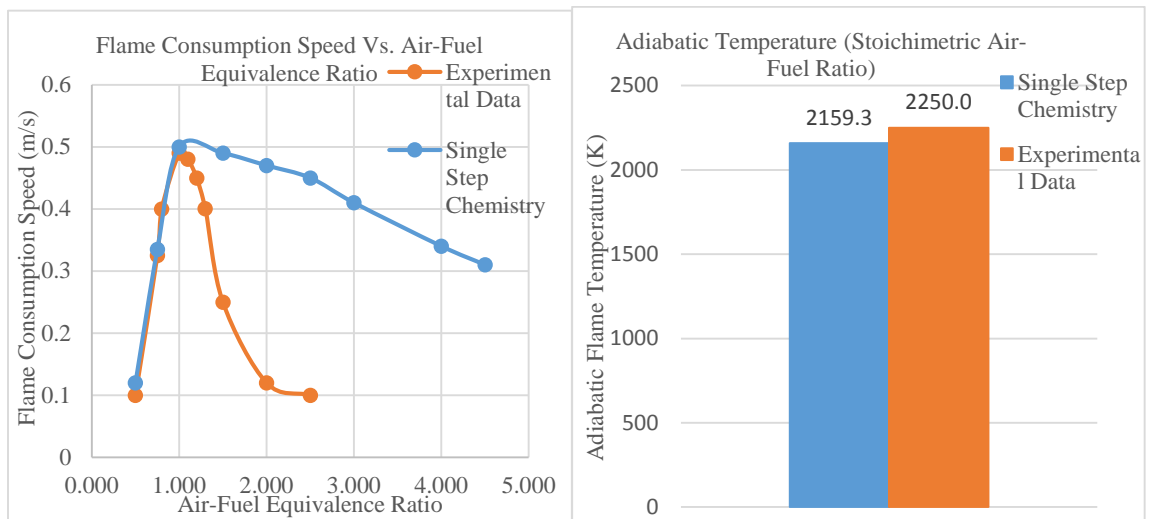


Figure 8: Graph of Validation of Flame Consumption Speed and Adiabatic Temperature

Based on the above graph, the single step chemistry model correctly predicts the flame consumption speed when air-fuel equivalence ratio ranges from 0.5 to 1. On the other side, the result shows a serious errors for rich mixtures. The adiabatic flame temperature of single step chemistry is 2159.3 K and the adiabatic temperature of experiment is

2250.0 K. Therefore, we can predict the simulation results of following model for lean mixtures are more accurate.

4.2 Flame Propagation in a Two Dimensional Channel with Adiabatic Walls

Under the presence of adiabatic walls at the bottom and top, fluid flow is predicted to be affected by the walls. There is no heat flux across the walls. The velocity of gas is zero as gas propagating normal to the walls. The left side is set to be inlet and right side is set to be outlet. The gas is free to flow in x and y direction at the open boundary condition. The boundary condition is shown in the following table.

X Distance (m)	Y Distance (m)	X Direction Mesh	Y Direction Mesh	Thermal Dynamic Pressure (Pa)	Initial Unburned Gas Temperature (K)
0.0045	0.0020	750	300	101325	298

Table 3: Initial and Boundary Conditions of Two Dimensions Adiabatic Walls Flame Propagation

There are two tests cases associated this. One case has the fuel injection from the inlet. The other case does not have the fuel injection. These two cases are used to investigate how the presence of fuel injection affect distribution of flame properties.

4.3 Flame Propagation in a Two Dimensional Channel with Conductive Walls (Tube Quenching Model)

Tube quenching simulation is also conducted in the two Dimension domain. The initial condition and boundary condition are set as following, which is the same as adiabatic case except for the temperature of walls are set to be a constant value T_w . Also, there is no fuel injection from the inlet. The flame propagates from the middle of simulation domain towards the inlet. In reality, the temperature of walls varies within 10 K when quenching happens [6]. Therefore, it is a reasonable approximation of walls temperature.

X Distance (m)	Y Distance (m)	X Direction Mesh	Y Direction Mesh	Thermal Pressure (Pa)	Initial Unburned Gas Temperature (K)
0.0045	0.0020	750	300	101325	298

Table 4: Initial and Boundary Conditions of Two Dimensions Conductive Walls Flame Propagation

For each air-fuel equivalence ratio and unburned gas temperature, the quenching distance can be found by shrinking the diameter of tube till the flame stop propagating. However, this method is time-consuming especially for fine mesh and large size channel. In this case, the approximation method is utilized. Based on the flame propagation in one dimensional open space, the relationship between chemical reaction and temperature can be extracted. According to figure

