HEFAT2010
7th International Conference on Heat Transfer, Fluid Mechanics and Thermodynamics
19-21 July 2010
Antalya, Turkey

HEAT TRANSFER CORRELATION CHOICE FOR TWO-ZONE COMBUSTION MODEL OPTIMIZATION IN THE CASE OF NATURAL GAS SI ENGINES

Mohand Said LOUNICI*, Mohand TAZEROUT, Mourad BALISTROU.
Laboratoire des Systèmes Energétiques & Environnement,
Emn, Nantes Cedex 3, France
Laboratoire de Dynamique des Moteurs et Vibroacoustique,
Umbb, Boumerdes, 35000, Algérie
E-mail: mslounici@hotmail.com,

ABSTRACT

Owing to its both economical and environmental qualities, natural gas is one of the most interesting and promising available fuels for internal combustion engines. However, optimization of engine design requires extensive engine testing. Therefore, engine modeling codes are generally preferred for evaluating initial designs. Two-zone model is one of the most interesting simulation tools, especially for natural gas SI engines, due to the combustion type in this case. On the other hand, about one third of the fuel energy is transformed to heat loss from the chamber walls. Hence, the accuracy of the simulation depends on the precision of the heat transfer model. However, in the previous studies, using two-zone models, many choices are made for heat transfer evaluation and no choice influence study has been carried out. The current study aims to investigate the effect of the choice of the heat transfer correlation and provide an optimized choice for a more accurate two-zone combustion model in the case of natural gas SI engines. For this purpose, a computer simulation is used, and experimental measurements are used for comparison and validation.

INTRODUCTION

Natural gas has been recently used as an alternative to conventional fuels in order to satisfy some environmental and economical concerns and governments have been motivated to expand in natural gas infrastructures in order to be feasible to passenger vehicles as well as stationary engines [1]. However, to be more attractive and feasible, many aspects have to be improved for best performance and emissions.

On the other hand, optimization of engine design requires extensive engine testing. Therefore, engine modeling codes are generally preferred for evaluating initial designs. Computer models of engine processes are valuable tools for analysis and optimization of engine performance and allow exploration of many engine design alternatives in an inexpensive way.

Internal combustion engine modeling has been a continuing effort over the years and many models have been developed to predict engine performance parameters.

NOMENCLATURE

A	[m^2]	Instantaneous surface area exposed to heat transfer
B	[m]	Cylinder bore or diameter
Gr	[-]	Grasshof number
h	[J.kg^{-1}]	Enthalpy
h_g	[W/m^2 K]	Gas-wall heat transfer coefficient,
k_g	[W/m K]	Thermal conductivity of the gas
m	[kg]	Mass of the gas
N	[rpm]	Engine rotational speed
Nu	[-]	Nusselt number
P	[pa]	Instantaneous cylinder gas pressure,
Pr	[-]	Prandtl number
q	[W/m^2]	Heat flux rate (heat transfer per unit area)
Q	[J]	Exchanged Heat
Re	[-]	Reynolds number
t	[s]	Time
T	[K]	Temperature
T_w	[K]	Wall surface temperature
u	[J.kg^{-1}]	Internal energy
V	[m^3]	Instantaneous cylinder volume
V_d	[m^3]	Volumetric capacity
V_{mp}	[m/s]	Mean piston speed
x_b	[-]	Burned gas mass fraction

Special characters

α	[° c.a]	Ignition advance,
θ	[° c.a]	Crank angle
$\Delta\theta_b$	[° c.a]	Combustion duration,
λ	[-]	Quotient of Connecting road length to crank radius,
$Ø$	[-]	Equivalent air-fuel ratio,

Subscripts and superscripts

a	Admission
b	Burned
e	Exhaust
u	Unburned

Abbreviations

EVO	Exhaust valve opening
EVC	Exhaust valve closing
IVO	Inlet valve opening
IVC	Inlet valve closing
CA(V)	Crank angle

Zero Dimensional (Zero-D) models are the most commonly preferred analytical tools for internal combustion engine development [2]. They are one of the simplest and fastest methods to model engine combustion processes. Engine designers may find that experimentally based Zero-D codes are more useful for design and development applications. If an experimental model is developed based on an engine's experimental data, this model can be used for new engines with similar design in a predictive manner.

