Study the Effect of Initial Temperature and Equivalence Ratio on the Pre – mixed Flame Propagation

Sadoun Fahad Dakhil, Tahseen Ali Gabbar

Technical college / Basrah

Abstract:

In this study a two dimensional, steady state propagation of the laminar premixed flame was numerically and experimental are investigated. The energy, momentum, continuity equations for species and global reaction mechanism with equation of stat for ideal gases were solved. Constant temperature boundary condition is applied on axi – symmetric in y – direction domain.

The governing equations were discretized by using computation fluid dynamics (CFD) and finite-volume method - central differencing scheme, then solved using Gauss-Seidel Iteration method on uniformed grid with VISUAL BASIC code. Effects of equivalence ratio and initial temperature of fresh gases (air – fuel mixture) were investigated for three types of fuel Methane, Propane and Butane. Also the flame speed and flame temperature were experimentally measured for air – fuel (Methane, Propane, Butane and LPG) mixtures. The burning velocity was calculated depending on the flame speed and flame temperature measuring. The flame speed and flame temperature were measured by using optical technique. Effects of equivalence ratio and initial temperature of fresh gases (air – fuel mixtures) on flame speed and flame temperature are investigated experimentally. Results were generated for the detailed description of the local fluid flow and heat transfer characteristics including temperature, axial velocity, density and mole fractions profiles. For example the flame speed reached 274.4 cm/s as a maximum value at $\emptyset=1.1$ for propane air mixture, and flame temperature comparison give a good agreement between theoretical and experimental results at rich mixtyre

دراسة تأثير درجة الحرارة الابتدائية ونسبة الخلط المكافئة على أنتشار اللهب مسبق الخلط

سعدون فهد داخل   تحسين علي جبار

الكلية التقنية – البصرة

الخلاصة:

تم في هذا البحث دراسة انتشار اللهب الطباقي ثنائي البعد، مستنقر، مسبق الخلط. حيث تم حل كل من معادلة الطاقة ومعادلة الزخم والاستمرارية ثنائية البعد لتفاعل كلي بالإضافة إلى استخدام معادلة الحالة للغازات المثالية. أن الدراسة كانت لحالة متناظرة وثبوت درجة حرارة السطح السفلي.

أن هذه المعادلات تم تحويلها إلى معادلات خطية باستخدام نظرية الحجوم المحددة وحلها بطرية تقارب كاوس لحجوم ثابتة ومتساوية وبرمجتها بلغة فجول بيسك. خلال البرنامج أخذ بنظر الاعتبار تأثير النسبة المكافئة(Φ) ودرجة حرارة الخليط الابتدائية وذلك للخلائط ميثان، بروبان وبيوتان – هواء.

كذلك تم دراسة وقياس سرعة جبهة اللهب ودرجة حرارته وحدد عملياً لخلائط (الميثان، البروبان ، البيوتان و غاز المسال – هواء). من خلال قياس سرعة جبهة اللهب ودرجة حرارته ثم حساب سرعة الاتقاد. أن سرعة جبهة اللهب ودرجة حرارة اللهب تم قياسها باستخدام تقنية بصرية (Photocell). كذلك أخذ بنظر الاعتبار تأثير كل من النسبة المكافئة ودرجة حرارة الخليط الابتدائية على سرعة جبهة اللهب ودرجة حرارته وبالتالي على سرعة الاتقاد. تم مقارنة النتائج النظرية مع العمليه لتوزيع درجات الحراره وكانت متطابقه للميثان عند المخلوط الغني كما ان اعلى قيمه لسرعة جبهة اللهب عند المخلوط بنسبة (1.1).

Symbols

Symbol	Definition	SI Units
A	Area , constant	m^2
C_P	Specific heat at constant pressure	$J/kg.K$
D	mass diffusivity	m^2/s
Δh_f^o	Enthalpy of formation	$kJ/kmol$
K	Thermal conductivity	$W/m.K$
K_p	Water – gas shift equilibrium constant	-
$KG(T)$	Bimolecular rate confident	-
L	Computation domain length	m
MW	Molecular weight	$kg/kmol$
MW_i	Molecular weight of species i	$kg/kmol$
MW_{mix}	Molecular weight of mixture	$kg/kmol$
n_i	Number mole of species i	$kmole$
N_{total}	Total number moles for mixture	$kmole$
s	Stoichiometric air / fuel ratio	-
S_L	Burning velocity	m/s
T	Temperature	K
T_e	Activation temperature	K
T_{in}	Temperature of fresh gases	K
T_2	Flame temperature	K
u	Velocity in x-direction	m/s
v	Velocity in y-direction	m/s
V	Vorticity	$1/s$
α	Number of carbon atoms in fuel	-
x_i	Mole fraction of species i	-
β	Number of hydrogen atoms in fuel	-
μ	Dynamic viscosity	$N.s/m^2$
Φ	Equivalence ratio	-
δ	Flame thicknesses	m
ρ	Density	kg/m^3
adi	adiabatic	--
ω_i	rate formation of specie i	$kmol/m^3.s$

Subscript:

Symbol	Definition
n	North
s	South
w	West
e	East
i	Speeches i
mix	Mixture
in	Unburned
adi	Adiabatic

INTRODUCTION

Most of the energy approximately (88%) is produced by the combustion of fossil fuels like oil, natural gas, and coal. According to the U.S. Department of Energy reports, the demand in the industrial sector is projected to increase by 0.8% per year to the year 2020[1].

When a combustible gas is confined in a channel, the flow induced by the thermal expansion of the gas in the combustion wave is restricted by the tube wall. The flow attains much higher velocities than flow under conditions of free expansion in an open flame. The thermal expansion with the wave generates continuous flow of burnt gas toward the open end i.e. owing to viscous drag. This flow is retarded at the wall and accelerated in the centre of the tube. The later acceleration produces as thrust which pushes the unburned gas in the central region of the tube toward the closed end. Since the gas cannot escape there, it is forced to reverse its direction so that it flows along a curved path away from the center and toward the open end. Figure (1) shows the dashed lines mark the path of various particles, and the arrows represent the particle velocity. It is seen that near the center, the unburned gas flow (at velocity is known as unburned gases velocity which different from flame speed and burning velocity) away from the combustion wave and near the wall it flows toward the wave, so that near center the wave is driven toward the closed end and near the wall it is driven toward the open end. The wave surface becomes curved as shown in figure (1).
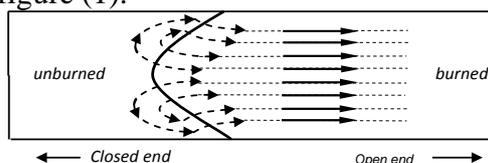
Figure (1) view of direction of flow and particle velocity for laminar combustion wave propagation from open end to closed end of tube.

The elementary case of a one-dimensional laminar flame propagating into a premixed gas (fuel mixed with air) is a basic problem in combustion, both for analytical and numerical techniques. The flame speed also called the burning velocity, normal combustion velocity, or laminar flame speed is more precisely defined as the velocity at which unburned gases move through the combustion wave in the direction normal to the wave surface.