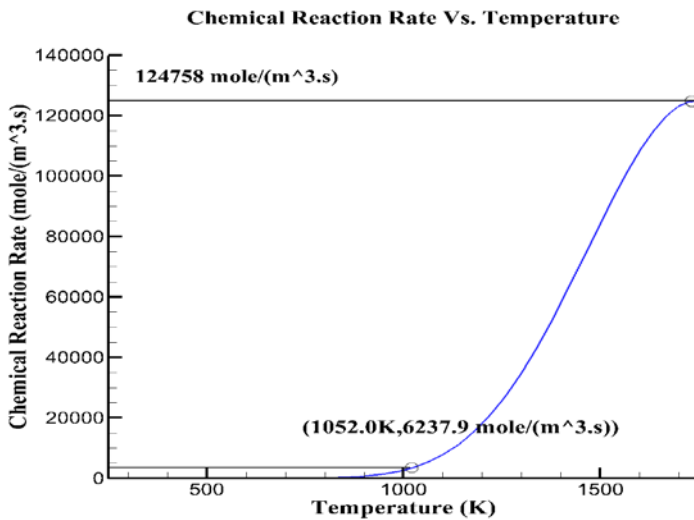


Figure 9: Relationship between the Chemical Reaction Rate and Temperature

9, the maximum chemical reaction rate is 124758 mole/ (m³.s). We assume that the temperature that corresponds to 5% of the maximum chemical reaction rate is the temperature for flame quenching. The quenching temperature is 1052.0K. Based on this, we can find the distance from walls to this temperature region. The quenching distance is twice of this distance. The test is aim to research the relationship between quenching distance and air-fuel equivalence ration and unburned gas temperature.

Chapter 5: Test Cases and Result Analysis

5.1 Flame Propagation in a Two Dimensional Channel with Adiabatic Walls

The presence of adiabatic walls around flame would affect the flame propagation.

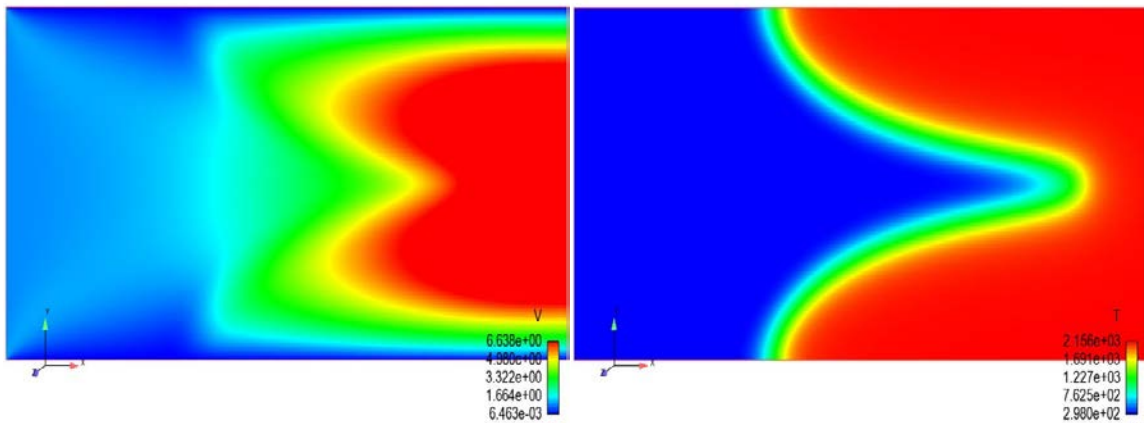


Figure 10: Fluid Velocity and Temperature Profiles in a Two Dimensional Channel with Adiabatic Walls and Fuel Injection (Unit:m/s, K, Stoichiometric n-heptane Air-Fuel Equivalence Ratio, $T_{unburned}=298K, P_{Initial}=1atm, Simulation Time:1.80050E-03 s$)

The fluid velocity boundary layer is formed near the walls due to viscosity, which stretch the temperature profile. Due to the area of flame front increases caused by the stretching, the consumption speed increases significantly, which is equal to 1.03m/s for Stoichiometric n – heptane air – fuel equivalence ratio.

With fuel injection from the inlet, the low temperature unburned gas push the high temperature burned gas towards to the outlet and form a V shape. Without fuel injection, the depth of V shape is smaller compared with the case with fuel injection. The V shape is only due to the velocity boundary layer that forms along vertical direction

