

Experiments and Simulations of a Lean-Boost Spark Ignition Engine for Thermal Efficiency Improvement

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ABSTRACT

Primary work is to investigate premixed laminar flame propagation in a constant volume chamber of iso-octane/air combustion. Experimental and numerical results are investigated by comparing flame front displacements under lean to rich conditions. As the laminar flame depends on equivalence ratio, temperature, and pressure conditions, it is a main property for chemical reaction mechanism validation. Firstly, one-dimensional laminar flame burning velocities are predicted in order to validate a reduced chemical reaction mechanism. A set of laminar burning velocities with pressure, temperature, and mixture equivalence ratio dependences are combined into a 3D-CFD calculation to compare the predicted flame front displacements with that of experiments. It is found that the reaction mechanism is well validated under the coupled 1D-3D combustion calculations. Next, lean experiments are operated in a SI engine by boosting intake pressure to maintain high efficiency without output power penalty. The peak indicated thermal efficiency are finally achieved under $\lambda=1.3$ with intake manifold absolute pressure 150 kPa in experiment. Data of in-cylinder pressure and rate of heat release from the 3D-CFD simulations combined with the validated chemical reaction mechanism are reproduced. NOx emissions from experiment and simulation are also in good agreements under the lean-boost combustion. Further thermal efficiency improvements of the lean-boost SI engine are investigated numerically by using dilution rate, high induced in-cylinder flow, and high knock resistant fuel. The peak indicated thermal efficiency and load of the SI engine is achieved. In addition, methods to prevent knock for high efficiency spark ignition engine are also discussed.

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INTRODUCTION

While fuel economy and environmental impacts are becoming major concerns, increased pressure and regulations are being put on engine and vehicle industries to reduce fuel consumption and follow stringent emission standards. Then, automotive manufacturers have been trying to improve thermal efficiency of gasoline engines as competitive as that of diesel engines for a better performance and emission. Certain measures are being researched thoroughly by weight reduction for efficient body and chassis, engine transmission, and engine design. Beside these opportunities, improving combustion technologies are thought to be advantageous for fuel consumption and emission reductions. Works of Southwest Research Institute [1] developed a new ignition system that facilitates the extension of lean-limit with a potential use in combination with high diluted engine. The concept, so-called high-efficiency dilute gasoline engine (HEDGE), also uses another ignition technology including microwave assisted spark plug [2] to extend lean stability limit of gasoline engines. Another technology used in SI engines by single and direct injections (DI) of fuel is successfully attempted in term of fuel consumption reduction, but NOx emission is still high due to

local rich zone of the mixture. A multiple injection strategy shows good performance compared to the single one but the SI engines still require NOx after treatment [3].

The present investigations are by introducing lean combustion in spark ignition engines and lowering emission of nitrogen oxide NOx using conventional ignition system. This combustion mode is operated by inducing high excess air into cylinder so that peak combustion temperature is lower compared to stoichiometric condition which also lowers formation of the NOx emission. Furthermore, heat loss from charge mixture to walls is also reduced when the combustion temperature decreases. Simultaneously, operating the SI engines under lean combustion mode has a possibility to increase the thermal efficiency. However, lone operation under the lean combustion has a constraint on engine performance limitation due to output power loss caused by a reduction of injected fuel mass. The authors in this work firstly conducted SI engine experiments from stoichiometric to lean limit mixtures under natural aspirated condition. Saturated thermal efficiency is obtained at excess air ratio $\lambda=1.6$ but engine load drastically drops compared to the stoichiometric combustion. In real engine application, the drop of

engine power is undesirable and inoperable when engine load and speed are transient (i.e. high acceleration at inclined road condition). To provide potential benefit in term of engine power, boosted operation either by supercharger or turbocharger provides additional power output depending on the enthalpy from exhaust stream. Boosting technology is now commercially available with SI engines, and combined with limited exhaust gas recirculation (EGR) [4]. However, a main challenge to combine boosting strategy with lean combustion operation is to reduce knock while maintaining optimum efficiency and performance. The knock limits engine performance and efficiency due to retarded spark ignition timing to reduce high pressure rise rate, otherwise, it is likely to damage engine components at certain cycles of operation. Schematic illustration of knock in unburnt zone before flame arrival in SI engines is shown in Figure 1.

