Prediction of Spark Ignition Performance in an Industrial Gas Turbine Combustor

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Abstract

Spark ignition performance of an annular combustor has been analyzed using computational modeling approach. Main steps of this approach include: (1) LES of the combustor non-reacting flow field, (2) using time-averaged LES results in a stochastic code in order to identify probable propagation behavior of the flame front using Lagrangian particle tracking, and (3) repeating the computations by an engineering approach and prediction of the combustor lean light-off (LLO) limit. By using this approach, effects of the ignition system location and specifications, fuel type and composition, and operating conditions on the gas turbine ignition performance can be evaluated effectively.

Introduction

Ignition is the process of transferring a flammable mixture from the non-reacting state to the self-sustaining combustion. Ignition is a transient phenomenon in which a complex interaction of chemical and physical processes occurs. Ignition in a flammable environment can be achieved by two general methods: forced ignition and self-ignition. During the forced ignition, a small volume of the mixture is ignited by an external force and the flame spreads across the whole system if the flame propagation conditions exist. In self-ignition or auto-ignition process, there is no local ignition source and the whole mixture ignites simultaneously [1].

Currently, practical gas turbine engine ignition systems include spark ignition (electrical spark and laser-induced spark or LIS) and torch systems (including both flame and plasma torches) [2]. Despite impressive progress in development of laser-induced spark ignition systems [3]-[6], and also flame and plasma torches [7]-[9], electrical spark is the most common type of ignition in gas turbine engines. A generic electrical spark ignition system consists of three main elements including voltage generating unit (exciter), connecting cables (leads), and spark plug(s). Stored energy in exciter reaches plug tip through the lead and energy discharge in the plug causes the formation of a hot plasma core and consequently a self-propagating flame core (kernel) in the reactive mixture. Main feature of this method is the possibility of concentrating energy discharge in a small volume during a very short time. Moreover, frequency, duration, and amount of energy can be easily controlled in electrical spark ignition systems [1].

Different studies have shown that the spark ignition in an annular gas turbine combustor is conducted at three phases. At the first phase, with starting discharge process and through breakdown, arc discharge, and glow discharge stages a flame kernel with sufficient size and temperature forms. At the second phase, this flame core spreads and the continuous spread of the flame from this kernel fills the primary combustion zone of the ignition burner. At the third phase, the flame is propagated from the ignited burner to other burners and the whole combustion space is filled with the flame. At the end of this so-called cross-ignition or light-around phase, the flame becomes stable for all burners in the combustion chamber [1].

Ignition is an intrinsically stochastic phenomenon in a turbulent flow field and consequently its quantification and evaluation needs proper statistical tools. To do so, two statistical variables of minimum ignition energy (MIE) and ignition probability (P_{ign}) are commonly used [5]. The MIE is a probabilistic variable indicating 50% successful ignitability of the ignition energy which can only be determined statistically by repeating many ignition experiments under the same experimental conditions. This variable is usually used in evaluating spark ignition performance in premixed systems [10]-[11]. On the other hand in non-premixed environments, such as gas turbine engines, the P_{ign} is usually used and is defined as the number of successful ignition events divided by the number of trials [12].

Computational evaluation of ignition probability in turbulent flow fields has been usually performed using two different approaches. In the first approach, ignition process from the end of energy discharge phase (flame kernel formation) until final flame stabilization is directly stimulated [13]-[23]. Since several repeated reacting flow computations are needed for different conditions, this method is computationally demanding or even prohibitive for industrial systems. In the second approach, only a single numerical simulation of non-reacting flow field is used and ignition probability is computed using the outputs of this single CFD simulation [24]-[29]. Although various assumptions are included in this method, previous studies have showed that using this approach leads to acceptable engineering results. Although, in another studies, performance of the gas turbine ignition system have been evaluated based on the MIE parameter [30]. Recently, a model has been proposed for evaluating time evolution of spark kernel under gas turbine working conditions which is based on modeling spark kernel with a perfectly stirred reactor

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and tracking this reactor volume outside the flammability zone and during rapid entrance of this volume into the flammable zone [31].