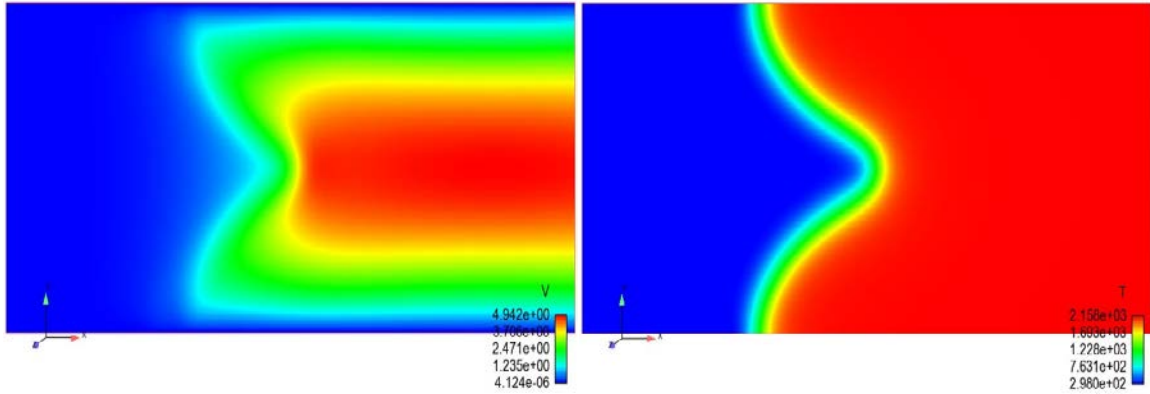


Figure 11: Fluid Velocity Profiles and Temperature in a Two Dimensional Channel with Adiabatic Walls and no Fuel Injection (Unit:m/s,K, Stoichiometric n-heptane Air-Fuel Equivalence Ratio, $T_{unburned}=298K$, $P_{Initial}=1atm$, Simulation Time: $2.00000E-3$)

5.2 Flame Propagation in a Two Dimensional Channel with Conductive Walls (Tube Quenching Model)

The presence of conductive walls surrounding the flame would significantly affect the temperature profile along vertical axis. The fluid velocity boundary layer forms near the walls due to viscosity. The thermal boundary layer also forms near the walls due to heat transfer. As the flame further develops, the high temperature region will shrink. At the steady state, the area of high temperature region and the temperature gradient will become constant.

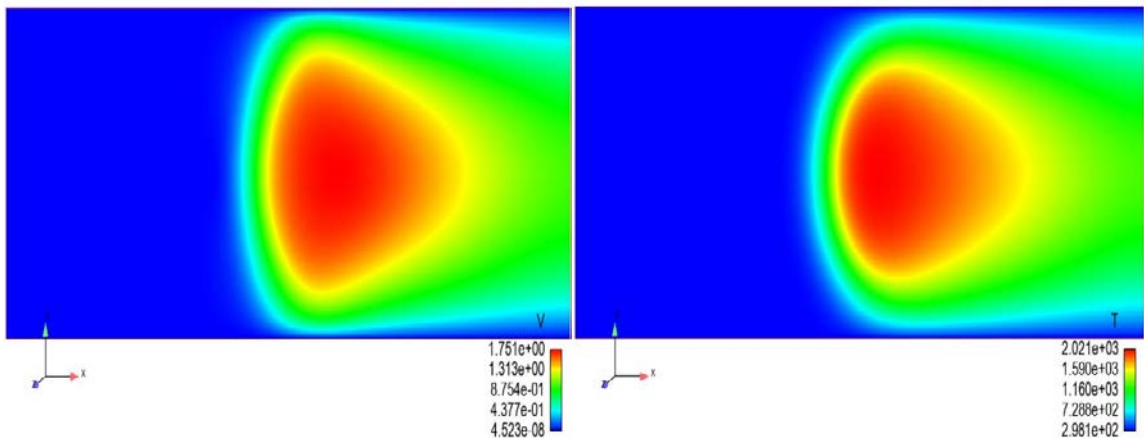


Figure 12: Fluid Velocity and Temperature Profiles in a Two Dimensional Channel with Conductive Walls (Unit:m/s,K, Stoichiometric n-heptane Air-Fuel Equivalence Ratio, $T_{unburned}=298K$, $T_{wall}=298K$, $P_{Initial}=1atm$, Simulation Time: $1.70000E-3s$)