On the other hand, heat transfer is particularly important in the combustion chamber energy balance. The gases temperatures can reach about 2800 K and heat flux induced can reach several tens megawatts per square meter for some engines. Heat transfers occupy a capital place in the combustion chamber heat release analysis, since they account for approximately 30 to 40 % of the energies in consideration [3]. For a small–scale 125 cm^3 two strokes SI engine, Franco [4] found that approximately 50% of the fuel energy is converted to heat loss. Therefore, heat transfer evaluation has a significant effect in the model accuracy.

However, in the previous studies, using two-zone model for natural gas SI engines, many choices are made for heat transfer evaluation. For instance, Amr and co-workers [1] used Woschni correlation, Caillol and co-workers [5] [6] used Hohenberg correlation and Soylu and co-workers [2] [22], used Annand's. However, no justification for the choice has been given in the literature. Thus, this study aims to investigate the effect of the choice of the heat exchange correlation and provide an optimized choice for a more accurate two-zone combustion model in the case of natural gas SI engines. For this purpose, a computer simulation is developed, and experimental measurements are carried out for comparison and validation.

NUMERICAL MODEL DESCRIPTION

1. Model assumptions

The following assumptions and approximations are considered for simplification:

1. The contents of the cylinder are fully mixed and spatially homogeneous in terms of composition and properties during intake, compression, expansion, and exhaust processes.
2. For the combustion process, two zones (each is spatially homogeneous) are used. The two zones are the burned and the unburned zones. The two zones are always separated by an infinitesimally thin flame.
3. Until the start of combustion, the model is a single zone and undergoes no pre-flame reactions.
4. All gases are considered to be ideal gases during the engine thermodynamic cycle.
5. The cylinder pressure is assumed to be the same for the burned and unburned zones.
6. The heat transfer between the two zones is neglected.
7. The cylinder walls temperature is assumed to be uniform and constant (400 K) [1][7]
8. The intake and exhaust manifolds are assumed to be infinite plenums containing gases at constant temperature and pressure.
9. All crevice effects are ignored, and the blow-by is assumed to be zero.
10. The engine is in steady state such that the thermodynamic state at the beginning of each thermodynamic cycle (two crankshaft revolutions) is the same as the end state of the cycle.

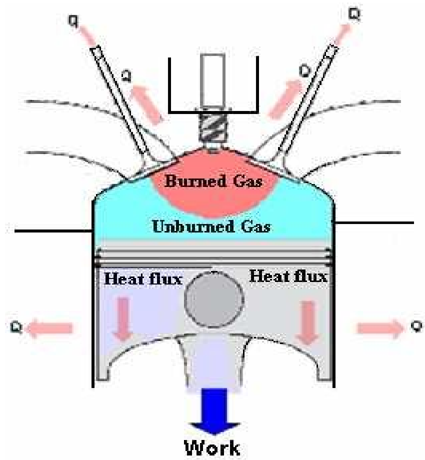


Figure 1 Schematic of the two-zones combustion modeling

2. Model equations

The main equations governing the two zone- model are the energy conservation equation applied to an open system (burned and unburned zones), the equation of ideal gases, the conservation of the mass, the evolution of volumes and different sub-models allowing the simulation of the thermodynamic cycle (sub-models of combustion, heat transfer, mass transfer during the open phases of the combustion chamber and formation of pollutant) [1][8][9].

The total mass is assumed to be constant, since valve leakage and blow-by are neglected.

$$m = m_u + m_b \tag{1}$$

The volume of the two zones is equal to the total cylinder volume, which is a function of the cylinder geometry and crank angle.

$$V = V_u + V_b \tag{2}$$

In each zone, assuming ideal gases and the same pressure, the equation of state gives.

$$P \cdot V_u = m_u \cdot R_u \cdot T_u \tag{3}$$

$$P \cdot V_b = m_b \cdot R_b \cdot T_b \tag{4}$$

The energy equations were written for each zone as follows.

$$\frac{d(m_u u_u)}{d\theta} = -P \cdot \frac{dV_u}{d\theta} + \sum_i \frac{dQ_{ui}}{d\theta} - h_u \cdot \frac{dm_u}{d\theta} \quad (5)$$

$$\frac{d(m_b u_b)}{d\theta} = -P \cdot \frac{dV_b}{d\theta} + \sum_i \frac{dQ_{bi}}{d\theta} + h_u \cdot \frac{dm_u}{d\theta} \quad (6)$$

Combustion sub-model

The S-shaped burned mass fraction profile, the Wiebe function, is often used to determine the burning rate. For SI engines, a simple function with four parameters allows to describe the different configurations of application.

$$x_b = 1 - exp\left[-a_w \cdot \left(\frac{\theta - \theta_0}{\Delta\theta_b} \right)^{m_w + 1} \right] \quad (7)$$

where θ is the crank angle, θ_0 is the crank angle at the start of combustion, $\Delta\ \theta_b$ is the total combustion duration (from $x_b = 0$ to $x_b \approx 1$), and a_w and m_w are adjustable parameters which fix the shape of the curve.

