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Numerical study on the minimum ignition energy of a methane-air mixture

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Abstract

In this paper, a semi-detailed mechanism is used to calculate the minimum ignition energy (MIE) of methane-air mixtures, with all calculations conducted using ANSYS Fluent software. Ignition behaviors between success and failure were analyzed, and the effect of isothermal and adiabatic wall condition, equivalence ratio and electrode wall temperature on the MIE were also numerically investigated. From time histories of temperature and species profiles it is easy to determine when a successful ignition occurs. There are big differences between the MIE predictions using the adiabatic wall and isothermal wall conditions. For different equivalence ratio results, we consider our numerical results to be considerably more accurate than previous studies.

Keywords: Spark ignition, Methane-air mixture, Minimum ignition energy, Numerical analysis

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21 **1. Introduction**

22 Despite the rapid development of new alternative sources of energy, fossil fuels still play an
23 important role in energy conversion [1]. Methane (CH₄), which is the main component of
24 natural gas, is a relatively clean fossil fuel compared to large hydrocarbons present in
25 conventional gasoline and diesel fuels. Since it can be made by the anaerobic digestion of
26 biomass, methane is regarded as a renewable energy source and is widely used in industry and
27 in our daily lives [2].

28 Spark ignition is a complex phenomenon involving the coupling of physical and chemical
29 processes. After years of research, the mechanism of ignition is not completely understood. It is
30 known that the formation of the ignition kernel is affected by many factors such as mixture
31 composition, the initial temperature and heat loss to the electrodes. Hill showed that these
32 factors not only change the ignition kernel, but also have an impact on the propagation behavior
33 of a flame [3]. Ballal et al. [4-7] measured the MIE of gaseous and heavy fuels at different
34 equivalence ratios, quenching distances, pressures and turbulence intensities. Cui et al. [8]
35 measured the MIE of methane/air mixtures at elevated temperatures and pressures, and
36 determined that the MIE has a high-order linear relationship with $1/p^2$, but has an
37 approximately linear relationship with $1/T$. However, there are not enough accurate results for
38 many factors including electrode size, electrode gap distance, and spark duration, among
39 others, and sometimes it is not easy to study these experimentally. Therefore, numerical
40 simulation is a good way to capture the transient processes that occur during ignition kernel
41 growth.

42 Several studies have reported on the growth process of the flame kernel. According to Kravchik
43 et al. [9], kernel growth is divided into two phases; in the first short phase, the pressure wave
44 and the expansion of the plasma kernel control the transformation of mass and energy; in the
45 second phase, diffusion and thermal conduction control mass and energy transfer. Ma et al. [10]
46 established a physical model for the spark ignition of combustible gases, and the MIE of
47 combustible mixtures was predicted by numerical analysis. Sarli et al. [11, 12] performed 3-D
48 large eddy simulations and PIV measurements and found that large-scale vortices play a
49 dominant role in the evolution of the flame structure along the path after ignition. Sarli et al.
50 also found that unsteady flame propagation is very sensitive to the characteristics of the initial
51 flame kernel, in particular its size and turbulence. Turquand d'Auzay et al. [13] simulated the
52 forced ignition of a methane-air mixture at different turbulence intensities using a three-
53 dimensional direct numerical simulation method. Han et al. [14] used simulations to study the
54 effect of different parameters on the MIE including electrode size, electrode gap distance,
55 equivalence ratio, and spark duration. For many numerical studies, natural convection, heat
56 transfer by radiation and ionic species are ignored as they have little effect on kernel growth
57 and flame propagation [15, 16]. Even though many CFD studies have been conducted, many
58 details still need to be calculated, and more accurate computations performed. There are few
59 simulation papers on the effect of heat loss and electrode wall temperature on the MIE of spark
60 ignition. Also, the relative error between simulation and experimental results needs to be
61 further reduced with improved methods and mechanisms.

62 In this study a methane/air mixture was used to computationally investigate the spark ignition
63 process using a skeletal mechanism reduced from a new chemical kinetic mechanism [17, 18].

64 The ignition behavior of methane, between success and failure is firstly compared. The effects
 65 of several factors on MIE are conducted; (i) isothermal wall condition, (ii) adiabatic wall
 66 condition, (iii) equivalence ratio and (iv) electrode wall temperature.

67 2. Computational domain and numerical methods

68 2.1 Governing equations

69 In the spark ignition process, chemical and physical processes are very complex at certain
 70 times. To characterize the entire process, the following equations are used.

71 (1) The continuity equation:

$$72 \frac{\partial \rho}{\partial t} + \frac{\partial}{\partial x}(\rho v_x) + \frac{\partial}{\partial r}(\rho v_r) + \frac{\rho v_r}{r} = 0 \quad (1)$$

73 The mass conservation equation is shown in Eq. (1). In Eq. (1), ρ is the density, x is the axial
 74 coordinate, r is the radial coordinate, v_x is the axial velocity, and v_r is the radial velocity.

75 (2) Momentum conservation equations

$$76 \frac{\partial}{\partial t}(\rho v_x) + \frac{1}{r} \frac{\partial}{\partial x}(r \rho v_x v_x) + \frac{1}{r} \frac{\partial}{\partial r}(r \rho v_r v_x) = -\frac{\partial p}{\partial x} + \frac{1}{r} \frac{\partial}{\partial x} \left[r \mu \left(2 \frac{\partial v_x}{\partial x} - \frac{2}{3} (\nabla \cdot \vec{v}) \right) \right] \\ + \frac{1}{r} \frac{\partial}{\partial r} \left[r \mu \left(\frac{\partial v_x}{\partial r} + \frac{\partial v_r}{\partial x} \right) \right] \quad (2)$$

$$77 \frac{\partial}{\partial t}(\rho v_r) + \frac{1}{r} \frac{\partial}{\partial x}(r \rho v_x v_r) + \frac{1}{r} \frac{\partial}{\partial r}(r \rho v_r v_r) = -\frac{\partial p}{\partial r} + \frac{1}{r} \frac{\partial}{\partial x} \left[r \mu \left(\frac{\partial v_r}{\partial r} + \frac{\partial v_x}{\partial r} \right) \right] \\ + \frac{1}{r} \frac{\partial}{\partial r} \left[r \mu \left(2 \frac{\partial v_r}{\partial r} - \frac{2}{3} (\nabla \cdot \vec{v}) \right) \right] - 2 \mu \frac{v_r}{r^2} \frac{2 \mu}{3 r} (\nabla \cdot \vec{v}) + \rho \frac{v_x^2}{r} \quad (3)$$

