International Journal on "Technical and Physical Problems of Engineering" (IJTPE)
Published by International Organization of IOTPE

ISSN 2077-3528
IJTPE Journal
www.iotpe.com
ijtpe@iotpe.com

June 2022	Issue 51	Volume 14	Number 2	Pages 1-6

TURBULENT FLAME SPEED IN SPARK IGNITION ENGINES FOR BIOFUEL AND GASOLINE BLEND

N.S. Khider Al Eniz [1] A.H. Mohammed [1] Y.S. Mahmood Al Jamiaa [1]
R.A. Mahmood Al Nuaimi [2]

1. Mechanical Engineering Department, College of Engineering, University of Mosul, Mosul, Iraq
nooralelln2017@uomosul.edu.iq, abidhabbo20@uomosul.edu.iq, yaseralmola@uomosul.edu.iq
2. Department of Mechanical Engineering, College of Engineering, University of Zakho, Zakho, Duhok, Iraq
raid.ahmed@uoz.edu.krd

Abstract- In order to more understanding the nature of physical phenomena, the simulations, mathematical equations and their solution are used to describe those physical phenomena. They also describe the combustion process with the help of mathematical and chemical equations for an ignition engine by spark. Alternative fuels and air are mixed at a molecular level prior to combustion which occurs as a flame front. This paper presents the combustion simulation of spark ignition for an engine with four-stroke cylinders and a compression ratio (12:1). Zero-dimension model is used to predict the speed of turbulent flame, and mixture (10%) of ethanol and gasoline expresses the fuel used in this study. The turbulent flame speed is predicted with different parity ratios (0.6-1.1), different engine speed (500-2500) RPM and different compression ratios (7:1 - 10:1) as well as the timing of the different spark timing up to (5-30 btdc), by using the visual basic program.

Keywords: Spark Ignition Engine, Ethanol, Gasoline, Turbulent Flame Speed, Rotational Speed of Engine, C.R., Ratio of Equivalence, Ignition Timing.

1. INTRODUCTION

Flame speed is defined as the speed of the an combust gases through the combustion wave in the natural surface direction of the wave. The most combustion processes are considered almost turbulent on a Bunsen Burner in spark ignition engines. Also considered the main factor that controls the rate of increasing pressure and thus has an important effect on the engine's performance. Finally, the combustion speed inside the engine cylinder can be considered of great importance to engine designers as it gives a good representation of the engine system [1].

Ghanauti, et al., simulate the combustion process of a combustion engine with a spark using the GT-Power program and the flame rate model to determine the Laminar Flam Speed and using (50% MFB) of the mixture other than the burning speed under the different temperatures and pressures of the unburned mixture, in addition to a different ratio of the rotational speed of the engine. They concluded that the highest combustion was at the equivalent ratio of the ideal ratio (Stoichiometric) [2].

For the sake of the ability to display a flame velocity model to determine the combustion phases and to predict the combustion spark, the researchers (Nafis Ahmad, Rashid Ali) developed a (Zero dimensional Thermodynamic) model to estimate the turbulent flame velocity in a combustion engine with a spark using a mixture of Iso-octan and air for a wide range of engine operating parameters that included the effect of engine's rotational speed (600-1160) RPM, ratio of equivalent (0.7-1.1), temperature for unburned mixture (532-650 Rankine), compression ratio (5-8) as well as the effect of timing of firing spark plug (5-30 btdc). Then they compared the practical and theoretical results with the results. The model followed in this study is that of (Cakir) and (Malik), where they concluded that the results obtained from the model used have a good compatibility with the (Cakir) and (Malik) models [1].

The researchers Anurag Mani Tripathi, Parth Panchal, Vidhyadhar Chaudhari [3] performed a simulation of a four-stroke single-cylinder engine to measure the velocity of turbulent flame in it, by using methane gas as a fuel at different operating conditions, the equivalent ratio ranged from (0.6-1.2) and the rotational speed of the engine ranged from (1500-3000 RPM). The researchers found an increase in the turbulent flame speed (7.1053-8.0386 m/s) with an increase in both the equivalent ratio and the rotational speed of the motor.