Heat transfer sub-models

- Eichelberg's Correlation (1939)

$$h_g = 7.67 \cdot 10^{-3}\ (V_{mp})^{1/3} \cdot (P.T_g)^{1/2} \quad (8)$$

- Woschni's Correlation (1965-68) [15]

$$h_g = C_0 \cdot \left[B^{-0.2} \cdot P^{0.8} \cdot \left((C_1 \cdot V_{mp}) + C_2 \cdot \frac{V_d \cdot T_1}{P_1 \cdot V_1} \cdot (P - P_{mot}) \right)^{0.8} \cdot T^{-0.53} \right] \quad (9)$$

P is the instantaneous pressure, in bar. C_0= 110-130. C_1 and C_2 are given in table 1

Table 1: C_1 and C_2 Coefficients For Woschni's Correlation

Phase	C_1 [-]	C_2 [m/s.K]
Intake-Exhaust	6.18	0
Compression	2.28	0
Combustion-Expansion	2.28	$3.24\ 10^{-3}$

- Hohenberg Correlation (1979) [19]

$$h_g = C_1 \cdot C_u^{-0.06} \cdot P^{0.8} \cdot T^{-0.4} (C_2 + V_{mp})^{0.8} \quad (10)$$

P is the instantaneous pressure, in bar. The numerical values C_1 = 130 and C_2 = 1.4 appearing in (10) are constants established on base representative of six diesel engines.

- Sitkei correlation [18]

$$h_g = 2.36 \times 10^{-4} \cdot (1+b) \cdot \frac{(P \cdot V_{mp})^{0.7} A^{0.3}}{T^{0.2} \cdot (4V)^{0.3}} \quad (11)$$

With b = 0 - 0.35

- Annand's Correlation [13] [14]

$$h_g = a \cdot \frac{k_g}{B} \cdot Re^{0.7} + b \cdot \frac{(T_g^4 - T_w^4)}{(T_g - T_w)} \quad (12)$$

With a = 0.35-0.8 and b = $4.3 \cdot 10^{-9}$ $W/m^2.K^{-4}$ for SI engines.

3 Model Integration

The preceding equations produce a system of first order differential equations of the form: M (t, y) . $y' = F$ (t, y) The numerical integration of this system, during the combustion process, with crank angle as the independent variable, is obtained by using a Runge-Kutta type method, to determine the following variables $m_u(\theta)$, $m_b(\theta)$, $V_u(\theta)$, $V_b(\theta)$, $P(\theta)$, $T_u(\theta)$ and $T_b(\theta)$.

A Matlab program is developed to simulate the engine operation. The program allows the use of a variable increment to allow an acceptable accuracy with a minimized calculation time.

For the initial values, the thermodynamic cycle simulation starts with assumed guesses of the values of pressure and temperature of the contents within the cylinder at the instant the intake valve opens. After two crankshaft revolutions (720 crank angle degrees), the calculated values of pressure and temperature are compared to the initial guesses. If the calculated values are not within an acceptable tolerance to the initial guesses, the simulation is repeated using the final calculated values as initial guesses.

RESULTS AND DISCUSSION

In order to study the effect of the correlation choice in the accuracy of a two-zone thermodynamic model, several operating conditions have been investigated. Two cases are exhibited (figure 2a, b). It appears clearly (figure 2-4), the influence of the heat transfer correlation choice in the resulting engine cycle performances. To explain this influence the heat transfer coefficient and the burned gas zone heat flux have been traced (figure 5, 6). Hohenberg heat transfer coefficient appears (figure. 5) as a mean function between overestimated (Annand08) and underestimated (sitkei) coefficients. Almost the same trend is given by Annand's correlation with tuning factor equal to 0.3. During compression, woshni and Eichelberg underestimates the heat transfer. However, during combustion Hohenberg over-estimates it. This is due to the specific gas velocity used. In fact, this correlation separates the gas velocity into two parts: the unfired gas velocity that is proportional to the mean piston speed, and the time-dependent, combustion induced gas velocity that is a function of the difference between the motoring and firing pressures.