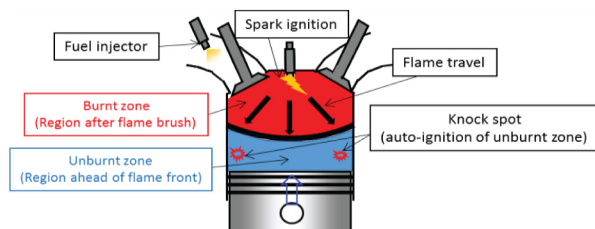


Figure 1. Knock concept in spark ignition engines

To avoid knock, one solution is to use low compression ratio SI engines so peak in-cylinder pressure and temperature can be deducted. This method is not acceptable because engines with low compression ratio reduce theoretical thermodynamic efficiency as seen equation 1:

$$\eta = 1 - \frac{1}{\epsilon^{\gamma-1}} \quad (1)$$

Where η - thermal efficiency

ϵ - Compression ratio

γ - Specific heat ratio

Other possible methods are enriching the fuel-air mixture, thus lowering the combustion temperature and retarding spark ignition timing until the knock does not generate. These methods are clearly questionable and undesirable from the viewpoint of thermal efficiency. Another technique is by introducing cooled dilution to mix with fresh mixture. Cooled dilution or exhaust gas recirculation (EGR) is beneficial for reducing peak temperature and increasing specific heat ratio γ . When the peak temperature is reduced, it is beneficial for NO_x reduction and SI engines can be operated under advanced spark ignition timing without excessive pressure rise rate. The consequence of using EGR results in slow burning rates which likely causes misfire or partial burn [5]. Optimal EGR rate shall be chosen. Several researches have done to optimize the efficiency improvement by 1D simulation. Thermodynamic models [6] presented comprehensive methods for engine design and operating concept leading from conventional to improved methods for the SI combustion engines proving that efficiency increases at two load

conditions. The simulation results show that significant gains in efficiency compared to the conventional SI engines can be achieved by using lean combustion, high EGR, high compression ratio, and high intake pressure.

The other extensive parametric studies by 1D simulation model [7] intend to explore fundamental thermodynamic operation under ranges of lean, boost, and EGR conditions in enabling high efficiency engines. In particular, they studied the tradeoff between engine thermodynamics, application of turbochargers, and downsizing. Significant gain in fuel economy is obtained from stoichiometric, lean-diluted, high boost, and downsized SI engine. Researchers of [8 and 9] predict the thermal efficiency and performance of SI engines accounted for knock avoidance and reduction of wall heat transfer loss between in-cylinder gas and combustion chamber walls by applying heat insulation coating to the combustion chamber wall surface. Reducing the wall heat loss potentially gains the thermal efficiency but increasing knock tendency. Advantageously, the models well predict engine efficiency and performance by minimizing heat loss while optimizing advanced spark ignition timing for knock avoidance.

Previous mentioned literatures demonstrate varieties of thermal efficiency and performance improvements for SI engines by stoichiometric, lean-diluted, and lean-boost conditions. The literature studies are based on one-dimensional numerical method while calculation with detailed/semi-detailed chemical reaction mechanisms provides a deeper and better understanding of combustion, and emission regions as well as knock location which possibly enables engine manufactures to optimize engine design for better performance and emission. Numerical calculations with detailed chemistry are also being extensively used to reduce cost of experiment and improve time consuming in both experiment and design. Researches done by [10, 11] investigate 3D simulation for knock prediction regions. They also measured combustion chamber wall temperature and predicted temperature swings using thermal fluid calculation of engine structures such as engine head, cylinder block, and head gasket. The simulation predicts knock onset location and timing and they suggested methods of cooling and choosing gasket shape to reduce local gas temperature, thus improve brake thermal efficiency.

The main objective of this paper is to explore thermal efficiency improvement by experiment to find optimal boosting for a lean SI engine which considers high thermal efficiency and output power under high load and high speed conditions. Firstly, the lean burnt concept is studied in a constant volume combustion chamber (CVC) for validation of chemical reaction mechanism by combining 1D premixed laminar burning velocity into 3D-CFD combustion simulation. The 3D-CFD simulations with semi-detailed chemistry from lean to rich mixtures are conducted and compared with experimental laminar flame displacements of iso-octane as representative of gasoline fuel. The experiments and simulations are extended to lean-boost SI engine under high speed and full load. Good results of in-cylinder pressure and rate of heat release are reproduced from 3D-CFD calculation combined with the validated chemical reaction mechanism. After validating the simulation model with various experimental data, further investigations for thermal

efficiency improvement are numerically conducted by using simulated cooled EGR, increasing burning velocity by high induced engine swirl, and using high octane-number gasoline.