After comprehensive review of different spark ignition modeling methods for gas turbine applications and investigating the advantages and disadvantages of these approaches [1], an engineering model for evaluation of ignition system performance has been selected and implemented in this work. This method has been already used for evaluating spark ignition system in an aero gas turbine combustor [32]-[33]. In this model, the results of time-averaged cold flow simulation of the combustor are generated as input data for a stochastic ignition modeling code. Afterwards, the stochastic code simulates the kernel propagation and identifies the full range of probable movements of flame particles following spark and models flame spread phase using a Monte Carlo particle tracking approach. Governing equations of the particle movement are standard equations of Lagrangian description of turbulent mixing. More details on this model and its validation in laboratory scale burners have been presented in our previous studies [1] and [34], and the results of evaluating the ignition performance of an industrial gas turbine by this model has been presented in the following sections.

The aim of this study is to predict the lean light-off (LLO) limit in an annular industrial gas turbine combustor. This combustor consists of 18 second-generation EV burners. A schematic of a single-burner sector of the combustor has been presented in Fig. 1. CFD simulation of the combustor has been performed in a three-burner sector (60 degrees sector of the hole combustor without diffuser) in order to capture the main ignition-related fluid dynamic features of the flow field. The importance of the three-burner combustor sector in evaluating ignition performance of annular combustors has been examined in previous studies [35]-[37].

Evaluated environmental conditions include ambient temperature between -10 to +50 ^{o}C and ambient pressure between 0.8 to 1.0 bar with standard humidity level (60%) in all calculations. Also, natural gas fuel of the turbine has been supposed to be pure methane.

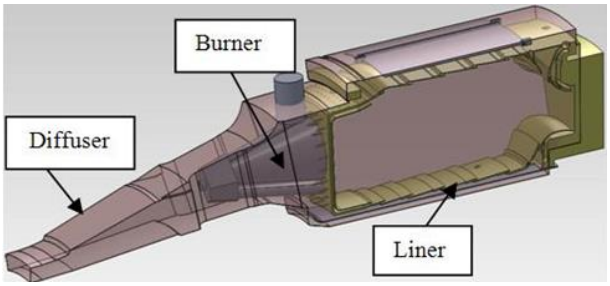

Fig. 1. The model of single burner gas turbine combustion chamber with diffuser

Our previous studies showed that the pure methane fuel has the lowest reactivity compared with all Iranian pipeline natural gas compositions [38] and therefore assuming natural gas as pure methane will yield the minimum ignition probability relative to all families of the Iranian pipeline natural gas fuels.

The target of the presented computational approach for LLO determination is to mimic the experimental procedure as closely as possible. In future sections, the details of the ignition system, and modeling procedure and results are presented.

Spark Ignition System

Two ignition systems are evaluated in this study, both of them are high-energy and low-voltage (2kV) capacitor type systems. The first ignition system is the product of Vibro-Meter Company. Exciter energy of this system is 8J and its sparking rate is 2 sparks per second with a sunken fire surface discharge plug. The second ignition system is the product of SMITSVONK Company whose exciter energy and sparking rate are 12J and 3 sparks per second, respectively with a flush fire surface discharge plug. Fig. 2 shows details of the both ignition systems and their plume sizes.

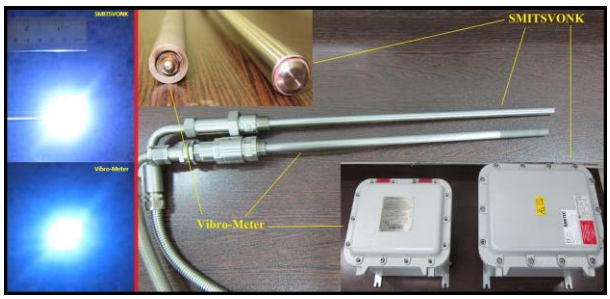

Fig. 2. Comparison of the Vibro-Meter and SMITSVONK ignition systems and their plume sizes

It is assumed in all computations that about 25% of the exciter energy is transferred to the spark. The initial size of the spherical kernel could be calculated with simple thermodynamic calculations. Using the assumptions and approach of [39], the initial kernel diameters for 8J and 12J systems are about 25 mm and 29 mm respectively. Furthermore, following the method described in [24], it has been shown that due to high energy of both systems, the initial flame kernel has been always formed. Moreover, in this study the ignition of a single burner (middle burner in 3-burner sector) has been considered and the light-around of the whole combustor is not studied. In other words, the focus of this study is on the 2nd phase of spark ignition process.