The thermal theories date back to Mallard and Le Chatelier as cited in [1] , who proposed that flame propagation with heat back through layers of gas which is controlled mechanism in flame propagation by using the energy equation as the basis for the development of the thermal theory.

The theory of particle diffusion was first advanced in 1934 by Lewis and von Elbe as cited in[1] in dealing with the ozone reaction. Tanford and Pease cited in [2] carried this concept further by postulating that it is the diffusion of radicals that is all important, not the temperature gradient as required by the thermal theories. In the thermal theory, the higher the ambient temperature, higher the final temperature and therefore the faster the reaction rate and flame velocity. Similarly, in the diffusion theory, the higher the temperature, the greater is the dissociation, the greater is the concentration of radicals to diffuse back, and therefore the faster is the velocity. Consequently, data obtained from temperature and pressure effects did not give conclusive results.

Later, there were improvements in the thermal Flammability limited theories. These theories are two types of flammability limits: Lean limit and Rich limit. Lean flammability limited (LFL) can defined as the minimum composition of a fuel in air over which a flame can propagate. But the rich flammability limited (RFL) is the maximum composition of fuel in air under which a flame can propagate.

Many researchers steady this phenomenon some of them M. Vidal et al [3] and Thierry Poinsot [4]

Finally there are many ways to compute laminar flame structure and speed depending on the complexity of the chemistry and transport descriptions. However, when chemistry and transport are suitably simplified, analytical or semi-analytical solution may be developed and these shed an essential light on the behavior of flame and on the numerical challenges to face when on tries to compute them in simple or complex situations.

The aim for the present work is to study the effect of initial temperature of mixture on the laminar flame speed through experimental measurement and calculation with the effective factors:

Theoretical Analysis:

The physical model studied is shown in the schematic Fig. (2) which represents the laminar two - dimensional axe-symmetric combustion problem. The domain of solution have length ($L=10mm$) and width ($W=3mm$). The fluid enters with a uniform velocity u_{in} equal to laminar burning velocity and uniform temperature T_{in}.
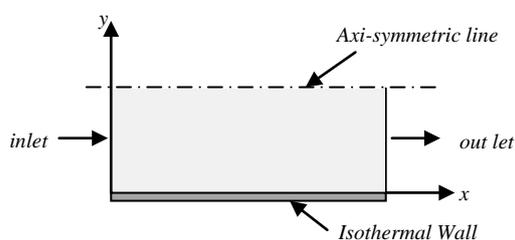
Figure(2) sketch of solution domain

Representation the combustion of an arbitrary hydrocarbon as:

$$C_\alpha H_\beta + a(O_2 + 3.76 N_2) \rightarrow bCO_2$$
$$+ cCO + dH_2O + eH_2 + 3.76aN_2 \qquad (1)$$
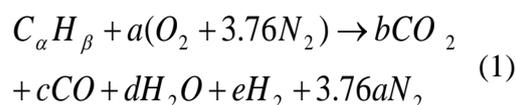
Which for lean or stoicometric conditions ($\Phi <= 1$) becomes

$$C_\alpha H_\beta + a(O_2 + 3.76N_2) \rightarrow bCO_2$$
$$+ dH_2O + fO_2 + 3.76aN_2 \qquad (2)$$
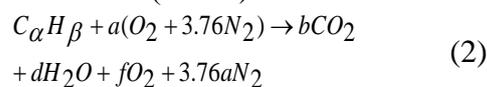
And for rich conditions ($\Phi > 1$) becomes as equation (1):

Since the coefficient *(a)* represents the ratio of the number of moles of O_2 in the reaction to the number of moles of fuel, we can relate *(a)* to the equivalence ratio as [5]:

$$a = \frac{\alpha + \beta/4}{\Phi} \qquad (3)$$

The mole fraction defines as:

$$x_i = \frac{number\ mole\ of\ species\ (i)}{sum\ numbre\ of\ moles\ for\ mixture}$$
$$= \frac{n_i}{N_{Total}} \qquad (4)$$

To find the mole fraction of all of the product species, we will study as:

For lean combustion The coefficient *c* and *e* in equation (2) are zero because there is sufficient O_2 to have all the fuel C and H react to form CO_2 and H_2O, respectively. The coefficient *b, d* and *f* can be found by *C, H* and *O* atom balance, respectively

The total number of moles of products (per mole of fuel burned) can be found by summing the above coefficients together with **3.76a** moles of N_2.

$$N_{Total} = \alpha + \beta/2 + (\frac{\alpha + \beta/4}{\Phi})$$
$$* (1 - \Phi + 3.76) \qquad (5)$$

The mole fractions are then determined by divided each of the coefficients above by N_{Total} for example.

$$xN_2 = 3.76(\alpha + \beta/4)/(\Phi N_{Total}) \qquad (6)$$

For rich combustion ($\Phi > 1$), no oxygen appears, so the coefficient *f* is zero and four unknowns (*b, c, d* and *e*). To solve for these, we employ the three element balances (*C, H,* and *O*) and the water-gas shift equilibrium

$$K_p = \frac{(P_{CO_2}/P).(P_{H_2}/P)}{(P_{CO}/P).(P_{H_2O}/P)} = \frac{b.e}{c.d} \qquad (7)$$

Where: $P_{CO_2}, P_{H_2O}, P_{H_2}, P_{CO}$ represent partial pressure of CO_2, H_2O, H_2 and CO respectively

P Total pressure of mixture then equation (7) can be written as [7].

$$K_P = \exp\left(-4.33 + \frac{4577.8}{T}\right) \qquad (8)$$

Mole fractions and mass fractions are readily converted from one to another using the molecular weights of the spices of interest and of the mixture:

$$Y_i = x_i MW_i / MW_{mix} \qquad (9)$$

$$x_i = Y_i MW_{mix} / MW_i \qquad (10)$$

Where: Y_i, x_i mass and mole fraction respectively and MW_i, MW_{mix} are the molecular weight of species and mixture respectively.

After that, we can calculated the mass fraction of fuel in the fresh gases(air-fuel mixture) as [3]:

$$Y_f = \frac{1}{1 + \dfrac{s}{\Phi}(1 + 3.76\dfrac{MW_{N_2}}{MW_{O_2}})} \qquad (11)$$

Where: s stoichiometric ratio equal to:

$$s = \left(\frac{Y_O}{Y_f}\right)_{st} = \frac{aMW_{O_2}}{MW_f} \qquad (12)$$

The rate, at which the fuel is consumed, can be expressed as [8]:

$$\frac{d[Y_f]}{dt} = A \ \exp\left(\frac{-T_e}{T}\right)[Y_f]^n [Y_{ox}]^m \qquad (13)$$ Where the notation [x] is used to denote the molar concentration (kmol/m^3), and m, n are constants f [8]:

A: is a constant termed the pre-exponential factor or the frequency factor. (*Te*) activation temperature (K). The parameter pre-exponential, activation temperature and m, n constants are shown in the table (1)

Table(1)single– step reaction parameter[8]

Fuel	Pre-exponential factor A	Activation temperature T_e (K)	m	n
CH$_4$	$1.3*10^8$	24358	-0.3	1.3
C$_3$H$_8$	$8.6*10^{11}$	15098	0.1	1.65
C$_4$H$_{10}$	$7.4*10^{11}$	15098	0.15	1.6