For the theoretical analysis, a hypothetical internal combustion engine operating with a spark plug and a four-stroke system was selected to study the effect of the engine's operational parameters on the turbulent speed of the flame as shown in Table 1.

Table 1. Engine specification [5]

Engine Type	Fray man A30 marine diesel engine
Compression Ratio (*CR*)	5:1 to 18:1
Bore (*B*)	95 mm
Stroke (*L*)	82 mm
Cycle	4-stroke
System of fuel suppled	Carburetor
Spark Timing	30 btdc to 10 atdc

2. FOUR-STROKE CYCLE MODE OF ENGINE

Figure 1 shows the four-stroke cycle mode of the engine which starts from point 6 to 1 representing the intake stroke where fuel and air is enter to the cylinder. Compression stroke begins at point 1 and ends at point 2 where fuel, air and residual gases are compressed. Combustion period is important in the cycle, which is started at point 2 and ended up at point 3. Expansion stroke begins at point 3 and ends at point 4. Finally, the exhaust stroke starts at point 4 and ends up at point 6 where exhaust gases are pushed out of the cylinder.

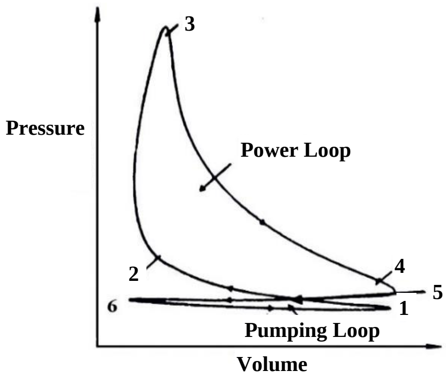

Figure 1. Diagram of the pressure-volume relationship for a four-stroke cycle [4]

3. ASSUMPTIONS SUPPORTED

The following assumptions were adopted during the mathematical operations:

- The effects of cracks and grooves in the cylinder walls are neglected.
- The combustion process starts before the upper dead-point at a specified angle ranging between (5-30 btdc).
- Assuming the calculation increment is one degree of crankshaft.
- Considering the combustion chamber as a burnt zone, and that the temperature and pressure inside the combustion chamber are homogeneous.
- Neglecting changing the work with changing the temperature.

4. THEORETICAL ANALYSIS

A Four-Stroke cycle engine was simulated using a visual basic software that calculates the combustion products as well as the performance parameters of the engine running on a spark based on the thermodynamic model. After determining the controlled operational conditions, the program calculates the temperature of the mixture of air, fuel and residual gases using the following relationship [4]:

$$T_1 = (1-f)T_i + f\left[1-\left(\frac{P_i}{P_e}\right)^{(K-1)/K}\right]T_e \quad (1)$$

where, T_e and P_e are the temperature and pressure of the exhaust gases and, T_i and P_e are the initial temperature and pressure. Note that the percentage of residual emissions f was calculated by the following Equation (4):

$$f = \frac{1}{CR}\left[\frac{P_e}{P_4}\right]^{1/K} \quad (2)$$

where, CR is compression ratio, and P_4 is pressure at the start of opening exhaust valve.

To find the pressure and temperature for mixture state, the compression stroke is divided into intervals equal to arithmetic steps. The total cylinder volume at each calculation step is calculated by the angle of the crankshaft by the following relationship (4).

$$V(\theta) = V_C + \frac{V_S}{2}\left[\frac{2L}{S} + 1 - \cos(\theta) - \left\{\left(\frac{2L}{S}\right)^2 - \sin^2(\theta)\right\}^{1/2}\right] \quad (3)$$

The temperature (T_2) and pressure (P_2) are calculated at the end of the calculation step using the following Equations (4) and (5).