78 Where

79
$$\nabla \cdot \vec{v} = \frac{\partial v_x}{\partial x} + \frac{\partial v_r}{\partial r} + \frac{v_r}{r}$$
 (4)

80 Eqs. (2) and (3) are the axial and radial momentum conservation equations, where μ is the
81 molecular viscosity and p is the pressure.

82 (3) Energy conservation equation

83
$$\frac{\partial}{\partial t}(\rho E) + \nabla \cdot (\vec{v}(\rho E + p)) = -\nabla \cdot \left(\sum_j h_j \vec{J}_j \right) + S_h$$
 (5)

84 (4) Species mass conservation equation

85
$$\frac{\partial}{\partial t}(\rho Y_i) + \nabla \cdot (\rho \vec{v} Y_i) = -\nabla \cdot \vec{J}_i + R_i + S_i$$
 (6)

86
$$\vec{J}_i = -\rho D_{i,m} \nabla Y_i - D_{T,i} \frac{\nabla T}{T}$$
 (7)

87 Eqs. (5) and (6) are the energy and species mass conservation equations, respectively, where E
88 is the internal energy, S_h is the energy source from heating, Y_i is the local mass fraction of each
89 species i , where R_i is the net rate of production of species i by chemical reaction and S_i is the
90 rate of creation by addition from the dispersed phase plus any user-defined sources. \vec{J}_i is the
91 diffusion flux of species i . In Eq. (7), $D_{i,m}$ is the mass diffusion coefficient for species i in the
92 mixture, and $D_{T,i}$ is the thermal (Soret) diffusion coefficient.

93 2.2 Assumptions for simplicity of computations

94 To accelerate simulation work and reduce the number of calculation, some assumptions were
95 made according to the following four assumptions from the work of Han et al. [14].

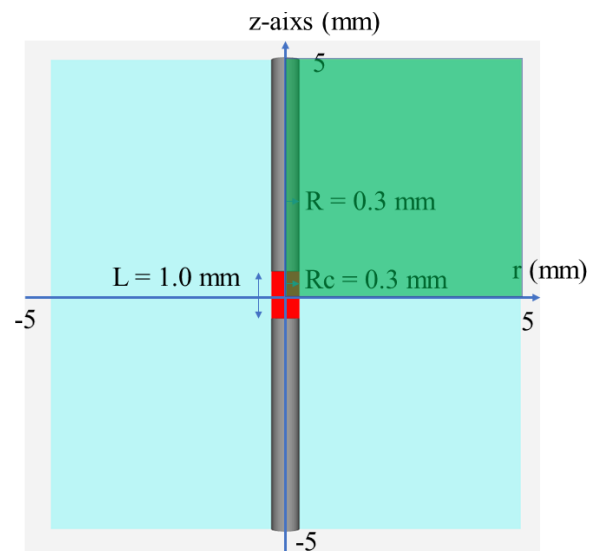
- 96 (1) The mixture gases are considered as ideal gases
- 97 (2) The Soret, Dufour, and pressure effects are negligible
- 98 (3) The influence of the magnetic and electrical fields can be ignored
- 99 (4) Heat transfer by radiation, ionic and electronic species is neglected

100 2.3 Initial and boundary conditions

101 In this paper, there is no flow before ignition, and the spark in the quiescent combustible
102 mixture is supposed to be cylindrical. If a 3-D simulation were conducted, the differences in the
103 results in the circumferential direction would be negligible. For the 2-D axisymmetric
104 simulations, accurate axial and radial results can be obtained using a reduced number of meshes
105 and with a much shorter computational time. According to Han et al. the initial shape of the
106 flame in gas mixtures is axisymmetric, and the 2-D axisymmetric model in ANSYS Fluent
107 software was used [19]. As shown in Fig. 1, the computational region is 5×5 mm in green, the
108 electrode radius $R = 0.3$ mm, the spark channel radius $R_C = R = 0.3$ mm, and the electrode gap
109 distance $L = 1.0$ mm. According to Sloane et al. [20], when R_C is smaller than 0.3 mm, the MIE
110 is almost constant, and so 0.3 mm was chosen for R_C .

111 The initial gas was a CH_4/air mixture, at an initial temperature of 300 K and at a pressure of 1
112 atm. For most cases the initial gases were stoichiometric mixtures, and the compositions were
113 changed when equivalence ratios were studied. There were no surface reactions on the
114 electrode boundary, the no-slip condition was used, and all gradients of the species on the
115 surface were set to zero. The temperature of the electrode was changed according to different
116 computational conditions.

117 In this paper, a certain constant ignition energy density q was used to simulate the constant
118 value of the electric power during the ignition process. The ignition energy, Q , was calculated
119 using $Q = q \times \tau_i \times V$, where q is ignition energy density, τ_i is the spark duration and V is the
120 volume of the spark channel. $50 \mu\text{s}$ was chosen for τ_i for most cases, since an optimum value of
121 the MIE can be obtained for spark durations in the range $3 - 100 \mu\text{s}$ [21]. The z-axis was
122 considered to axisymmetric, the r-axis was considered as the line of symmetry and other two
123 boundaries were treated as outlets.