The combustion process is governed by mass, momentum, energy and mass- fraction conservation together with an equation of state. In atmospheric combustion(at atmosphere pressure), the influence of pressure variation on density (ρ) may be neglected: therefore, density is a function of temperature (T) only [4], according to the equation of the state. This property is used to reduce the system of equations by introducing a stream function-vorticity formulation [4].For laminar steady state tow – dimensions premixed flames, the conservation equations are:

The total mass conservation equation is unchanged compared to non – reacting flows (combustion dose not generate mass).

$$\frac{\partial(\rho u)}{\partial x} + \frac{\partial(\rho v)}{\partial y} = 0 \qquad (14)$$

The species conservation equation for species (i) can be written as[3] :

$$\rho u \frac{\partial Y_i}{\partial x} + \rho v \frac{\partial Y_i}{\partial y} = \frac{\partial}{\partial x}\left(\rho D_i \frac{\partial Y_i}{\partial x}\right)$$
$$+ \frac{\partial}{\partial y}\left(\rho D_i \frac{\partial Y_i}{\partial y}\right) + \dot{\omega}_i \qquad (15)$$

A. *The vorticity* (V) is defined as [4]:

$$V = \frac{\partial u}{\partial y} - \frac{\partial v}{\partial x} \qquad (16)$$

Within the calculation domain the vorticity (V) is found from the momentum equation [4]:

$$\rho u \frac{\partial V}{\partial x} + \rho v \frac{\partial V}{\partial y} = \mu \left(\frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial y^2} \right)$$
$$+ \frac{\partial \rho}{\partial x} \frac{\partial \zeta}{\partial y} - \frac{\partial \rho}{\partial y} \frac{\partial \zeta}{\partial x} \qquad (17)$$

Where $\zeta = (u^2 + v^2)/2$

***B. The stream function (ψ)* is** defined so that flow field solution always satisfies the continuity equation [4].

$$\rho u = \frac{\partial \psi}{\partial y}$$

$$\rho v = -\frac{\partial \psi}{\partial x}$$

The stream function is calculated from the vorticity definition equation (17):

$$\frac{1}{\rho} \frac{\partial^2 \psi}{\partial x^2} + \frac{1}{\rho} \frac{\partial^2 \psi}{\partial y^2} = V \qquad (18)$$

C. Conservation of energy
The conservation energy equation can be written as [3] :

$$u \frac{\partial T}{\partial x} + v \frac{\partial T}{\partial y} = \frac{K}{\rho CP} \left(\frac{\partial}{\partial x}(\frac{\partial T}{\partial x}) + \frac{\partial}{\partial y}(\frac{\partial T}{\partial y}) \right)$$
$$- \sum_{i=1}^{N} \frac{(h_i^o \omega_i MW_i)}{\rho Cp} \qquad (19)$$

The flame thickness expressed as [6]:

$$\delta = \frac{K_{in}}{\rho_{in} Cp S_L} = \frac{D_{th}^1}{S_L} \qquad (20)$$

Where: K_{in} = unburned conductivity, D_{th} is thermal diffusivity.

Boundary Conditions
The boundary conditions for the computational domain can be specified, based on the previous assumptions[3]

$x = 0 \qquad u = S_L , v = 0$
$\qquad\qquad T = T_{in} , \rho = \rho_{in}$

$x = L \qquad u = S_L(1 + (T_{adi}/T_{in})) , \frac{\partial v}{\partial x} = 0$
$\qquad\qquad T = T_{adi} , \rho = \rho_{in}(T_{in}/T_{adi})$

$y = 0 \qquad u = 0, \quad v = 0$
$\qquad \frac{\partial T}{\partial y} = 0 , \frac{\partial Y_i}{\partial y} = 0, \frac{\partial \rho}{\partial y} = 0$

$y = W \qquad \frac{\partial u}{\partial y} = 0, \frac{\partial v}{\partial y} = 0$
$\qquad \frac{\partial T}{\partial y} = 0 , \frac{\partial Y_i}{\partial y} = 0 , \frac{\partial \rho}{\partial y} = 0$

Numerical Modeling:

A computation fluid dynamics (CFD) procedures for the governing differential equation, are explained by using finite volume method [FVM], through solution of the mass , species, momentum and energy equations to obtain the discretization equations for the velocity, mass fraction and temperature fields.

The discretization equations are solved in a formula of the Central scheme. A computer program of the operation is built according to excplict scheme procedure in VSUAL BASIC Language.
For example the finite volume method is applied on each terms of the energy equation.

$$\int_w^e \frac{\partial}{\partial x}(uT)A + \int_s^n \frac{\partial}{\partial y}(vT)A = \int_w^e \frac{K}{\rho Cp}\left(\frac{\partial^2 T}{\partial x^2}\right)A$$
$$+ \int_s^n \frac{K}{\rho Cp}\left(\frac{\partial^2 T}{\partial y^2}\right)A - \sum_{i=1}^{N} \frac{(h_i^o \omega_i MW_i)}{\rho Cp}A \qquad (21)$$

Result of integration is

$$\left\{ \frac{1}{\Delta x}\left[\begin{matrix}(u_e T_e A_e)\\ -(u_w T_w A_w)\end{matrix}\right] + \frac{1}{\Delta y}\left[\begin{matrix}(v_n T_n A_n)\\-(v_s T_s A_s)\end{matrix}\right] \right\} =$$

$$\left[\frac{1}{\Delta x}\left[\begin{matrix}A_e\left(\frac{K}{\rho Cp}\frac{\partial T}{\partial x}\right)_e\\-A_w\left(\frac{K}{\rho Cp}\frac{\partial T}{\partial x}\right)_w\end{matrix}\right] + \frac{1}{\Delta y}\left[\begin{matrix}A_n\left(\frac{K}{\rho Cp}\frac{\partial T}{\partial y}\right)_n\\-A_s\left(\frac{K}{\rho Cp}\frac{\partial T}{\partial y}\right)_s\end{matrix}\right] \right] - sum_i A \qquad (22)$$

The discretized equation can be written as:

$$\alpha_P T_P = \alpha_N T_N + \alpha_S T_S + \alpha_E T_E + \alpha_W T_W + b \qquad (23)$$

Where:

$$\alpha_N = -\frac{v_n A_n}{2\Delta y} + \frac{K_n A_n}{\rho_n Cp_n \Delta y^2}$$

$$\alpha_S = \frac{v_s A_s}{2\Delta y} + \frac{K_s A_s}{\rho_s Cp_s \Delta y^2}$$

$$\alpha_E = -\frac{u_e A_e}{2\Delta x} + \frac{K_e A_e}{\rho_e Cp_e \Delta x^2}$$

$$\alpha_W = \frac{u_w A_w}{2\Delta x} + \frac{K_w A_w}{\rho_w Cp_w \Delta x^2}$$

$$\alpha_P = \begin{bmatrix} \dfrac{u_e A_e}{2\Delta x} - \dfrac{u_w A_w}{2\Delta x} + \dfrac{v_n A_n}{2\Delta y} - \dfrac{v_s A_s}{2\Delta y} \\[2mm] + \dfrac{K_n A_n}{\rho_n Cp_n \Delta y^2} + \dfrac{K_s A_s}{\rho_s Cp_s \Delta y^2} \\[2mm] + \dfrac{K_e A_e}{\rho_e Cp_e \Delta x^2} + \dfrac{A_w}{Pe\Delta x^2} \end{bmatrix}$$

$$b = -sum_i A$$

With the same procedure species, vorticty, energy equations can be solved.