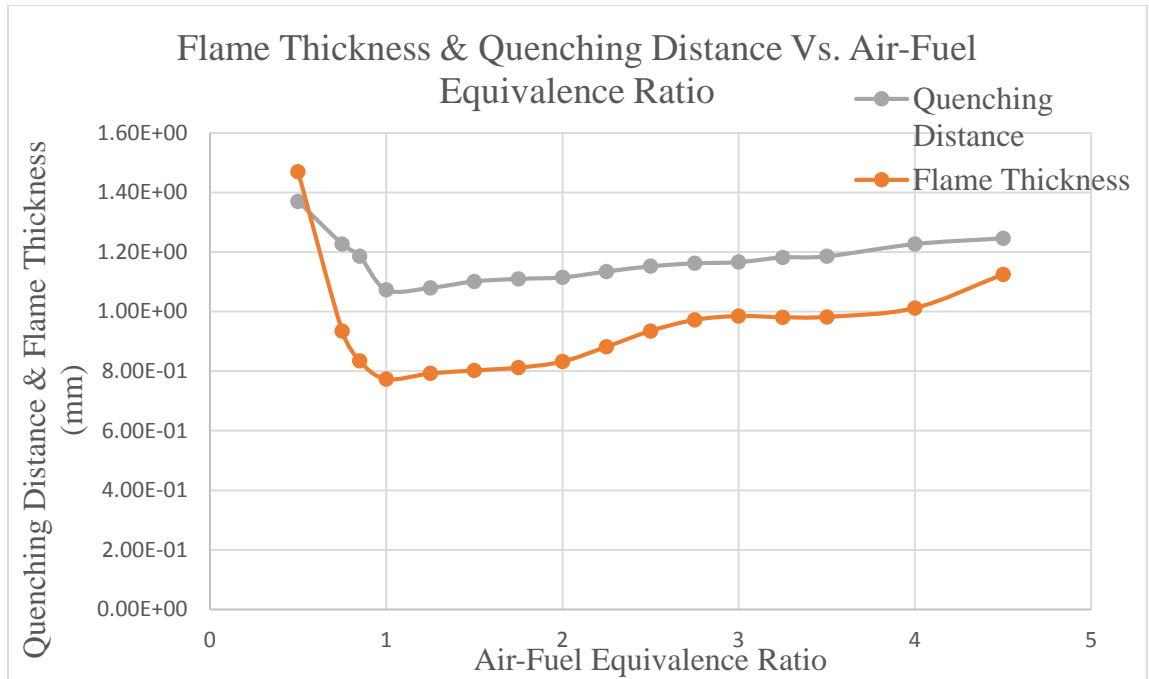


Figure 13: Relationship between Quenching Distance & Flame Thickness and Air-Fuel Equivalence Ratio ($T_{unburned} = 298K$, $T_{wall} = 298K$, $P_{Initial} = 1atm$)

The magnitude of quenching distance ranges from 1.10 mm to 1.37 mm. The minimum quenching distance occurs when the air-fuel equivalence ratio is equal to 1. The quenching distance increases as the air-fuel equivalence ratio deviates from 1. The flame thickness ranges from 0.77 mm to 1.47 mm, which is smaller than the quenching distance under every air-fuel equivalence ratio except for 0.5. The flame thickness has the same trend as quenching distance. This result also meets equation 15 that indicates that the quenching distance is predicted larger than the flame thickness.

Chapter 6: Summary & Conclusions

6.1 Flame Propagation in a Two Dimensional Channel with Adiabatic Walls

Based on the simulation result, the presence of adiabatic walls makes the fluid velocity boundary layer form near the walls. Due to the non-uniform velocity profile, the area of flame front increases, which increases the flame consumption speed. The temperature profile is also stretched to be a V shape because the high speed and low temperature unburned gas near the centerline blows the high temperature burned gas backward.

When there is no fuel injection into the combustion channel, the temperature and velocity profiles are similar to those for the case with fuel injection. Since there is no low temperature unburned gas, the depth of the V shape of both the temperature and velocity profiles are smaller than the case with fuel injection.

6.2 Flame Propagation in a Two Dimensional Channel with Conductive Walls (Tube Quenching Model)

Based on the relationship between the quenching distance and air-fuel equivalence ratio, the minimum quenching distance occurs when air-fuel equivalence ratio is equal to 1. The quenching distance will increase as the air-fuel equivalence ratio deviates from 1. The flame thickness is smaller than the quenching distance and has the same trend as the quenching distance. This result also meets the theoretical relationship between the quenching distance and flame thickness.

Chapter 7: Recommendation and Future Work

This research focuses on using n-heptane as the fuel. In the future, the fuel type can be changed. Furthermore, this research only studies the flame propagation in a two dimensional domain. Future research will explore three dimensional cases, which is more practical.

In addition to the tube quenching, head on quenching is another important flame wall interaction model. It is typically used for studying the auto ignition inside the engine, which is also called engine knock. In the future, the head on quenching model can be built based on the current model. The inlet will be replaced by a conductive wall with a constant temperature. The flame can propagate towards the wall perpendicularly. During the propagation, the quenching will happen as the flame approaches the wall. In order to study the quenching, the chemical reaction rate of the flame front will be extracted. We can predict that the chemical reaction rate will decrease as the flame front approaches the wall. During this study, we can change the fuel type, initial pressure, unburned gas temperature, wall temperature, size and geometry of the simulation domain and air-fuel equivalence ratio. Based on those factors, we can map how each factor affects the head-on-quenching. The head on quenching research can be done in both two and three dimensional domains.