METHODOLOGY

Constant Volume Combustion Experiments

Figure 2 shows an illustration of a constant volume chamber experimental setup. The stainless steel vessel has diameter of 80 mm and width of 18 mm which is equivalent to 90 cc of chamber volume. Firstly, the chamber wall is heated by electrical heaters to a desired wall temperature. As the wall temperature increases, ambient temperature inside the combustion chamber also reaches a desired point. A thermocouple is mounted on the bottom side of the chamber for the wall temperature measurement, while an extra one is on the top to measure ambient temperature inside the chamber. Fuel is injected manually by micro-syringe from the fuel inlet port shown in the figure. As spark ignition starts at the side of the chamber, pressure rise due to combustion is measured by two pressure transducers (Kistler 6053 Bsp, 6053 Csp). A side quartz window is used for optical measurement of flame displacement by a high speed video camera (NAC MEMRECAM GX-1) at a frame speed of 2000 frame/sec and 1280 × 1024 resolutions.

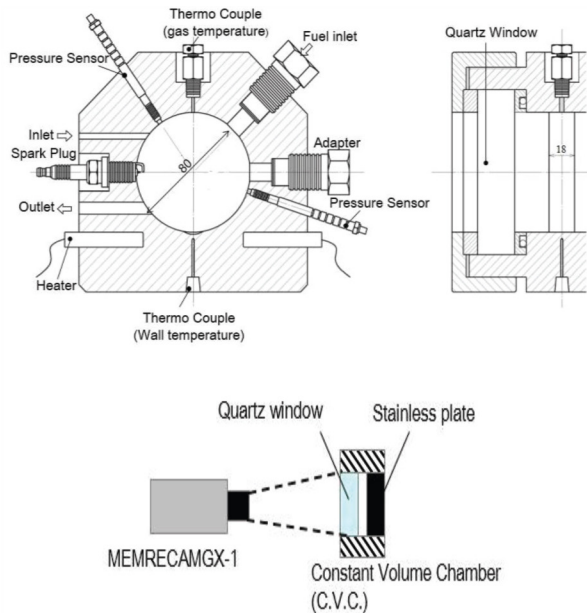


Figure 2. Schematic illustration of CVC experiment (top-left: front view, top-right: side view, bottom: direct exposure high speed video camera)

Lean Burnt Experiments in CVC

In order to validate the chemical reaction mechanism, premixed laminar flame propagation is firstly studied in constant volume combustion in CVC from lean to rich condition at elevated pressure and temperature. Iso-octane is used as a representative of gasoline fuel in the experiments from lean to rich combustion flame propagations. The oxidizer is pure air containing 79% nitrogen and 21% oxygen. The experiments are conducted at elevated pressure and temperature of 0.3 MPa (abs.) and 420 K, respectively. The chamber wall is heated to 445 K. Summarized lean to rich experimental

conditions of pure iso-octane/air mixture are listed in Table 1. Data of experimental pressure from conditions in Table 1 are plotted in Figure 3. Under $\phi = 1.1$, the combustion pressure is the most advance compared to that of other cases. This is due to the peak of adiabatic flame temperature under slightly rich condition [12] which generates the fastest pressure rise and enhances burning rate. The slowest combustion rate is at the leanest mixture $\phi = 0.8$.

Table 1. Lean burnt experiment and simulation conditions

Condition	Experiment and Simulation
Fuel	iC ₈ H ₁₈
Oxidizer %	N ₂ : 79, O ₂ : 21
Ambient pressure MPa	0.3 (abs)
Ambient temperature K	420
Wall temperature K	445
Equivalence ratio ϕ	0.8-1.4 (0.1 step)

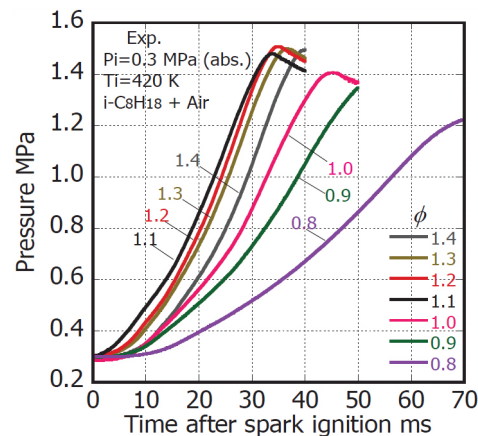


Figure 3. Experimental pressure from lean to rich iso-octane/air mixtures $\phi = 0.8-1.4$, $P_i = 0.3$ MPa, $T_i = 420$ K