The procedure of solution can be described as:

1. First: the initial temperature, type of fuel (all properties of fuel automatically will be known in the program) and equivalence ratio are input to program.
2. Analysis of production gases by moles.
3. Calculation the heat of combustion.
4. Calculation the adiabatic flame temperature.
5. Calculation the burning velocity.
6. Starting in two – dimension solution.
7. Evaluated all properties.
8. Calculation the MError and comparison with true value.

Experimental analysis:

The study of the flame speed needs accurate measuring system, because the time to occur it is very small and it does not exceed thousand part from second, In this study, In this study, an optical technique to measuring the flame speed are used. This method is allocating the position of the flame front by using photocell.

The combustion tube (the combustion chamber) shown in figure (3) made of copper at length (225cm), inside diameter (10cm) and thickness (6mm)

and external spark through the inlet main valve(1)..

The photocell sensor is a (B-P-X-65) type has a circle shape and (5 mm) diameter. It to be distinguished by the generate voltage from this sensor is line proportional with the strong falling light on it, and it to be characterized by the high response too.
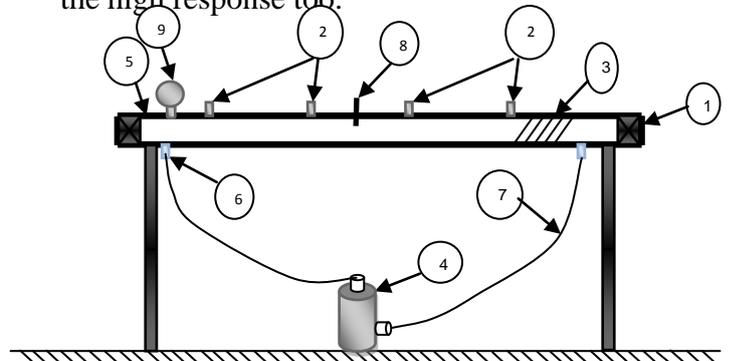
Figure (3) Sketch the combustion tube and attached

No:.	Name		
1	Main valve	5	Body of combustion tube
2	Sensor (photocell)	6	Secondary valve
3	External heater	7	Tube
4	Compressor	8	Thermocouple sensor
		9	Vacuum pressure gauge

The experimental proceeding of an experiment and the manner used in measuring the flame speed can be described as follows adaptation and magnification by the attachment electronic circle.

1- Preparation of the fuel - air mixture
2- Measuring the initial temperature of mixture
3- Flame speed measurement
4- Flame temperature measurement, which measured by photocell technique.
5- Calculation burning velocity

Results and Discussion:

The results can be described as of computational results for fluid flow, thermodynamic and mass species for two dimension laminar premixed combustion. On other hand, laminar flame speed and flame temperature were experimentally studied. Since the computations of this work include

study the effect of equivalence ratio, initial temperature and differences type of gases fuel on combustion.

Figure (4) shows the mole fractions of burned mixture variation with equivalence ratio. One can see the maximum value of mole fraction of water vapor occur at stiocheometric such as carbon dioxide. This behavior can be explained as:

In lean side, since one mole of fuel burned, then the number moles of air are changed (decreased) with equivalence ratio. The number moles of carbon dioxide and water vapor productions are constant depend on number of atoms carbon and hydrogen in fuel form. For example, when burned one mole of Methane, the number moles production of CO_2 and H_2O equal 1 and 4 respectively, with change of equivalence ratio. In other hand, with increase equivalence ratio decreasing number moles of access Oxygen. Since the mole fractions define in equation (5), therefore the mole fraction of CO_2 and H_2O increase.

In the rich side, some of carbon is formed as CO_2 and the other CO because of Oxygen limited. When CO is found, the water – gas shift doing, therefore Hydrogen is formed. Then the mole fractions of H_2O and CO_2 decreasing.

The Oxygen mole fraction is changed linearly with the equivalence ratio in lean side in both figures (4). It disappears in rich side because of all Oxygen conserves to water. One can see it only in lean side as shown in the figures. The Oxygen mole fraction is decreased with increasing equivalence ratio because the linear relationship between them. This behavior is similar for Propane and Butane except the gradient $(\alpha+\beta/4)$ of equation (4) depend on type fuel. The mole fraction of CO and H_2 depends on the water gas shift, therefore these mole fractions to undergo it. The water gas shift described by equation (8) and (9).

From equation (9) can be concluded it depend on adiabatic flame temperature, therefore the water gas shift is decreased with increase equivalence ratio. Return to equation (9) the number moles of CO and H_2O increase as a result of decrease water gas shift.

Figure (5) shows the species mole fractions profile for Methane. These figures refer to the change of the mole fractions along axial distance. Fuel and Oxygen mole fractions in interning region are unchanged (no reaction) depend on equivalence ratio and type of fuel. The mole fraction of fuel was reduced according to the equation (3) specified fuel consumption. Oxygen mole fraction reduced by the same value that reducing of fuel multiply by stiocheometry ratio, which depends on equivalence ratio. In lean combustion figures (5,a) shows mole fraction of Oxygen through the exit. But in stoicheomtry and rich combustion (figures 5,b) at exit region Oxygen molar is eliminated. Hydrogen and carbon monoxide mole fractions can be seen in figures (5,c) in rich combustion only.

Figure (6) shows temperature distribution with change equivalence ratio for Methane at T_i =300K. The flame temperature increases with increasing equivalence ratio in the lean side until stoichiometry, then the flame temperature decreasing with increasing equivalence ratio in rich side. This behavior can be explained as:

In lean side, when increasing equivalence ratio, the mole fraction of CO_2 and H_2O increase too. This means although the product moles number of CO_2 and H_2O remain constant but the moles number of O_2 and N_2 in products decreases with increasing equivalence ratio according to the equation (4). The heat of combustion define as heat of reaction, its remains constant .Amount of this heat is per-heated the excess moles of Oxygen and Nitrogen. The

other part of this heat is used to rise the flame temperature.

In rich side of combustion, with increasing of equivalence ratio, the mole fraction of CO and H_2 are increased but the mole fraction of CO_2 and H_2O decreased depended on water – gas shift reaction. Also CO_2 and H_2O are reacted in endothermic reaction (to absorb heat reaction), then the heat release by the combustion in this case is decreased, therefore the flame temperature is decreased.

The axial velocity is evaluated depending on stream function-vorticity formulation. To explain the behaviors of axial velocity can be depend on continuity equation (combustion dose not generate mass). Therefore, when gradient found is in density, must be gradient in velocity. Figure (7) explained the maximum axial velocity occur at stoicheometry combustion. The results show the change in contour of axial velocity with equivalence ratio in case burned Methane. This figure explained the maximum axial velocity occur at stoicheometry combustion.