Each correlation results will be discussed alone, taking into considerations several aspects.

- Woschni: Like noted by Hohenberg [19], this correlation underestimates the heat transfer coefficient during compression and over-estimates it during combustion. However, the effect on the cycle performance is negligible. Its need to motoring pressure makes it difficult to use. It also needs more calculation time comparing to other correlations.

- Hohenberg: It is the best choice. It gives the more accurate results. They are the average results between overestimated (Annand08 heat coefficients) and underestimated (sitkei ones). It's easy to use, and the calculation time is minimized. Almost no tuning has to be performed.

- Eichelberg: Except for compression stroke (figure. 5), like mentioned in reference [4], this correlation results are acceptable. Like Hohenberg's correlation, it's easy to use, and the calculation time is minimized. Almost no tuning has to be performed. So, it's a second choice.

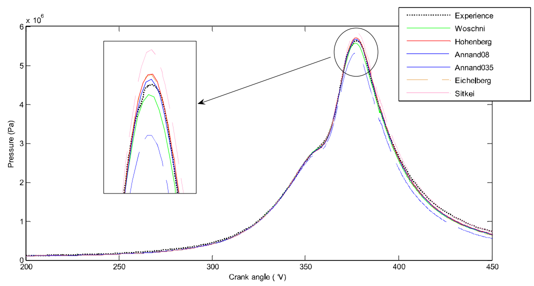
- Annand: This correlation, comparing to others, includes radiation term. However, this term doesn't have a big influence. Its tuning factor has a big effect on the heat transfer coefficient and the corresponding engine cycle performance. In fact, if the (*a*) parameter is set to 0.8 the heat transfer coefficient is overestimated (figure 5) and hence the engine cycle performance is underestimated (figure 2-4). This makes it not very interesting to use in thermodynamic models. Using a phenomenological combustion model, a calibration has to be performed in the combustion sub- model.

- Sitkei: It shows a bad accuracy. The heat transfer coefficient is underestimated (figure 5) and the engine cycle performance is consequently overestimated (figure 2-4).

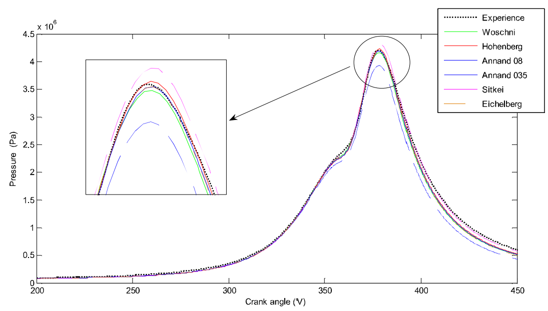
The following table summarizes those comparing elements:

Table 3: Correlations comparison

	Accuracy	Use	Calculation time	Tuning
Hohenberg	Good	Easy	Good	No tuning
Eichelberg	Acceptable	Easy	Good	No tuning
Woschni	Acceptable	Difficult	More	Need
Annand	Depends on tuning	Easy	Good	Big influence
Sitkei	Bad	Easy	Good	Need

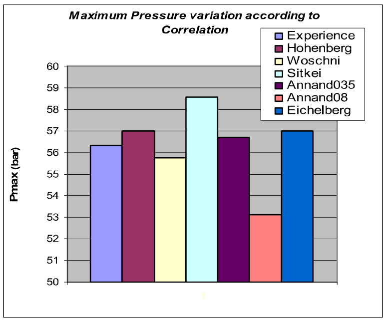


(a)

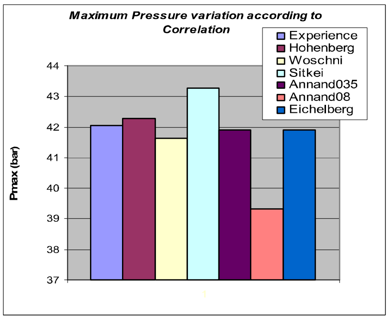


(b)

Figure 2 Pressure comparison for different correlations within two cases:
(a) Ø=1,α=9° c.a, Full load
(b) Ø=1,α=9° c.a, Partial load

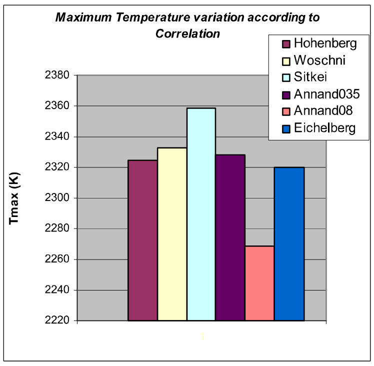


(a)