Modeling Approach

Main steps of ignition modeling approach have been described in [25]. First, a structured computational grid is produced with a specific mesh size resulting from numerical solution of the flow field. Each cell in this grid could have one of the cold, burnt, or out the flammability zone states. Initially, all cells in flammability zone are at the cold state. Simulation starts by defining a spark volume in the solution domain. This primary volume of the spark results from the ignition system characteristics or experimental data. Cells with common boundaries with spark volume turn into burnt

state (if these cells are in the flammable zone) and each of them emits a flame (active) particle. If a cell has a common boundary with spark and is outside the flammable zone, the cell will produce an inactive particle with high temperature. Active particles are tracked using the simplified Langevin stochastic differential equation (SDE) within the cold flow field. For numerical solution of this equation, Euler-Maruyama approach is used in this study.

For inactive particles, in addition to this equation, a so-called thermal memory equation is also solved. In this case, the particle gains a chance to return to the flammable zone before quenching. In fact, the particle will not quench until it has a temperature above a pre-assumed critical temperature. This time depends on the particle speed of movement and mixture fraction which have mean and instantaneous values determined from CFD solution.

In any point of particle path, the possibility of particle quenching exists. In the case of quenching, particle calculations will terminate. Quenching criterion for active particles is defined based on a Karlovitz number whose computation basis is local turbulence features and mixture equivalence ratio. When a particle meets a grid cell in clod state, its state changes into burnt state and a new particle with its own speed and mixture fraction is emitted from the cell center. In this condition, if flame particle gets out of the flammability zone, it turns into an inactive particle for which thermal memory equation is solved.

During simulation, ratio of number of burnt cells to total cells in the domain is computed as a time dependent function. This parameter is called ignition progress factor (Π_{ign}). At the end of the simulation, Π_{ign} is compared with a threshold value to evaluate successful ignition. By repeating these computations for different spark locations, ignition probability map is produced for the entire combustor. For a fixed ignition point, the effects of different factors like ignition system specifications, fuel type and composition, and operating and environmental conditions of turbine on ignition system performance can be examined. Moreover, in the current modeling approach flame-wall interactions have been considered and the effects of head on quenching (HOQ) and side-wall quenching (SWQ) of the flame have been studied. The Peklet number values of 3 and 7 for maximum loss of heat flux in turbulent case have been selected for HOQ and SWQ respectively. By applying these values in our code and calculation of flame thickness using these parameters, a limit value for approaching the particles to the wall is achieved. If the particle distance from the wall becomes below that value, the particle returns to quenched state. For implementing this model, firstly, the time-averaged LES solution of the combustor cold flow field is generated using the commercial FLUENT software. Afterwards, the ignition performance of the combustor has been evaluated using a stochastic ignition code that has been developed in the MATLAB environment [1].