SI Engine Experiments

Experimental investigations of lean-boost SI engine are conducted using a four stroke single cylinder engine. The engine is operated under high speed at 4000 rpm and load variation is operated by changing angle of intake air throttle valve. To neglect the effect of ambient humidity and temperature, a precision air conditioner (PAU-A2600S) is used for ambient humidity and temperature at 50 % and 25 °C. Cooling water volumetric flow rate is set at 25 dm³/min and its temperature is set at 90 °C. The experiment is conducted under lean-boost combustion. An electrical supercharger with intercooler system is used for boosting the intake pressure. Pressure data of intake, exhaust, and in-cylinder are measured by three water-cooled pressure transducers mounted directly on intake port, exhaust port, and cylinder head, respectively. The data of intake and exhaust pressure are recorded by one crank angle degree while in-cylinder pressure is measured by 0.5 crank angle degree. A standard emission bench is used for measurement of exhaust gas emissions. All experimental pressure data are recorded to be averaged from 300 consecutive cycles. Engine specification is listed in Table 2 and the experimental bench is illustrated in Figure 4.

Table 2. Spark Ignition engine specifications

Engine type	4-stroke single cylinder
Number of valve	4
Bore x Stroke mm	73 x 82
Compression ratio	13.1
Fuel supply	Port-fuel injection
Experimental fuel type	Primary reference fuel (PRF90)

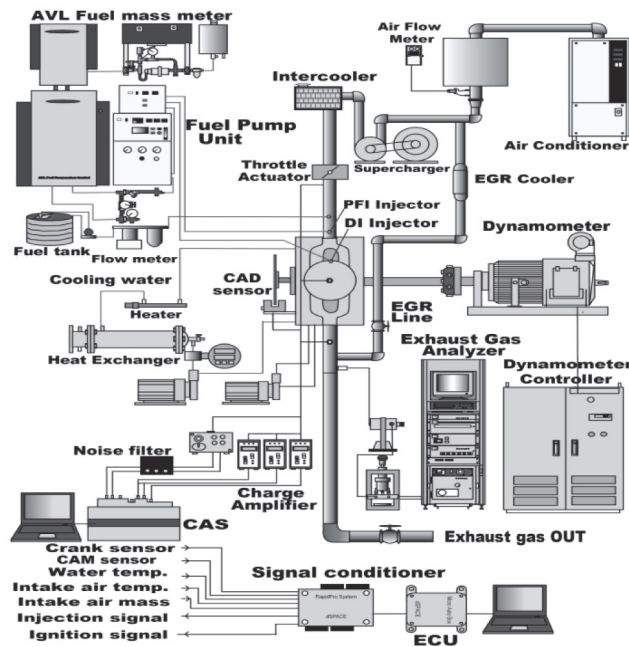


Figure 4. Experimental bench of the gasoline spark ignition engine

Table 3. Lean-boost spark ignition engine experiments

Case	1	2	3	4	5
Engine speed rpm	4000/wide-open throttle				
Engine load IMEP kPa (abs)	891	942	978	962	1112
Excess air ratio λ	1.0	1.2	1.3	1.4	1.3
Fueling rate mg/cyc	18.87	18.6	18.19	18.17	23.94
MAP kPa	95	107	117	123	150
Spark timing bTDC (knock limit)	10	12	14	16	5
EGR %	0.0				
Intake air temp. deg.C	25				
Intake air humidity %	50				
Coolant temp. deg.C	90				
Cooling flow rate [L/min]	25				
Oil temp. deg.C	85				