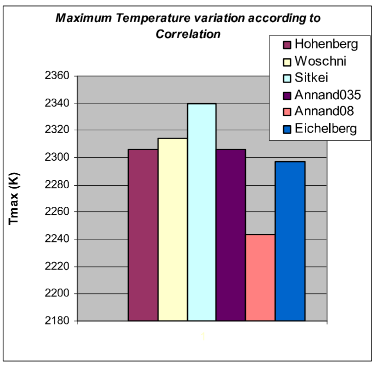


(b)

Figure 3 Maximum Pressure comparison for different correlations within two cases:
(a) Ø=1,α=9° c.a, Full load
(b) Ø=1,α=9° c.a, Partial load

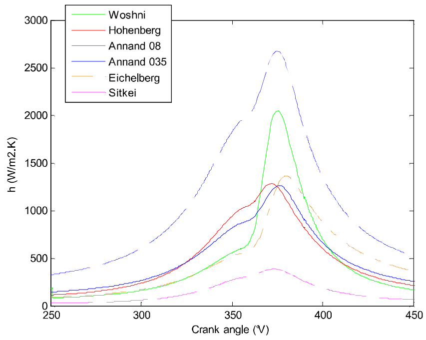


(a)

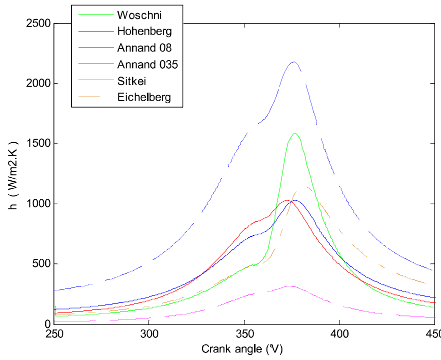


(b)

Figure 4 Maximum temperature comparison for different correlations within two cases:
(a) Ø=1,α=9° c.a, Full load
(b) Ø=1,α=9° c.a, Partial load

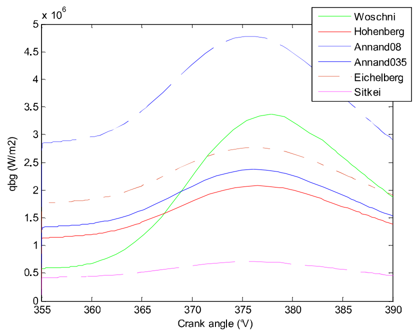


(a)

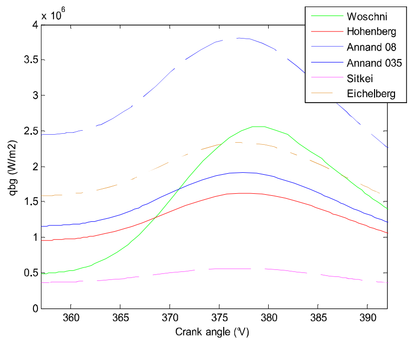


(b)

Figure 5 Heat transfer coefficient comparison for different correlations within two cases:
(a) Ø=1,α=9° c.a, Full load
(b) Ø=1,α=9° c.a, Partial load



(a)



(b)

Figure 6 Burned gas zone heat flux density comparison for different correlations within two cases:
(a) Ø=1,α=9° c.a, Full load
(b) Ø=1,α=9° c.a, Partial load

CONCLUSION

The accuracy of an engine simulation depends on the precision of the heat transfer estimation.
In previous works using two-zone model for natural gas SI engines, many heat transfer correlations have been used and no choice justification has been given.
In the present study, the effect of the heat transfer correlation choice has been investigated. The most known correlations have been tested and compared considering different aspects, and using experimental results. It's found that Hohenberg's correlation is the best choice. It gives the more accurate results. It's easy to use, and the calculation time is minimised. Almost no tuning has to be performed. Even though not like the first one, Eichelberg's correlation is also an acceptable choice.