Numerical Details

As mentioned in the previous section, the numerical simulation of the three-burner sector of the combustor has been performed using the FLUENT software. For this means, a computational grid with about 20 million cells was produced for three-burner combustor. Distribution of cells was in a way that the number of cells in the middle section (No. 6 or ignition burner) became twice the number of the cells in neighboring sections. Fuel is injected only from the main fuel path of the central burner. In air entrance slots of the burner, proper profiles were used and all properties were considered temperature dependent. For numerical solution of the non-reacting flow field, first the RANS simulation of the combustor has been performed using realizable k-ε turbulence model. All space discretizations have been done by QUICK method, except for pressure variable whose discretization is second-order. After convergence of the steady solution, LES simulation has been conducted. In LES simulation, energy and momentum space discretizations have been performed using bounded-central differencing (BCD) method, pressure with second-order method, and other variables were discretized by QUICK method. Time discretization was done by the second-order method. For convergence of each time step, 10 iterations were considered and the whole computation time is 0.02 s (about 3 times of the through-flow time or L/U). During the unsteady solution, firstly about 0.02 s solution has been performed in order to produce turbulent fluctuations properly and also eliminate the effects of preliminary solution. Afterwards the simulation continued for further 0.02 s and during this time period, sampling from solution field and time-averaging was activated. All unsteady simulations were conducted with 2×10^{-5} s time step (non-dimensional time step of about t/U.D=0.01). In this way, 1000 time steps before time-averaging and 1000 time steps during time-averaging were solved. The size of computational grid was carefully selected to model key phenomena. Thus, maximum mesh size in the burner and its output and also in shear layers and mixing zones of the combustor is about 1 mm. The mesh size is less than 2 mm in flame stabilization zone. Also, at least about 10 cells were considered in output of fuel injection holes in order to model fuel injection and mixing accurately. Maximum dimensionless size of the grid (Δ_{max}/D) in the primary combustion zone was selected to be below 0.02. In LES simulations, sub-grid scale modeling of turbulence has been done by the Smagorinsky-Lilly model. Boundary conditions include mass flow inlet for fuel holes, velocity inlet with pre-defined profiles in the burner air inlet slots, mass flow inlet in cooling holes and pressure outlet in the combustor exit plane. All the walls have been modeled as adiabatic assuming the no-slip condition on the walls. In all of the simulations, governing equations were solved by the SIMPLE method based on pressure-based approach. Calculations were conducted using a system consisting of 96 AMD

processors with process capability of 2.4 GHz and memory of 128 GB.

An outline of the computational grid is shown in Fig.3. Formation of the central recirculation zones in three-burner sector colored by fuel mixture fraction is shown in the Fig. 4.

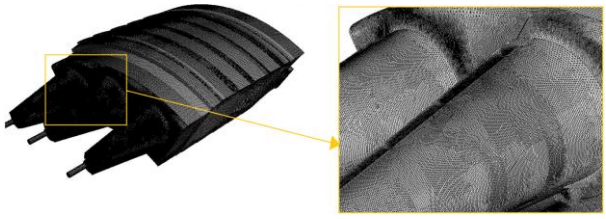

Fig. 3. Computational grid for LES of three-burner sector

During the ignition period, fuel is just injected from the main fuel path of the middle (No.6) burner. In Fig. 5 mean and instantaneous values of fuel mixture fraction and axial velocity has been presented in the central plane of the No.6 burner in three-burner sector.

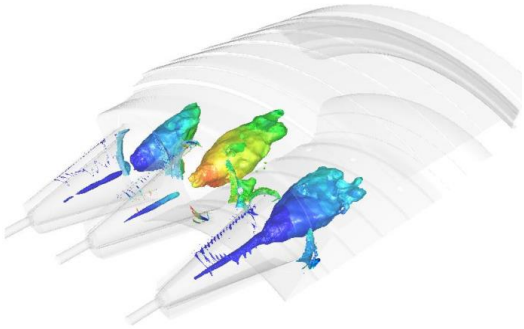

Fig. 4. Central recirculation zones in three-burner sector colored by fuel mixture fraction (in the ignition period, fuel is just injected from the main fuel path of the middle burner)

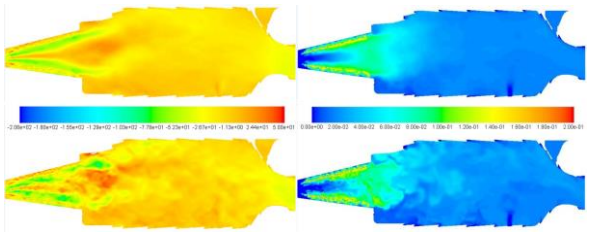

Fig. 5. Mean (top) and instantaneous (bottom) values of fuel mixture fraction (right) and axial velocity (left) in the central plane of the three-burner sector

The structured grid of the combustor has been generated using the snappyHexMesh module of the OpenFOAM code. In order to reduce the computation time, only the primary ignition zone was regarded in computations. This area includes the main ignition zone of the No.6 burner and its characteristics will be presented in the following section. After extracting the needed data from the CFD solution, results are transferred to the structured grid and stored in MATLAB software. This structured data is stored as the input for the stochastic ignition modeling code. This code has been validated extensively in our previous studies [1].