Figure (8) shows variation of flame thickness with equivalence ratio at three deferent initial temperatures of fresh gases. Flame thickness was calculated from equation (19). Flame thickness, at lean side, decreasing with increasing equivalence ratio until it reaches minimum value at stoicheometry condition. Then, flame thickness increases with increasing equivalence ratio in the rich side. This behavior cause by the increase the flame temperature which increased the reaction rate which results increased the burning velocity then the flame thickness decreasing. Then can be concluded the flame thickness is inversed proportional with burning velocity.

Measuring of the flame speed was performed by delimitation the flame front by using photocell along the tube for two points. The distance between them is known (90cm). The

flame front is moved as a result for burning velocity and velocity of unburned gases which moving in the flame front. The flame speed depends on chemical rate which depend on flame temperature, where the flame speed increasing when the chemical rate increasing. But the velocity of unburned gases creates from thermal expansion, as a result of heat transfer from flame front by conduction, convection and radiation to that unburned gases. Heat transfer in combustion wave depends on deference between the flame temperature and fresh gases temperature. Therefore the velocity of unburned gases depend on the flame temperature where it's increasing when the flame temperature increasing. Then can be concluded the flame speed increasing when the flame temperature increasing too. In figure (9), can be shown the flame speed change with equivalence ratio in polynomial relationship where the flame speed increasing with increasing equivalence ratio until maximum value near the stoichiometry value ($\Phi=1$). The flame speed increased with increasing equivalence ratio due to increase the flame temperature.

In lean side ($\Phi<1$), with increasing equivalence ratio the flame temperature increases due to increase the mass of fuel, therefore the flame speed increases as a result of increasing burning velocity and velocity of unburned gases. In rich side, the increases in equivalence ratio indicates an increasing the mass of fuel in mixture (or decreased the mass of oxygen in mixture). This leads to the formation of CO instead of CO_2, which leads to a decrease in the release heat from combustion then a decrease in the flame temperature and burning velocity. Also, in rich side, products (CO, CO_2, H_2O) at high temperature, these products will react in endothermic chemical reaction

(endothermic reaction) which decreased the flame temperature too.

Also, in the rich side, the mixture has high thermal properties especially thermal conductivity which contributes to raise release heat of combustion to cause a decreased of the flame temperature. That cause the maximum value of flame speed occur in stoicheometric condition ($\Phi=1$), but figure (9) shows the maximum value of flame speed occur in the rich side. This phenomena was created a result of dissociation which occur at higher temperature.

Fig (10) presents the comparison between theoretical and experimental results for flame temperature with different fuel and equivalence ratio Φ. The results show acceptable agreement for both propane and butane fuel, while a good agreement for methane fuel at rich mixture due to high temperature releases at rich mixture.

Fig (11) presents the comparison between theoretical and experimental results for burning velocity with different fuel and equivalence ratio Φ. The results show acceptable agreement for both propane and butane fuel , while a good agreement for methane fuel at rich mixture due to high temperature releases at rich mixture.

Conclusions:

1. The maximum adiabatic flame temperature occurred at stoicheometry condition due to high temperature releases at rich mixture for the theoretical calculations for all fuel types .For example maximum temperature reach to 2176 K for methane

2. The maximum flame temperature occurring at rich side in experimentally study as in Fig.(10).

3. Also, the maximum flame speed and burning velocity occurred at the same point which maximum flame temperature occurring on it at same equivalence ratio in numeric and experiment study.

4. Good agreement for methane fuel at rich mixture due to high temperature releases at rich mixture.

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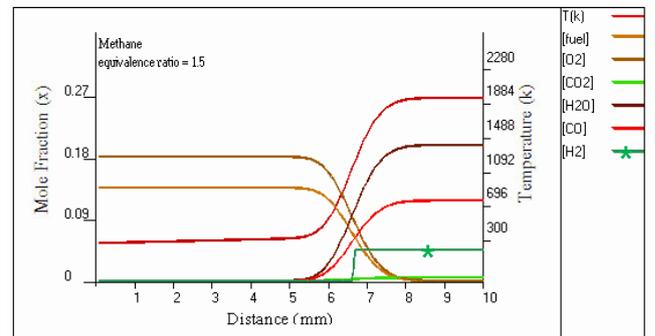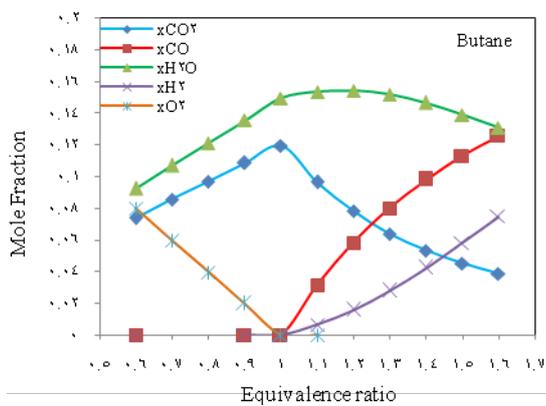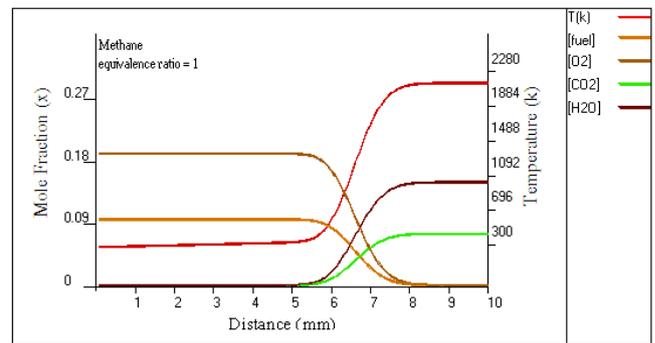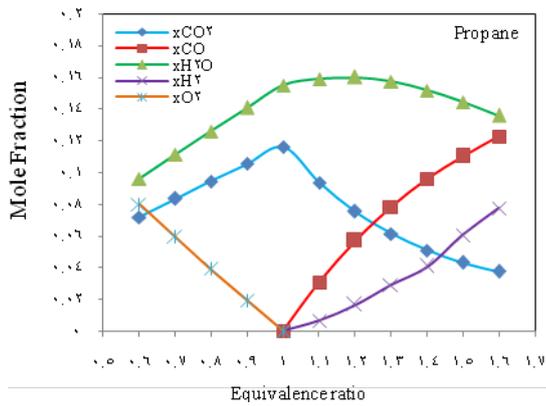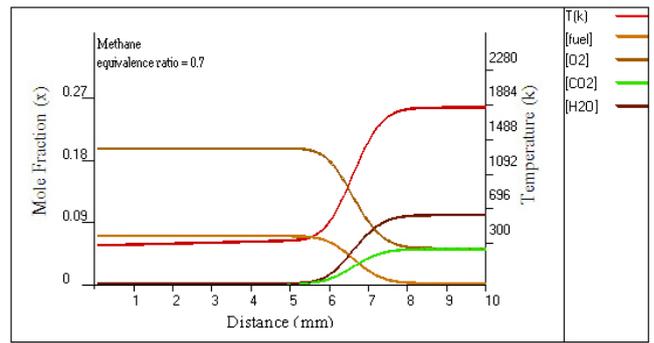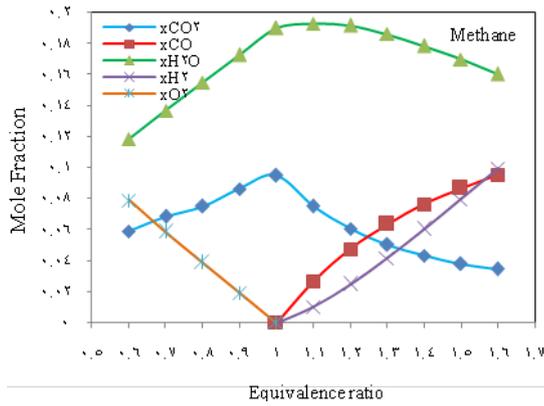
Figure (4) The mole fractions variation with equivalence ratio