124

125 Fig. 1. Scheme of computational domain.

126 2.4 Numerical method

127 A semi-detailed mechanism containing 39 species and 231 reactions was used[18]. The specific
128 heat, thermal conductivity, viscosity and mass diffusivity were calculated using the
129 CHEMKIN-CFD Solver [19], and mixture averaged mass diffusivities for each species were
130 used in the mass diffusivity calculation.

131 In this paper, the ANSYS Fluent software [19] was used, and finite-rate model was conducted

132 by CHEMKIN-CFD solver in Fluent. For the finite-rate kinetic model, general reaction-rate
 133 expressions are used to calculate the chemical source terms. The net source of a chemical
 134 species i due to reaction is computed as the sum of the reaction sources over the N_R reactions
 135 that the species participate in:

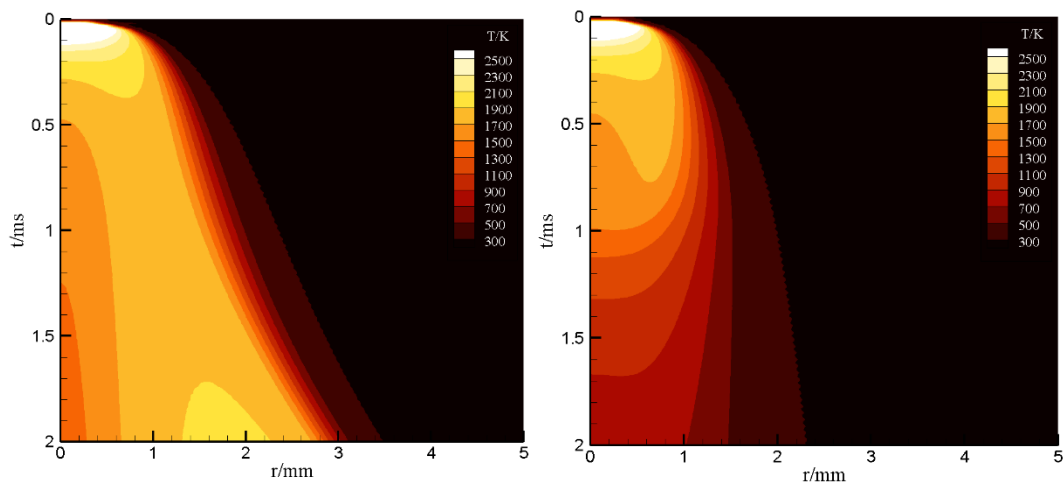
$$136 \quad R_i = M_{w,i} \sum_{r=1}^{N_R} \hat{R}_{i,r} \quad (8)$$

137 where $M_{w,i}$ is the molecular weight of species i and $\hat{R}_{i,r}$ is the molar rate of reaction/destruction of
 138 species i in reaction r . For more details about finite-rate modeling can be found in the work by Sloane
 139 and Rooney [10].

140 The grid size was set at 0.05 mm and the time step to 1 μ s according to Han et al. [14, 22]. The
 141 coupled method was used to adjust the velocity and pressure fields, and a second-order upwind
 142 scheme was employed for convective terms [23].

143 3. Results and discussion

144 3.1 Ignition behavior between success and failure



145

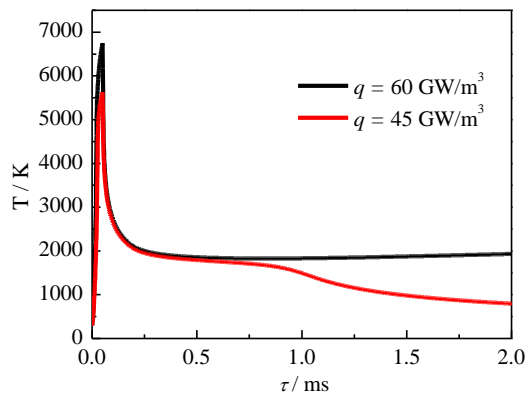
146
147

(a) $q = 60 \text{ GW/m}^3$ (b) $q = 45 \text{ GW/m}^3$

Fig. 2. Time history of temperature at $z = 0 \text{ mm}$ section.

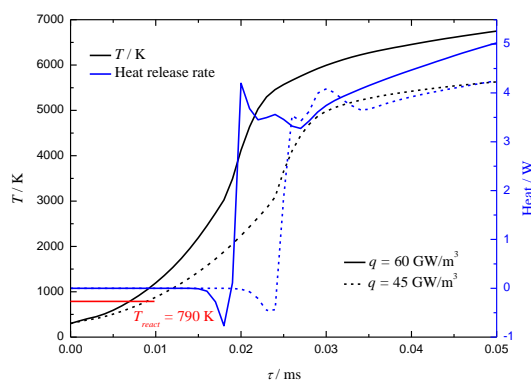
148 When computing the MIE, the definition of ignition success is very significant. Successful
149 ignition is defined as the time when a flame kernel is formed at a certain place and it can
150 propagate steadily [14]. Based on the temperature time history at the $z = 0 \text{ mm}$ section, the state
151 of ignition can be judged. On the basis of Fig. 2(a), the high temperature area can expand to the
152 ambient cold region with time using $q = 60 \text{ GW/m}^3$ ($Q = 0.848 \text{ mJ}$); Meanwhile Fig. 2(b),
153 using $q = 45 \text{ GW/m}^3$ ($Q = 0.636 \text{ mJ}$), shows that it cannot extend and it even shrinks with time
154 (τ), and moreover the maximum temperature decreases. In this way, the MIE can be simulated
155 in different cases.

156 Maximum temperature histories are shown in Fig. 3. As we can see, the temperature increases
157 quickly with the start of ignition, and after $\tau = 50 \mu\text{s}$, it decreases sharply to 2000 K. At $\tau = 2$
158 ms, the maximum temperature using $q = 45 \text{ GW/m}^3$ is lower than 1000 K, which means the
159 flame has quenched; while the maximum temperature using $q = 60 \text{ GW/m}^3$ remains steady at
160 2000 K.