$$T_2 = T_1\left[\frac{V_1}{V_2}\right]^{(K-1)} \quad (4)$$

$$P_2 = \left[\frac{V_1}{V_2}\right]\left[\frac{T_2}{T_1}\right]P_1 \quad (5)$$

The value of K is calculated from the relationship (6).

$$K = \frac{R_o}{C_v(T_1)} + 1 \quad (6)$$

where, R_o represents the general constant of gases, and C_v as the specific heat of a particular mixture can be obtained from the following Equation (7):

$$C_v(T) = R_o \sum_{i=1}^{i=N} \frac{w_i}{w_m}\left[\left\{\sum_{j=1}^{j=5} jU_{ij}T^{(j-1)}\right\} - 1\right] \quad (7)$$

Compression stroke calculations continue until the angle of the crankshaft is equal to ($\theta = \theta_{sprk}$), then the (Delay Time) calculations begin. Burned temperature (T_b) during this phase depends on Equations (8) and (9), respectively, depending on the equivalent ratio (ϕ) the temperature of the unburned charge [8].

$$T_b = T_{un} + 2500\phi \;,\; \phi \le 1.0 \quad (8)$$

$$T_b = T_{un} + 2500\phi - 700(\phi - 1) \;,\; \phi > 1.0 \quad (9)$$

The accurate calculation of the spread speed of the flame needs a briefing of the reactant properties (physical and chemical). However, turbulence combustion inside the combustion chamber deflects this, as it crimps and breaks the flame front, making it difficult to predict how fast the flame will spread. Therefore, the researchers used to present experimental equations and these equations included the influential elements on the velocity of flame propagation. Perhaps the most common equation is the experimental (Kuehl) equation which has been adopted to calculate the velocity of the flame flow (Laminar flame speed) [11].

$$U_L = \frac{1.089\times 10^6}{\left\{\left(\frac{10^4}{T_b}\right)+\left(\frac{900}{T_{un}}\right)^{4.939}\right\}} P^{-0.09876} \tag{10}$$

It is difficult to express the turbulent state of the flame speed in the combustion chamber in terms of (Laminar flame speed), so it was necessary to find the turbulent speed of flame which is calculated through the following Equation (12).

$$U_T = U_L \times ff \tag{11}$$

As for the calculation of the turbulent flame factor value (*ff*), the following relationship was used (13):

$$ff = 1+0.0018N \tag{12}$$

The delay period was calculated in terms of the rotational speed of the crankshaft using the following relationship [9].

$$(\Delta\alpha)_{delay} = \frac{360NR_f}{U_T} \tag{13}$$

which R_f represents the radius of the flame front.

The volume of the cylinder at the beginning of combustion can also be expressed as Equation (14):

$$V_P = \frac{2}{3}\pi R_f^3 \tag{14}$$

The value of R_f is calculated by Equation (15).

$$R_f = \left[\frac{0.001V_c}{\frac{2}{3}\pi}\right]^{1/3} \tag{15}$$

Since the combustion period is proportional with the engine speed, the combustion period can be guessed by the following relationship [10].

$$\theta_b = 40+5\left(\frac{N}{600}-1\right)+166\left(\frac{Y_{cc}}{Y}-1.1\right)^2 \tag{16}$$

where, Y_{cc} represents the amount of oxygen needed, Y is the number of oxygen moles in the mixture.

While the calculation of fuel mass (m'_f) used as a mixture with the air entering the engine during the intake stroke. first requires calculating the added air mass within relationship (17) [11]:

$$m_1 = \frac{CD_t A_t P_o}{6N(RT_o)^{0.5}} \tag{17}$$

$$m'_a = m_1\left(\frac{P_c}{P_o}\right)^{1/K}\left[\frac{2K}{(K-1)}\left\{1-\left(\frac{P_c}{P_o}\right)^{(K-1)/K}\right\}\right]^{0.5} \tag{18}$$

where, CD_t is the discharge coefficient in the neck of the venture, P_o and T_o represent the atmospheric pressure and temperature, respectively.