Figure (5) Species moles fraction and temperature profile of Methane at Ti=300K
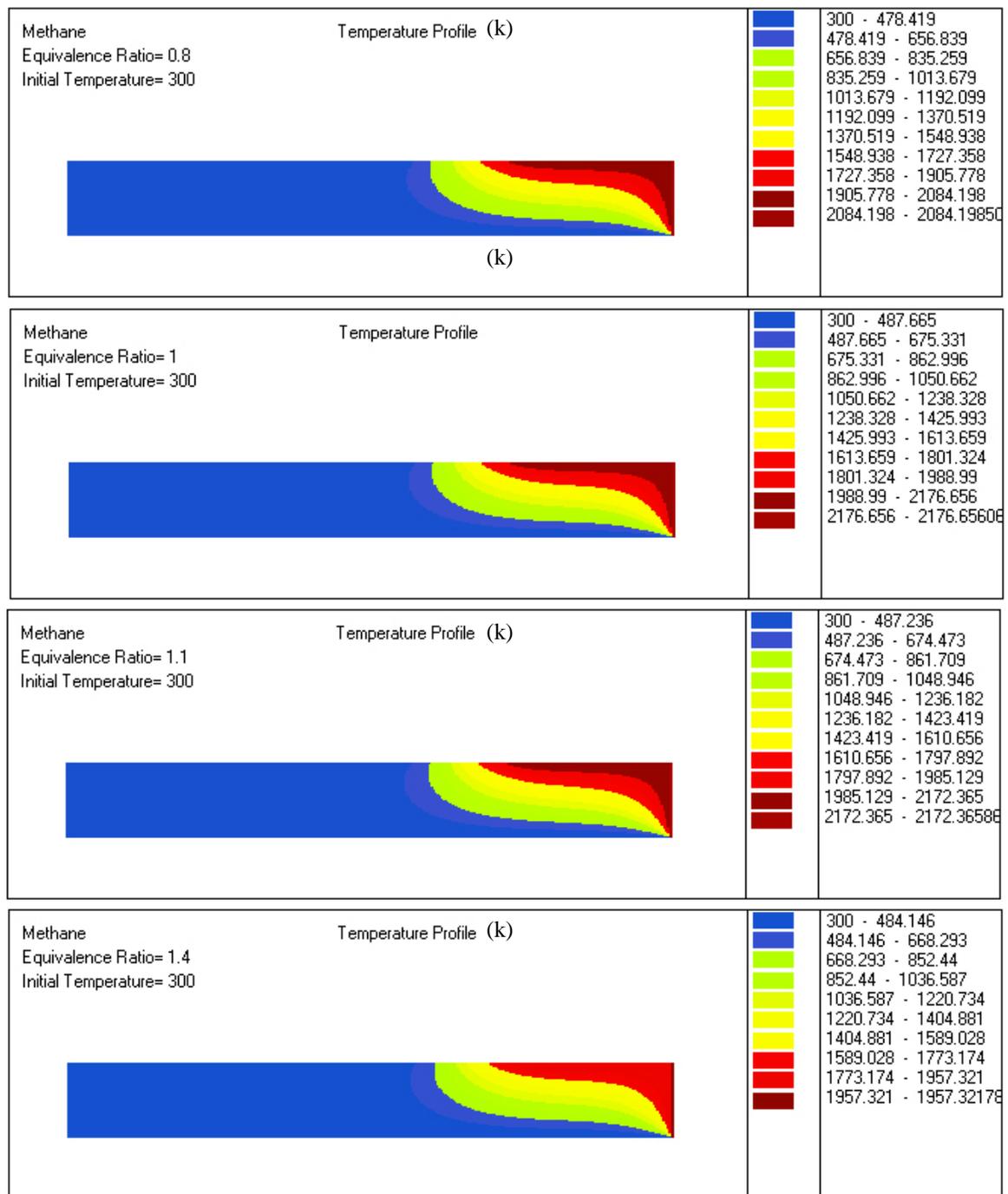
Figure (6) Temperature distribution with equivalence ratio for Methane at T_i =300K
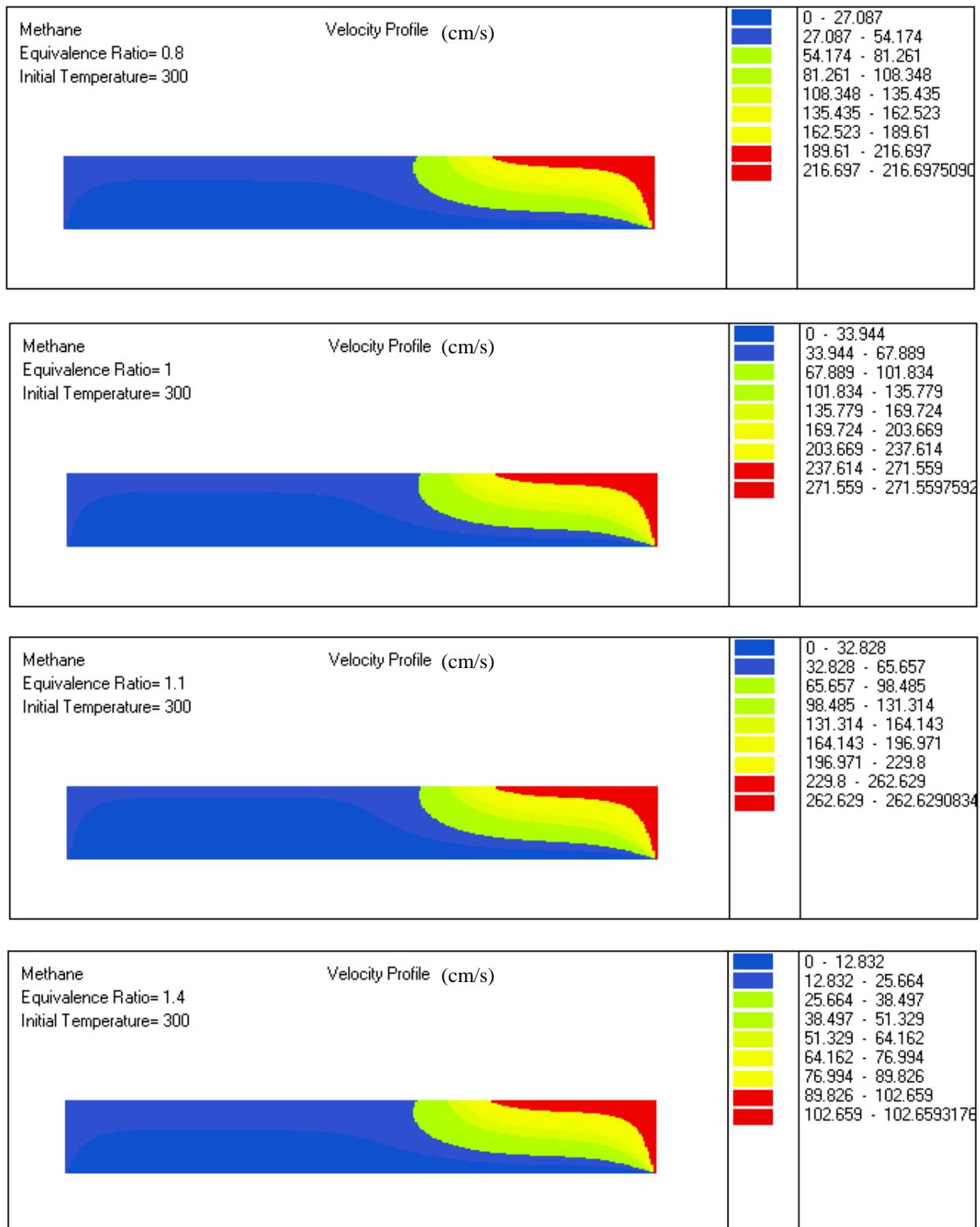
Figure (7) Axial velocity distribution of Methane with change of equivalence ratio at