161
162

Fig. 3. Maximum temperature history for $q = 60 \text{ GW/m}^3$ and $q = 45 \text{ GW/m}^3$.



163

164

Fig. 4. Maximum temperature and heat release ratio histories during the first 50 μs .

165

Figure 4 shows the maximum temperature and heat release ratio histories during first 50 μs .

166

The red line shows the temperature ($T = 790 \text{ K}$) at which reaction begins. In the initial phase,

167

reaction begins with the reaction $\text{CH}_4 + \text{O}_2 = \dot{\text{C}}\text{H}_3 + \text{H}\dot{\text{O}}_2$, after which, $\dot{\text{C}}\text{H}_3$ radicals react with

168

O_2 to produce $\text{CH}_3\dot{\text{O}}_2$ radicals $\dot{\text{C}}\text{H}_3 + \text{O}_2 = \text{CH}_3\dot{\text{O}}_2$. At low temperatures the reaction $\text{CH}_3\dot{\text{O}}_2 +$

169

$\dot{\text{C}}\text{H}_3 = \text{CH}_3\dot{\text{O}} + \text{CH}_3\dot{\text{O}}$ is important in promoting reactivity. The temperature gradually increases

170

due to a constant energy input, before $\tau = 18 \mu\text{s}$ for $q = 60 \text{ GW/m}^3$ ($\tau = 24 \mu\text{s}$ for $q = 45 \text{ GW/m}^3$)

171

most reactions are endothermic, and the mixture burns at $\tau = 20 \mu\text{s}$ ($\tau = 26 \mu\text{s}$ for $q = 45 \text{ GW/m}^3$)

172

with a sharp increase of heat release ratio. After a few microseconds, a flame front is formed,

173

and propagates steadily towards the unburned mixture region.

174

Figure 5 shows the profiles of some major species in radial direction at different times. At $\tau =$

175

40 μs the temperature is high due to heat release. At high temperatures ($\geq 2000 \text{ K}$), it is easy to

176

produce radical species such as $\ddot{\text{O}}$, $\dot{\text{O}}\text{H}$ and $\dot{\text{C}}\text{H}_3$ in the spark channel. Since there is a difference

177

in energy density, the mass fraction of radical species in the solid lines are higher than the short

178

dashed lines at $r = 0 - 0.5 \text{ mm}$. Figure 5(a) shows that the spark kernel of the high energy

179

density ignition can propagate quickly. At $\tau = 1500 \mu\text{s}$ the maximum temperature of $q = 60$

180

GW/m^3 is approximately 2000 K, and the flame front propagates to $r = 2.2 \text{ mm}$.

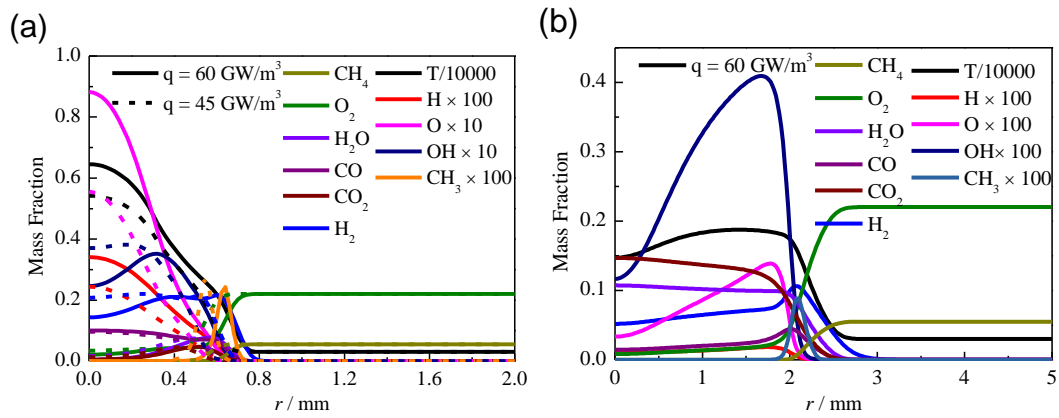
181

From the temperature time history and the profiles of the major species, there are significant

182

differences between successful and failed ignition events, so the state of ignition can be clearly

183 defined.



184

185 Fig. 5. Profiles of some major species at (a) $\tau = 40 \mu\text{s}$ and (b) $\tau = 1500 \mu\text{s}$.

186 3.2 Ignition behavior between isothermal and adiabatic wall condition

187 In order to study heat loss during the ignition process, two simulations, Case 1 and Case 2 were
 188 performed, with details of the conditions given in Table 1.

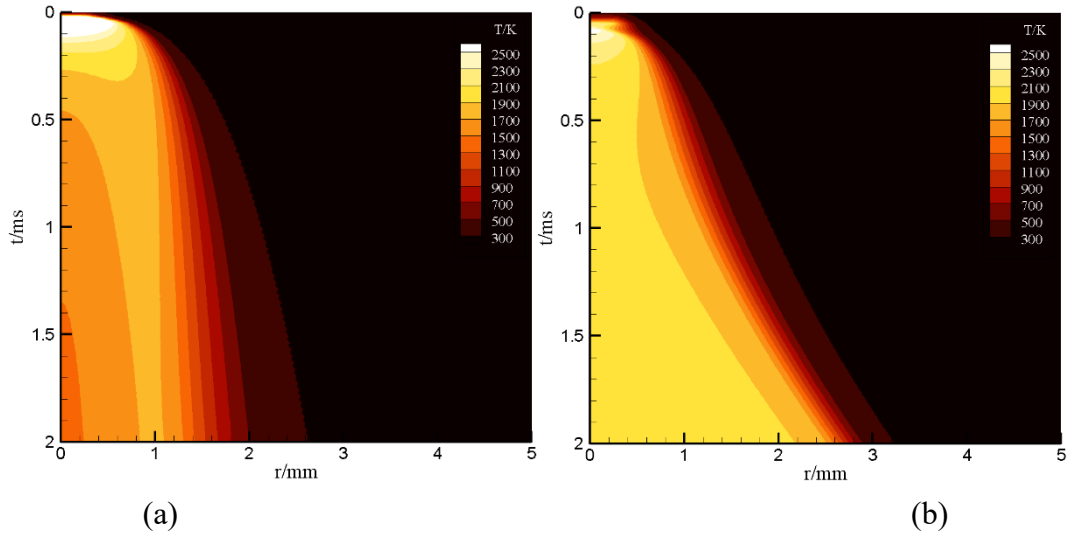
189 Table 1. Case conditions

Name	Wall condition	MIE / mJ
Case 1	Isothermal, $T = 300 \text{ K}$	0.707
Case 2	Adiabatic	0.254