Results and Discussion

In the present study, fuel type and igniter location are assumed to be fixed. Therefore, computations were performed based on the following variables: (1) ignition system type (Vibro-Meter or SMITSVONK), and (2) operating conditions of turbine (temperature, pressure, and mass flow rate of input air) expressed as a loading parameter [35]. In the previous similar studies [32], this modeling approach was used for computing ignition probability. But in this study, the model has been used in a different approach in order to generate the LLO map the combustor. The proposed approach follows these steps. For a certain ignition system and a set of defined operating conditions, sparking process is simulated (see e.g. [1] and [34]). After repetitive simulating of sparking process (50 times in this study) ignition probability is computed. Since acceptable time for turbine ignition is about 10 s and sparking frequency of combustion system is also known, assuming the worst conditions (i.e. success of the last spark in 10 s of turbine ignition system operation), minimum acceptable ignition probability in related conditions can be estimated. If calculated ignition probability by ignition code becomes lower than acceptable value, fuel mass flow rate will be increased a little and computations will be repeated. Finally, the minimum acceptable fuel mass flow rate for successful ignition can be identified. By repeating this trend, LLO map of the turbine can be generated. Calculations may be initiated from the turbine lean blowout (LBO) limit [40] and the fuel flow rate increased from this initial value gradually. If no data exists for LBO limits of the turbine, in premixed combustion systems, primary point of fuel mass flow rate can be selected in a way that global equivalence ratio of the primary combustion zone falls in the stoichiometric range. In non-premixed systems with liquid fuel, fuel mixture fraction in spark location and on the path of flame spread toward stabilization zone should be considered [41]-[44].

To alleviate the time of repetitive calculations for derivation of the LLO map, two methods have been proposed and implemented in this study. The first approach is reducing the computation domain. For this purpose, only primary ignition area of the ignition burner (No.6) was studied. Fig. 6 shows the geometry of three-burner sector with igniter location and the volume of the primary ignition zone.

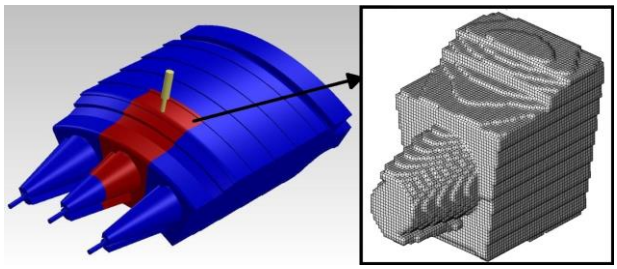

Fig. 6. Geometry of the three-burner sector with schematic position of igniter in front of the No. 6 burner (left) and primary ignition volume and its structured computational grid with 4 mm mesh size (right)

In this way, the computational domain of the ignition code decreases significantly. This volume has been selected in a way that it has minimum effects on the process of particle tracking and prediction of the ignition probability.

The second approach is related to reducing the computation time by reducing the LES simulation runs. Gradual increasing of the fuel mass flow rate needs repeating LES solution for each value of the fuel mass flow rate. This issue increases computational time significantly, terminating the engineering nature of developed tools. For decreasing the computational load, a method has been proposed which is based on the small amount of fuel compared to air in the combustor. By changing the fuel flow rate, general features of flow field like velocity distribution and turbulence features do not change significantly. The hypothesis is that the fuel flow rate of all grid cells in the ignition zone change linearly with the amount of the total input fuel. To test the accuracy of this hypothesis, three LES simulations were conducted: Nominal amount of fuel, ±50% of nominal amount of fuel. Afterwards, fuel distribution was compared for two states: (a) direct output of LES solution in nominal conditions of fuel mass flow rate, and (b) computed mean values of nominal fuel conditions based on the results of LES solution related to ±50% of nominal fuel mass flow rate.