$T_i = 300K$
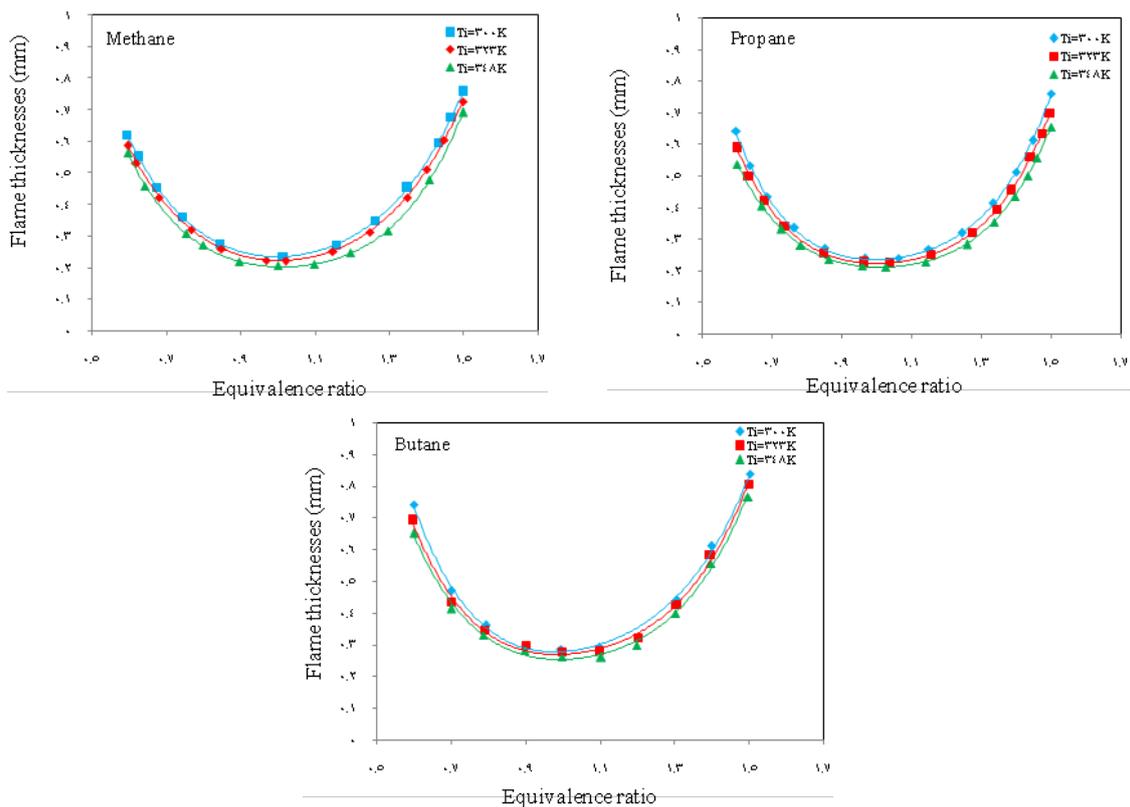
Figure (8) Variation flame thickness with equivalence ratio at three deference initial temperature of fresh gases for: Methane ,(b)Propane (c)Butane and (d)LPG
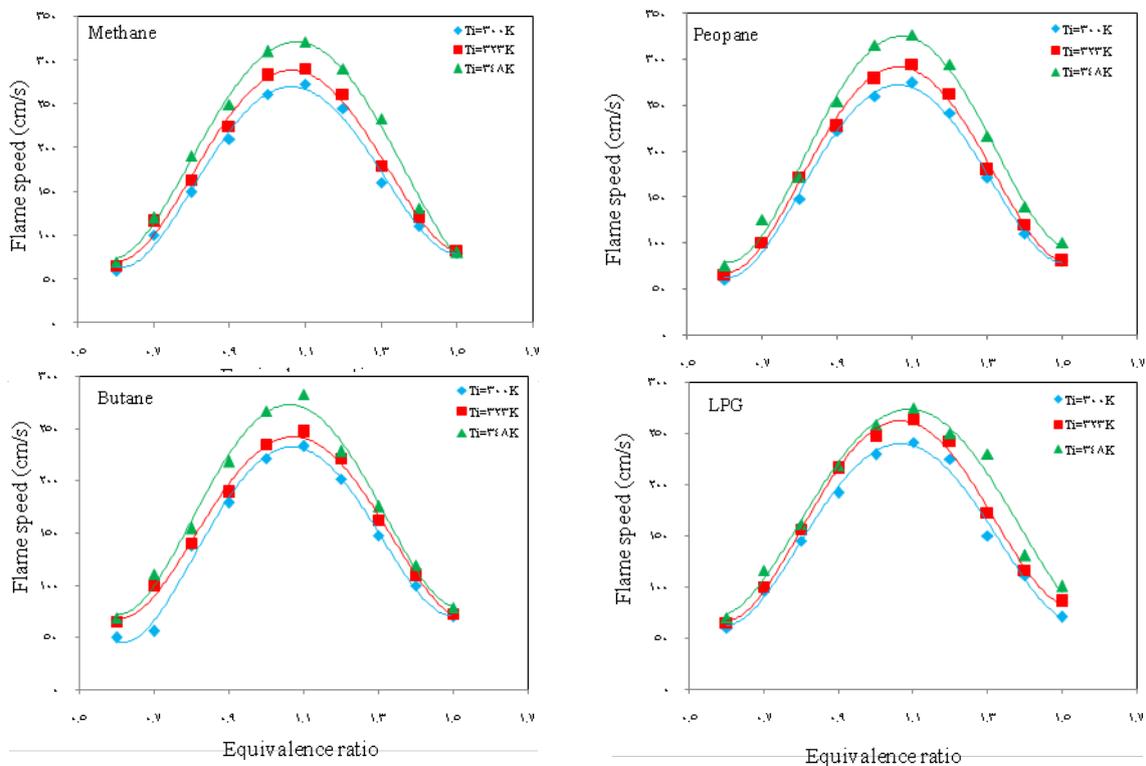
Figure (9) Variation of flame speed with equivalence ratio with change