Appendix

A1) Distribution of Species and Density for One Dimensional Open Space Flame Propagation (Stoichimetric n – hepatne Air –

Fuel Equivalence Ratio, $T_{\text{Unburned}} = 298\text{K}$, $P_{\text{Initial}} = 1\text{atm}$,

Simulation Time: $5.7595\text{E} - 03\text{ s}$)

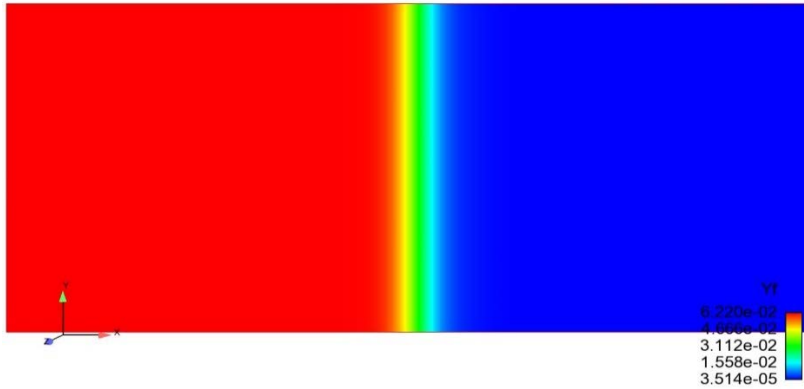


Figure 14: Distribution of Fuel Mass Fraction

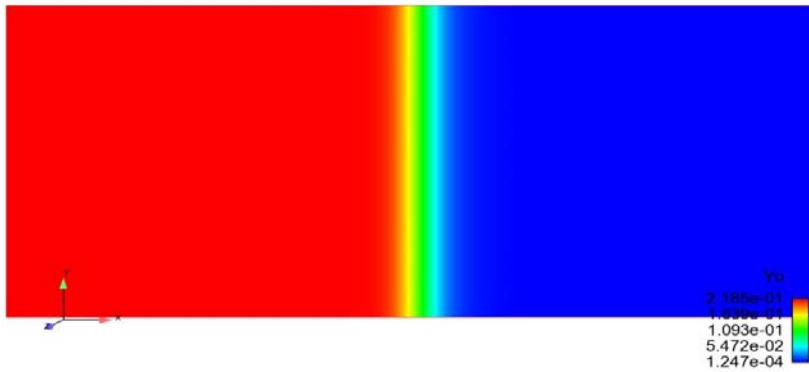


Figure 15: Distribution of Oxidizer Mass Fraction

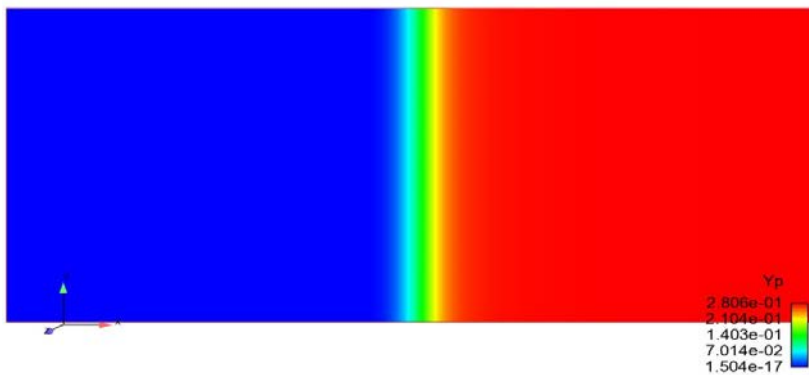


Figure 16: Distribution of Products Mass Fraction

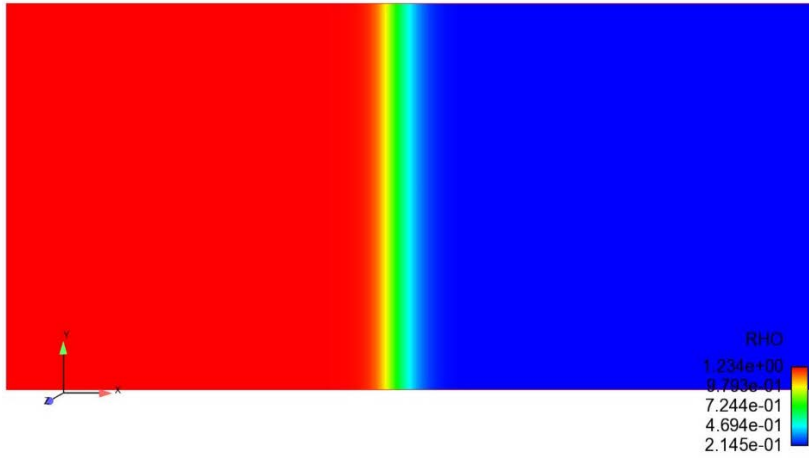


Figure 17: Distribution of Density (Unit: m^3/s)

A2) Distribution of Species and Density for Two Dimensional Channel with Adiabatic Walls Flame Propagation with Fuel Injection (Stoichimetric n – hepatne Air – Fuel Equivalence Ratio, $T_{Unburned} = 2, P_{Initial} = 1atm$, Simulation Time: $1.80050E - 03$ s)