Distribution of mean and RMS of the fuel mixture fraction estimation error using this linear change assumption with inlet fuel flow rate has been shown in the Fig. 7. Mean and RMS of fuel mixture fraction prediction error in primary ignition zone were below 12% and 22% respectively. Moreover, as seen in Fig.7, distribution of this error in primary ignition zone was less than fuel injection and mixing zones.

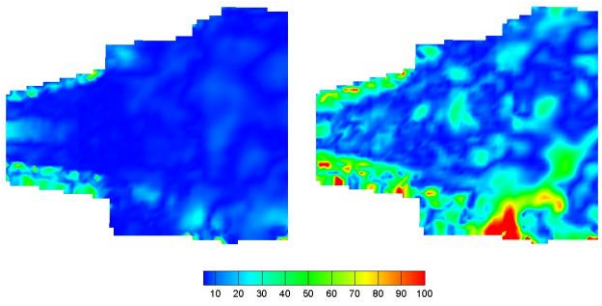

Fig. 7. Distribution of mean (left) and RMS (right) of the fuel mixture fraction estimation error using the linear change assumption with inlet fuel flow rate

Thus, the hypothesis suggesting linear change of fuel mixture fraction of computational cells with changes in input fuel mass flow rate is accepted. By this hypothesis, the number of LES computations for each set of operating conditions is limited to two solutions and there is no need to repeating LES solution for each value of fuel mass flow rate. Since the results of LES computations (not presented here) showed that the mean value of the turbulence dissipation rate and kinetic energy in primary ignition zone are about 100,000 m^2/s^3 and 200 m^2/s^2 respectively, according to the theory presented in [25], the structured grid size of 4 mm and time step of about 0.5 ms has been selected for all of the calculations in the ignition code. Table. 1 includes a sample of LLO limit calculation results. Calculated LLO is close to the turbine data and the relative error is less than 20% (only Vibro-Meter data is available).

Table 1. LLO limit for a set of operating conditions

Ignition System	T_3 (k)	P_3 (kPa)	$m_{air,total}$ (kg/s)	$m_{fuel,LLO}$ (kg/s)	
				Calculated	Turbine Data
Vibro-Meter	308	127	0.45	0.0298	0.0250
SMITSVONK				0.0281	-

Fig. 9 shows a sample of calculation results for particle tracking during a successful ignition event. By repeating the proposed process, the complete LLO limit of the turbine could be generated.

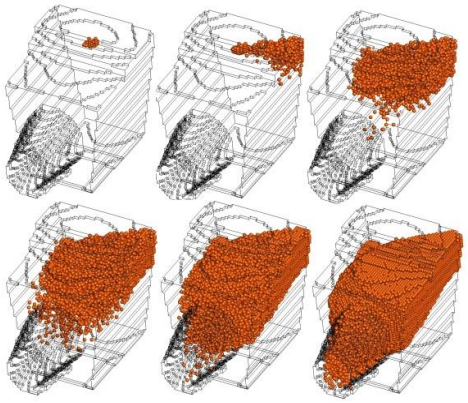

Fig. 8. A successful ignition sequence

Conclusions

Spark ignition performance of an annular combustor analyzed using a computational modeling approach. This method has the potential to predict the effects of the ignition system location and specifications, fuel type and composition, and operating conditions on the gas turbine ignition performance and LLO limit with acceptable level of accuracy and manageable computation time.

References

[1] M.A. Soroudi and M. Timaji, Development and Evaluation of the Spark Ignition Process Modeling Algorithm in the Gas Turbine Combustion Chamber, National Iranian Gas Turbine Project Report, Turbotec Company, Tehran, Iran, 2012 (in Persian).

[2] A. Berg, Flexible Ignition System for a Gas Turbine, MSc Thesis, KTH, 2012.