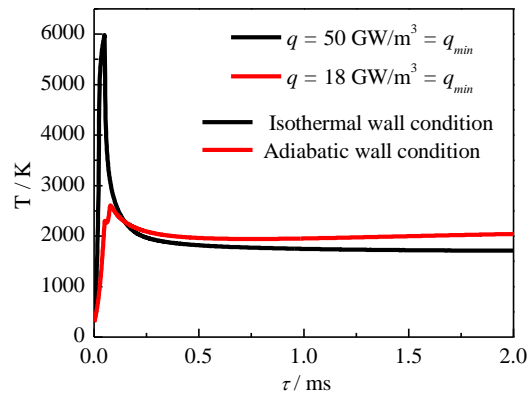
190 Figure 6 shows the time histories for these two cases. The flame kernel for Case 1 spreads
 191 quickly initially, and at $\tau = 50 \mu\text{s}$, the high temperature region expanded at approximately $r = 1$
 192 mm; while the boundary position of the adiabatic flame kernel was approximately $r = 0.4 \text{ mm}$.

193 Adiabatic ignition can propagate stably after $\tau = 50 \mu\text{s}$ and the temperature of burned region is
 194 higher than for isothermal ignition. Figure 7 shows the maximum temperature history and it
 195 reflects the different character of these two wall-condition ignitions. Case 1 attains a higher

196 temperature than Case 2 initially, but the temperature in Case 1 falls sharply after the energy
 197 supply is stopped.



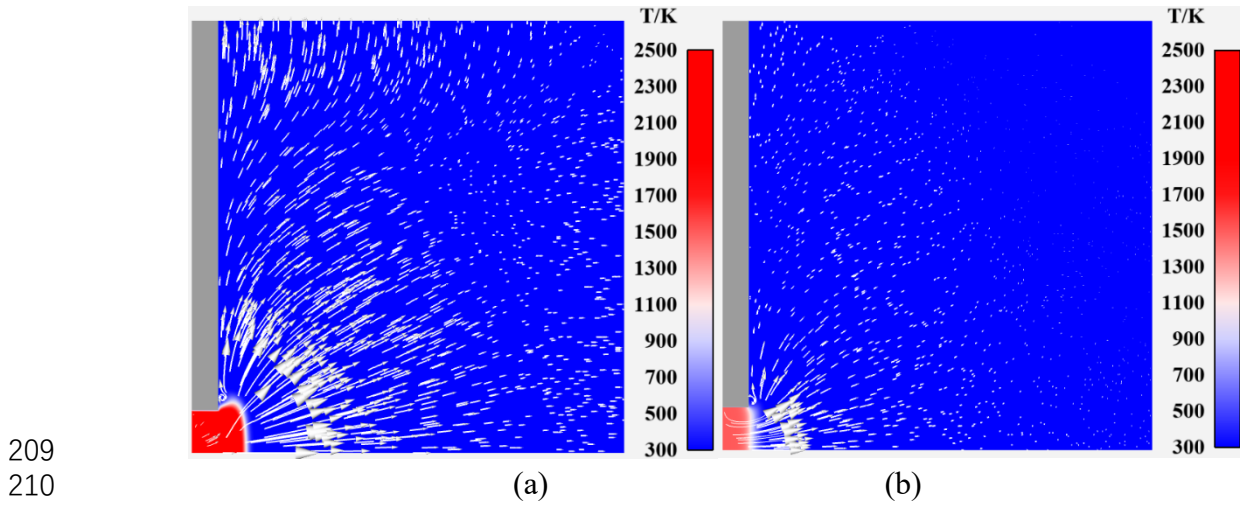
198
 199 (a) (b)
 200 Fig. 6. Temperature/time histor at the $z = 0$ mm section. (a) Case 1: isothermal wall condition
 201 and (b) Case 2: adiabatic wall condition.



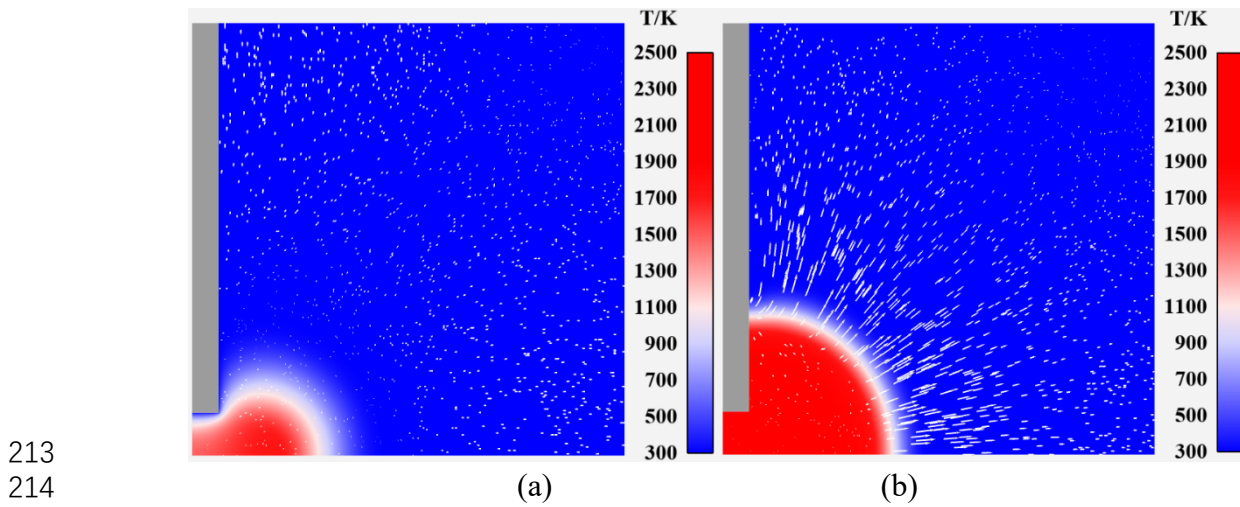
202
 203 Fig. 7. Maximum temperature histories for two cases.