The A_t represents the amount of flow in the neck of the venture which is calculated by the following relationship (12):

$$A_t = \pi D_v L_v \tag{19}$$

where, D_v represents the diameter of the intake valve, and L_v represents the distance traveled by the intake valve which is calculated by the following relationship (12):

$$L_v(\theta) = \frac{L_{iv,\max}(1-\cos\varphi)}{2} \tag{20}$$

where, $L_{iv,\max}$ is amount of lifting the intake valve at the full opening of the valve.

The φ (coefficient of charge intake valve angle) can be obtained from the following Equation (12):

$$\varphi = \frac{\pi(I_{VO}-I_{VC}+2\theta+540)}{I_{VO}+I_{VC}+180} \tag{21}$$

where, I_{VO} and I_{VC} represent the angle of the crankshaft when opening and closing the intake valve. The general equation for combustion of hydrocarbon fuels with air can be described as follows [13].

$$a(C_nH_mO_r)+\frac{1}{\varphi}\left(n+\frac{m}{4}-\frac{r}{2}\right)\left[O_2+\frac{79}{21}N_2\right]\rightarrow\sum_{i=1}^{i=N}Y_iZ_i \tag{22}$$

where, a represents the total number of moles of fuel.

5. RESULTS AND DISCUSSION:

The turbulent flame speed obtained from the mathematical model at different engine operating parameters including the engine speed, compression ratio, and equivalent ratio as well as the spark timing angle has been presented and discussed.

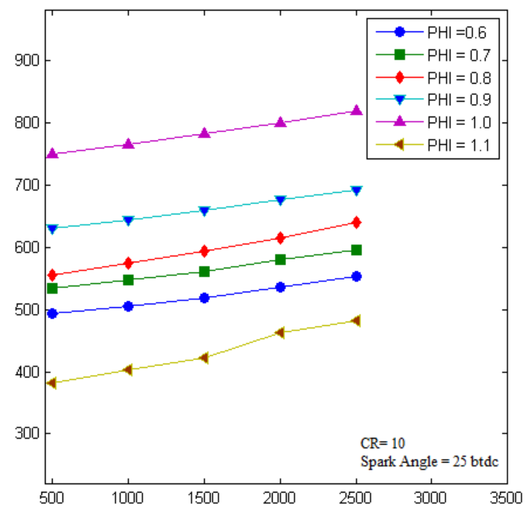

Figure 2. The engine speed and equivalence ratio effect on turbulent speed of flame

Figure 2 shows the effect of the engine speed on the turbulent flame speed at different equivalent ratios, increasing the speed of flame with an increase in the engine speed at all equivalent ratios. As for the effect of the equivalent ratio when the engine speed is fixed, we notice a gradual increase in the flame speed with an increase in equivalent ratio reaches its highest value at the Stoichiometric equivalent ratio ($\varphi = 1.0$), then the flame velocity decreases when increasing the equivalent ratio ($\varphi > 1.0$). The reason for this is referred to the state of incomplete combustion.

Figure 3 shows the effect of the equivalent ratio on turbulent flame speed at different compression ratios of the engine. It also shows increases on turbulent flame speed with increasing the equivalent ratio to its highest value at the Stoichiometric equivalent ratio ($\varphi = 1.0$), then the flame speed decreases when the equivalent ratio increases ($\varphi > 1.0$).

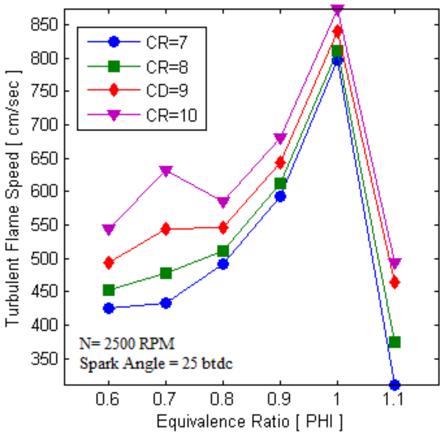

Figure 3. The equivalence ratio and compression ratio effect on turbulent speed of flame