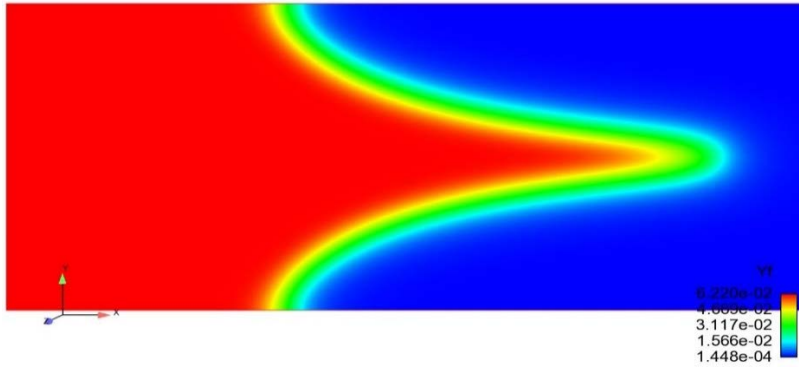


Figure 18: Distribution of Fuel Mass Fraction

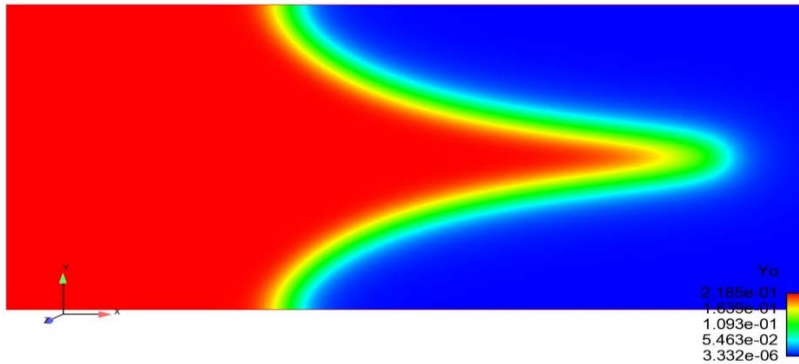


Figure 19: Distribution of Oxidizer Mass Fraction

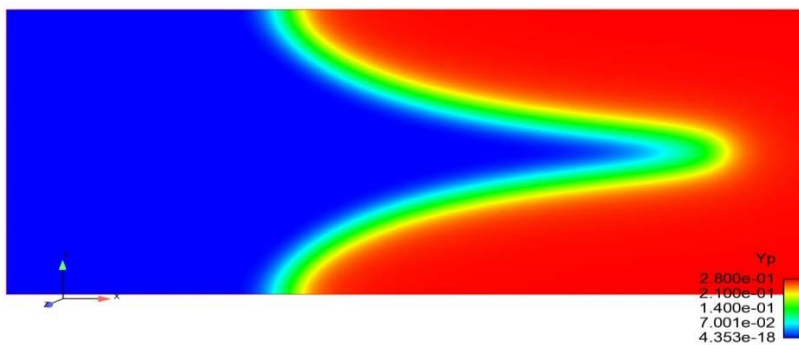


Figure 20: Distribution of Products Mass Fraction

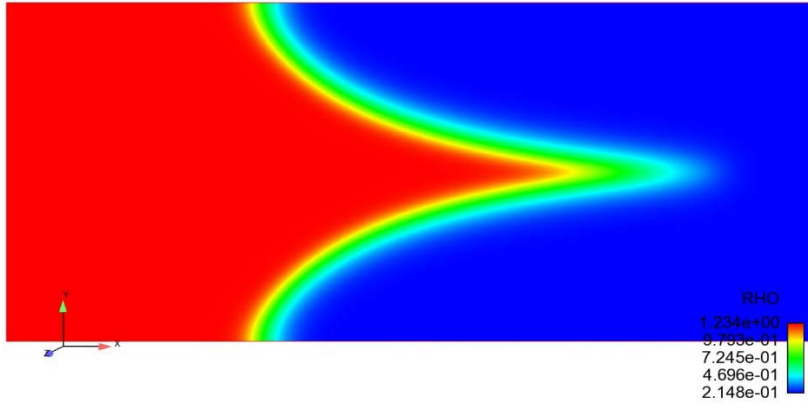


Figure 21: Distribution of Density (Unit: m^3/s)

A3) Distribution of Species and Density for Two Dimensional Channel with Adiabatic Walls Flame Propagation without Fuel Injection (Stoichimetric n – hepatne Air – Fuel Equivalence Ratio, $T_{Unburned} = 2, P_{Initial} = 1atm$, Simulation Time: $2.00000E - 03$ s)