204 Temperature and velocity contours are shown in Fig. 8 and Fig. 9. Initially the energy density
 205 of Case 1 is $q_{min} = 50 \text{ GW/m}^3$ which is higher than for Case 2 where $q_{min} = 18 \text{ GW/m}^3$. The
 206 high temperature region of Case 1 extends out at high velocity. Due to heat loss from the
 207 electrode surface, the fall in temperature in Case 1 was big, and the flame propagation velocity

208 becomes slow. For Case 2, after successful ignition, the flame burned steadily.

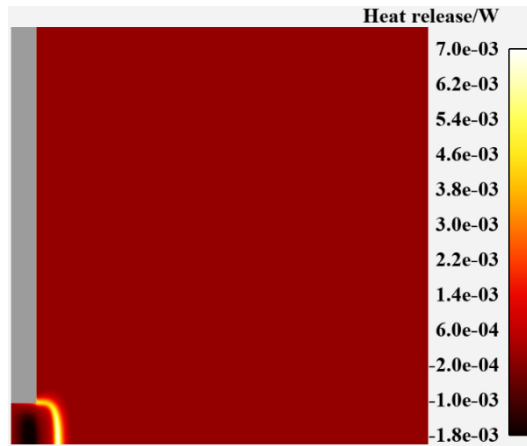


211 Fig. 8. Temperature and velocity contour at $\tau = 40 \mu\text{s}$. (a) Case 1: isothermal wall condition and
212 (b) Case 2: adiabatic wall condition.



215 Fig. 9. Temperature and velocity contour at $\tau = 1500 \mu\text{s}$. (a) Case 1: isothermal wall
216 condition and (b) Case 2: adiabatic wall condition.

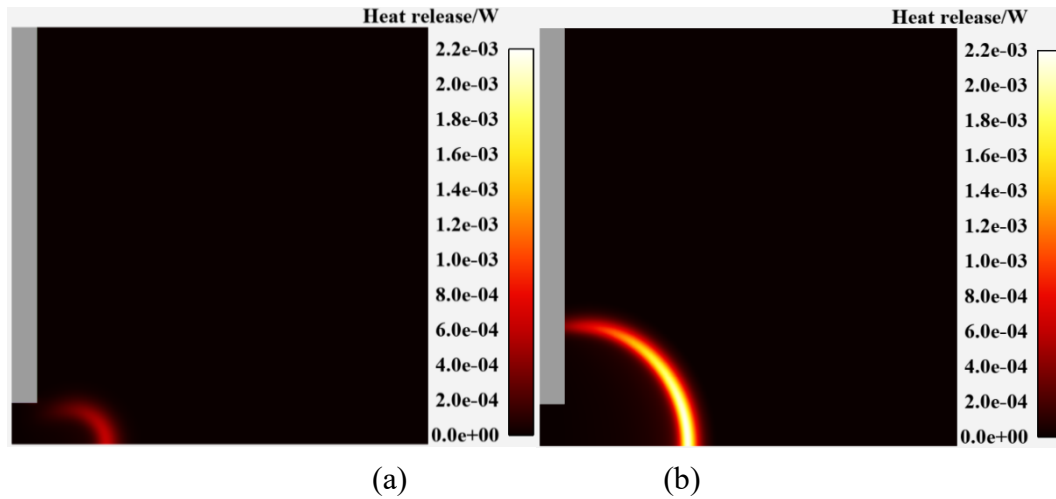
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218

Fig. 10. Heat release contour at $\tau = 40 \mu\text{s}$ in Case 1: isothermal wall condition.

219



220

221

Fig. 11. Heat release contour at $\tau = 1500 \mu\text{s}$. (a) Case 1: isothermal wall condition and (b) Case

222

2: adiabatic wall condition.

223

Figures 10 and Figure 11 show heat release contours at $\tau = 40 \mu\text{s}$ and $\tau = 1500 \mu\text{s}$. At $\tau = 40 \mu\text{s}$,

224

for Case 1 there is a strong heat release, while there is no heat release for Case 2 since the

225

temperature between the electrodes is low. At the propagation phase the heat release become

226

weak for Case 1, while the heat release of Case 2 is very strong.

227

Comparing the MIE for Cases 1 and 2, more energy is consumed for the isothermal wall

228

condition, and the MIE of Case 2 is 35.93% that of Case 1, Fig. 12. Moreover, 64.07% of the

229 energy consumed is used to heat both the mixture and the electrode wall.