[3] K.G. Moesl, K.G. Vollmer, T. Sattelmayer, J. Eckstein and H. Kopecek, J. Engr. Gas Turb. Power 131 (2009) 021501.

[4] J. Griffiths, M. Riley, A. Borman, C. Dowding, A. Kirk and R. Bickerton, Opt. Lasers Eng. 66 (2015) 132-137.

[5] M. Cordier, A. Vandel, G. Cabot, B. Renou and A.M. Boukhalfa, Combust. Sci. Technol. 158 (2013) 379-407.

[6] S. Barbosa, P. Scouflaire, S. Ducruix and G. Gaborel, Comparisons Between Spark Plug and Laser Ignition in a Gas Turbine Combustor, Proceedings of the European Combustion Meeting, ECM2007, 2007.
[7] A.H. Lefebvre and D.R. Ballal, Gas Turbine Combustion: Alternative Fuels and Emissions, 3rd Ed., CRC Press, New York, 2010.
[8] V.V. Mikhailov, V.V. Lebedev and A.N. Mukhin, Russian Aeronautics 53 (2010) 349-352.
[9] A. Starikovskiy and N. Aleksandrov, Prog. Energy Combust. Sci. 39 (2013) 61-110.
[10] C. Cardin, B. Renou, G. Cabot and A. Boukhalfa, Comptes Rendus Mecanique 341 (2013) 191-200.
[11] S.P.M. Bane, J.L. Ziegler, P.A. Boettche, S.A. Coronel and J.E. Shepherd, J. Loss Prev. Proc. Ind. 26 (2013) 290-294.
[12]. E. Mastorakos, Prog. Energy Combust. Sci. 35 (2009) 57-97.
[13]. A. Triantafyllidis E. Mastorakos and R.L.G.M. Eggels, Combust. Flame 156 (2009) 2328-2345.
[14]. V. Subramanian, P. Domingo and L. Vervisch, Combust. Flame 157 (2010) 579-601.
[15]. M. Boileau, G. Staffelbach, B. Cuenot, T. Poinsot, and C. Berat, Combust. Flame 154 (2008) 2-22.
[16]. G. Lacaze, E. Richardson and T. Poinsot, Combust. Flame 165 (2009) 1993-2009.
[17]. W.P. Jones, and A. Tyliszczak, Flow Turbulence Combust. 85 (2010) 711-734.
[18]. W.P. Jones and V.N. Prasad, Proc. Combust. Inst. 33 (2011) 1355-1363.
[19]. J.M. Boyde, P.C. Le Clercq, M. Di Domenico and M. Aigner, Combust. Flame 160 (2013) 351-365.
[20]. J.M. Boyde, P. Le Clercq, M. Di Domenico, M. Rachner, G.C. Gebel, T. Mosbach and M. Aigner, J. Engr. Gas Turb. Power 135 (2013) 011506.
[21]. J-F. Bourgouin, D. Durox, T. Schuller, J. Beaunier and S. Candel, Combust. Flame 160 (2013) 1398-1413.
[22]. M. Philip, M. Boileau, R. Vicquelin, E. Riber, T. Schmitt, B. Cuenot, D. Durox and S. Candel, Proc. Combust. Inst., Article in Press.
[23]. L. Escalpez, E. Riber, and B. Cuenot, Proc. Combust. Inst., Article in Press.
[24]. A. Eyssartier, B. Cuenot, L.Y.M. Gicquel and T. Poinsot, Combust. Flame 160 (2013) 1191-1207.
[25]. A. Neophytou, E.S. Richardson and E. Mastorakos, Combust. Flame 159 (2012) 1503-1522.
[26]. J. Weckering, A. Sadiki, J. Janicka, E. Mastorakos and R.L.G.M. Eggels, Proc. Combust. Inst. 33 (2011) 2919-2925.
[27]. G. Linassier, A. Bruyat, P. Villedieu, N. Bertier, C. Laurent, O. Rouzoud, R. Lecourt, H. Verdier and G. Lavergne, Comptes Rendus Mecanique 341 (2013) 201-210.
[28]. N.G. Rosa, G. Linassier, R. Lecourt, P. Villedieu and G. Lavergne, Heat Trans. Eng. 32 (2011) 949-956.