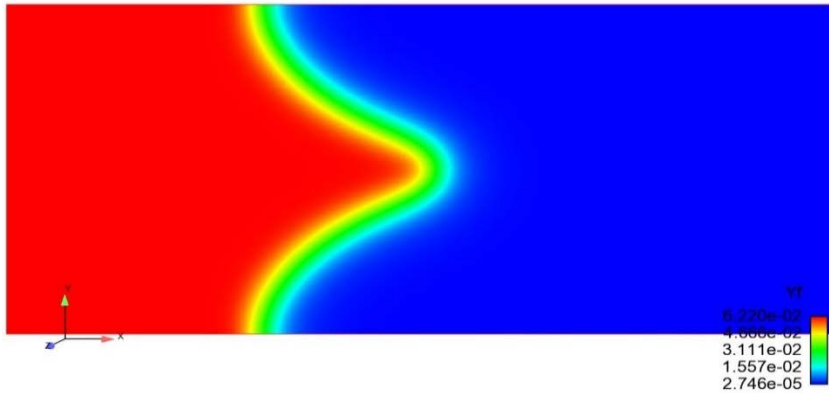


Figure 22: Distribution of Fuel Mass Fraction

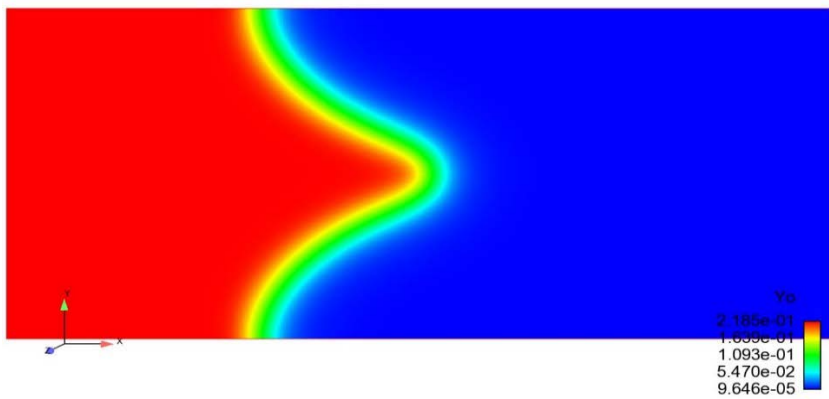


Figure 23: Distribution of Oxidizer Mass Fraction

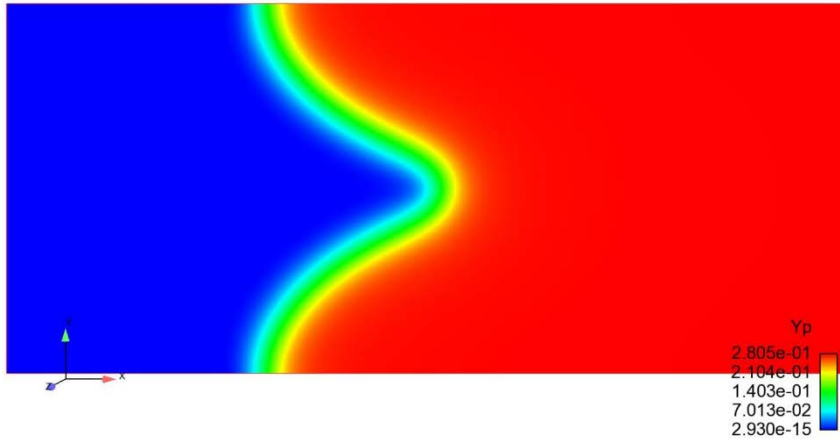


Figure 24: Distribution of Product Mass Fraction

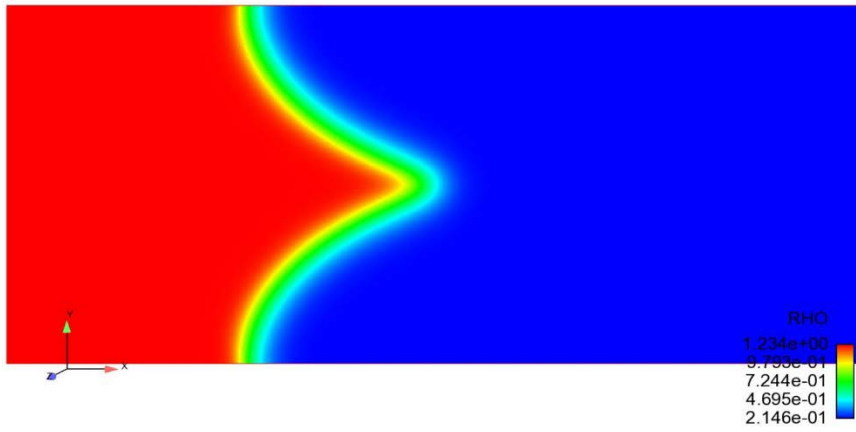


Figure 25: Distribution of Density (Unit: m^3/s)

**A4) Distribution of Species and Density for Two Dimension Conductive Walls
 Flame Propagation (Stoichimetric n – hepatne Air –
 Fuel Equivalence Ratio, $T_{Unburned} = 2$, $T_{Walls} = 298K$, $P_{Initial} = 1atm$,
 Simulation Time: $2.00000E - 03 s$)**