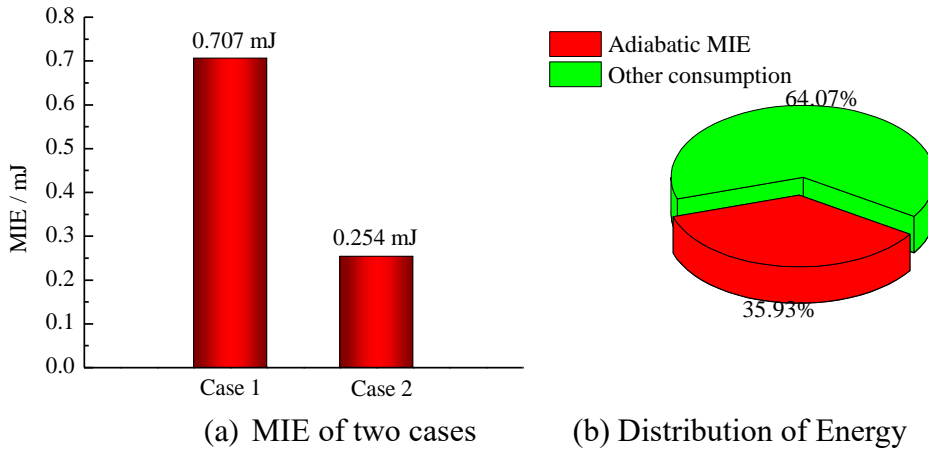
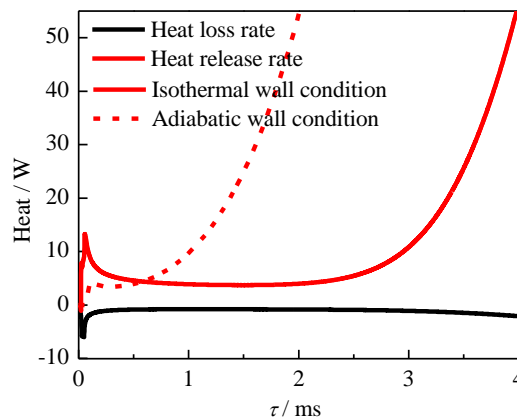


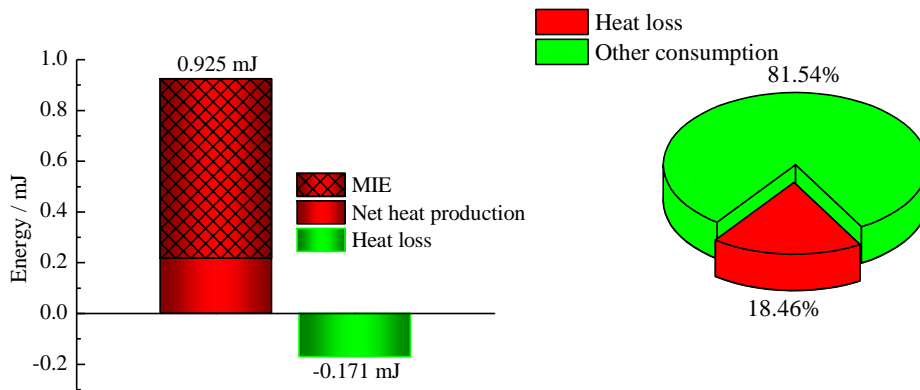
Fig. 12. MIE comparison of two cases.

233 The heat loss and the release rate of both cases are shown in Fig. 13. For the isothermal wall
234 condition the heat release rate and the heat loss rate remain low in the first 3 ms, while it
235 increases rapidly after 1ms for the adiabatic wall condition. The total energy, including heat
236 production and energy input, of the isothermal wall condition ignition is 0.925 mJ over 50 μ s,
237 and heat loss is 0.171 mJ. 18.46% of the energy is lost to the wall, with the remaining 81.54%
238 of the energy used to heat and ignite the mixture. Therefore, when designing a spark igniter, an
239 electrode wall composed of a lower thermal conductivity material is more effective in
240 decreasing the MIE and improving ignition performance.



242

Fig. 13. Heat loss rate and heat release rate of two cases.



243

244

(a) Energy production and loss in 50 μs (b) Energy distribution in 50 μs

245

Fig. 14. Energy distribution of isothermal wall condition ignition.

246 3.3 Effect of the equivalence ratio on the MIE

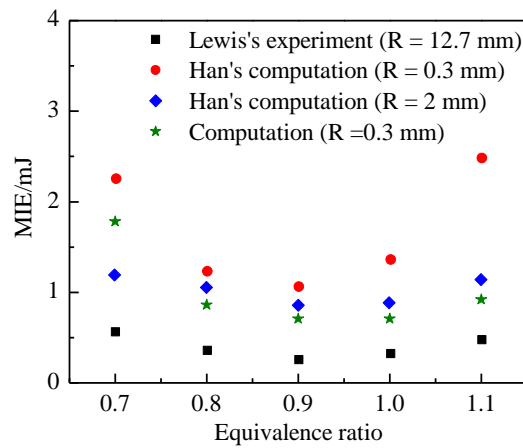
247 Simulations of the effect of equivalence ratio on the MIE were conducted, and these results
248 were compared with Han's computations and Lewis' experimental work [14, 24]. For $\phi = 0.7$,
249 $100 \mu\text{s}$ was chosen for τ_i to ensure a converged solution, so some numerical problems with
250 high energy density addition for $\tau_i = 50 \mu\text{s}$ could be avoided. The simulation results in Fig. 15
251 show a very good trend compared to the experiment data; the minimum MIE of different
252 equivalence ratios was close to 1.0 in the fuel-lean mixture region. Compared to Han's
253 computation, the simulations in this paper are more accurate. In Han's paper, a simplified
254 transport model was used. The effective thermal conductivity, λ , was calculated from C_p and T
255 using a polynomial fit, and the effective diffusivity of species i was calculated using the Lewis
256 number. In this paper, for material properties such as specific heat, viscosity, thermal
257 conductivity, mass diffusivity and thermal diffusion, they are computed using the CHEMKIN-
258 CFD solver and the mechanism files. Mixture averaged mass diffusivities for each species were
259 used in the calculations. Results are improved using this methodology.