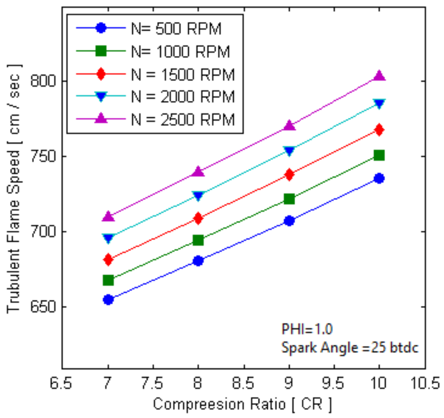

Figure 4. The compression ratio and engine speed effected on turbulent speed of flame

Figure 4 shows the compression ratio effect on the speed of the flame at different engine speeds and increasing the compression ratios to reach their highest value at the highest compression ratio. The reason is attributed to the increase in the pressure of the cylinder, which makes combustion in it completely combustible.

Figure 5 shows the spark timing effect on turbulent flame speed at different equivalent ratios. From the figure, noticed that the flame velocity is gradually increased with the equivalent ratio increase and reaching its highest value at the Stoichiometric equivalent ratio ($\varphi = 1.0$) , and then the flame speed decreases, when the equivalent ratio is increased ($\varphi > 1.0$) at each spark timing, relative to the effect of the given spark timing, when the equivalent ratio is constant, noticed that flam speed is gradually decreased with an increase in the timing of given the spark, and the reason is attributed to the state of incomplete combustion inside the cylinder.

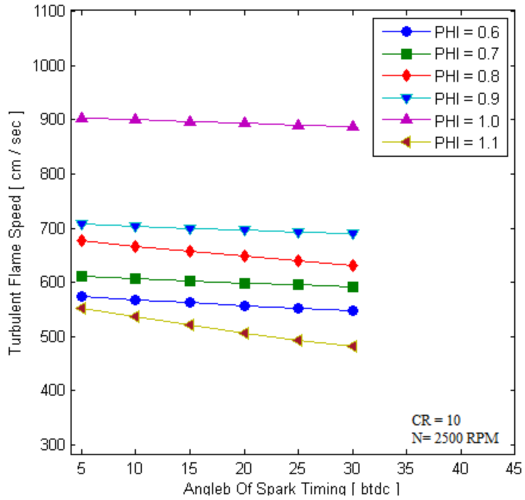

Figure 5. The ignition timing and equivalence ratio effect on turbulent speed of flame

6. CONCLUSIONS

- The flame speed increases from (550 cm/sec) to (880 cm/sec) when the equivalent ratio is increased from (0.6) to (1.0), from which it is concluded the turbulent speed of flame increases with the increase of equivalent ratio.
- The flame speed increases from (715 cm/sec) to (810 cm / sec) when the compression ratio is increased from (7:1) to (10:1), from which it is concluded the turbulent speed of flame increases with increase in the compression ratio.
- Flame speed increases from (750 cm/sec) to (820 cm/sec) when the rotational speed of the engine is increased from (500 RPM) to (2500 RPM), from which it is concluded that the turbulent speed of flame increases with increase rotational speed of the engine.
- The flame speed decreases from (905 cm / sec) to (880 cm/sec) when the spark ignition timing is increased from (5 btdc) to (30 btdc), from which it is concluded that the turbulent speed of flame decreases with increase the spark ignition timing. This conclusion is consistent with the conclusion reached by the researcher (K. Rezapour) that the mechanical output decreases with the increase in the spark ignition timing [19]. Above results were validated with the work of K.A. Malik [20] and a number of other researchers.