Firstly, lean burnt experiments are performed from stoichiometric $\lambda=1.0$ to the lean limit $\lambda=1.6$ of the present SI engine under natural aspirated condition. As the mixture goes leaner, spark ignition timing can be advanced so that the thermal efficiency increases. However, the engine load termed as indicated mean effective pressure (IMEP) drastically drops because the fuel is reduced to maintain a desired excess air ratio. The main objective of these investigations is to achieve high thermal efficiency and load of the SI engine. Next, the lean combustion experiments are further investigated under boosted operations to increase engine load while maintaining high thermal efficiency. Intake and exhaust pressures under boosted operation are controlled equally. The experimental points are chosen at high speed 4000 rpm and under full load (wide-open throttle) considering that the thermal efficiency under high speed and full load is higher than the low and partial load conditions because ratios of cooling loss, mechanical loss, and pumping loss under the high load are lower than that of the low load. The operating conditions of the lean-boost experiments are listed in Table 3. Engine load IMEP increases from the cases 1-3 ($\lambda=1.0, 1.2$, and 1.3) and drops when operated at $\lambda=1.4$. Therefore, operating points of case 3 ($\lambda=1.3$, $P_i=117$ kPa [abs]) is chosen for optimizing boosted pressure and maximum boost pressure $P_i=150$ kPa (abs) is optimized with acceptable maximum pressure rise rate (MPRR).

Simulation Methods

Numerical simulations are performed to be validated with experiments for lean combustion in CVC and lean-boost SI engine. The simulations are divided into two parts. The first one is for the lean combustion in the CVC for the validation of chemical reaction mechanism to be utilized in the turbulent combustion simulation in lean-boost SI engine. The 3D numerical calculation for combustion flame propagation in CVC and SI combustion are summarized in Figure 5 using FORTÉ CFD, a commercial RANS-based multidimensional computational fluid dynamics (CFD) with detailed chemistry [13]. Numerical domains are required in order to perform the combustion simulations. For the CVC simulation case, due to simple cylindrical form, the chamber shape can be made by creating block geometry and specification of the combustion chamber in an input, so-called iprep file [14]. This input file contains three spatial coordinates (XYZ) of the chamber geometry. A grid generator, k3prep [14], converts the iprep to a readable geometry, itape17, from the specification for 3D CFD solvers. The experimental conditions of the CVC combustion are at elevated pressure and temperature and beyond that of SI engines, it is necessary to predict laminar burning velocity from a one dimensional (1D) premixed flame solver. A wide range of laminar flame speed based on different temperature, pressure, and equivalence ratio conditions are performed by the 1D laminar burning velocity solvers, CHEMKIN-PRO [15]. A mechanism of gasoline surrogate model by Golovitchev [16] is used as a reaction mechanism shown in Table 4.

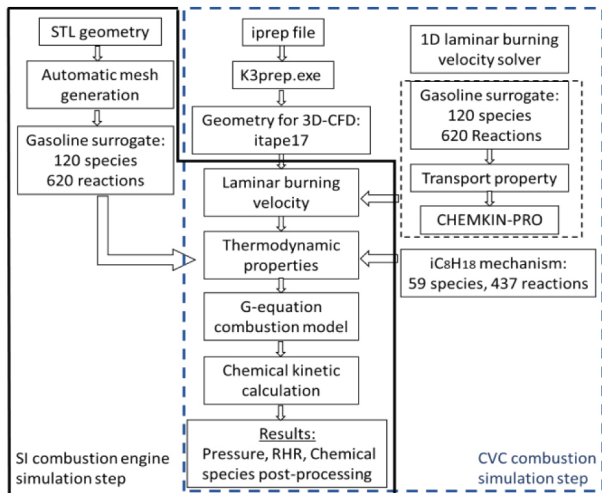


Figure 5. 3D simulation procedures for laminar flame propagation in CVC and SI turbulent combustion engine

Table 4. Summary of reaction schemes of gasoline surrogate [16] and iso-octane [13] for CFD simulation

Reaction mechanism	Gasoline surrogate	iso-octane
Composition	iC ₈ H ₁₈	iC ₈ H ₁₈
	nC ₇ H ₁₆	
	C ₇ H ₁₆	
Number of species	120	59
Number of reactions	620	437

In the 3D CFD calculation, the SI combustion model highly depends on laminar burning velocity. There are options to use laminar flame speed from 1-D solver or two conventional laminar flame speed correlations [17, 18]. However, using the semi-empirical flame correlations requires difficult coefficient tunings based on the dependences of fuel compositions, unburnt temperature, and pressure. Then, a wide range of laminar burning velocity predicted from the 1D solver based on the flame speed dependences is coupled with the 3D solver for the premixed combustion simulation in the CVC. Iso-octane reaction mechanism of 59 species and 437 reactions (see Table 4) is used for chemical reaction calculation in the 3D combustion calculation.