initial temperature for (a) Methane, (b) Propane,(c) Butane, (d) LPG
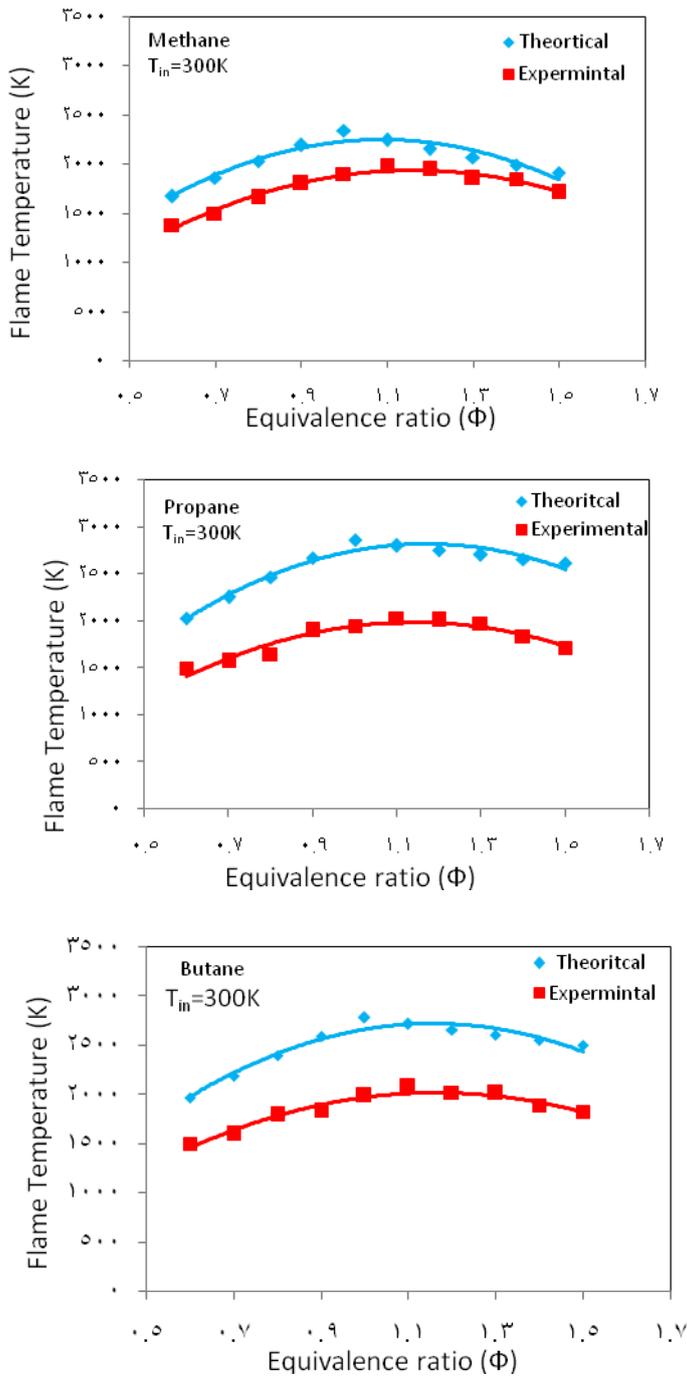
Figure (10) Comparison between theoretical and experimental results for flame Temperature
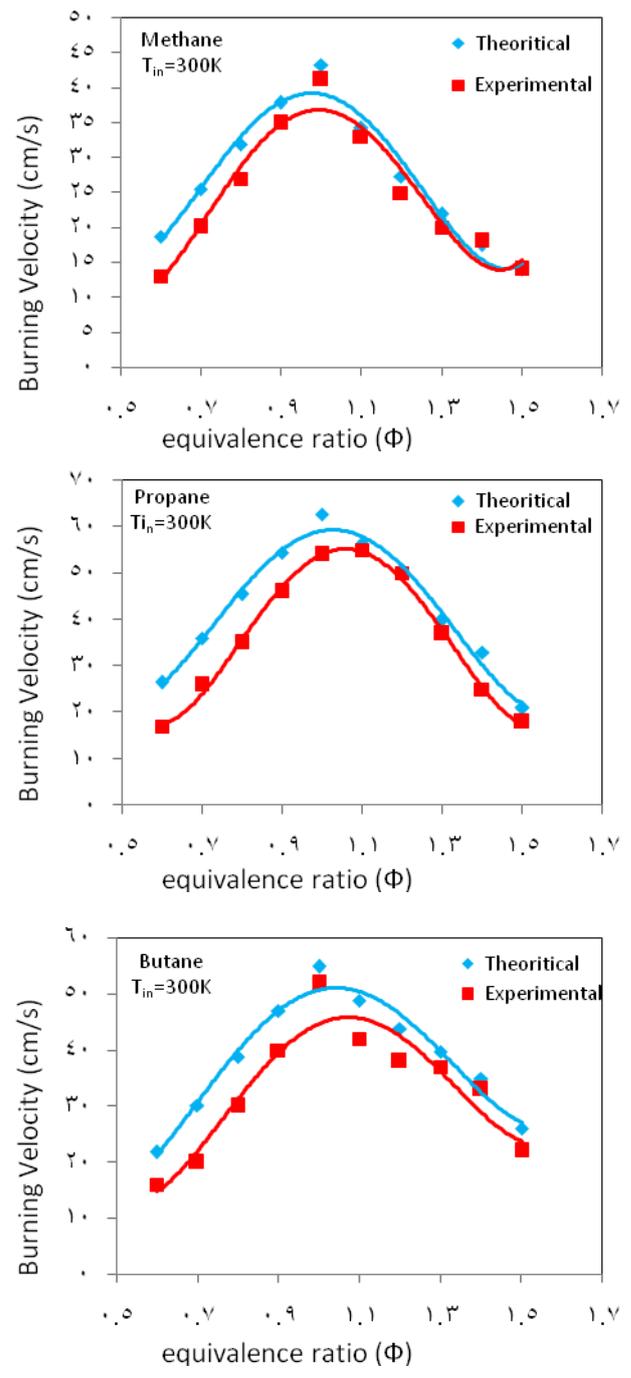
Figure (11) Comparison between theoretical and experimental results for burning velocity