NOMENCLATURES

1. Symbols / Parameters

P_e: Exhaust gases pressure
T_e: Exhaust gases temperature
P_i: Initial pressure of charge
T_i: Initial temperature charge
CR: Compression ratio
f: Percentage of residual gases
$\Delta\theta$: Computational step
P_4: Pressure at opened the exhaust gases valve
P_2: The Pressure at the end of the computational step
T_2: The temperature at the end of the computational step
m'_f : Mass of fuel
Y_{cc}: The amount of oxygen needed

$L_{iv,\max}$: The amount of lifting the intake valve at the full opening of the valve
D_v: Diameter of intake valves
I_{VC}: Intake valve close angel
I_{VO}: Intake valve open angel
R_o: Universal gases constant
a: The number of moles of the total fuel used
w_i: The number of moles of a compound
C_v: Specific heat of gases
θ_{park} : Spark ignition angle
w_m: Total number of moles
T_b: Burned charge temperature
θ: Crank shaft angle
ϕ: Equivalent ratio
T_{un}: Unburned charge temperature
N: Engine rotational speed
R_f: Radius of the flame front
C_{Dt}: Discharge coefficient in venture tube
Y: The number of moles of oxygen in the mixture
φ: Coefficient of charge intake valve angle
L_v: Represents the distance traveled by the intake valve

REFERENCES

[1] R. Ali, N. Ahmad, "Turbulent Flame Velocity Model for SI Engine", International Journal of Engineering Research and Applications (IJAME), Vol. 1, Issue 2, pp. 2231-5950, 2011.
[2] A. Ghanaati, I.M. Darus, M.F. Muhamad Said, A.M. Andwari, "A Mean Value Model for Estimation of Laminar and Turbulent Flame Speed in Spark Ignition Engine", International Journal of Automotive and Mechanical Engineering (IJAME), Vol. 11, pp. 2224-2234, June 2015.
[3] A.M. Tripathi, P. Pancchal, V. Chaudhari, "Turbulent Flame Speed Prediction for SI Engine using Methane as Fuel", International Journal of Engineering Research and Applications (IJERA) Vol. 3, Issue 4, pp. 248-254, August 2013.
[4] R. Stone, "Introduction to Internal Combustion Engines", University of Basra, Basra, Iraq, 1988.
[5] P.W.E. Foss, "Variable Compression Engine, Test Rig", Operating Instruction, Tec Equipment limited, 1981.
[6] C.R. Ferguson, A.T. Kirkpatrick, "Internal Combustion Engines Applied Thermosciences", The 2nd Edition, John Wiley's sons Inc., 2001.
[7] R.S. Benson, "Advanced Engineering Thermodynamics", The 2nd Edition, Oxford, 1976.
[8] M. Mustafa, W. Kiha, "Study of the Effect of Incoming Charge Temperature on the Performance and Pollutants of Internal Combustion Engine Working Spark", Tikrit University, Tikrit, Iraq, 2009.
[9] Y.S. Philip, "Simulation of a Single Cylinder Engine Including Intake and Exhaust System", Master Thesis, University of Technology, Baghdad, Iraq, 1989.
[10] D.K. Kueh, "Laminar Burning Velocities of Propane -Air Mixture", The 8th Symposium International on Combustion, p. 510, 1962.
[11] H.M. Mahdi, "Simulation Model for Exhaust Emission and Performance of Single Cylinder Spark Ignition Engines", A Thesis Submitted for Degree of Master in the University of Technology, Baghdad, Iraq, 1990.
[12] W.W. Pulkrabek, "Engineering Fundamentals of The Internal Combustion Engine", Prentice Hail-Inc., pp. 235-237, 1997.
[13] R.A. Khalefa, "The Effect of Spark Timing on The Spark Ignition Engine Performance", Tikrit Journal of Eng. Sciences, Vol. 18, No. 2, pp. 62-71, June 2011.
[14] W.J.D. Annand, "Heat Transfer in The Cylinder of Reciprocation Internal Combustion Engines", Proc. Inst. Mech. Eng, Vol. 177, p. 973, July 1963.
[15] S. Sitthiracha, "An Analytical Model of Spark Ignition Engine for Performance", Prediction, Mongkutk's Institute of Technology North Bangkok, Thailand, 2006.
[16] S.R. Gollahalli, "Emission and Efficiency of Spark Ignition Engine", International Journal power Generation Conference, Florida, USA, July 2000.
[17] S. Banday, M. Maroufwani, "Investigation on a Single Cylinder Spark Ignition Engine Using Ethanol Gasoline Blends for Power Generation", (IJMERR) All Rights reserved, Vol. 7, No. 12, April 2015.
[18] S. Chapra, R.P. Canale, "Numerical Methods for Engineers", The 4th Edition, McGraw Hill, January 2002.
[19] K. Rezapour, "Availability Analysis of a Bi-Fuel SI Engine Model for Improvement its Performance", International Journal on Technical and Physical Problems of Engineering (IJTPE), Issue 11, Vol. 4, No. 2, pp. 115-121, June 2012.
[20] K.A. Malik, "A Theoretical Model for Turbulent Flame Speed, Based on Turbulent Transport Process for Spark Ignition Engine", Indian Journal of Technology, Vol. 29, pp. 217-220, 1991.