[29]. C.W. Wilson, C.G.W. Sheppard and H.C. Low, A Novel Technique for Predicting the Ignition Performance of an Aero Gas Turbine Combustion Chamber, Gas Turbine Engine Combustion, Emissions and Alternative Fuels, Lisbon, Portugal, 1996.
[30]. O. Antoshkiv, S. Bake and H.P. Berg, Spray Phenomena and Their Influence on the Ignition Performance of a Modern Aeroengine Combustor, ILASS Paper ILASS08-11-1, 2008.
[31]. B. Sforzo, J. Kim, J. Jagoda and J. Seitzman, J. Engr. Gas Turb. Power 137 (2015) 011502.
[32]. A. Neophytou, E. Mastorakos, E.S. Richardson, S. Stow and M. Zedda, A Practical Model for the High-Altitude Relight of a Gas Turbine Combustor, MCS7, Sardinia, Italy, 2011.
[33]. T. Soworka, M. Gerendas, R.L.G.M. Eggels and E. Mastorakos, Numerical Investigation of Ignition Performance of a Lean Burn Combustor at Sub-Atmospheric Conditions, ASME Paper GT2014-25644, 2014.
[34]. M.A. Soroudi, M. Timaji, N. Rasooli and H. Khaledi, Estimation of Ignition Limit of a Model Gas Turbine Combustor Using Monte Carlo Particle Tracking Method, 2nd National Gas Turbines Conference, Iran University of Science and Technology, Tehran, Iran, 2013 (in Persian).
[35]. G. Linassier, C. Viguier, H. Verdier, R. Lecourt, G. Heid and G. Lavergne, Experimental Investigations of the Ignition Performances on a Multi-sector Combustor under High Altitude Conditions, AIAA Paper 2012-0934, 2012.
[36]. C. Hirsch, T. Kuenzi, H.P. Knopfel, B. Paikert, C. Steinbach, W. Geng and K. Dobbeling, An Annular Combustor Natural Gas Ignition Model Derived from Atmospheric Sector Experiments, ASME Paper GT2002-30073, 2002.
[37]. M. Kobayashi, H. Ogata, T. Oda, R. Matsuyama, A. Horikawa and Y. Kinoshita, Improvement on Ignition Performance for a Lean Staged Low NOx Combustor, ASME Paper GT2011-46187, 2011.
[38]. M.A. Soroudi, and N. Rasooli, Natural Gas Composition Effects on DLE Gas Turbine Combustor Operability and Emissions, CSME International Congress, Toronto, Canada, 2014.
[39]. J.M. Boyde, M. Di Domenico, B. Noll, and M. Aigner, Spark Ignition Simulations and the Generation of Ignition Maps by Means of a Turbulent Flame Speed Closure Approach, ASME Paper GT2010-22211, 2010.
[40]. M.A. Soroudi, S. Hadi Bafekr, M. Timaji and N. Rasooli, A Priori Calculation of Lean Blowout Limit in an Industrial Gas Turbine Combustor, Proceedings of the European Combustion Meeting, ECM2013, 2013.
[41]. R.W. Read, J.W. Rogerson, and S. Hochgreb, AIAA J. 48 (2010) 1916-1927.
[42]. R.W. Read, J.W. Rogerson and S. Hochgreb, J. Prop. Power 29 (2013) 961-974.
[43]. X. Wang, Y. Lin, H. Hu, C. Zhang and Y. Kang, Effect of Swirl Cup's Venturi Shape on Spray Structure and Ignition Process, ASME Paper GT2014-25216, 2014.
[44]. M. Majcherczyk, N. Zarzalis and F. Turrini, Influence of the Turbulence Length Scale and Intensity on Spark Ignition of Kerosene Jet-A1-Air Mixtures at High Altitude Relight Conditions, ASME Paper GT2014-25332, 2014.