260 Figure 16 shows heat loss ratios of different equivalence ratios. In Fig. 16,

$$261 \quad \text{Ratio 1} = \frac{q_{\text{Heat loss}}}{MIE + q_{\text{Heat release}}} \quad (7)$$

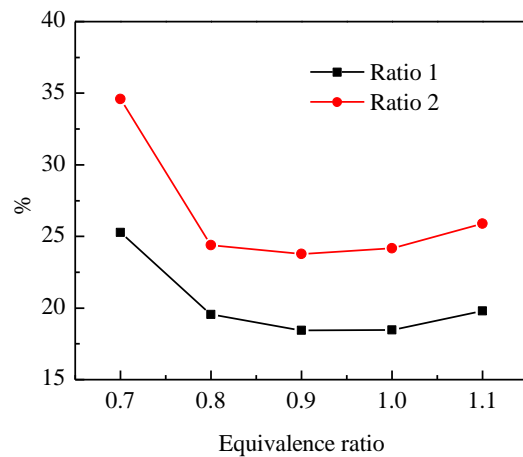
$$262 \quad \text{Ratio 2} = \frac{q_{\text{Heat loss}}}{MIE} \quad (8)$$

263 For both Ratio 1 and Ratio 2, minimum values are obtained at $\varphi = 0.9$, with average values of
 264 Ratio 1 being about 20% and average values of Ratio 2 are 5% higher than for Ratio 1. Figure
 265 16 indicated that a lower MIE will lose less energy to the electrode wall.



266

267 Fig. 15. Effect of the equivalence ratio on the MIE.



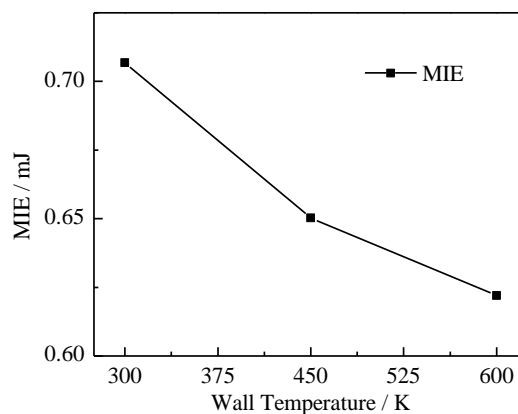
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269

Fig. 16. Heat loss ratios during ignition period.

270 3.4 Effect of the electrode wall temperature on the MIE

271 Figure 16 shows that 25% of the MIE is consumed by heat loss to the wall, so wall temperature
272 effects on the MIE were studied. The effect of wall temperature on the MIE is shown in Fig. 17,
273 and it is observed that as the temperature increases, the MIE decreases. Figure 18 shows the
274 heat loss rate to the wall at wall temperatures of $T = 450$ K and $T = 600$ K are higher than for
275 the mixture at temperature of 300 K, because initially heat transfers from the wall to the
276 mixture. When the mixture temperature becomes higher than the wall, energy is transferred
277 back to the wall. At higher wall temperatures, a lower MIE is needed. Moreover, on heating the
278 electrode wall, ignition becomes easier. Due to the contribution of the electrode wall
279 temperature, ignition under extreme conditions, such as high altitude relight and plateau
280 ignition, can be achieved. So it is much better for a designer to add a heating system on the
281 electrode wall of the spark igniter to decrease the MIE of the mixture and enhance ignition
282 performance.



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Fig. 17. Effect of the wall temperature on the MIE.

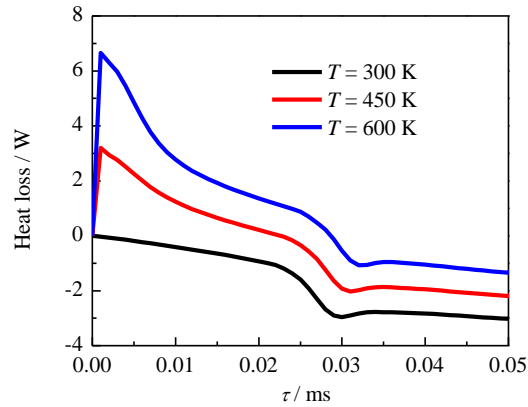


Fig. 18. Heat loss to the wall at different wall temperatures.

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287 4. Conclusions

288 Ignition behaviors between success and failure were numerically investigated for isothermal
 289 and adiabatic wall conditions, at different equivalence ratios and electrode wall temperatures to
 290 determine the effect on the MIE for methane/air mixtures. From our computational results, the
 291 following conclusions were reached.

292 (1) Successful ignition can be determined from the temperature/time history at the $z = 0$ mm
 293 section. Maximum temperature histories and species profiles were determined with
 294 significant differences being observed between successful ignitions and failures.

295 (2) For the adiabatic wall condition, ignition costs much less energy than for the isothermal
 296 wall condition. A sharp change in the temperature history was observed for the isothermal
 297 wall condition ignition while in the adiabatic condition it was very flat. For the
 298 stoichiometric methane/air mixture at 1 atm, 18.46% of the energy was taken up by heat
 299 loss to the wall, and the remaining 81.54% was used to heat and ignite mixture. Thus, an
 300 electrode wall with a lower thermal conductivity material is very helpful in decreasing the
 301 MIE required and improving the ignition performance of a spark igniter.

302 (3) At different equivalence ratios, the minimum MIE is close to 0.9, and numerical results
303 with the CHEMKIN-CFD solver and the latest skeletal mechanism are in good agreement
304 with the experiment data and they are better than previous simulation works. A lower MIE
305 will lose less energy to the electrode wall.

306 (4) For higher wall temperatures, a lower MIE is needed. And after heating the electrode, it is
307 much easier to ignite a mixture. When a heating system is added to the electrode wall of the
308 spark igniter, the MIE of the mixture is smaller and the ignition performance is enhanced.

309 The MIE of a methane-air mixture was studied using CFD in this paper. Some methods such as
310 choosing a lower thermal conductivity material and adding a heating system to the electrode
311 wall are proposed to improve ignition performance. However, there are still some factors that
312 affect the MIE such as the ways the energy is discharged, the turbulence intensity and initial
313 conditions of temperature and pressure that should be studied in flow and real engine
314 conditions.

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