Different simulation steps are conducted for spark ignition engine simulations. A STL (STereoLithography) format geometry is used for automatic mesh generation. Gasoline surrogate mechanism validated in the investigation of the CVC calculations is used as thermodynamic properties for turbulent combustion model with chemical reactions. The laminar burning velocity model is based on the semi-empirical laminar flame correlations [17, 18]. Finally, results of cylinder pressure, rate of heat release, and post-processing of chemical species are obtained to be validated with experimental data.

Numerical Domain of CVC

Figure 6 is the numerical domain of the CVC made by mesh generation from iprep to itape17 as mentioned previously. The chamber diameter is 80 mm and 18 mm of chamber width, corresponding to total chamber volume of 90 cc. Equal mesh size is 2 mm for the calculation.

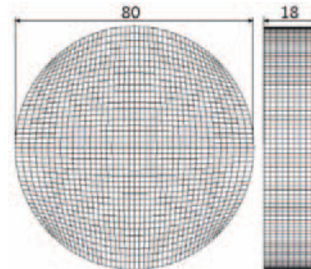


Figure 6. Numerical domain of the constant volume chamber. Diameter=80mm, Chamber width=18mm, Total combustion chamber volume=90cc

Numerical Domain of SI Engine Combustion Chamber

Combustion chamber geometry of the SI engine is shown in Figure 7. A STL format which the piston is at bottom dead center is shown on the left of the figure and the right side is the numerical domain at top dead center shown in grid format. Due to consideration on computational time calculations, the simulations are performed from the intake valve closure (IVC) to the exhaust valve open (EVO) in which mixture at IVC is assumed to be homogeneous. The domain contains 127024 cells when the piston is at the IVC position. The geometry is a pent-roof shape with global mesh size 1.5 mm. A mesh refinement method is applied for all walls at a half of the global mesh size considering the effect flame-wall interaction.

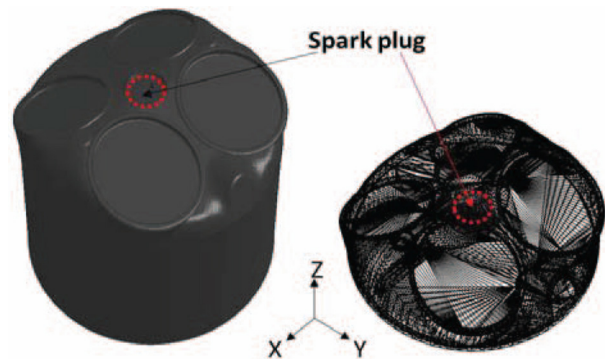


Figure 7. Iso-surface geometry STL geometry at intake valve closure (left) and its numerical domain at top-dead center (right), and location of spark plug

Combustion Model

A brief explanation of the combustion model is described hereafter. A level-set method or G-equation is used for tracking flame propagation in combustion chamber. The model uses a correlation equation (4) of laminar and turbulent flame speed [19] and solves with detailed kinetics. The spark kernel growth at early stage development is very small compared to the average global mesh size (2-5 mm). To track this ignition kernel growth, one may reduce computational mesh size around the ignition region to a very small size typically less than 1

mm (spark plug gap < 1 mm) [5]. However, this is not desirable considering computational efficiency. Then the spark kernel can be tracked by a Discrete Particle Kernel Growth (DPIK) [20, 21] which is expressed as following

$$\frac{dr_k}{dt} = \frac{\rho_u}{\rho_k} (S_t + S_{plasma}) \quad (2)$$

ρ_u, ρ_k are unburnt gas density and density of kernel particle, respectively. S_{plasma} is the kernel growth rate by plasma obtained from energy balance as following.

$$S_{plasma} = \frac{\eta_{eff} \dot{Q}}{4\pi r_k^2 \left[\rho_u (u_k - h_u) + p \frac{\rho_u}{\rho_k} \right]} \quad (3)$$

Spark discharged energy efficiency η_{eff} and energy heat release rate $\dot{Q}$ are chosen at 30 % and 20 J/s, respectively [5].