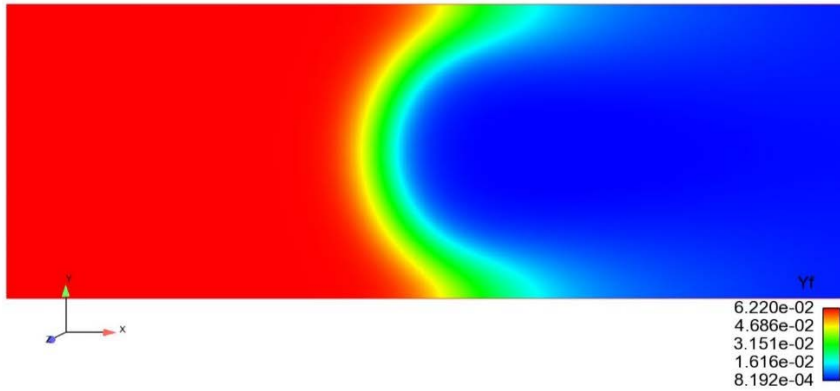


Figure 26: Distribution of Fuel Mass Fraction

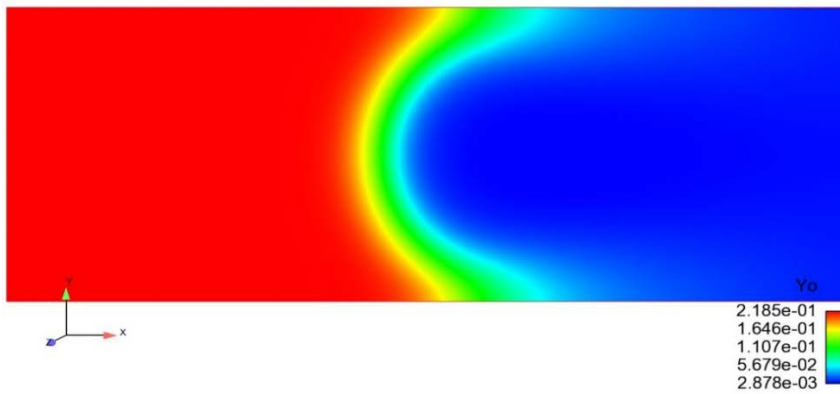


Figure 27: Distribution of Oxidizer Mass Fraction

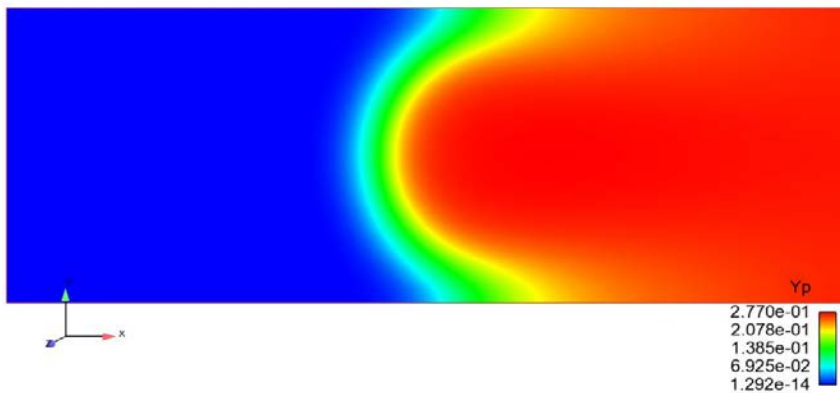


Figure 28: Distribution of Products Mass Fraction

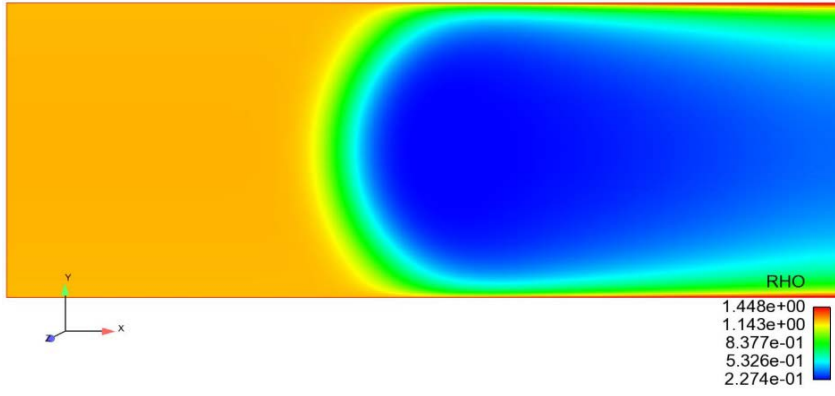


Figure 29: Distribution of Density (Unit: m^3/s)

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