REFERENCES

1. Amr Ibrahim and Saiful Bari Optimization of a natural gas SI engine employing EGR strategy using a two-zone combustion model, *Fuel*, *2008.*
2. Seref Soylu and Jon Van Gerpen Development of empirically based burning rate sub-models for a natural gas engine, *Energy Conversion and Management 45, pages 467-481, 2004.*
3. Guibert P., Modélisation du cycle moteur – Moteurs à allumage commandé, *Technique de l'ingénieur BM 2511*
4. Yuh-Yih Wu, Bo-Chiuan Chen, Feng-Chi Hsieh, Heat transfer model for small-scale air-cooled spark-ignition four-stroke engines, *International Journal of Heat and Mass Transfer 49,3895–3905, 2006.*
5. Caillol C., Berardi G., Brecq G., Ramspacher M., Meunier P., A simulation tool for evaluating gas compoition effects on engine performance. *International Gas Research Conference , Novembre 2004.*
6. Caillol C, Porterie B, Delorme T, Modélisation phénoménologique de la combustion dans un moteur à gaz fonctionnant en mélange pauvre: Application à la prédiction des émissions polluantes. *12ème Journées Internationales de thermique, Novembre 2005.*
7. Ollivier E., Contribution à la caractérisation des transferts thermiques dans les moteurs à allumage commandé. Application à la détection du cliquetis, *PhD thesis ENSTIM de Nantes*, 2006.
8. Guibert P., Modélisation du cycle moteur -Approche zérodimensionnelle, *Technique de l'ingénieur BM 2510.*
9. Lounici M S, Tazerout , Balistrou M, Effet de changement de combustibles sur les émissions de GES par les automobiles en zones urbaines en Algérie, *4ème édition du Colloque francophone en énergie, environnement, économie et thermodynamique, Cofret-08, 2008*
10. Heywood, J., Internal combustion engine fundamentals, McGraw-Hill New York, USA 1988.
11. Junseok Chang, Orgun Guralp, Zoran Filipi, Dennis Assanis, T Kuo, P Najt and R Rask, New Heat Transfer Correlation for an HCCI Engine Derived from Measurements of Instantaneous Surface Heat Flux. *Society of Automotive Engineers, 2004.*
12. Eichelberg, G., Some New Investigations on Old Combustion Engine Problems, *Engineering 148, 463-446.547-560, 1939.*
13. Annand, W. J. D., Heat Transfer in the Cylinder of Reciprocating Internal Combustion Engines, *Proc Instn Mech Engrs*, *Vol. 177, pp 973-990, 1963.*
14. Annand, W. J. D., Ma, T. H., Instantaneous Heat Transfer Rates to the Cylinder Head Surface of a Small Compression-Ignition Engine, *Proc Instn Mech Engrs*, *Vol. 185, pp 976-987, 1970-1971.*
15. Woschni, G., A Universally Applicable Equation for the instantaneous Heat Transfer Coefficient in the Internal Combustion Engine, *SAE Paper 670931, 1967.*
16. Shiling, K., Woschni, G., Experimental Investigation of the Instantaneous Heat Transfer in the Cylinder of a High Speed Diesel engine, *SAE Paper 790833, 1979.*
17. Huber, K., Woschni, G., Zeilinger, K., Investigations on Heat Transfer in Internal Combustion Engines under Low Load and Motoring Conditions, *SAE Paper 905018, 1990.*
18. Sitkei, G., Ramanaiah, G. V., "Rational Approach for Calculation of Heat Transfer in Diesel Engines", *SAE Paper 720027, 1972.*
19. Hohenberg, G. F., Advanced Approaches for Heat Transfer Calculations, *SAE Paper 790825,1979.*
20. J. D. Trapy. Transferts thermiques dans les moteurs à combustion interne. Mesure et modélisation. *Revue Générale de Thermique*, *Vol. 233, p.385-390, 1981.*
21. G. Borman and K. Nishiwaki. Internal-combustion engine heat transfer. *Progress in Energy and Combustion Science*, *Vol.13, p.1-46, 1987.*
22. Seref Soylu, Examination of combustion characteristics and phasing strategies of natural gas HCCI engine, *Energy Conversion and Management 46, pages 101-119, 2005.*
23. Shapiro HN, Van Gerpen JH. Two zone combustion models for second law analysis of internal combustion engines. *Society of Automotive Engineers; 1989. SAE Paper no. 890823, 1989.*
24. Ferguson CR. Internal combustion engines. *Wiley & Sons; 1986.*
25. T. Oguri, On the coefficient of heat transfer between gases and cylinder walls of the spark-ignition engine, *Bull. JSME 3 (11)) 363–369,1960.*
26. P.J. Shayler, S.A. May, T. Ma, The determination of heat transfer from the combustion chambers of SI engines, *SAE Paper No. 931131, 1993.*