$$\frac{S_T}{S_L} = 1 + I_p \left\{ \frac{-a_4 b_3^2 l}{2b_1 l_F} + \left[\left(\frac{a_4 b_3^2 l}{2b_1 l_F} \right)^2 + a_4 b_3^2 l \frac{u'}{S_l l_F} \right]^{0.5} \right\} \quad (4)$$

$$I_p = \left[1 - \exp \left(-\frac{t}{C_m \tau} \right) \right]^{0.5} \quad (5)$$

$$r_{k2} \geq C_{m1} l = C_{m1} 0.16 \frac{k^{1.5}}{\varepsilon} \quad (6)$$

Correlation in (4) represents the turbulent/laminar flame velocity used for turbulent combustion model [19] and is used in the SI turbulent combustion simulation. Coefficients a_4, b_3, b_1 are tunable constants in the laminar/turbulent flame correlation. l, u' , and l_F are turbulent integral length scale, turbulent intensity and laminar flame thickness, respectively. I_p in (5) is a progress variable accounting for the effect of surrounding eddies on the flame kernel growth from laminar to fully developed turbulent stage and is taken into account in the SI turbulent combustion. $\tau = k/\varepsilon$ is timescale calculated from turbulent kinetic energy k and its dissipation rate ε . Coefficient C_m in (5) is a tuning constant. Equations (4 and 5) are used in the SI turbulent combustion model while in the premixed laminar flame propagation calculations in the CVC, it is assumed that there is no effect of turbulent eddies on the flame kernel growth so the correlation (4) becomes $S_T = S_L$. A criteria (equation 6) to activate the turbulent combustion model is when the initial flame kernel radius grows larger

than the turbulent integral length scale multiplied by a model constant C_{m1} and this constant is set at 2.0 for all simulations. Based on these equations for the turbulent/laminar correlation together with Reynolds-Averaged Navier-Stokes equations, these provide a complete set of equations for tracking premixed turbulent flame propagation based on level-set method. Details can be found in [19].

RESULTS AND DISCUSSIONS

Constant Volume Combustion

Validation of 1D Laminar Burning Velocity

Figure 8 presents the predicted 1D adiabatic free-propagating laminar burning velocities under reference states ($P_i=0.1$ MPa, $T_i=300$ K, 400 K) are compared with data from previous literatures. The validations are under iso-octane/air mixture from equivalence ratio $\phi=0.6-1.4$ (by 0.1 increment). Under leanest mixture $\phi=0.6$, the burning velocity is the lowest and it burns faster as the mixture goes richer. However, under extremely rich conditions ($\phi>1.1$), the burning velocity drastically drops because oxygen concentration in the mixtures are reduced. It can be seen from Figure 8 that the peak laminar burning velocity is under slightly rich mixture ($\phi=1.1$) because of its peak adiabatic flame temperature [22].

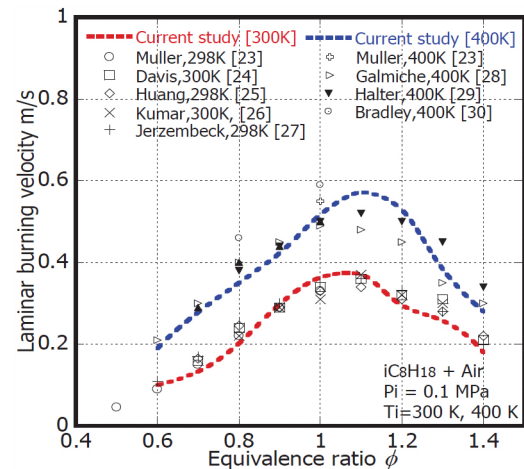


Figure 8. Comparisons of predicted laminar burning velocity with data from previous researches at different mixture equivalence ratio $\phi=0.8-1.4$ under reference states. Initial condition of pressure and temperature of the current studies are $P_i=0.1$ MPa, $T_i=300$ K, and 400 K

There are slight differences between the predicted laminar burning velocities and that of the previous literature studies. The differences in burning velocity are due to slight differences in initial conditions such as temperature and ratio of oxidizers, experimental setup, and methodology of flame burning rate extraction. Under the case of initial temperature $T_i=300$ K, good agreements between the predicted burning velocities and that of the previous literature studies are obtained. Slight differences are expected to be caused by lower initial temperature (298 K) in [23, 25, and 27]. Another case is a comparison under $T_i=400$ K, the burning velocities from [28] are slower than that of the present studies which are likely caused by higher fraction of oxidizer ($N_2/O_2=79.5/20.5$) used in the literature.