BIOGRAPHIES

Nooraldeen Saleh Khider Al Eniz was born in Talafer, Mosul, Iraq, in 1975. He is an assistant lecturer at Department of Mechanical Engineering, College of Engineering, University of Mosul, Mosul, Iraq. He received the Bachelor degree in Mechanical Engineering, Energy and Heating System and Master degree in Fluids and Heating, Internal Combustion Engine from Mosul University in 2000 and 2018, respectively. He is a faculty member at Mechanical Engineering Department, College of Engineering, University of Mosul since 2006. He is also a member of Iraqi Engineering Association since 2000. His areas of interests are heat transfer and internal combustion engine, combined power plants and air pollutant, theoretical study of spark ignition engine performance using a blend of gasoline and oxygenated fuels, effect of combustion duration on the performance and emission of four stroke spark ignition engine using gasoline - biofuel blend and mechanical vapor compression refrigeration system: environment and performance challenge.

Abdulrahman Habbo Mohammed was born in Sinjar, Mosul, Iraq, in 1964. He received a four-years degree in Mechanical Engineering from University of Mosul, Mosul, Iraq in 1987. He received a M.Sc. degree in combustion and energy from Leeds University, UK in 1990. He carried the research at University of Leeds in the field of internal combustions engines. He awarded a Ph.D. degree in thermal power (I.C. Engines and gas turbine) in 2004. He is a faculty member at university of Mosul, and taught many courses including internal combustions engines, power plant, advance thermodynamics, heat transfer, and pollution. He supervised the M.Sc. students in research experimentally and theoretically under including various topics such as improving gas turbine performance in hot climate, combined cycles and using alternative fuel in spark ignition engines.

Yaser Shuker Mahmood Al Jamiaa was born in Mosul, Iraq, in 1974. He received the B.Sc. degree in Mechanical Engineering, from Mosul University, Mosul, Iraq in 2004 and M.Sc. degree in Advanced Mech. Engineering (Thermal Power-Fluidized Bed) from Mechanical Department, Faculty of Engineering, University of Mosul. He was a manager of maintenance department in Mosul University from 2004 to 2008, teaching duties, and lecturer as one of staff members in mechanical engineering department from 2008 till now. His scientific researches are in mechanical vapor compression refrigeration system, environment and performance challenge, and state of art single-objective optimization of small-scale cylindrical cavity receiver.

Raid Ahmad Mahmood Al Nuaimi was born in Iraq, in 1978. He completed the Bachelor degree in Mechanical Engineering in 2001. He received the Ph.D. qualification in Mechanical Engineering from (University of Southern Queensland) USQ, Toowoomba, Australia in 2018. He is the expert in two-phase flow and CFD simulation. He has a combination of practical (industry knowledge) and academic skills in thermo-fluid. He is a skilled mechanical engineer with demonstrated experiences in manufacture, system modification, research duties, solar system, air conditioning and refrigeration fields.